

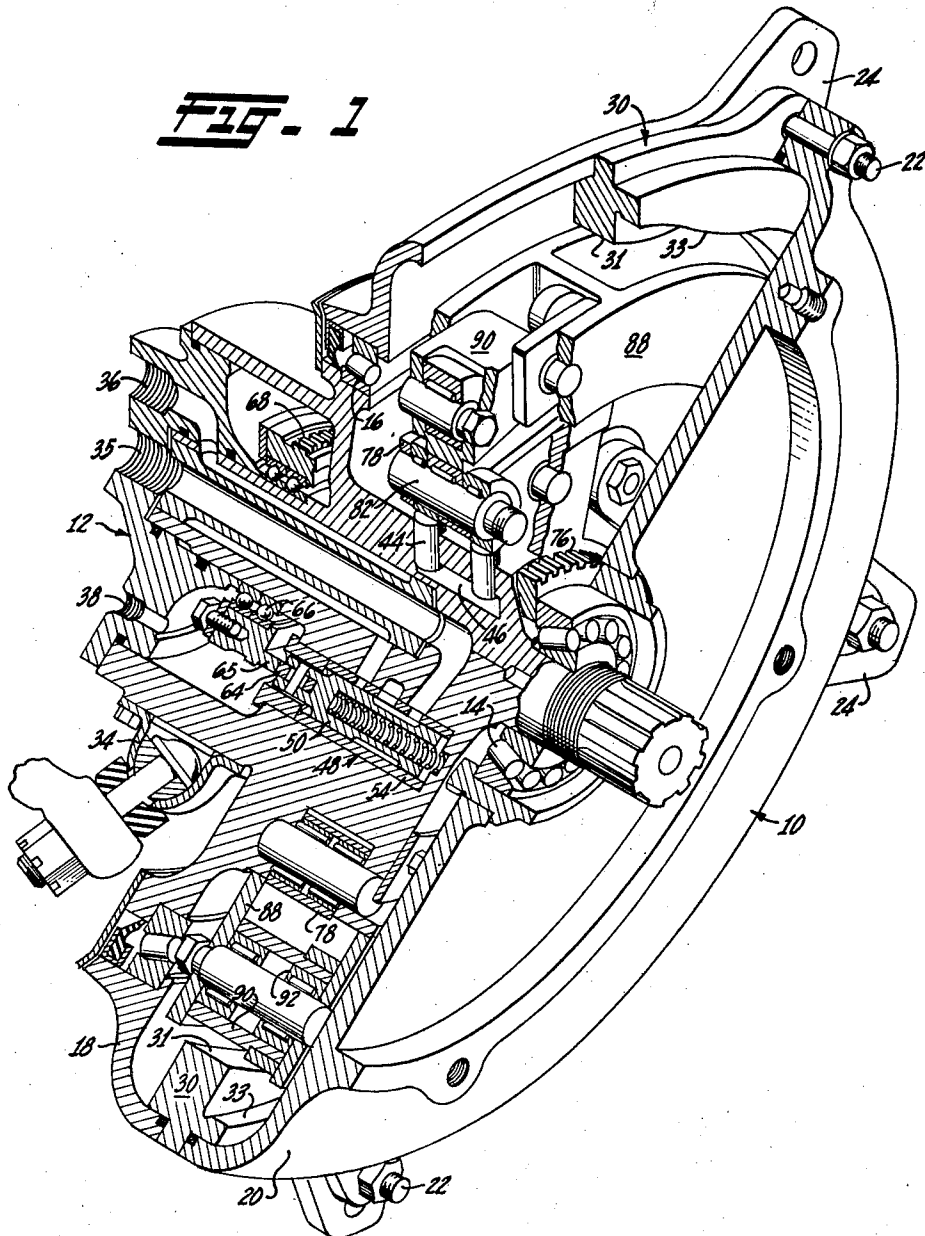
May 5, 1964

E. ORSHANSKY, JR
HYDRAULIC MOTOR OR PUMP

3,131,604

Filed April 13, 1962

7 Sheets-Sheet 1



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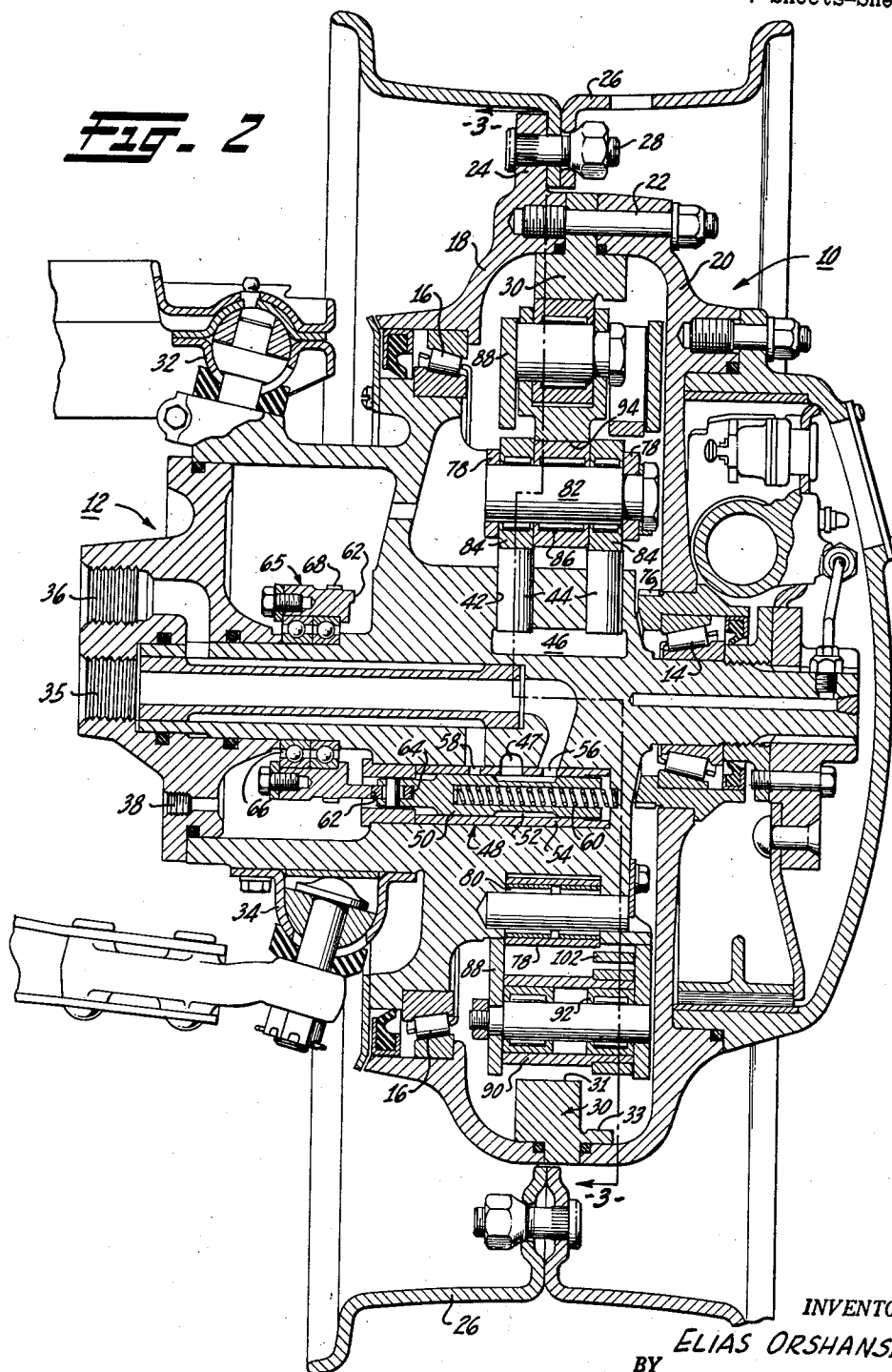
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FIG. 2



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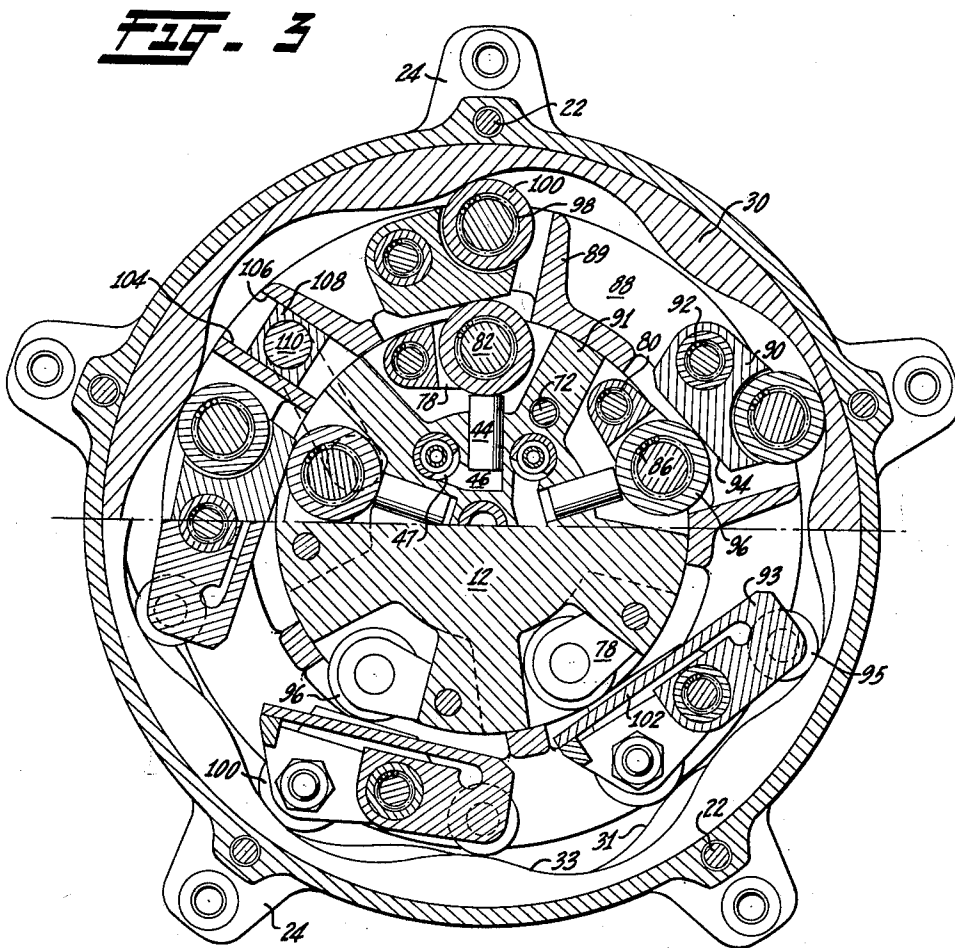
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7 Sheets-Sheet 3



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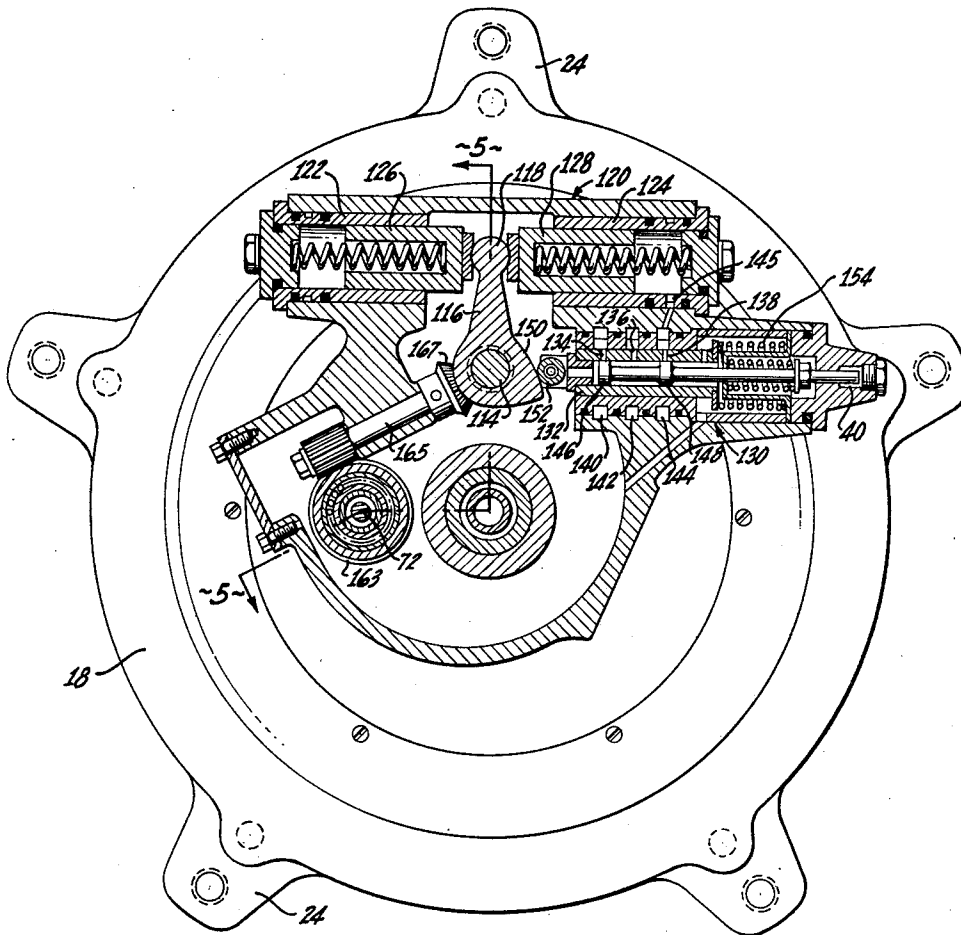
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Fig. 4



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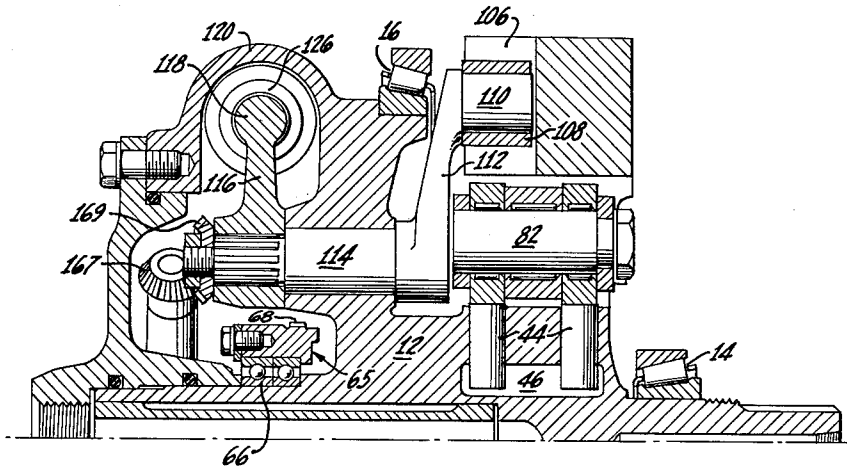


FIG. 5

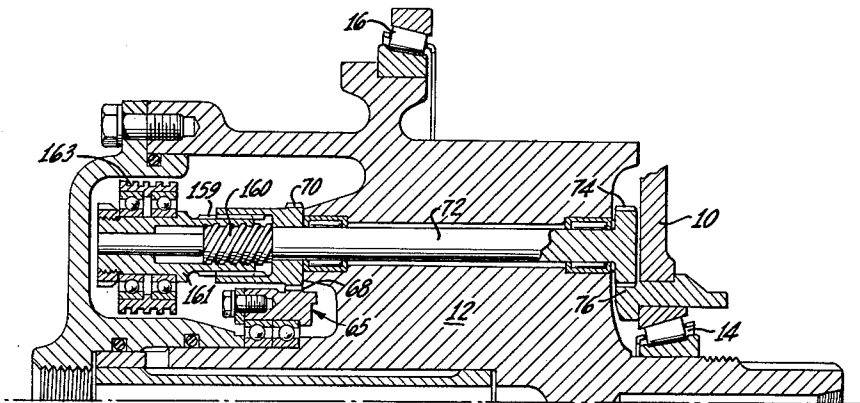


FIG. 6

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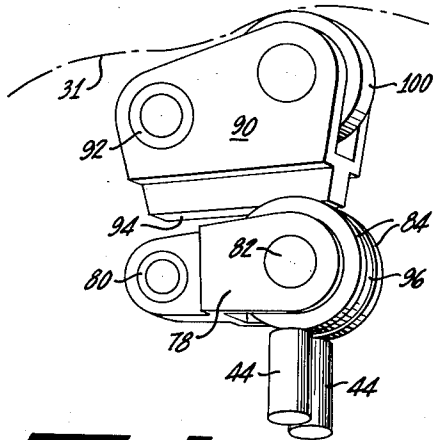


FIG. 7

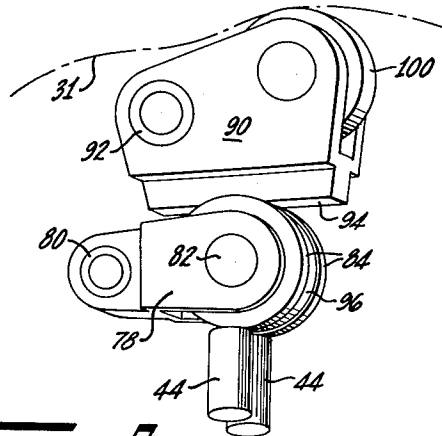


FIG. 8

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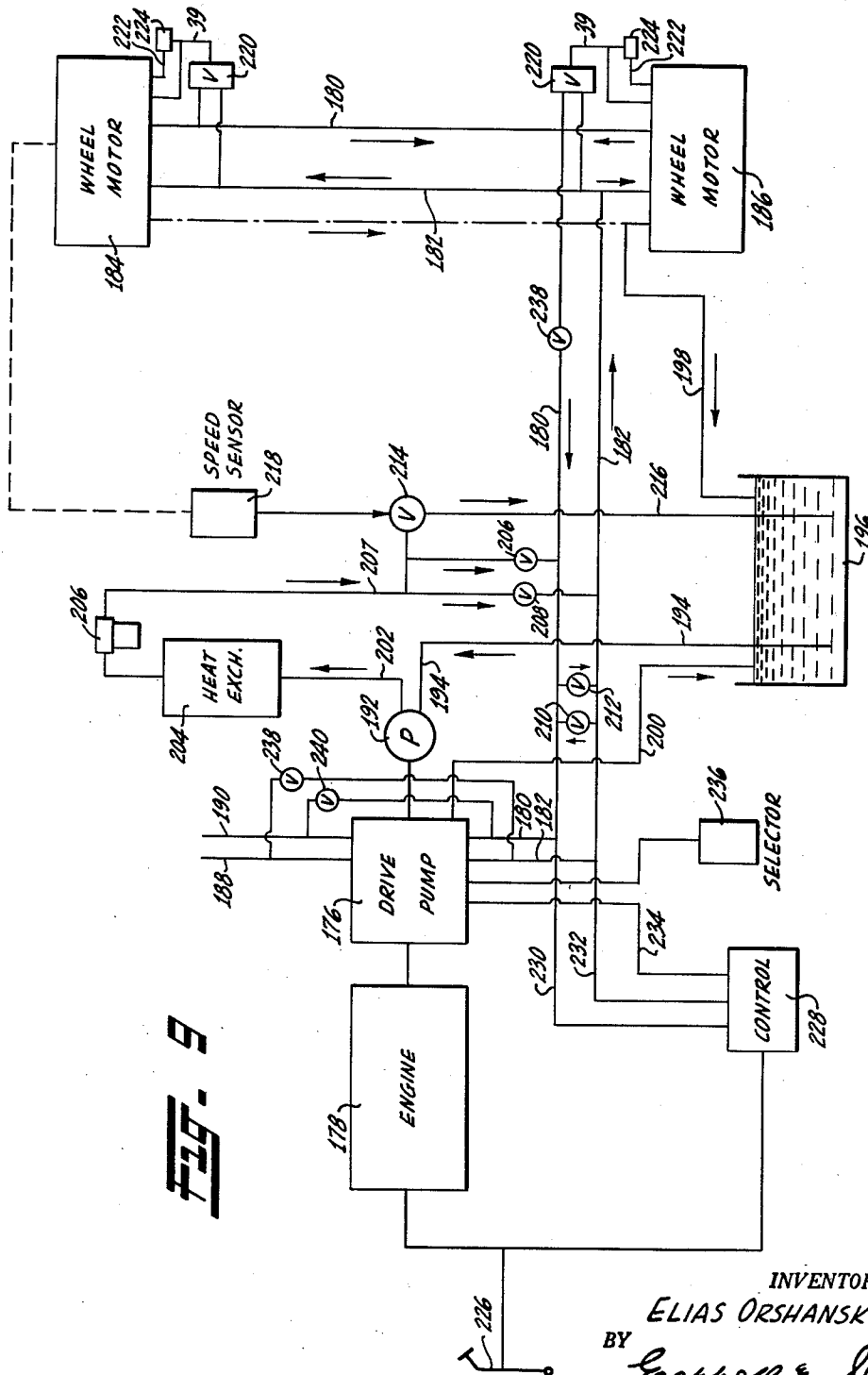


FIG. 9

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3,131,604

HYDRAULIC MOTOR OR PUMP

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10 Claims. (Cl. 91-180)

This invention relates to a hydraulic motor and more particularly relates to a hydraulic motor which is adapted to be mounted within a wheel rim. In conventional drive systems, a gear shifting mechanism is necessary together with drive shafts, universal joints, differentials and axles. All of these parts are eliminated with the drive of the present invention. Perhaps the most important factor is that the fluid drive of the present invention eliminates the gear shifting and provides both an efficient and powerful drive at low speeds, as well as a smooth transmission into high-speed operating conditions.

In accordance with the present invention, a hydraulic drive system is provided having a multiple lobe cam so that there is a large number of driving strokes per revolution of the wheel. Further, there is a smooth flow of power and low radial loads on the main bearings since a large number of pistons with cross-firing are employed.

The drive of the present invention also provides a novel system wherein anti-friction bearings can be used throughout and one in which there is no substantial side thrust on the pistons.

The drive of the present invention also embodies a system whereby the stroke, and therefore the effective displacement, of the motor can be smoothly varied over an infinite ratio. This is accomplished by a lever system later described in detail.

The hydraulic motor of the present invention utilizes individual spool valves.

Other features of the invention will be apparent from the specification which follows.

In the drawings forming part of this application:

FIGURE 1 is a perspective view of a motor embodying the present invention with certain of the parts cut away to show the internal construction.

FIGURE 2 is an end view in section of a motor embodying the present invention.

FIGURE 3 is a sectional view on the line 3-3 of FIGURE 2.

FIGURE 4 is a sectional view through the cam link cage adjusting mechanism.

FIGURE 5 is an enlarged partial sectional view on part of the line 5-5 of FIGURE 4.

FIGURE 6 is an enlarged partial sectional view on the lower part of line 5-5 of FIGURE 4.

FIGURE 7 is a diagrammatic view of the cam action showing the position taken by the parts when the motor is operating at a relatively low speed under maximum load.

FIGURE 8 is a view similar to FIGURE 7 but showing the parts shifted to a position occupied when operating at higher speed under a lighter load.

FIGURE 9 is a flow diagram of the flow of oil to and from the hydraulic unit of the present invention.

Turning now to a description of the drawings by reference characters, the motor comprises a rotating housing 10 which is mounted for rotation on a stationary hub 12 by means of a first tapered roller bearing 14 and a second

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tapered roller bearing 16. Preferably, the housing is made in two segments, namely, an inner housing 18 and an outer housing 20. The two housing segments are held together by means of a suitable bolts 22 and the inner housing 18 has a plurality of lugs 24 thereon to which a vehicle rim 26 may be fastened by suitable bolts 28. Clamped between the inner segment 18 and the outer segment 20 is the double cam ring 30 having a main or driving cam track 31 and a secondary or follower cam track 33. In the embodiment illustrated, each cam track has seven lobes but other numbers might be employed as is hereinafter disclosed in detail.

The hub 12 has suitable attachments 32 and 34 for attaching the hub to the body of a vehicle or the like and the hub is provided with an inlet 35 and an outlet 36 for supplying and removing the actuating fluid. It will be understood, of course, that in reverse operation the outlet and inlet openings serve the opposite function, but these designations are used for convenience. The hub is also provided with an outlet 38 for the removal of any oil which may by-pass the pistons or valves. Further, a line 39 is provided leading to chamber 40 for use as a control circuit, as is shown in FIGURE 9.

The hub 12 has formed therein a plurality of cylinders 42, each of which contains a piston 44. Preferably, the pistons are arranged in pairs and five pairs are arranged equally around the periphery of the hub, although other numbers may be employed. Each pair of cylinders 42 leads to a chamber 46 connecting the two cylinders to each other and the chambers 46 in turn lead to chamber 47 of a spool valve assembly generally designated 48. The valve 48 comprises a sliding valve body 50 having a central recessed portion 52, the valve body being adapted to slide in a cylinder 54 having one opening 56 communicating with the inlet 35 and a second opening 58 communicating with the outlet 36. It will be understood that the central recessed portion 52 of the valve body 50 is at all times in communication with the passage 47 while said recessed portion 52 is alternately in communication with the passages 56 and 58, depending upon the position of the valve body 50. A spring 60 normally biases the valve body 50 to the left in FIGURE 2, so that the roller 64 on valve body 50 is held in contact with the face 62 of rotating cam ring 65. The cam ring 65 is mounted for rotation on the bearings 66 on the hub 12, and is driven by the annular gear 68 which is driven by the pinion 70 on the shaft 72, which in turn is driven by the pinion 74 at the opposite end of the shaft which engages the annular gear 76 mounted on the rotating housing 10. Provision is made, as is hereinafter described in detail, for slightly changing the position of pinion 70 with respect to the shaft 72 for proper valve timing under varying displacement conditions.

Mounted on the hub 12 are a plurality of piston reaction links 78, there being one bifurcated link for each pair of pistons. The piston reaction links 78 are pivoted to the hub by means of anti-friction bearings 80 and the links carry a shaft 82 which is provided with two outer anti-friction bearings 84, the races of each of which are in contact with one of the pistons 44. Shaft 82 also has an anti-friction bearing 86 with an outer race 96 located between bearings 84. An adjustable cage 88 is mounted for rotation on the hub 12 by means of cage cross members 89 which ride on arms 91 of the hub 12.

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The cage 88 generally is stationary with respect to the hub, but a slight angular adjustment can be imparted thereto to vary the displacement of the motor.

The cage 88 carries a plurality of cam reaction links 90, there being one cam reaction link 90 for each piston reaction link 78. The cam reaction links 90 are pivoted to the adjustable cage 88 by means of anti-friction bearings 92 and have a flat bottom portion 94 which is adapted to be in rolling contact with the outer race 96 of the anti-friction bearing 86 located on the corresponding piston reaction link. At the opposite end of the cam reaction link is an anti-friction bearing 98 having an outer race 100 adapted to bear against the main driving cam track 31. The cam reaction links are preloaded to exert a constant preload since otherwise the parts would rattle and be subjected to undue strain under conditions of too low oil pressure. This is accomplished by the secondary links 93 which correspond in number and pivot points to the reaction links 90. The secondary links have a follower roller 95 riding on the secondary track 33 and are connected to the cam reaction links by leaf spring 102. It will thus be seen as the pistons reciprocate, they cause a corresponding movement in the piston reaction links, which in turn causes a corresponding movement in the cam reaction links, resulting in turning movement of the outer housing 10. Because of the manner in which the anti-friction bearings are employed, there is no rubbing contact of any of the parts so that the reciprocating motion of the pistons is transferred to the rotating movement of the hub only through rolling motion. It will be apparent that it is possible to effectively change the length of stroke necessary for the pistons to cause a whole stroke of the cam reaction links by varying the position of the adjustable cage carrying the cam reaction links. This is more clearly shown in FIGURES 7 and 8. In FIGURE 7, the pistons 44, the piston reaction link bearings 84, the cam reaction link bearing 98 and outer race 100, and the main driving cam track 31 are substantially in alignment. Thus, a given movement of the pistons 44 will result in substantially the same amount of movement on the cam reaction bearing 98 against the main driving cam 31. On the other hand, it will be noted that in FIGURE 8 the cam reaction link 90 has been shifted somewhat to the right by re-positioning the adjustable cage, the cage not being shown in this view. Here it is apparent that the amount of movement imparted to the roller 100 will be multiplied since the fulcrum point provided by the bearing 92 of the cam reaction link 90 had been shifted. Thus, FIGURE 7 illustrates a typical operation when maximum torque is required since here a full piston stroke is necessary to produce a full stroke against the cam. On the other hand, FIGURE 8 represents high-speed operation since here the piston need move only through a portion of its path of travel to produce a full stroke at the cam surface.

The method by which the adjustable cage is adjusted to change the effective displacement of the motor is shown in FIGURES 3, 4, 5 and 6. The adjustable cam ring 88 has surfaces 104 and 106 with a sliding block 108 between them. The sliding block 108 has a shaft 110 attached to a lever arm 112 formed on shaft 114, the opposite end of shaft 114 carrying a driving arm 116. The driving arm terminates in a ball 118 near the center of the cylinder support 120. Cylinder support 120 has a first cylinder 122 and a second cylinder 124, each of which is provided with a piston, respectively 126 and 128. The heads of the pistons 126 and 128 are adapted to press against the ball 118, thus adjusting the cage by their action. In order to actuate the pistons 126 or 128 and thus move the adjustable cage through the linkage heretofore described, a spool valve generally designated 130 is employed. The spool valve has an inner sleeve 132 having three openings therein, as illustrated. These openings, designated 134, 136 and 138, line up with

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openings 140, 142 and 144, respectively, in the valve housing 130. It will be noted that the openings 134, 136 and 138 are smaller than the mating openings 140, 142 and 144 and that the path of travel of the sleeve 132 is such that the sleeve 132 will always have its openings lined up with the openings of the housing, because of the limited travel of the sleeve 132. The opening 140 leads to the cylinder 122 by a connection not shown, while the opening 144 leads to the opening 145 to the cylinder 124. The opening 142 leads to the high pressure line which acts as a control pressure. Mounted within the sleeve 132 is a spool valve having an open center and lands 146 and 148. The amount of movement imparted to the spool is such that either opening 134 or opening 138 can be placed in communication with opening 136 or both 134 and 138 can be blanked off. Thus, either of the pistons 126 or 128 can be actuated by the position of the spool valve or both can be rendered inactive. When the spool valve is at the extreme of its path of travel, the cylinder 122 or 124 can drain through one end or the other of the spool valve. Further, the arm 116 has a follow-up cam 150 which forms a part thereof, while the sleeve 132 of the spool valve has a cam follower 152 which is urged into contact with the cam by means of a spring 154. Thus, when the command to operate the adjustable cage is given, the cam 150 acting on the follower 152 acts as a follow-up mechanism so that the openings of the valve is restored to its neutral position because of the relative shift of the spool and the sleeve after the command has been executed, even though the spool is still displaced from what would normally be its neutral position.

Since the angular adjustment of the cage 88 changes the position of the rollers 100 on the cam 30, the valve cam 64 must be similarly adjusted to make the inlet and outlet openings of the main valves 48 coincide with the relative position of the cam reaction links 90 with the cam track 31. This adjustment is accomplished by the mechanism in FIGURE 6. As has been previously described, gear 76 rotates shaft 72 which in turn rotates gear 68, which forms part of the timing cam. However, pinion 70 is not rigidly fixed to shaft 72 but is coupled thereto by splines 159 on member 161, which in turn is coupled to the shaft 72 by helix 160. A cylindrical rack 163 forms the end of the member 161. Thus, as rack 163 is moved longitudinally, a small adjustment is made of the relative positions of gears 68 and 76. Rack 163 is moved by the action of shaft 165 having gear 167 which meshes with gear 169 on shaft 114.

The motor of the present invention may be used with various hydraulic pumps and control circuits and a typical diagram is shown in FIGURE 9. Here, the drive includes a variable displacement and reversible pump 176 driven by prime mover 178. The pump has an inlet line 180 and an outlet line 182. The line 182 connects with inlets of motors 184 and 186, and the line 180 connects with outlets of the motors.

The pump illustrated has dual inlets and outlets, so that a second inlet 188 and outlet 190 are provided leading to two additional motors. Thus, four motors are driven by the pump; each pair of motors is supplied by a separate inlet and outlet line. The pump has two banks of cylinders independently controlled. The control system associated with one pair of motors only is shown, the other being identical.

Thus, each of the two inlets and outlets can be independently regulated. This is required if, for instance, a vehicle of the track type is to be steered by the motors, e.g., one side may be slowed down and the other side speeded up. Also, in a four wheel drive vehicle, beyond certain speeds two of the motors may be cut out by reducing the output of one block of the pump to zero, and the drive may be entirely transmitted by the remaining two wheels.

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The line 182 is normally under high pressure for forward motion, and the line 180 is under low pressure. The line 180 is supercharged by low pressure pump 192, which has an inlet 194, which receives oil from reservoir 196. The drain lines 198 and 200 from the motors and pump return leakage oil to the reservoir, from which the pump 192 picks it up and returns it to the circuit by means of outlet line 202. The oil flows through heat exchanger 204 for cooling of the entire system, and through filter 206. It is then supplied to lines 180 or 182, whichever is under low pressure, by means of check valves 206 and 208 and line 207. High pressure relief valves 210 and 212 are supplied to bypass liquid from line 180 to 182 or vice versa if the pressure in either line should become excessive. The pressure in line 207 is regulated by regulator valve 214, which bypasses the excess oil through line 216 to the reservoir. The pressure regulator is controlled by the speed sensor 218, which senses the speed of the motors and therefore the speed of the vehicle. Thus, the pressure in line 207 is regulated as a function of vehicle speed. This variation in pressure is used to control the displacement of the motors by means of line 39, which is connected to either line 180 or 182, whichever is the low pressure line, by shuttle valves 220. Thus, the pressure acting on piston 40 (FIGURE 4) is determined by the pressure regulator 214, and therefore the vehicle speed. This avoids the necessity of providing additional control lines to the motors.

Since the quantity of oil supplied by pump 192 is very small for leakage make-up, due to the extremely high efficiency of the motors, the quantity passing through the cooler is not sufficient to cool the system. An excess capacity is therefore supplied through line 222 and restrictor 224 to the motor case, where it serves to cool the motor and is thereafter returned by drain lines 198 to the reservoir.

The speed and torque control of the drive system is entirely automatic, and the driver achieves all control functions by operating foot throttle 226, which simultaneously controls the engine and provides a power demand input into control 228. At this point, the power demand input is integrated with the resistance encountered by the motors by means of lines 230 and 232, which connect with lines 180 and 182 respectively. The resultant control signal is supplied to the pump by output signal 234, which regulates the displacement of one of the pump cylinder blocks. A similar control regulates the second pump cylinder block.

A selector for forward, neutral, park and reverse is shown at 236.

For braking of the vehicle, the line 180 is throttled by the throttling valve 238, which is operated by a conventional brake pedal, not shown.

The flow in lines 188 and 190, and 182 and 180, is independently controlled, and for normal straight-ahead driving is evenly divided. However, during turning, one set of motors has to operate at a slightly different speed from the second set of motors. For that reason, valve 238 is provided between line 188 and line 182, and a similar valve 240 is provided between lines 190 and 180. These valves permit a small amount of flow between the lines, but when the flow exceeds a predetermined limit, they shut off intercommunication between these lines.

The motor has been described as having five pistons per bank and seven lobes on each of the cams. These numbers have been established by trial and error as being optimum for a particular class of military vehicles but it will be obvious to those skilled in the art that a smaller or larger number of pistons or lobes might be used. The purpose of having many cam lobes is to permit the pistons to make many strokes per revolution so that the motor develops much more torque than a motor would of similar dimensions wherein only one stroke per revolution is made. However, there is a

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practical upper limit to the number of lobes which can be used since the greater the number of lobes the sharper the cam peaks and the greater the tendency of the cam follower to jump off of the peaks as speed is increased. Likewise, the number of cylinders can be varied and generally it is preferred to use an odd number of cylinders since this results in a smoother fluid flow than an even number. If fewer cylinders are employed, the bearing loading must be larger and forces acting upon the cam reaction links and piston reaction links become high, requiring large diameter cam followers. Generally speaking, neither the number of lobes nor the number of pistons should be divisible by a whole number since the flow will become very rough and the operation of the motor will be equivalent to that of one having only two or three pistons, since several of the pistons will be firing at the same time. There should not be the same number of pistons as there are lobes since this is equivalent to a one-cylinder motor which can stop on dead center. By selecting a proper number of cam lobes and cylinders, the radial loads on the main bearings are substantially balanced out.

Although the pistons have been described as being paired, a single piston or more than two could be used in each bank. It is preferred to have the pistons paired since the size can be kept small and the load on the piston reaction links is distributed to avoid side thrust.

The device has been described as a motor, although it will be obvious that it could also be used as a pump.

I claim:

1. In a fluid-operated device wherein a force is transmitted between a piston and a cam, the improvement comprising a first pivoted lever arm transmitting motion to and from the piston to a second pivoted lever arm, said first pivoted lever arm preventing side thrust on the piston and said second pivoted lever arm transmitting motion to and from the cam and means for adjusting the second pivoted lever arm whereby the mechanical advantage between the piston and the cam can be varied.

2. The device of claim 1 wherein a valve-actuating cam is provided to control a valve regulating the flow of oil to and from the piston and wherein a linkage is provided between said cam and said means for adjusting the second pivoted lever arm whereby an adjustment of the second pivoted lever arm keeps the valve timing in proper relationship with piston motion.

3. A fluid-operated motor comprising in combination a stationary hub and a housing mounted for rotation on said hub, a plurality of radial cylinders in said hub, pistons in said cylinders, said pistons extending outwardly from said cylinders, a piston reaction link pivoted on said hub a short distance circumferentially from each of said pistons, and each piston reaction link having a free end extending over a piston, each of said piston reaction links having a bearing surface on its free end in contact with a piston face, an adjustable cage mounted for limited rotational movement on the outside of said hub, said adjustable cage carrying a plurality of cam reaction links corresponding in number to the number of piston reaction links, a driving cam mounted within the rotary housing, each of said cam reaction links having a first bearing surface bearing against its mating piston reaction link and a second bearing surface bearing against said cam and means for rotatably adjusting the position of the adjustable cage relative to said piston reaction links whereby the effective length of stroke required by said pistons to produce a full stroke against said cam by the cam reaction links can be adjusted.

4. The motor of claim 3 wherein the pistons are paired with a piston reaction link provided between each pair of pistons.

5. The motor of claim 3 wherein each piston reaction link is provided with an anti-friction bearing in contact with a piston face.

6. The motor of claim 3 wherein each cam reaction

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link has a first plain bearing surface in contact with a piston reaction link and a second anti-friction bearing in contact with the cam.

7. The motor of claim 3 wherein two cams are provided, a first driving cam and a second follow-up cam, with a follower in contact with the second cam and a spring connection between said follower and a cam reaction link whereby the second cam tends to force a piston into its cylinder.

8. The motor of claim 3 wherein a spool valve and a cam to actuate the spool valve are provided to supply fluid under pressure to a piston.

9. The motor of claim 8 wherein the phase of the valve actuating cam is adjusted relative to the phase of the driving cam by a linkage between said valve actuating cam and the adjustable cage carrying the cam reaction links.

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10. In a fluid-operated device wherein a force is transmitted between a piston and a cam, the improvement comprising a first pivoted lever arm transmitting motion to and from the piston to a second pivoted lever arm, said first pivoted lever arm preventing side thrust on the piston and said second pivoted lever arm transmitting motion to and from the cam.

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