Feb. 3, 1970 H. H. PLATT 3,492,948

HYDRAULIC PUMP/MOTOR

Filed April 8, 1968 2 Sheets-Sheet 1

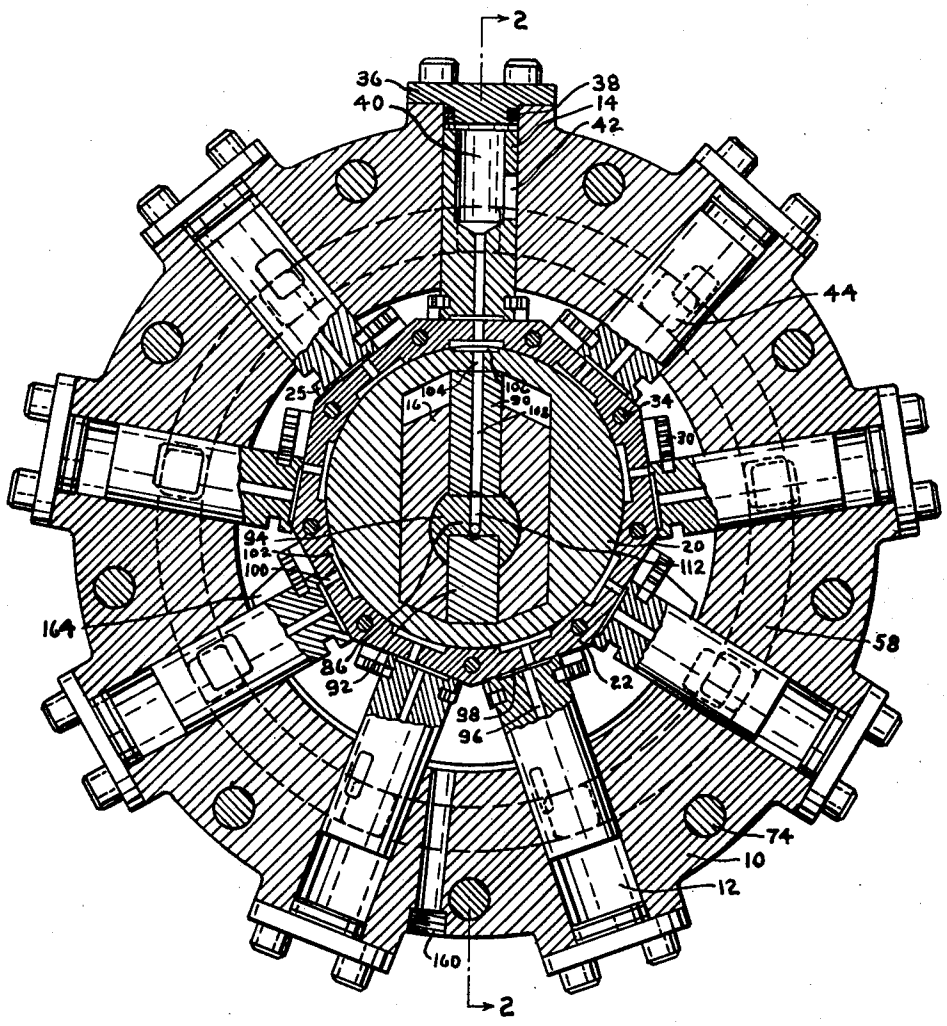


FIG 1

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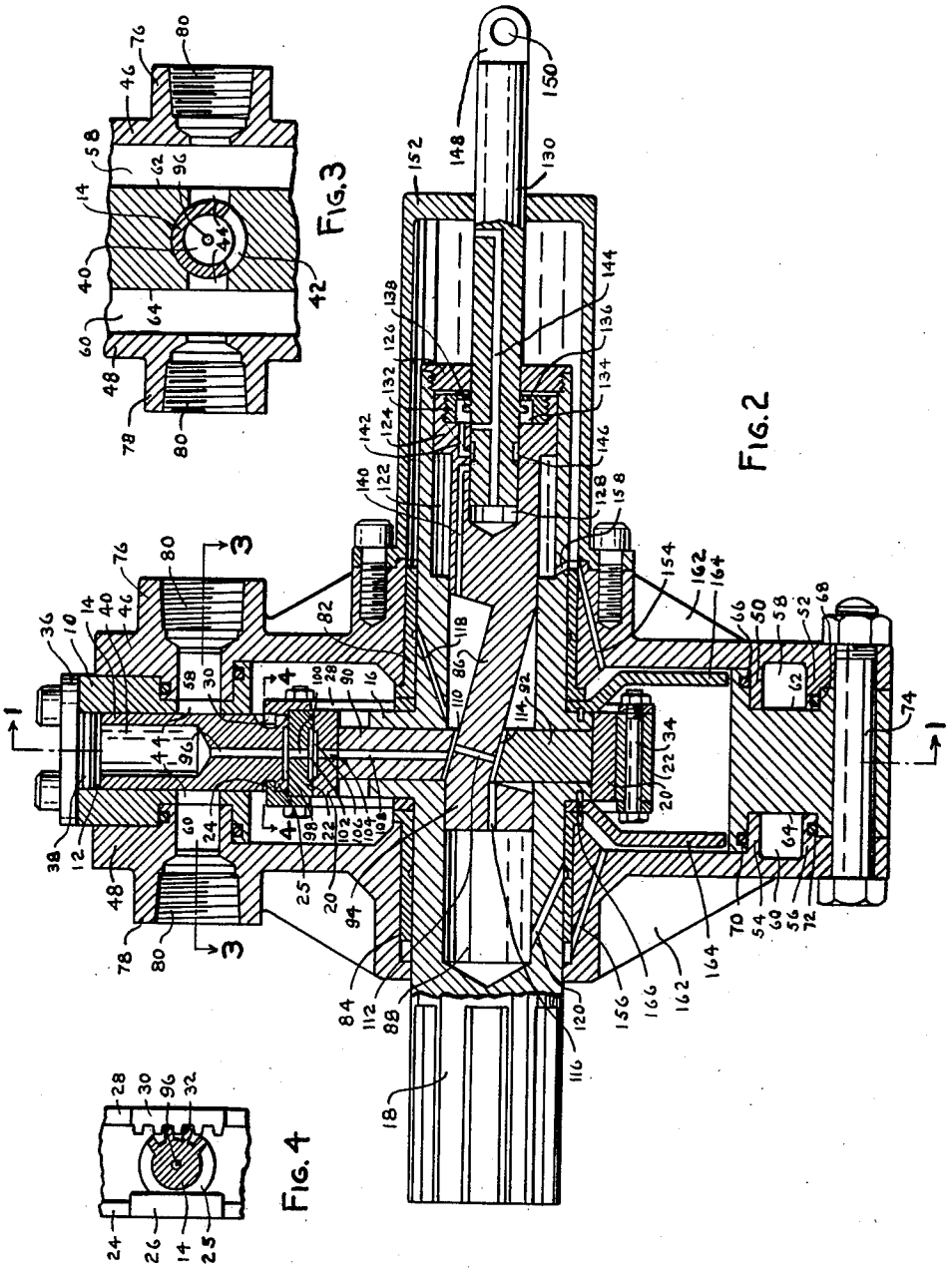


FIG. 2

FIG. 3

FIG. 4

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HYDRAULIC PUMP/MOTOR

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ABSTRACT OF THE DISCLOSURE

The device disclosed is a radial cylinder hydraulic unit characterized by the utilization of rotational reciprocation of each plunger to effect valve actuation. In the embodiment shown a ring assembly surrounding a driving eccentric and held against rotation with said driving eccentric is fitted with toothed racks adapted to engage gear teeth on the plungers. The portion of the ring assembly adjacent each plunger, while not rotating with the eccentric, does undergo a transverse reciprocation relative to the plunger. This reciprocation is translated into rotary reciprocation by engagement of the rack with the plunger teeth.

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.

The objects of my invention may be summarized as improvements over hydraulic pumps and motors previously known in the following characteristics of performance:

Compactness
Low cost
Light weight
Long endurance
High speed capability
High power capacity
Simplicity of fabrication and assembly
Smoothness of fluid and power flow
High rigidity
High efficiency
Adaptability for automatic control
Choice of fixed or controllable displacement.

My invention accordingly consists in the features of construction, combination of elements and arrangement 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 the embodiment of my invention at present favored by me.

FIG. 1 is a transverse sectional view taken along line 1—1 of FIG. 2 of one form of power unit embodying my invention illustrating particularly the arrangement of the plungers and cylinders;

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

FIG. 3 is a fragmentary sectional view taken along line 3—3 of FIG. 2 illustrating particularly the arrangement of valve parts and fluid ducts;

FIG. 4 is a fragmentary sectional view taken along line 4—4 of FIG. 2 illustrating particularly the means for imparting rotary motion to the plungers thereby to actuate the valve ports.

The present invention is an improvement over the hydraulic power unit which forms the subject of my United States Patent No. 3,255,707.

The basic principle of this improvement is the utilization of plungers for the dual functions of pressure pistons and of valves through imparting to each of said plungers two motion components phased ninety degrees apart, one of said motion components being axial reciprocation and the other rotational reciprocation, the axial motion serving the pressure function and the rotational motion actuating the valve ports in correct timing for the entrance and discharge of hydraulic fluid.

In the embodiment illustrated nine cylinders are provided so as to furnish the maximum utilization of space and smoothness of flow, although any other number of cylinders may be substituted if preferred. The plungers are driven from or drive a drive-shaft depending on whether the unit is functioning as a pump or 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 drive-shaft and in the second instance power is taken off through the drive-shaft.

The cylinder ring **10** is bored radially to form the nine cylinders **12**. The plungers **14**, fitting slidably in cylinders **12**, are operatively connected to the driving block **16** integral with the drive-shaft **18**. The driving connection between block **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 block **16** an eccentric **20**. Surrounding the eccentric **20** is a slidably fitted ring member **22** the outer surface of which is formed with nine 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 **24** is also provided to hold the inner ends of plungers **14** in contact with the flat bearing surfaces of the ring **22** by engagement of the flanges **25** of plungers **14** with the tangentially disposed flanges **26** of ring **24**. A second ring member **28** is mounted at the other side of the eccentric **20**, ring **28** being similar to ring **24** except that, in place of the solid flanges **26**, toothed racks **30** are formed on it. The teeth of racks **30** are arranged to mesh with suitable gear teeth **32** formed on plungers **14** contiguous to their inner ends. Rings **24** and **28** are secured to ring member **22** by any suitable means such as bolts **34**. While rings **24** and **28** can well be integral parts they may each be divided into nine segments to facilitate fabrication.

The cylinders **12** are closed at their outer ends by cylinder heads **36** bolted to the cylinder ring **10** and sealed by suitable packing such as the O-rings **38**. Plungers **14** are bored out to provide the internal passages **40** and generally rectangular ports **42** are cut through the walls between passages **40** and the exterior surfaces of plungers **14**. The side faces of the cylinder ring **10** are pierced with oppositely disposed ports **44** adapted to register with the plunger ports **42**. Fitted at the sides of the cylinder ring **10** are the side plates **46** and **48**. Annular ring extensions **50** and **52** on side plate **46** and extensions **54** and **56** on plate **48** form ring channel manifolds **58** and **60**, one of which is shown by dashed lines on FIG. 1. Adapted to receive closely flanges **50**, **52**, **54** and **56** are annular grooves in the side faces of cylinder ring **10**, the parallel inner faces **62** and **64** of these grooves thus form the fourth sides of the annular manifolds **58** and **60**. Fluid sealing is provided by any suitable packing such as the O-rings **66**, **68**, **70** and **72**. Side plates **46** and **48** are secured to cylinder ring **10** by suitable retention means such as the bolts **74**. Bosses **76** and **78** on side plates **46**

and 48 provide connections from manifolds 58 and 60 to external tube lines. Any of the available types of connection may be used such as the pipe-threaded sockets 80. Side plates 46 and 48 are bored centrally to retain bearing bushings 82 and 84 adapted to support with a suitable running fit drive-shaft 18.

A variety of means for controlling the displacement by changing the throw of the eccentric are available. The type here shown and at present preferred is one in which displacement of the solid eccentric 20 is actuated by a linear cam 86 slidable within a centrally bored cylinder 88 in drive-shaft 18, transmission of motion from cam 86 to eccentric 20 being effected by the two rods 90 and 92 which are slidable in a transversely bored hole in driving block 16. Preferably, for purposes of stiffness and guidance, cam 86 is formed in the space between two opposed inclined slots in the cylindrical plunger 94, as shown particularly in FIG. 1.

Lubrication of all bearing surfaces, as well as actuation of cam 86, are effected by hydraulic pressure derived from the pressure cylinders. To this end a small diameter drilled hole 96 extends from passage 40 to the center of the inner face of each plunger 14 where it connects with a circular recess 98. The recess 98 serves the double function of distributing oil to the bearing face of plunger 14 and of providing a balancing pressure and thus reducing friction between plunger 14 and the ring 22. A similar duct 100 extends from the center of each flat on ring 22 through the wall of ring 22 and connects with a counterbored recess 102. While the opening of duct 100 reciprocates with relation to recess 98 and plunger 14 the proportions are such that it is always in connection with recess 98 and never leaves the confines of the end surface of plunger 14. The counterboard recess 102 serves to provide a balancing pressure, thus reducing friction between eccentric 20 and ring 22, and also has an oil distributing function. Eccentric 20 has also a single drilled duct 104 substantially in line with its eccentricity axis and central of the circular recess 106. As the eccentric rotates, recess 106 connects successively with each recess 102 and is thus almost continuously in connection with one or another of the passages 40. The diameter of recess 106 is slightly less than the space between two contiguous recesses 102 and thus avoids direct connection between cylinders 12 at any time. Rod 90 is drilled axially to provide the duct 108 which is adapted to lead fluid to the bearing face of rod 90 on cam 86. Transverse groove 110, passage 112, groove 114 and passage 116 afford oil connections to all parts of cylinder 88 at all times. Leads 118 and 120 are adapted to supply high-pressure lubricating oil to bushings 82 and 84.

A servo control provision effects actuation of displacement change. For this purpose cylinder 88 in shaft 18 terminates in an enlarged cylinder 122 in which is slidably fitted an enlarged portion 124 of plunger 94 capable of acting as a piston in cylinder 122, which is closed by a cylinder head 126 secured by screw threads, as shown, or otherwise to the end of shaft 18. A small-diameter cylindrical recess 128 is formed in the end of plunger 94 in which is slidably fitted the end of servo control rod 130. Piston 124 is recessed at its outer end to receive a threaded plug 132 which is inwardly flanged to form an annular recess 134. Plug 132 with its flange 136 serves to confine a flange 138 on rod 130 loosely so as to permit a small amount of end play. Flange 136 does not contact rod 130 but allows fluid passage from recess 134 to the outer face of piston 124 and plug 132. A passage 140 in plunger 94 leads to a point in the wall of recess 128 and from a point closely adjacent a second passage 142 leads from the wall of recess 128 to the annular recess 134. In control rod 130 a passage 144 leads from its inboard end to two points on its surface, one in position to cooperate with recess 134 and the other outside of cylinder head 126, which fits rod 130 snugly but slidably. An annular groove 146 on rod 130 is adapted to cooperate with passages 140 and 142 by connecting and disconnecting them.

Rod 130 terminates in a flat portion 148 with a hole 150 for attachment to a suitable control lever or the like, not shown, adapted to prevent rotation of rod 130. Alternately rod 130 may be allowed to rotate with the drive-shaft, the connection at 148 being arranged for rotation. A cylindrical casing 152 bolted to side plate 46 provides a substantially fluid-tight closure. Passages 154 and 156 provide drainage in to the interior of cylinder ring 10 for the small amount of fluid that may pass bushings 82 and 84 and rod 130, as well as that discharged in operation of the servo system. Passage 158 drains the cylinder 122, thus maintaining pressure behind piston 124 always at the low pressure in casing 152. Drainage from cylinder ring 10 may be provided, as at 160. Outlet 160 may lead to a gravity sump or to a small auxiliary pump capable of returning the fluid to the operative hydraulic system, as in common practice. Rigidity of the unit is enhanced by the stiffening ribs 162 integral with side plates 46 and 48. Alternatively, corrugation of the side plate walls may be substituted for the ribbing shown.

An optional feature is the counterweighting provided by the segmental plates 164 which are extensions of the end bearing washers for bearings 82 and 84. The plates 164 are constrained to rotate with shaft 18 by pins 166. For the constant displacement power unit the counterweights may have a weight and radius which exactly balance the weights of the eccentric and plungers. For the variable displacement unit an exact balance is attained for only one displacement setting. The minimum unbalance is attained by selecting counterweight mass which balances completely at a displacement of one-half the maximum. Then the greatest unbalance occurs at zero displacement and maximum displacement, being in each case one-half the amount that would obtain at full displacement without counterweights. If the unit is intended for use most of the time at some particular displacement the counterweighting can be adjusted accordingly. On account of the small eccentricity and light weight of the rotating and reciprocating parts the amount of residual unbalance is in any case unobjectionable for most installations. With slight added complication automatically adjustable counterweights may be provided to accomplish complete balance under all conditions of operation.

In order to illustrate more clearly the valve action, only the operative edges of the valve and cylinder ports 42 and 44 are shown in FIG. 1. It will be noted that the plunger ports 42 on the right side in FIG. 1 are in register with the cylinder ports 44 of manifold 58, as indicated by their dashed-line outlines, while ports 42 on the left hand side register with the ports 44 of manifold 60, as indicated by the solid outlines. Thus, if eccentric 20 is rotated clockwise fluid flows into those cylinders 12 which are to the right of the eccentric axis from manifold 58 and discharges from the cylinders on the left into manifold 60. For anti-clockwise rotation the sequence is reversed. Thus the connections to manifolds 58 and 60 may alternate as to high or low pressure depending on direction of rotation and operation as pump or motor. It may be noted that the port opening area can be increased by elongating ports 42 and 44 axially. This, on the other hand, increases size and weight. Therefore a considerable latitude of design variation is available in seeking optimum proportions for different requirements of speed and pressure. Another consideration in this connection is that of side force on the plungers due to the unbalanced port area. With the proportions shown friction losses due to this cause are very slight. Port elongation will however increase them.

Displacement variability is controlled by changing the eccentricity of eccentric 20. The driving block 16 is displaced radially from the shaft center a distance equal to one-half the designed maximum eccentricity, which is in turn one-half the maximum stroke of plungers 14. Ec-

centric **20** has freedom to slide on the block **16** through a distance again equal to one-half the maximum eccentricity. Thus, in the position shown in FIGS. 1 and 2 the eccentric is at its maximum eccentricity, while when at the opposite end of its sliding range the eccentricity is zero. Thus by moving eccentric **20** through the short distance shown a complete range of plunger stroke from maximum to zero is accomplished. Control of eccentricity and consequently of displacement is effected by axial sliding of control plunger **94**, motion to the left in FIG. 2 serving to reduce displacement. It will be noted that valve timing remains correct for all eccentricity settings.

In operation, recess **106** in the periphery of eccentric **20** sweeps past recesses **102** in ring **22** as the shaft **18** revolves, thus connecting passage **108** successively with passages **96** of all plungers **14**. During each of said connections passage **108** connects through recess **40** and port **42** for one-half its traverse with manifold **60** and the other half with manifold **58**. The pressure in duct **96** thus fluctuates nine times per revolution between the low and high pressure lines, the effective pressure in shaft cylinder **88** being therefore generally one-half that of the high pressure line. Thus pressure in the bearings throughout is established at a satisfactory value for low friction without excessive leakage. Still another requirement is satisfied by the periodic connection of the duct system to low pressure. If cylinder **88** were continually at high pressure motion of plunger **94** would be blocked. With periodic opening to the low pressure side this difficulty is avoided since the raising of pressure above average in cylinder **88** will permit an outflow allowing motion of plunger **94**. The hydraulic fluid in common use is nearly but not quite incompressible. The possible loss of power which might be caused by compression and expansion of the fluid in cylinder **88** and its connections has been investigated and it has been found that the damping effect of the narrow passages at the normally high frequency of pressure fluctuation ensures negligible loss from this cause.

As shown particularly in FIG. 2 the position of control rod **130** is such as to block passage of fluid from cylinder **88** and from recess **134**, the eccentric being thus locked in position for maximum displacement. A short shift of rod **130** to the left causes groove **146** to connect ducts **140** and **142**, thus admitting fluid under pressure to recess **134** and, through it, to the outboard end of cylinder **122** where it exerts a force on piston **124** tending to move it and all of plunger **94** to the left. This force is opposed by the pressure on the smaller end of plunger **94** in cylinder **88**. However, cylinder **122** has approximately twice the cross-sectional area of cylinder **88** and consequently plunger **94** and cam **86** are moved to the left, thus reducing eccentricity. If the displacement of rod **130** is stopped at any point the motion of plunger **94** cuts off the connection between ducts **140** and **142** and thus arrests the displacement change. A subsequent motion of control rod **130** to the right connects recess **134** and through it the outboard end of cylinder **122** with the unpressurized space in casing **152** through passage **144**. Pressure in cylinder **88** then becomes predominant and forces plunger **94** to the right, thus increasing eccentricity. This motion is again arrested by the shutting off of the connection if rod **130** is stopped. It is thus evident that the cam **86** closely follows the motion of the control rod. Furthermore, since all forces except friction are balanced, only a very small force is required to effect control. This is important for operation by an automatic external mechanism. Flange **138** affords direct actuation of the control when the system is not pressurized. Reverse of rotation is effected by a suitable valve in the external lines, not shown.

The three radial bearings, that of ring **22** on eccentric **20** and the two bushings **82** and **84**, are of the hydrostatic type which is recognized in the art as having a longer endurance life than the more usual ball and roller types of bearing, while having comparatively low coefficients of friction. In the hydrostatic bearings, as exemplified here, the high pressure of the lubricating oil maintains a film between the metal surfaces which prevents direct contact and thus minimizes friction and wear. It is to be noted in this connection that the counterbores **102** greatly reduce pressure on the eccentric, thus further diminishing friction. The same principle may be applied to the drive-shaft bearings. To this end the drive-shaft may be relieved in the bearings over a considerable area on the side opposite the point of application of the normal pressure resultant force. This expedient, not illustrated, is effective for only one set of operating circumstances. For example, if the unit is operated as a pump the bearing relief will be correct for only one direction of rotation. For most installations most or all running is confined to one direction of shaft rotation and therefore bearing relief is advantageous in reducing friction.

While the power unit which has been described and illustrated herein is adapted for variable displacement it is evident that the basic principles are applicable to the operation of a unit having constant displacement also. For this purpose it is required only to omit the control provision and fix the eccentric to the drive-shaft.

In the use of hydrostatic bearings flexure of the shaft and bearing supports has been found to be of vital importance. This is so because a very slight misalignment destroys the uniformity of thickness of the oil film in the bearings, which may result in excessive friction and wear. For this reason the very short distance between bearings and the stiffness of the bearing supports make the hydraulic unit of the present invention outstanding in endurance and efficiency.

A further great advantage inheres in the simplicity of the various parts and the facility of their assembly. The result is a considerable saving in production cost.

The drawings illustrate a unit having low production cost for service in which weight is not a prime consideration. Accordingly, the cylinder ring **10** may advantageously be of cast iron which is known to have good bearing and machining properties. Even with this relatively heavy metal units of the present invention compare favorably in lightness with units of other designs. However, for extreme lightness the cylinder ring **10** and also the side plates **46** and **48** may be of light metal such as aluminum or magnesium alloys. If soft metals are used the cylinders **12** may be fitted with thin-walled liners of hardened steel.

It may be noted from examination of FIG. 1 that the flat portions on ring **22** are at all times held in contact with plungers **14**. Consequently ring **22** does not rotate with eccentric **20**. Each flat however, as the result of the eccentric rotation, has a reciprocating sliding component of motion which is transverse of the plunger axis. This motion is phased ninety degrees from the axial oscillation of the plunger. This relationship is readily apprehended by inspection of FIG. 1 in which it is seen that racks **30**, which are fixed on ring **22**, are central of plungers **14** when the eccentric **20** is in line with a cylinder axis while they have a maximum displacement when the eccentric is at right angles to a cylinder axis, the displacement being equal and opposite on the right and left sides of FIG. 1. The reciprocating motion of the racks **30** is converted into rotational reciprocation of the plungers **14** by meshing with the plunger teeth **32**.

No means of support for the unit is illustrated. However, it is evident that either of the side plates **46** and **48** provides excellent opportunity for the fitting of brackets and the like. Bolts **74** may then provide secure attachment.

While only one form of 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.

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 rotationally fixed on said shaft, a casing supporting said shaft, radial cylinders in said casing, plungers in said cylinders, each of said plungers being hollow at its outer end, a port in the wall of each of said plungers, ports in said casing adapted to register with said plunger ports, a ring member mounted on and surrounding said eccentric, flats on the outer surface of said ring member adapted to contact the inner ends of said plungers, gear teeth formed on the inner end of each of said plungers and rack teeth mounted on said ring member adjacent each of said flats, said rack teeth being adapted to engage the gear teeth on said plungers whereby rotation of said eccentric causes both axial and rotational reciprocation of said plungers.

2. In a hydraulic power unit as set forth in claim 1, said casing comprising a cylinder ring and two end plates, said end plates having annular grooves forming with the cylinder ring fluid manifolds connecting said cylinder ports on each side of said cylinder ring.

3. In a hydraulic power unit as set forth in claim 1, said end plates comprising also bearings for said shaft.

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U.S. Cl. X.R.

91—233; 103—38, 174