

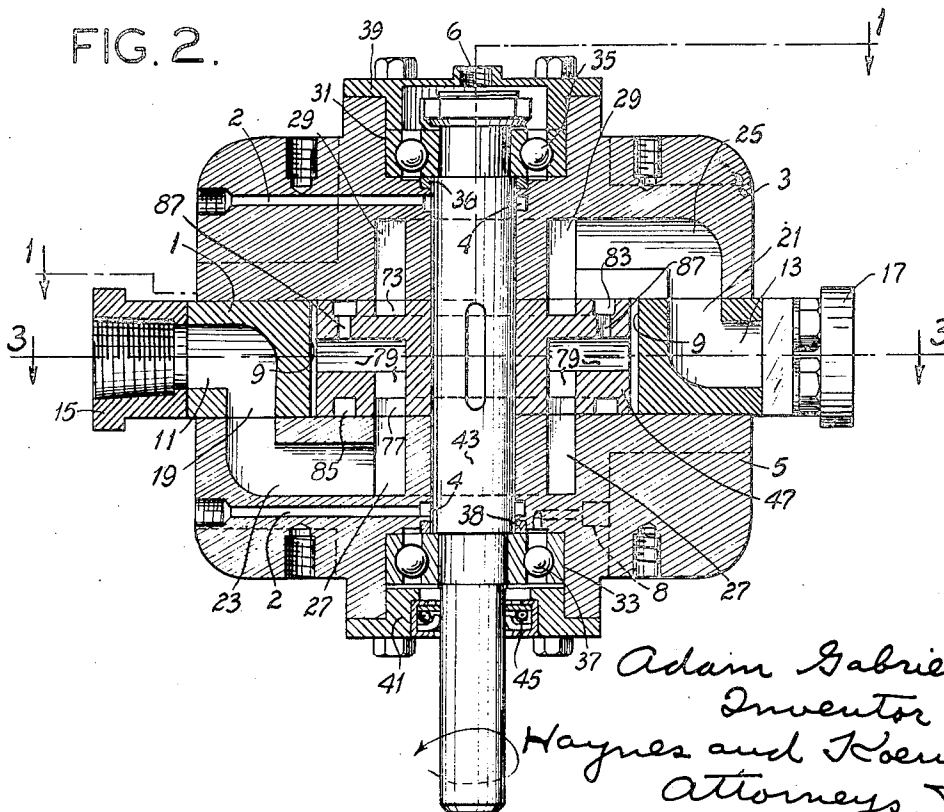
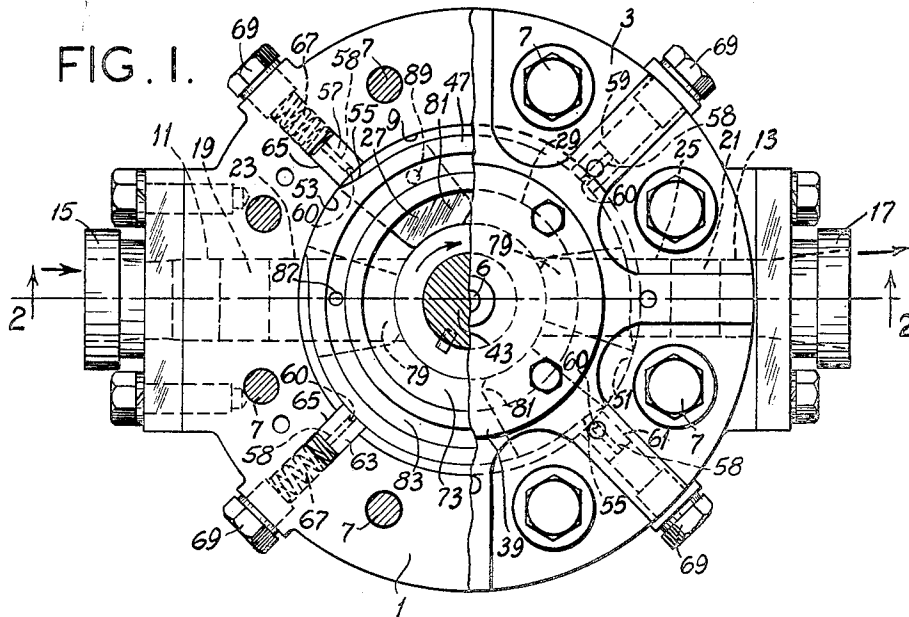
May 9, 1950

A. GABRIEL
HYDRAULIC ROTARY MOTOR OF THE RADIALLY
SLIDING ABUTMENT TYPE

2,507,151

Filed May 10, 1946

4 Sheets-Sheet 1



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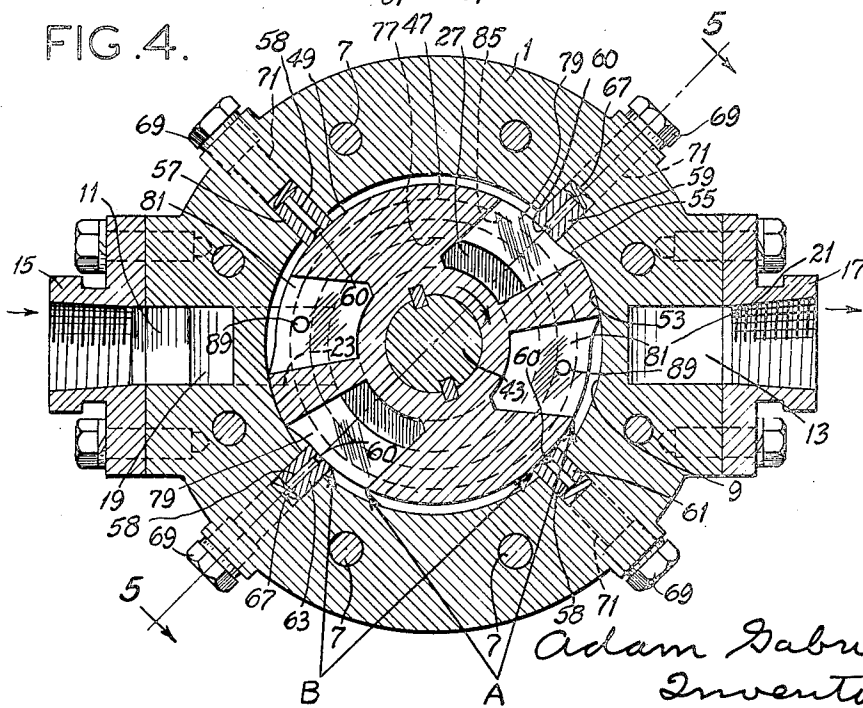
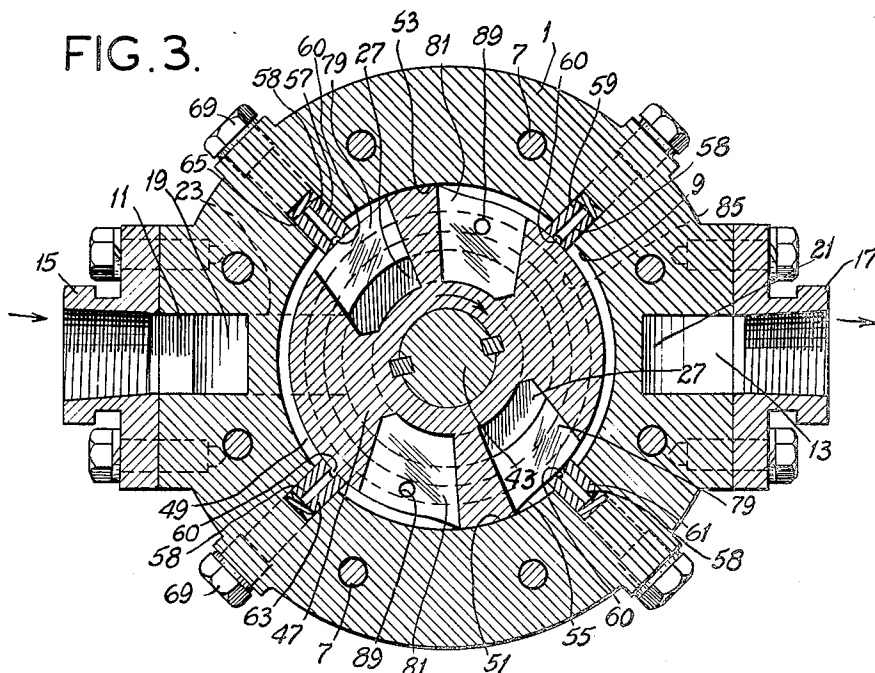
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2,507,151

Filed May 10, 1946

4 Sheets-Sheet 2



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Filed May 10, 1946

4 Sheets-Sheet 3

FIG. 5.

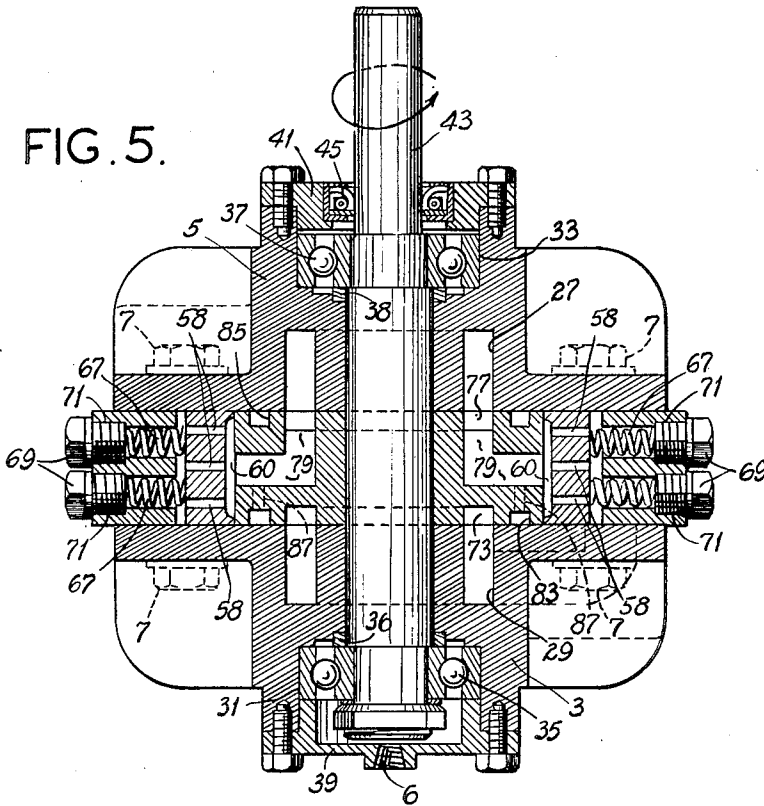


FIG. 7.

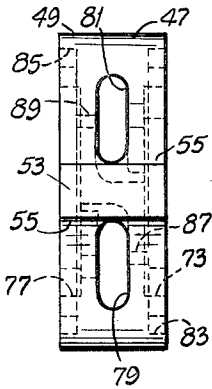


FIG. 6.

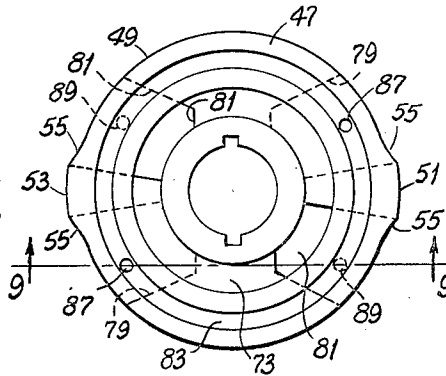


FIG. 8.

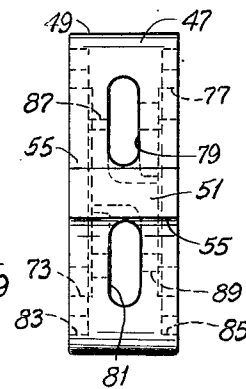
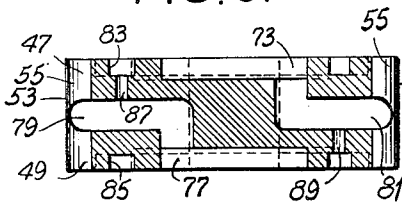


FIG. 9.



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FIG. 10.

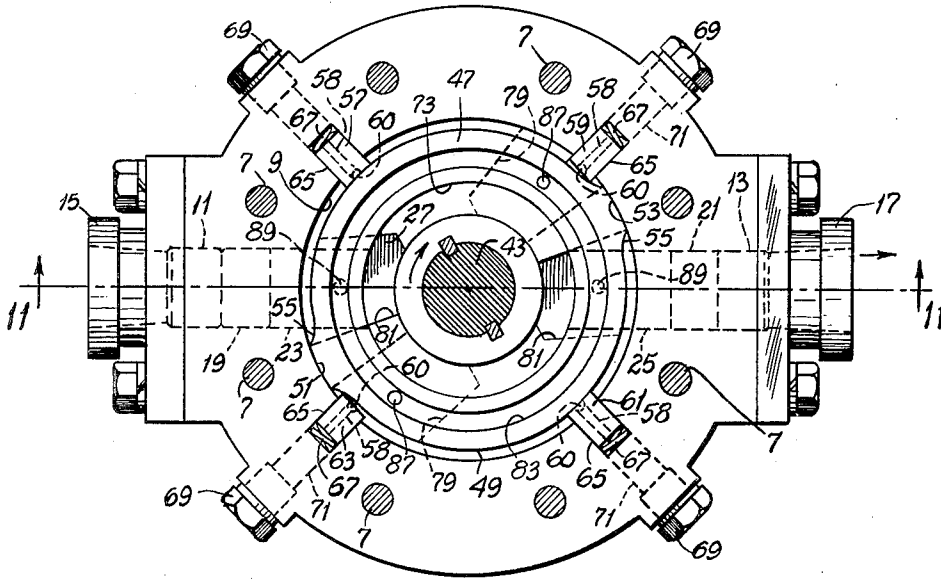
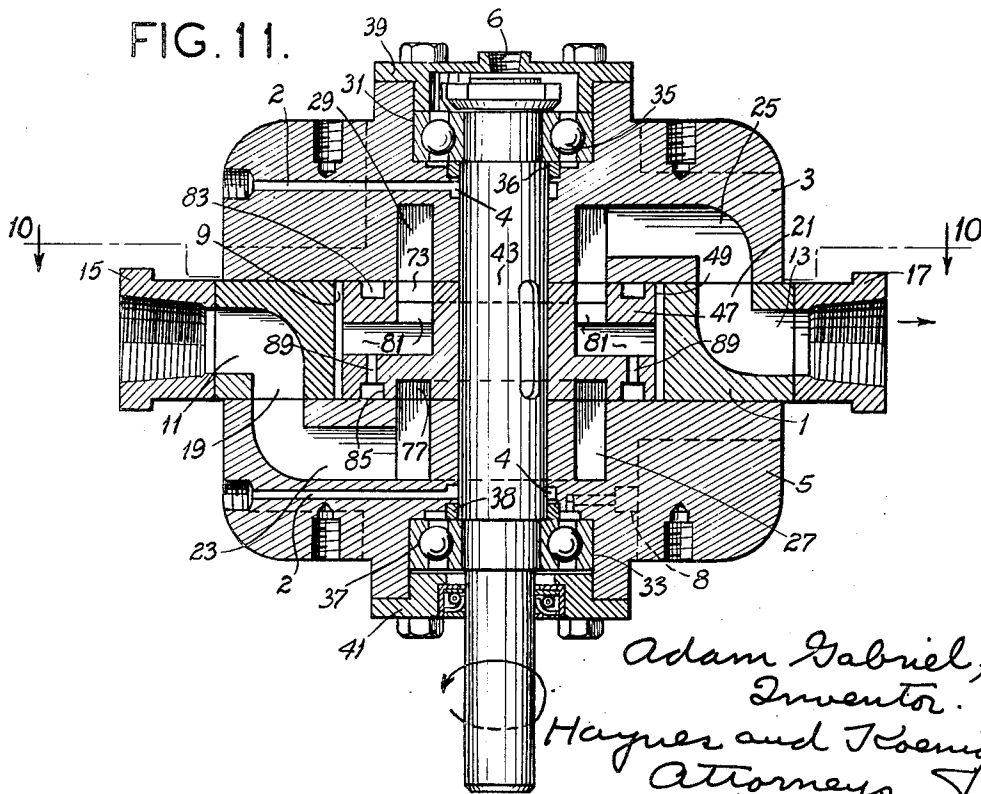


FIG. 11.



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UNITED STATES PATENT OFFICE

2,507,151

HYDRAULIC ROTARY MOTOR OF THE
RADIALLY SLIDING ABUTMENT TYPE

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mesne assignments, to Acme Industrial Hy-
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Application May 10, 1946, Serial No. 668,647
In Canada April 19, 1946

6 Claims. (Cl. 121-77)

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This invention relates to hydraulic motors, and with regard to certain more specific features, to a radially sliding abutment type of rotary motor of this class.

This application is a continuation-in-part of my United States patent application, Serial No. 595,370, filed May 23, 1945, for Hydraulic motor, now abandoned.

Among the several objects of the invention may be noted the provision of a reversible hydraulic motor of high efficiency which will deliver a high starting torque and high torque over a substantial speed range; the provision of a motor of this class having substantially symmetrical part forms which may be manufactured at low cost; the provision of a motor of the class described which is axially balanced in all of its forms; and the provision of a motor of the class described having normally lightly loaded power abutments subject to little wear but which abutments are at all points in their operating cycles positively held without knocking against their operating surfaces. Other objects will be in part obvious and in part pointed out hereinafter.

The invention accordingly comprises the elements and combinations of elements, features of construction, and arrangements of parts which will be exemplified in the structures hereinafter described, and the scope of the application of which will be indicated in the following claims.

In the accompanying drawings, in which one of various possible embodiments of the invention is illustrated,

Fig. 1 is an end elevation taken on line 1-1 of Fig. 2, parts being shown in section;

Fig. 2 is a horizontal section, looking upward, taken on line 2-2 of Fig. 1;

Fig. 3 is a vertical section taken on line 3-3 of Fig. 2;

Fig. 4 is a view similar to Fig. 3 showing advanced positions of certain parts;

Fig. 5 is an oblique section taken on line 5-5 of Fig. 4;

Fig. 6 is an end view of the rotor of the motor removed from other parts and showing its symmetry;

Fig. 7 is a left-end view of Fig. 6;

Fig. 8 is a right-end view of Fig. 6;

Fig. 9 is a sectional slice taken on line 9-9 of Fig. 6;

Fig. 10 is a vertical section taken on line 10-10 of Fig. 11; and,

Fig. 11 is a horizontal section taken on line 11-11 of Fig. 10.

Similar reference characters indicate cor-

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responding parts throughout the several views of the drawings.

Common faults of hydraulic motors are their low efficiencies and low starting torques. Many of them do not have the ability to carry high torque over a substantial speed range. The present motor overcomes these objections. Thus the motor may be started at full load and maintain it. Furthermore, efficiencies are held at substantial values.

Referring now more particularly to Figs. 1 and 2, there is shown at numeral 1 a central ring-shaped member upon which are end members or heads 3 and 5. The essential shapes of the members 3 and 5 are similar so that they may be made from similar patterns, requiring only modification as to detail in machining, and not much of that. The central ring-shaped member 1 and the end members are held together by draw bolts 7.

The ring-shaped member 1 is essentially of simple form, being interiorly circular, as indicated at 9, and having opposite cored ports 11 and 13 which may be identical. In the present description port 11 will be considered to be an inlet port and port 13 an outlet port, although this relationship may be reversed, as will appear.

Bolted over the inlet port is an inlet fitting 15. Bolted over the outlet port 13 is an outlet fitting 17. These fittings are preferably identical in form. As shown in Fig. 2, the ports 11 and 13 have opposite radial openings communicating with the fittings 15 and 17, respectively. Ports 11 and 13 also have opposite axial openings 19 and 21 for connections respectively with cored openings 25 and 23 in the end members 3 and 5, respectively. The openings 23 and 25 are also preferably identical and lead to identical annular manifold spaces 27 and 29 as shown.

The end members 3 and 5 are respectively counterbored as shown at 31 and 33 for ball bearings 35 and 37 enclosed by covers 39 and 41. The bearings 35 and 37 support a shaft 43, one short end of which is entirely closed by the cover 39. The other end of the shaft extends through a packing gland 45 and serves as a driving member.

The bearings 35 and 37 are protected against leakage by packings 36 and 38 and by channels 4 around the shaft leading to outlets 2. Lubrication for the bearing 35 is supplied by a fitting applied at an opening 6 and lubrication for the bearing 37 by a fitting applied to opening 8. As stated, the body members 3 and 5 are preferably made identical, except for such details as the

opening 8. As such these body members 3 and 5 may be from a single pattern. Furthermore, the opposite sides of the ring member 1 of the pump body are identical.

Keyed to the shaft 43 is a rotor 47 which is of an easily made symmetrical form. This rotor is shown per se in Figs. 6 to 9 and consists of a substantially circular outer form 49 having on opposite sides elevated lands 51 and 53 which are circularly machined for sealing cooperation with the inside of the cylinder 9 of the ring-shaped member 1 to divide the space between the rotor and cylinder into non-communicating liquid chambers. Between the circular form 49 and the lands 51 and 53 are sloping cam lift portions 55 which serve as lift means for lifting valves or abutments 57, 59, 61 and 63, all of like form. These abutments are positioned at 90° intervals around the periphery of the ring-shaped member 1 and slide radially in radial slots 65. These slots 65 pass from face to face of ring member 1 and are rectangular in cross section, as are also the abutments. The abutments are biased inward by springs 67 reacting from cap screws 69, the latter being threaded into openings 71 drilled in from the outside periphery of the ring 1. The springs 67 are also in the openings 71. Thus the cavities for the abutments 57, 59, 61 and 63 are formed as radial slots in the ring 1 with drilled cavities 71 for the springs 67; and the lengths of the abutments are such as fit movably between the opposite flat-machined heads 3 and 5. Such a structure is quite advantageous from a manufacturing viewpoint, since the cavities for the abutments are simply made as inside slots.

Each abutment 57, 59, 61 and 63 has radial bores 58 connecting with an inside groove 60 for allowing pressure behind each abutment to change at proper intervals and to equal that at its engaging end. The purpose of this will appear below.

Returning to the rotor shown in Figs. 6-9, it has on its opposite faces equal inner circular recesses 73 and 77. Recess 73 aligns with the manifold 29 and recess 77 aligns with the manifold 27. In the rotor 47 are two opposite L-shaped ports 79 which, through the recess 77 are always in communication with the circular manifold space 27 and the inlet 23 (Fig. 2). Also in the rotor are oppositely located ports 81 of L shapes which are in communication with the circular recess 73 and hence always in communication with the circular outlet manifold space 29 and the outlet 25 in the member 3 (Fig. 11).

The rotor also has on its opposite faces outer equal peripheral recesses 83 and 85. Openings 87 connect the recess 83 with the ports 79; and openings 89 connect the ports 81 with the recess 85. The width of outer recesses 83 and 85 is less than the width of inner recesses 73 and 77 so that the projected end areas of the outer recesses are substantially equal to the projected end areas of the inner recesses, which themselves are equal, thus counterbalancing the rotor in the axial direction.

The following points should be carefully noted in respect to Figs. 1, 3 and 4: Fig. 1 shows a first rotary position, Fig. 3 a second rotary position and Fig. 4 a third. However, Fig. 1 is taken on section line 1-1 of Fig. 2 which is higher than the section line 3-3 of Fig. 2. The difference makes the inlet 79 at the left of Fig. 1 dotted; whereas in Figs. 3 and 4 the corresponding rotated port is shown in solid section.

The driving fluid is preferably a good grade of lubricating oil.

Operation is as follows, assuming port 11 to be an inlet and port 13 to be an outlet. Pressure enters port 23, pressure manifold 27 and the ring-shaped recess 77 in the rotor 1, which thus constitutes a pressure-receiving recess. This places fluid under pressure in what are now inlet ports 79 (Fig. 2). This flows out to the space between the rotor and the ring 1. Assuming that the inlet ports 79 are in the position shown in Fig. 1, this will introduce pressure on the clockwise side of the abutments 63 and 59. The resulting pressure reacting against these abutments pushes against the counterclockwise (rear) sides of the lands 51 and 53 to force the latter clockwise, thus applying torque to the rotor to turn the shaft 43. This drives the lands toward the other abutments 57 and 61. Then the lands lift the latter abutment. Abutments 57 and 61 then drop down behind the lands (Fig. 3). However, the pressure charge between abutments 63 and 57 on the one hand and 61, 59 on the other hand, is momentarily still applied to the lands until finally the charge between abutments 63 and 57 is trapped without any intervening land and the power cycle as to abutments 63 and 59 ceases.

However, as soon as the inlets 79 completely pass the abutments 57 and 61 respectively, pressure is introduced on the clockwise side of the latter, thus for example continuing to force the land 53 clockwise in the space between abutments 57 and 59. Thus lands 53 and 51 will continue to be forced clockwise from the Fig. 3 position even after abutment 57 is met by the trailing edge of port 79. Thus application of torque to a given land is substantially constant throughout 360° of shaft movement.

When a land passes under an abutment its clockwise side acts to sweep out the charge trapped between adjacent pairs of abutments. As shown in Figs. 3 and 4, the ports 81 acting as outlets under such conditions receive the charges which are being pressed and deliver them to the recess 73, which constitutes an exhaust-receiving recess, and then to the outlet ports 25 and 13 via the exhaust manifold 29 (Fig. 11).

Thus a given land 53 or 51 is being pushed by reaction against an abutment just passed and is sweeping out liquid by compression against the abutment which is being approached. The action is the same on opposite sides of the machine. Momentarily the spaces between abutments become cut off from all ports 79 and 81. This is about to occur at the left in Fig. 3. The fluid which is then trapped offers very little resistance to rotation of the rotor and is swept out when the next land in order sweeps into the space. The condition lasts for the period required for a land edge to cross an abutment.

In either direction of rotation there is one outlet port ahead of each land 51 and 53 and one inlet port behind it. The arcuate distance on the rotor between adjacent inlet and outlet ports is, as shown by the arrows A in Fig. 4, slightly greater than the arcuate distance between adjacent abutments 63 and 61 as shown by the arrow B. The stated length A prevents an inlet port from ever being connected with the next trailing outlet port, thus preventing any short-circuiting of the fluid through the motor without delivering power.

A given inlet port furnishes fluid pressure on a given abutment even when that port is straddling the next abutment, said pressure providing

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the reaction for applying torque. However, as soon as the rear edge of the stated port passes the next abutment, then the former abutment is cut off and the reaction is against said next abutment. The result is that at no point in the cycle of 360° is there any absence of any applied torque.

Although an abutment such as 57 in Fig. 3 may have closed off the neutral space between abutments 57 and 63, driving torque is being applied to the land 53 in the chamber between abutments 57 and 59. This means that the space between the abutments 57 and 63, although not yet cut off from the inlet, does not apply negative torque, since the reaction is at this time against the abutment 63 until abutment 57 closes off the space between abutments 57 and 63. As soon as the abutment 57 closes off this space, then the reaction is applied to abutment 57. For this reason there is constant torque throughout the entire 360° of movement.

Since the device is of the positive displacement variety and moment is applied throughout 360° of motion, it will deliver full torque at any speed within its range, including zero speed. Thus it also has a high starting torque at any position. The torque only falls off when the speed becomes so high that the pressure cannot be adequately supplied through the ports. Falling off of the torque does not occur until rather high speeds are reached since the entire lengths of the inlet porting system and the outlet porting system are of ample cross section and compared to the fluid displacement in the space between the rotor 47 and the rings 1.

The motor is intended to operate at substantially high pressures, such as for example several thousand p. s. i., which means that at the radius provided for the lands 53 and 51, only a small net projective pressure area is needed for driving. This also means that the lift for the abutments may be low, which is conducive to high-speed operation without dynamic troubles from abutment operation. For example, a motor of approximately 10 inches in diameter by 12 inches length will develop 15 H. P. at a thousand R. P. M. with a pressure of 1,000 p. s. i., or approximately 30 H. P. at 2,000 p. s. i.

It will be seen that if it is desired to reverse the motor it is only necessary to reverse the connections at the fittings 15 and 17. Thus if the fitting 17 is made an inlet and the fitting 15 and outlet, the device will operate in reverse according to the same principles that it operated in the other direction. In this event what have been described as inlet ports above become outlet ports and the inlet ports become outlet ports. All of the stated advantages accrue in either direction of operation.

It will be noted that each abutment includes radial passages through it from front to rear, with a connecting cross passage on the rubbing face. This construction not only provides for lubrication behind each abutment, but also has the following functions: As any abutment is traversed by a land, the ports through the abutment become opened. This occurs as the particular abutment is traversed by the sloping cam portions next to a land. As an abutment is traversed by a slope 55, groove 60 is exposed. This causes a slightly unbalanced pressure on the opposite ends of the abutment, due to the small area in contact with the slope. This is desirable as it assures perfect action of the abutment. Furthermore, it is desirable as it relieves the

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spring from further duty once the pressure is set up in the motor. The springs are used only for initially setting the abutments in their proper positions. Once the motor is running under a slight pressure, the slightly unbalanced abutments relieve the springs from further duty. If there were not this slightly unbalanced abutment condition, there would be required very heavy abutment springs to prevent fluttering. If there were not a slightly unbalanced condition and very light springs were being used, the abutments would not remain on the cam. This would result in oil momentarily by-passing from the inlet to the outlet. This would cause a fluttering flow of the oil and also cause a vibration of the motor.

This feature of the opening of the ports through the abutments when they traverse the sloping cam portions is quite important. Without it there is a tendency for the abutments to have a fluttering action.

In view of the above, it will be seen that the several objects of the invention are achieved and other advantageous results attained.

As many changes could be made in the above constructions without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

I claim:

1. In a rotary hydraulic motor, a cylinder, a rotor in the cylinder having inner and outer annular recesses in opposite ends thereof, one of the inner recesses constituting a pressure-receiving recess, the other constituting an exhaust-receiving recess, end heads on the cylinder enclosing the rotor, one of said heads having an annular pressure manifold and the other an annular exhaust manifold opening into said pressure-receiving and exhaust-receiving recesses, respectively, said rotor having therein a pressure port connecting the pressure-receiving recess with one portion and an exhaust port connecting the exhaust-receiving recess with another portion of the periphery of the rotor, a passage connecting said pressure port with the outer annular recess on the end of the rotor including said exhaust-receiving recess, and a passage connecting said exhaust port with the outer annular recess on the end of the rotor including said pressure-receiving recess.

2. In a rotary hydraulic motor, a cylinder, a rotor in the cylinder having inner and outer annular recesses in opposite ends thereof, one of the inner recesses constituting a pressure-receiving recess, the other being substantially identical and constituting an exhaust-receiving recess, end heads on the cylinder enclosing the rotor, one of said heads having an annular pressure manifold and the other an annular exhaust manifold opening into said pressure-receiving and exhaust-receiving recesses, respectively, said rotor having therein a pressure port connecting the pressure-receiving recess with one portion and an exhaust port connecting the exhaust-receiving recess with another portion of the periphery of the rotor, a passage connecting said pressure port with the outer annular recess on the end of the rotor including said exhaust-receiving recess, and a passage connecting said exhaust port with the outer annular recess on the end of the rotor including said pressure-receiving recess, said outer annular recesses being substantially identical and of less radial width

than said inner recesses so as to have substantially the same areas as said inner recesses for axially balancing said rotor.

3. A rotary hydraulic motor comprising a cylinder, a rotor in the cylinder having peripherally spaced lands engaging the internal wall of the cylinder and dividing the space between said rotor and cylinder into non-communicating liquid chambers, said rotor having sloping cam surfaces adjacent said lands and being formed with inner and outer annular recesses in opposite ends thereof, one of the inner recesses constituting a pressure-receiving recess, the other constituting an exhaust-receiving recess, end heads on the cylinder enclosing the rotor, one of said heads having an annular pressure manifold and the other an annular exhaust manifold opening into said pressure-receiving and exhaust-receiving recesses respectively, said rotor having therein pressure ports and exhaust ports connecting said pressure-receiving and said exhaust-receiving recesses to said liquid chambers, passages connecting said pressure ports with the outer annular recess on the end of the rotor including said exhaust-receiving recess, and passages connecting said exhaust ports with the outer annular recess on the other end of the rotor, abutments mounted for radial sliding movement in the wall of said cylinder, and means for biasing said abutments so that their inner ends sealingly engage said rotor, said abutments being adapted to be moved radially outward by said cam surfaces.

4. A rotary hydraulic motor as set forth in claim 3, wherein said abutments are slidable in radial recesses in the wall of said cylinder, each abutment being provided with a radial port there-through adapted to be opened only as the abutment is moved radially outward by said cam surfaces to establish communication between said liquid chambers and the portions of the abutment recesses radially outward from said abutments.

5. A rotary hydraulic motor comprising a cylinder, a rotor in the cylinder having peripherally spaced lands engaging the internal wall of the cylinder and dividing the space between said rotor and cylinder into non-communicating liquid chambers, said rotor having sloping cam surfaces adjacent said lands and being formed with inner and outer annular recesses in opposite ends thereof, one of the inner recesses constituting a pres-

sure-receiving recess, the other being substantially identical and constituting an exhaust-receiving recess, end heads on the cylinder enclosing the rotor, one of said heads having an annular pressure manifold and the other an annular exhaust manifold opening into said pressure-receiving and exhaust-receiving recesses respectively, said rotor having therein pressure ports and exhaust ports connecting said pressure-receiving and said exhaust-receiving recesses to said liquid chambers, passages connecting said pressure ports with the outer annular recess on the end of the rotor including said exhaust-receiving recess, and passages connecting said exhaust ports with the outer annular recess on the other end of the rotor, said outer annular recesses being substantially identical and of less width than said inner recesses so as to have substantially the same areas as said inner recesses for axially balancing said rotor, abutments mounted for radial sliding movement in the wall of said cylinder, and means for biasing said abutments so that their inner ends sealingly engage said rotor, said abutments being adapted to be moved radially outward by said cam surfaces.

6. A rotary hydraulic motor as set forth in claim 5, wherein said abutments are slidable in radial recesses in the wall of said cylinder, each abutment being provided with a radial port there-through adapted to be opened only as the abutment is moved radially outward by said cam surfaces to establish communication between said liquid chambers and the portions of the abutment recesses radially outward from said abutments.

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REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
23,466	Hughes	Apr. 5, 1859
167,489	Adams	Sept. 7, 1875
349,888	Knebel	Sept. 28, 1886
381,287	Snevely	Apr. 17, 1888
494,138	Graham	Mar. 28, 1893
508,555	Petty	Nov. 14, 1893
708,328	Eek et al.	Sept. 2, 1902
723,226	Anderson	Mar. 24, 1903
963,207	Burleigh	July 5, 1910