

May 14, 1968

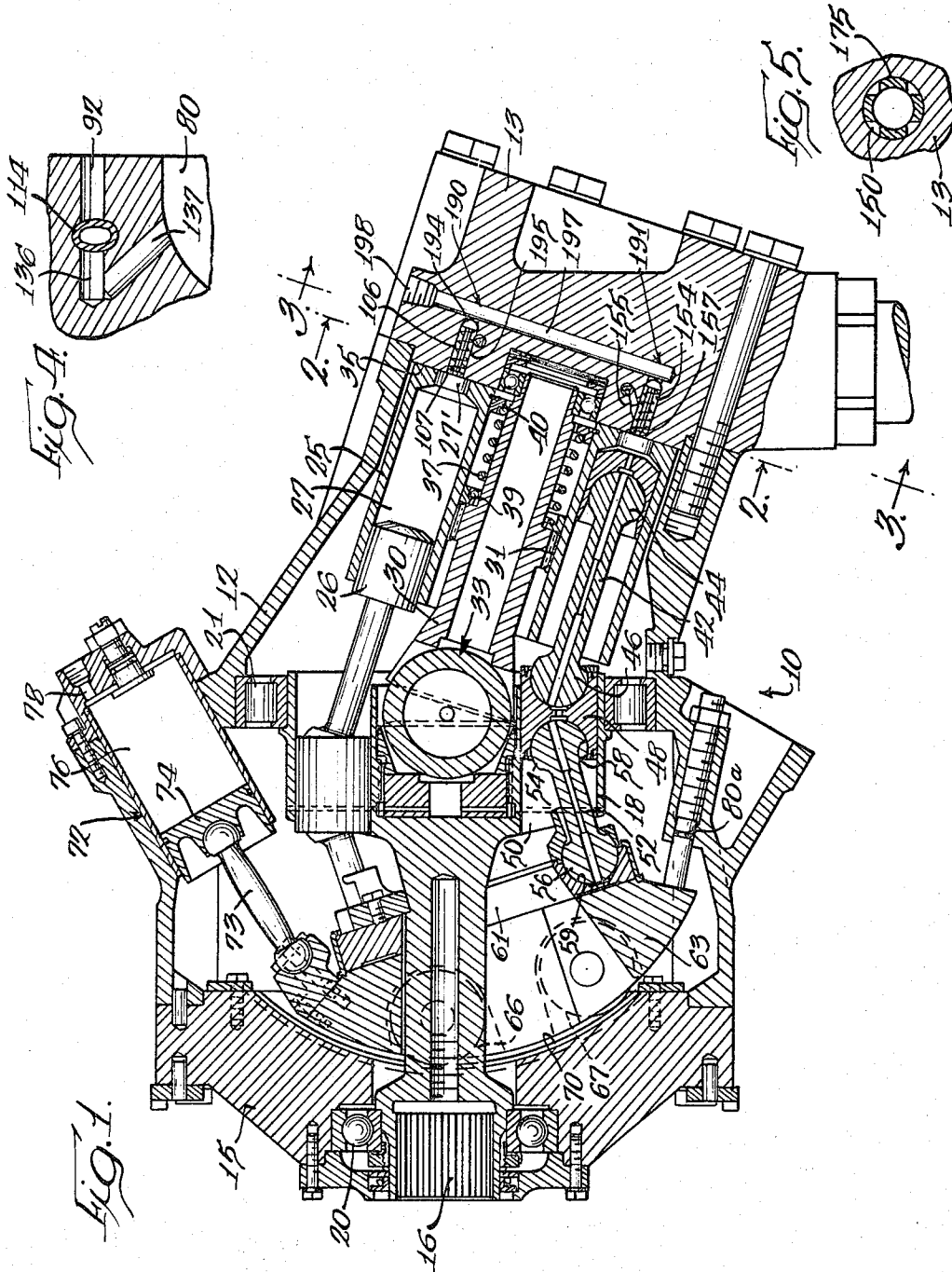
G. A. SCHAUER

3,382,813

HYDRAULIC PUMP OR MOTOR

Filed Feb. 15, 1966

3 Sheets-Sheet 1



Inventor:
George A. Schauer
By Hofgren, Wegner,
Allen, Stillman & McCord Attys

May 14, 1968

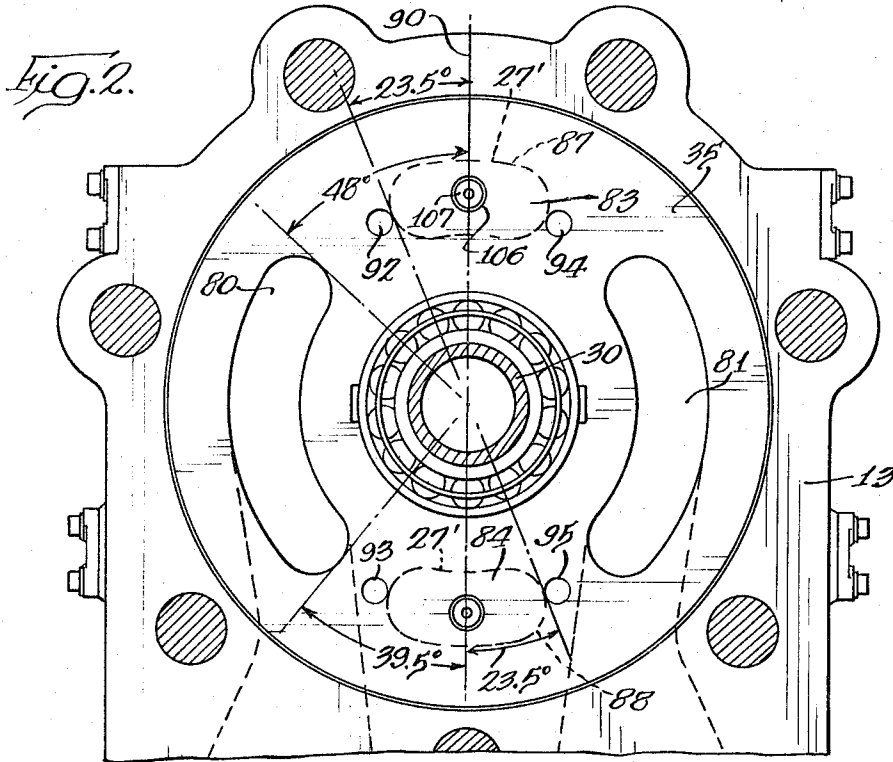
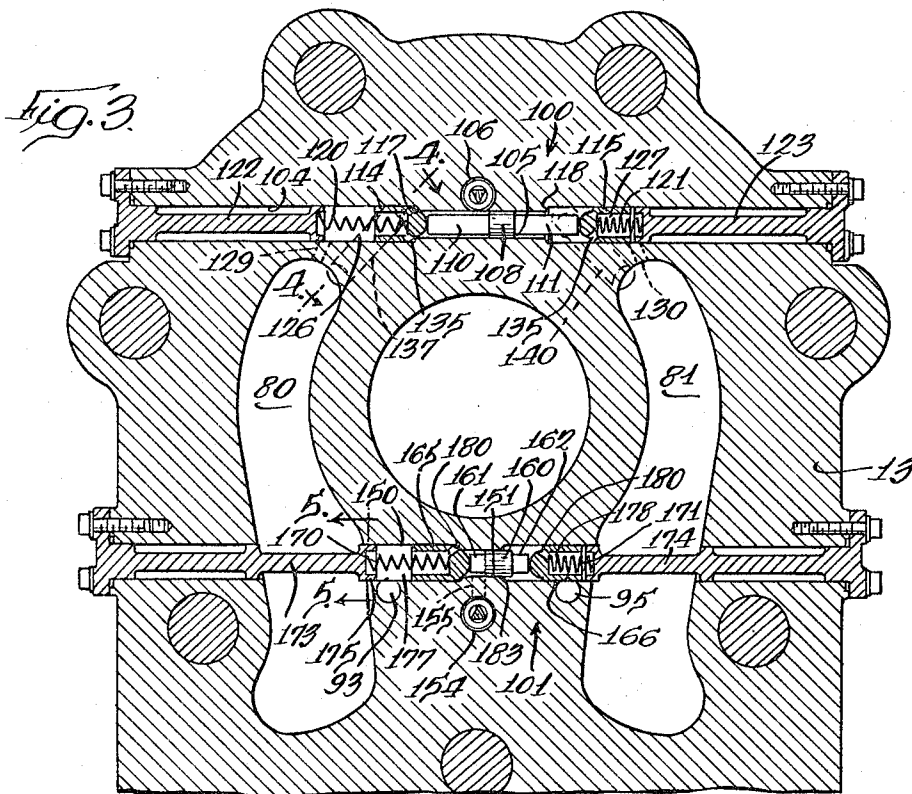
G. A. SCHAUER

3,382,813

HYDRAULIC PUMP OR MOTOR

Filed Feb. 15, 1966

3 Sheets-Sheet 2



May 14, 1968

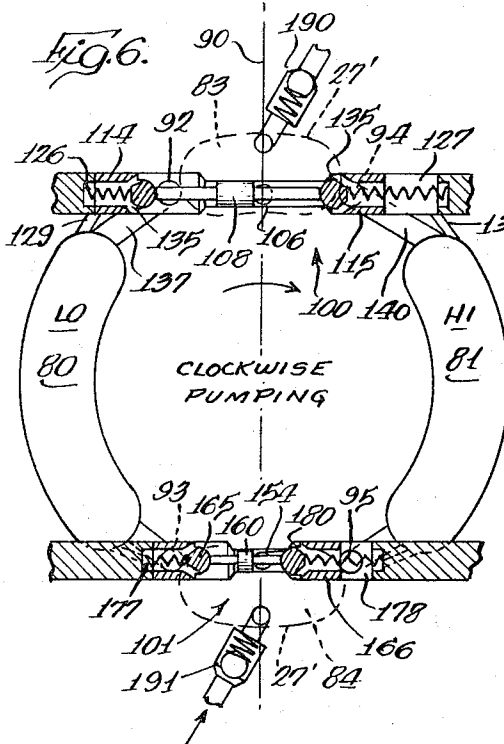
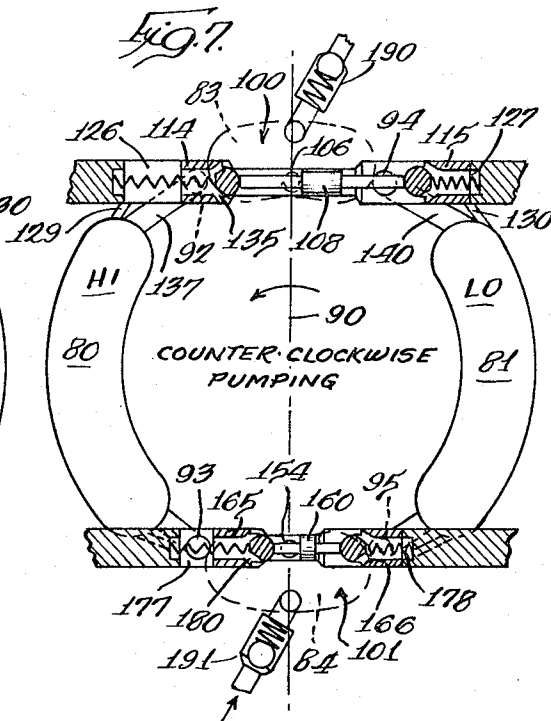
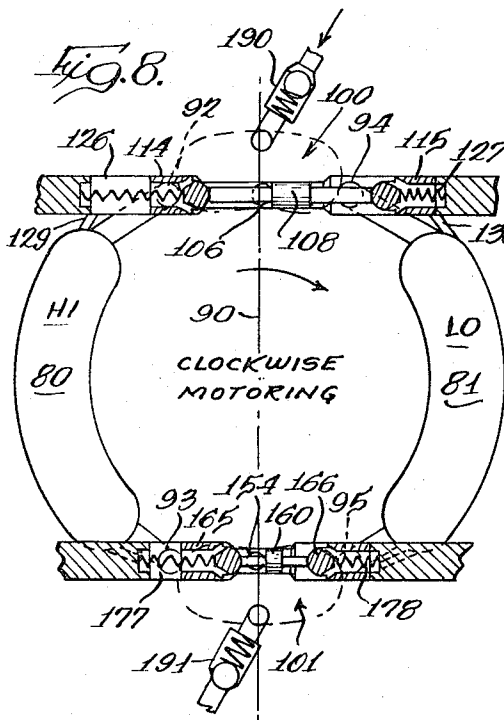
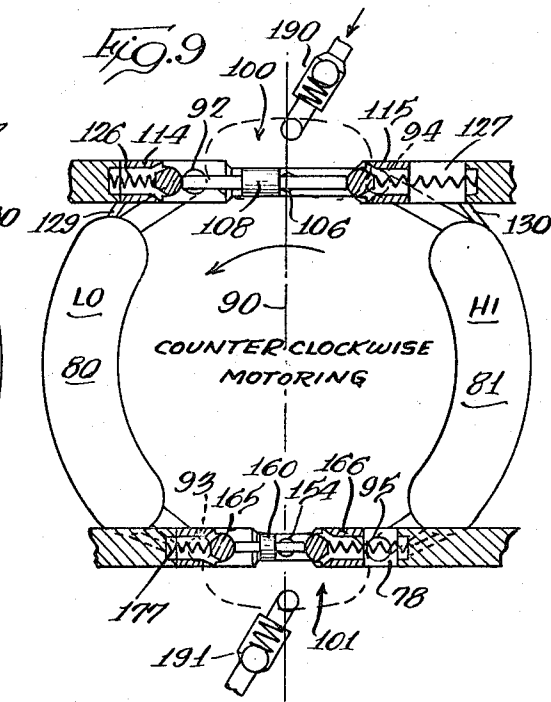
G. A. SCHAUER

3,382,813

HYDRAULIC PUMP OR MOTOR

Filed Feb. 15, 1966

3 Sheets-Sheet 3



1

3,382,813

HYDRAULIC PUMP OR MOTOR

George Schauer, Belvidere, Ill., assignor to Sundstrand Corporation, a corporation of Illinois
Filed Feb. 15, 1966, Ser. No. 527,540
15 Claims. (Cl. 103-162)

ABSTRACT OF THE DISCLOSURE

A hydraulic energy translating device in which noise and shock are minimized by equalizing the pressure between the cylinders and the approaching main ports through the use of (a) auxiliary ports for controlling pre-compression and preexpansion in the cylinders by varying the effective main port length, (b) high pressure relief ports for relieving excess pressure in the cylinders, and (c) low pressure relief ports for increasing the pressure in the cylinders when it becomes too low, with all of these functions being operable when the device operates as a pump or a motor in either direction of rotation.

The present invention relates to hydraulic energy translating devices and, more particularly, to a multiple piston rotary device constructed to operate either as a pump or a motor in either direction of rotation.

In units of this general type, a cylinder block is usually provided which rotatably engages a valve plate having inlet and outlet ports therein. Pistons slidably mounted in cylinders in the block are reciprocated by an inclined cam so that piston bottom dead center occurs as the cylinders pass over one crossover between the high and low pressure ports and piston top dead center occurs at the other crossover between the ports.

In hydraulic devices of this type, problems have resulted from the pressure differential between the high and low pressure ports and the approaching cylinders. This pressure differential results in inefficiency and noise, and endangers the integrity of the device. For example, when the hydraulic unit is operating as a pump, there may be insufficient piston travel as the cylinders approach the high pressure discharge port to precompress the fluid in the cylinders to the outlet port pressure. When these cylinders suddenly communicate with the high pressure port, shock occurs resulting in noise and inefficiency. On the other hand, excessive piston travel at this time may cause fluid pressure in the cylinder to exceed discharge port pressure and this as well produces a shock wave. Similar pressure differentials may occur in the cylinders approaching the low pressure port during pumping, as well as in the cylinders approaching both the high and low pressure ports during motoring.

In accordance with the present device, the pressures in the high and low pressure ports are substantially equal to the pressure in the approaching cylinders during either direction of rotation of the device whether it operates as a pump or as a motor. This vastly improves the efficiency, noise level, and the integrity of the equipment. The timing required to effect this result varies both with the direction of rotation of the hydraulic unit as well as in changing from pump to motor operation.

Toward this end, the start and duration of inlet and outlet flow are varied in the present device to permit the proper compression and expansion of the fluid to and from one pressure level to another. To effect this, auxiliary ports are provided adjacent both ends of the inlet and outlet ports in the valve member and the auxiliary ports which are selectively connected to the adjacent port to properly vary the timing of the device. Further, high and low pressure relief valves are provided to avoid over-expansion

2

and over-compression of the fluid in the cylinders. All of these functions are accomplished automatically for the various operating modes of a hydraulic energy translating device.

It is, therefore, a primary object of the present invention to provide a new and improved hydraulic energy translating device of the type described, in which the timing of the device is automatically varied to substantially equalize the pressure in the cylinders before crossover from one pressure port to the other.

Another object of the present invention is to provide a new and improved hydraulic energy translating device of the type described, in which timing is automatically varied as the operation of the device changes from pumping to motoring as well as when the direction of torque of the device reverses.

A further object of the present invention is to provide a new and improved hydraulic unit of the type described above, in which selectively operable pressure relief valves are provided for preventing over-compression of hydraulic fluid in the cylinders in both directions of operation of the unit regardless of whether the unit is operating as a pump or a motor.

A still further object of the present invention is to provide a new and improved hydraulic energy translating device of the type described above, in which a low pressure relief valve is provided for porting fluid to the cylinders during crossover when the pressure in the cylinders is below the pressure in the approached low pressure port thus, preventing over-expansion of fluid prior to communication with the low pressure port.

A more specific object of the present invention is to provide a new and improved hydraulic energy translating device of the type described, in which a shiftable valve assembly is provided in the valve member which functions to provide both the port length control and the over-compression relief function to properly time the device. The valve assembly includes two opposed spring biased pressure relief valves separated by a shiftable seal member. Portions of the relief valves also selectively block communication between the auxiliary ports and the adjacent one of the main ports. The shiftable seal is effective to provide communication between a relief port opening to the valve member crossover land and the pressure relief valve associated with the high pressure port. The pressure relief valves and the seal shift as a unit in response to a reversal in the high and low pressure main ports.

A still further object of the present invention is to provide a new and improved hydraulic energy translating device of the type described immediately above, in which an additional valve assembly is provided for controlling the auxiliary ports in the other crossover between the high and low pressure main ports providing substantially the same functions as the other valve assembly, so that the timing apparatus is fully reversible to effect proper timing during pumping and motoring in either direction of rotation.

Still another object of the present invention is to provide a new and improved hydraulic unit of the type described immediately above, and including under pressure relief valves communicating with each of the relief ports in the valve member crossover land for porting fluid to the cylinders when the pressure in the cylinders falls below that of a pressure source connected to the other side of the valves.

Another object of the present invention is to provide a new and improved hydraulic unit of the type described above, in which arcuate or kidney shaped ports are provided in the valve member with these ports being spaced closer to one another at the bottom than at the top, so that there is greater piston travel during crossover when

the pistons are retracted (adjacent bottom dead center) than when the pistons are extended adjacent piston top dead center. The purpose of this is to provide proper precompression or expansion of the fluid in the cylinders.

Further objects and advantages will become apparent from the following detailed description taken in connection with the accompanying drawings, in which:

FIGURE 1 is a longitudinal sectional view through an axial piston hydraulic unit incorporating the present invention;

FIGURE 2 is a cross section taken generally along line 2—2 of FIGURE 1 showing the valve plate configuration;

FIGURE 3 is a cross sectional view taken generally along line 3—3 of FIGURE 1 showing the timing valve assemblies in the valve plate;

FIGURE 4 is a fragmentary section taken generally along line 4—4 in FIGURE 3 showing one of the auxiliary port control passages;

FIGURE 5 is a fragmentary sectional view taken generally along line 5—5 in FIGURE 3 showing a portion of the lower valve assembly;

FIGURES 6 to 9 are schematic views of the timing valve assemblies during the various modes of operation of the hydraulic unit.

While this invention is susceptible of embodiment in many different forms, there is shown in the drawings and will herein be described in detail an embodiment of the invention with the understanding that the present disclosure is to be considered as an exemplification of the principles of the invention and is not intended to limit the invention to the embodiment illustrated. The scope of the invention will be pointed out in the appended claims.

Referring now to FIGURE 1, wherein an axial piston hydraulic unit 10 is shown incorporating the present invention, it should be understood that the principles of the invention may be incorporated into other types of multiple piston hydraulic units.

In a preferred construction which may be generally similar to that in Schauer application Ser. No. 292,267, filed July 2, 1963, and assigned to the assignee of this application, a housing member 12 is provided closed at one end by a valve or port plate 13 and at the other end by a generally annular housing member 15. A drive shaft 16 is rotatably supported within the housing and carries at one end thereof a crosshead 18; the other end of the shaft being mounted in bearing 20, while the crosshead 18 is rotatably supported in bearing 21. A cylinder block 25 is rotatably supported in the housing and carries a plurality of axial pistons 26 in cylinders 27. While the cylinders 27 extend generally parallel to the axis of rotation of the cylinder block as shown, some angularity is permissible and may even be desirable under certain circumstances.

To produce rotation of the cylinder block 25, a shaft 30 is splined thereto as at 31 and is secured by means of a universal joint 33 to the shaft 16. The universal joint may be similar to that which is the subject of copending application Ser. No. 453,405, filed by George A. Schauer on May 5, 1965, and assigned to the assignee of this application.

To hold the cylinder block 25 against port plate 35 formed on the inner face of valve member 13, there is provided a spring 37 compressed between a shoulder 39 on shaft 30 and a ring member 40 bearing against a snap ring or the like in the cylinder block 25.

Pivotally connected to each of the pistons 26 are links 42 each having a ball 44 at one end thereof pivotally mounted in a piston 26 and a second ball 46 at the opposite end thereof pivotally secured to one of a plurality of slide members 48 slidably mounted in bores 50 in crosshead 18.

A second series of links 52 are provided with balls 54 and 56 at opposite ends thereof engaging sockets 58

and 59 provided in the slide members 48 and in a rotatable cam ring 61, respectively. The cam ring 61 is rotatably supported by a cam member 63 movably carried on 2 sets of rollers 66 and 67 bearing against arcuate support means 70 formed on the inner face of the housing member 15. Movement of the cam member along the arcuate support means 70 is accomplished by means of a hydraulic piston and cylinder device 72 together with a link 73 having one end pivotally connected to the cam member 63, and the opposite end pivotally connected to piston 74 reciprocable within cylinder 76. Fluid under pressure may be supplied through a port 78 to the cylinder 76 to move the piston 74 to the position shown in the drawing. At its maximum stroke, the cam member bears against an adjustable stop 80a to limit the extent of its movement. Movement of the cam member upon withdrawal from the cylinder 76 is accomplished through the natural moment thereof tending to move the same toward minimum stroke position. The cam plate or swash plate mounting may be similar to that in the first copending Schauer application referred to above.

The side forces or side thrust which would otherwise be directed against the pistons 26 are for the most part taken up by the slide members 48 operating in bores 50. Thus, most of the radial loads in the unit are absorbed by the housing through the crosshead 18 and bearing 21. Furthermore, relatively large changes in amplitude in reciprocation of the pistons 26 can be accomplished through the movement of the cam member 63 due to the angular relationship between the shaft 16 and the axis of cylinder block 25.

Referring to FIGURE 2, wherein the valve plate 13 is shown in more detail, two generally arcuate main ports 80 and 81 are seen to be provided therein. Opposite ports are adapted to be either high or low pressure ports, and inlet or outlet ports depending upon the direction of rotation of cylinder block 25 in the operation of the hydraulic unit as a pump or motor as will appear hereinafter. The arcuate portion of the port plate 35 between the upper ends of main ports 80 and 81 may be termed an upper crossover 83, while the arcuate portion between the lower ends of the main ports may be termed the lower crossover 84.

At this point it may be helpful to briefly review the operation of the hydraulic unit as thus far described in its various modes of operation with a view toward understanding the pressure changes in the cylinders 27 and in the main ports 80 and 81.

When operating as a pump with shaft 30, and cylinder block 25 as seen in FIGURE 2, rotating in a clockwise direction, port 80 will be a low pressure inlet port, while main port 81 will be a high pressure discharge port. That is, as the cylinders in the cylinder block pass over port 80, the pistons 26 are withdrawn from cylinders 27 drawing hydraulic fluid from port 80 into the cylinders. The piston withdrawal may be effected by a piston return ring associated with cam member 63 and/or by charge fluid entering low pressure port 80 acting on the end of the pistons. The cam member 63 is arranged with respect to the port plate 35 so that the pistons reach maximum retraction (piston bottom dead center) when the cylinders are in the position designated 87 in FIGURE 2 midway across the upper crossover 83. After passing over this position, cam member 63 urges the pistons into the cylinders precompressing the fluid therein prior to communication with discharge port 81. The pistons reach the end of the pumping stroke (piston top dead center) when the cylinders are in the position designated 88 in FIGURE 2 midway across the bottom crossover 84.

When shaft 30 is rotated in a counter-clockwise direction during a pumping operation, the main port 81 becomes a low pressure inlet port while port 80 becomes the high pressure discharge port. When operating as a motor, and with shaft 30 rotating in a clockwise direction, port 80 is a high pressure inlet port, while port 81 is the

low pressure discharge port and upon rotation of shaft 30 in a counter-clockwise direction in the motoring mode of the unit, port 81 becomes the high pressure inlet port, and port 80 becomes the low pressure inlet port.

As seen in FIGURE 2, the upper ends of ports 80 and 81 are spaced a greater angular distance from the axis of symmetry 90 of the port plate than the lower ends of the ports. In one construction of this type, the upper ends of the ports were spaced 48 degrees from the axis of symmetry, while the lower ends of the ports were spaced 39½ degrees from the axis of symmetry. The purpose of providing this greater crossover length at crossover 83 than at 84 is to assist in assuring the proper compression and expansion of fluid in the cylinders as they approach the main ports 80 and 81.

More specifically, the amount of travel of the pistons 26 to properly compress or expand the fluid in the cylinders 27 prior to communication with the ports 80 and 81 is dependent upon the volume of fluid in the cylinders as they pass over crossovers 83 and 84. When cylinders 27 pass over the upper crossover 83, the pistons are substantially at full retraction for the particular stroke so that the volume of fluid in the cylinders consists of a clearance volume plus substantially the entire displaced volume. On the other hand, as the cylinders pass over crossover 84 the pistons 26 are substantially at no stroke so that the volume in the cylinders at this time is only equal to about the clearance volume. As greater piston travel is required to compress or expand the fluid as the cylinders pass over the upper crossover 83, than as they pass over crossover 84, the upper crossover 83 is longer than crossover 84 to achieve the proper compression or expansion of fluid in the cylinders.

In addition to the proper spacing of the ports 80 and 81, other means are provided in the present device to assure that the pressure in cylinders 27 substantially equals that in the ports 80 and 81 as the cylinders approach and communicate therewith. Toward this end, auxiliary ports 92, 93, 94 and 95 are provided in the port plate 35 in the path of the cylinder ports 27'.

In the exemplary construction shown, each of the auxiliary ports is spaced at 23½ degrees from the axis of symmetry 90. The auxiliary ports 92 to 95 serve to vary both the angularity of ports 80 and 81 with respect to the axis of symmetry 90 (piston top dead center and piston bottom dead center axis) and also the effective arcuate length of these ports. Ports 92 and 93 are selectively communicable (through valve assemblies described in more detail below) with port 80, while auxiliary ports 94 and 95 are selectively communicable with main port 81.

When port 80 is the low pressure port, auxiliary port 92 communicates therewith while auxiliary port 93 is blocked (FIGURES 6 and 9). The low pressure port thereby extends effectively throughout main port 80 and across auxiliary port 92. On the other hand, when port 80 is the high pressure port, port 93 communicates therewith while port 92 is blocked (FIGURES 7 and 8). Thus, the high pressure port extends across main port 80 and across auxiliary port 92. Note that the effective length of the port is longer when it is at low pressure than when it is at high pressure.

Similar port length control is provided with main port 81. That is, when port 81 is a low pressure port, auxiliary port 94 communicates therewith and port 95 is blocked (FIGS. 7 and 8). When 81 is a high pressure port, port 95 communicates with main port 81, while auxiliary port 94 is blocked. Thus, precompression in the cylinders, as they pass from the low pressure port to the high pressure port during pumping, is increased to properly precompress the fluid in the cylinders to approximate high pressure discharge. Port pressure in the net preexpansion of the fluid in the cylinders as they pass from the high pressure port to the low pressure port is increased. The same thing is true during the motoring mode, although the timing is different as will appear hereinafter.

Note that the auxiliary port 92 is positioned so that when main port 80 is the low pressure port and auxiliary port 92 communicates therewith, as explained above, and cylinder block 25 is rotating clockwise in a pumping mode, cylinder port 27' will remain in communication with the low pressure port 80 through auxiliary port 92 until the piston in cylinder 27' reaches the limit of its retraction stroke thereby preventing unnecessary expansion of fluid in the cylinder. The same is true during counter-clockwise rotation in the pumping mode. Note that auxiliary port 95 communicates with the cylinder port 27' until it reaches the end of the end of the pumping stroke to prevent any unnecessary precompression of fluid in the cylinders before they approach the low pressure inlet port 80. Ports 93 and 94 are positioned in the same manner to effect the identical result upon counter-clockwise pumping operation. Further, the position of the ports effects a similar result during both directions of rotation during motoring.

As shown in FIG. 3, combined overpressure relief and auxiliary port control valve assemblies 100 and 101 are provided for selectively controlling communication of the auxiliary ports 92 to 95 with their associated main ports 80 and 81, and for preventing the overcompression of fluid in the cylinders as they approach the high pressure one of the main ports 80 and 81. Valve assembly 100 includes a through bore 104 in valve plate 13 with a reduced central portion 105. The central reduced bore communicates with the port face 35 through a relief passage 106 which opens to the port face on the axis of symmetry 90.

Referring again to FIG. 1, the relief passage 106 is seen to have a washer 107 at the port plate end thereof which may be retained by a suitable snap ring. Washer 107 does not restrict flow through the passage 106, however.

The valve assembly 100 also includes a shiftable seal member 108 which selectively directs communication between the relief passage 106 and the left or right-hand portions of the central bore 105. Extending from the seal member 108 and defining a part thereof are triangular extensions 110 and 111. If desired, check valves 114 and 115 in bore 104 may be biased toward valve seats 117 and 118 in bore 104 by compression springs 120 and 121, respectively. The compression springs are seated against elongated seating members 122 and 123 fixed within the ends of bore 104. Alternatively, the check valves may be operated solely by fluid pressure, without springs.

The check valves 114 and 115 and the spring seats 122 and 123 define fluid chambers 126 and 127 in bore 104. These chambers communicate, respectively, with main ports 80 and 81 through passages 129 and 130 in valve plate 13 (shown in FIG. 3). Thus, chambers 126 and 127 remain at pressures equal to that in the associated main port.

Assuming that port 80 is the high pressure port and port 81 the low pressure port, fluid pressure in chamber 126 will be greater than that in chamber 127 and check valve 114 will shift to the right against seat 117 forcing seal 108 to the right which moves check valve 115 away from its seat 118. Thus, check valve 114 communicates with relief passage 106 across the triangular extension 110 of seal 108, while the right hand portion of bore 104 is sealed from relief port passage 106 by the seal member 108.

At this time, if pressure in any cylinder passing over relief passage 106 exceeds that of the fluid in port 80, the overcompression relief valve 114 will open permitting fluid to flow through passages 135 in the relief valve into chamber 126 through passage 129 and into high pressure port 80 thus, equalizing the pressure in the cylinders and the approached high pressure port. The springs 120 and 121 are light springs so that they do not result in a significant pressure differential between the cylinders and the high pressure port. On the other hand, if pressure in port 80 is higher than that in the cylinders, at this time,

check valve 114 will remain closed preventing any reverse flow.

If port 81 becomes the high pressure port, the fluid pressure in chamber 127 will shift the check valve 115 to the left from the position shown in FIG. 3, moving the seal 108 to the other side of passage 106 so that the projection 110 lifts relief valve 114 from its valve seat. In this mode, check valve 115 will open if pressure in the cylinders passing over relief port 106 exceeds that in the then high pressure port 81 permitting flow through passages 135 into chamber 127 and through passage 130 into port 81.

In addition to providing the overcompression function, relief valves 114 and 115 also serve to selectively communicate auxiliary ports 92 and 94 with the associated main ports 80 and 81, respectively. For the purpose of describing this function, reference is made to FIG. 4 which is a fragmentary section taken through check valve 114 in FIG. 3 illustrating passages 136 and 137 which connect the auxiliary port 92 with the main port 80. When relief valve 114 is closed engaging seat 117, it blocks communication between port 92 and main port 80. This is the position shown in FIG. 3. On the other hand, when the valve assembly 100 is shifted to the left, in FIG. 3, upon reversal of port 81 to the high pressure port, relief valve 114 will shift to the left opening passage 136 and provide communication between port 92 and main port 80. An identical passage configuration 140 is provided between auxiliary port 94 and main port 81 controlled by relief valve 115. That is, when relief valve 115 is in the position shown in FIG. 3, auxiliary port 94 will communicate with the main port 81. When valve 115 engages seat 118 communication between auxiliary port 94 and main port 81 will be blocked.

Thus, when main port 80 is the high pressure port auxiliary port 92 will be blocked by relief valve 114 while auxiliary port 94 will communicate through passage 140 with main port 81. When port 81 is the high pressure port, relief valve 115 will block auxiliary port 94, while relief valve 114 will be shifted to its open position and permit communication between auxiliary port 92 and main port 80.

The valve assembly 101 is generally similar to valve assembly 100 and functions to not only selectively control auxiliary ports 93 and 95, but also prevents overcompression of fluid in the cylinders as they pass over the lower crossover 84. This assembly includes a valve bore 150 in valve plate 13 with a reduced central portion 151 communicating with an axial pressure relief passage 154 through passage 155. In relief passage 154 is a washer 157 at the port plate end thereof similar to washer 107, as shown in FIG. 1.

Slidable within the reduced central portion 151 of bore 150 is a seal 160 which selectively communicates relief passage 154 with either the left or right side of bore 150. Extending from seal 160 are triangular projections 161 and 162. Overpressure relief valves 165 and 166 are provided similar to valves 114 and 115. If desired, these valves are biased by light springs 170 and 171, respectively. Springs 170 and 171 bear against spring seats 173 and 174, respectively. The ends 175 of the spring seat members are grooved, as shown in FIG. 5, to permit communication between the main port and fluid chambers 177 and 178 which are defined between the valves and the spring seats. Projections 161 and 162 maintain the relief valves 165 and 166 separated, so that one valve is open when the other is closed.

Thus, when port 80 is the high pressure port, the pressure fluid in chamber 177 will shift the valve 165 to the right to its position shown in FIG. 3, shifting the seal member 160 to the right of passage 165, so that projection 162 opens relief valve 166. In this mode, communication is established between relief passage 154 and relief valve 165. If the pressure in a cylinder passing crossover 84 exceeds that in port 80, passage 165 will open permitting

flow from the cylinders to port 80 through relief passage 154, passages 180 in relief valve member 165, chamber 177 and around the end of spring seat 173. Conversely, when pressure port 81 is the high pressure port, seal 160 will be shifted to the left of passage 155 thereby providing communication between passage 154 and check valve 166 which is then seated against seat 183. In this mode, if pressure in the cylinders passing over crossover 84 goes above the pressure in port 81, valve 166 will open permitting flow in a similar manner to port 81.

The check valves 165 and 166 control communication between ports 93 and 95 and main ports 80 and 81 in a similar fashion to valve assembly 100. As may be seen in FIGURE 3, port 93 extends axially and intersects bore 150 so that it communicates with the chamber 177. In a similar fashion, port 94 communicates with the right hand side of bore 150. As chambers 177 and 178 communicate with ports 80 and 81, respectively, auxiliary ports 93 and 95 are thereby communicable with the main ports. As shown in FIGURE 3, when check valve 165 is closed, i.e., seated against its associated valve seat, auxiliary port 93 communicates with main port 80. The valves are constructed, however, so that when opened by one of the extending portions 161 or 162 of the seal, the valves will close ports 93 and 95 preventing communication with the main cylinder. As shown in FIGURE 3, valve 166 is blocking port 95 preventing communication thereby with the main port 81.

Thus, when port 80 is the high pressure port, check valve 165 will remain engaged in its seat, as shown in FIGURE 3, and auxiliary port 93 will communicate with the main port 80 through chamber 177 while auxiliary port 95 will be blocked by relief valve 166. On the other hand, when main port 81 is the high pressure port, the valve assembly 101 will be shifted to the left by fluid acting in chamber 178, auxiliary port 95 will communicate with main port 81 while check valve 165 will block communication between auxiliary port 93 and main port 80.

Referring again to FIGURE 1, over-expansion relief valves 190 and 191 are provided to prevent over-expansion of the fluid in the cylinders 27 as they pass from the high pressure one of the main ports to the low pressure port. These valves also serve to prevent communication of the high and low pressure portions of the system. As these valves are identical, the description thereof will be referenced to valve 190 with the understanding that valve 191 is identical thereto. Valve 190 includes a check valve member 194 biased by a light spring 195 against a seat at one end of relief passage 106. Passage 106 communicates with a transverse bore 197 and valve plate 13 which communicates with a low pressure source of charging system through a suitable fitting 198. Thus, check valve 194 prevents flow from cylinders 27 into passage 197, but permits flow from passage 197 around check valve 194 when the pressure of fluid in cylinders 27 passing over relief passage 106 falls below the pressure of the low pressure source connected to fitting 198. Alternatively, passage 197 may be connected internally to the low pressure one of the main ports and achieve substantially the same function. Thus, if the pistons withdrawing from the cylinders as the cylinders pass over crossover 83 cause a pressure in cylinders 27 to fall below the pressure of the source, check valve 194 will open raising the pressures in the cylinders to that of the source, so that the pressure in the cylinders will not suddenly rise when communication is established with the approached low pressure port. Valve 191 achieves the same result in the cylinders as they pass across crossover 84.

The operation of the present invention will be described with reference to FIGURES 6 to 9 which disclose the following operational modes;

FIGURE 6, clockwise pumping;

FIGURE 7, counter-clockwise pumping;

FIGURE 8, clockwise motoring;

FIGURE 9, counter-clockwise motoring.

Clockwise pumping.—With drive shaft 30 being rotated by a source of power in a clockwise direction as viewed in FIGURE 2, and with cam 63 positioned so that the limit of piston retraction occurs in upper crossover 83, the valve assemblies will assume the position shown in FIGURE 6, with port 80 being the low pressure inlet port and port 81 being the high pressure discharge port. The resulting pressure differential in chambers 127 and 126 will shift valve assembly 100 to the left to the position shown. Check valve 115 blocks auxiliary port 94 while auxiliary port 92 freely communicates with the low pressure main port 80. A pressure differential in chambers 177 and 178 causes valve assembly 101 to shift to its left position shown, where auxiliary port 93 is blocked by check valve 165 and auxiliary port 95 freely communicates with high pressure main port 81.

As the cylinders pass over low pressure port 80, the pistons therein withdrawing, the fluid flows into the cylinders. The cylinders remain in communication with the low pressure port 80 through auxiliary port 92 until the pistons reach their bottom dead center position, as indicated by cylinder port 27' symmetrical about the timing axis 90. Thus, the position of auxiliary port 92 assists in preventing over-expansion of fluid in the cylinders at this time.

Continued rotation of the cylinder block causes cam 63 to begin movement of the pistons into the cylinders 27. As auxiliary port 94 is blocked, the pistons will effect a precompression of fluid in the cylinders raising the pressure therein at least to the pressure in high pressure port 81. If, during this inward movement of the pistons, the pressure in the cylinders rises above that in the high pressure port 81, check valve 115 will open permitting flow from the cylinders through relief passage 106, through passages 135 and the check valve 115, and through passage 130 into the high pressure port 81. The cylinder ports 27' remain in communication with the relief passage 106 until the cylinder ports communicate with main port 81. Thus, when the cylinders communicate with the high pressure port 81, the pressure in the cylinders is substantially equal to that in the high pressure port.

As the cylinders pass over high pressure port 81 the pistons therein discharge fluid into the port. The cylinders continue communication with the high pressure discharge port through auxiliary port 95 until the cylinders reach the bottom dead center position illustrated in dotted lines. This assists in preventing unnecessary compression of the fluid in the cylinders as they pass over lower crossover 84.

Continued rotation of the cylinder block 25 causes cam member 63 to begin the withdrawal of the pistons from the cylinders. This effects a preexpansion of fluid in the cylinders reducing the pressure therein. Auxiliary port 93 is blocked at this time so that crossover 84 effectively extends to the low pressure port 80. As auxiliary port 93 is blocked, the length of crossover 84 is sufficient to decrease the pressure in the cylinders at least to that in low pressure port 80. At this time, if the pressure in the cylinders falls below that in the low pressure port 80 or below that of the low pressure source connected to fitting 198 (FIGURE 1), over-expansion relief valve 191 will open permitting flow across the valve into the cylinder ports raising the pressure therein substantially to that in the low pressure port. It should be understood that the relief valves 190 and 191 are shown schematically in FIGURES 6 to 9, and that as described above with reference to FIGURES 1 to 5 the low pressure relief valves, when open, port fluid through relief passages 106 and 154. Passage 154 is positioned so that it remains in communication with the cylinder ports 27' until the cylinder port communicates with low pressure port 80 as shown in FIGURE 6. Thus, when the cylinder ports communicate with low pressure port 80 the pressure in the cylinders is substantially equal to that in the port.

Counter-clockwise pumping.—With the cam positioned on the same side of neutral as that described with reference to FIGURE 6, but with the shaft 30 being driven in a counter-clockwise direction, the valve assemblies 100 and 101 will assume the positions shown in FIGURE 7, with port 81 changing to the low pressure and port 80 becoming the high pressure discharge port. The pressure differential in chambers 126 and 127 causes valve assembly 100 to shift to its right position shown. In this position, check valve 114 blocks auxiliary port 92, and check valve 115 permits port 94 to communicate with low pressure port 81.

After discharge, through high pressure port 80, the cylinder ports remain in communication with the high pressure port 80 through auxiliary port 93 until the pistons in the cylinders reach the lower crossover 83. Further rotation of the cylinder block causes withdrawal of the pistons. At this time, low pressure or over-expansion relief valve 191 prevents cylinder pressure from falling below that in the low pressure port 81, and in a similar manner as that described above.

Clockwise motoring.—With the cam member 63 positioned so that piston bottom dead center occurs in crossover 83 in the same fashion as that described above and with high pressure fluid being delivered to port 80, the cylinder block will rotate in a clockwise direction and the hydraulic circuit and the valves 100 and 101 will assume the position shown in FIGURE 8. In this position, check valve 114 blocks auxiliary port 92 while auxiliary port 94 freely communicates with the low pressure port 81. In the lower crossover, auxiliary port 93 freely communicates with high pressure port 80, while auxiliary port 95 is blocked by check valve 166. Assuming the cylinder under consideration is passing over low pressure port 81, cam member 63 will at this time be moving the pistons into the cylinders discharging fluid out port 81. As auxiliary port 95 is blocked, when the cylinder ports 27' cease communication with low pressure port 81, the pistons therein continue inward movement in the cylinders a sufficient distance to precompress the fluid therein at least to that in the high pressure port 80. If the pressure in the cylinders at this time rises above that in high pressure port 80, check valve 165 will open permitting flow from the cylinders through passage 154, through valve 165, through chamber 177 and into high pressure port 80.

As auxiliary port 93 communicates with high pressure port 80 at this time, continued rotation of the cylinders past the top dead center position illustrated in dotted lines in FIGURE 8 causes the cylinders to communicate with high pressure port 80 through auxiliary port 93. This prevents any expansion of the fluid in the cylinder prior to communication with high pressure port 80.

As the cylinders pass over the high pressure port 80, fluid flows into the cylinders forcing the pistons outwardly of the cylinders thereby causing rotation of the cylinder block and shaft 16, which is then an output shaft.

The trailing edge of the high pressure port 80 ends at the upper end of the arcuate main port as auxiliary port 92 is blocked at this time. Therefore, as the cylinders cease communication with port 80 the pistons therein continue their withdrawal from the cylinders across crossover 83 thereby providing a preexpansion of the fluid lowering the pressure therein at least to that in low pressure port 81.

At this time, if the pressure in the cylinders falls below that in the low pressure port 81 or that of the constant pressure source connected to fitting 198 (FIGURE 1), check valve 190 will prevent this over-expansion by opening and porting fluid to the cylinders raising the pressure therein to substantially that in low pressure port 81. Further, rotation of the cylinder block and the cylinder port illustrated in the upper crossover in FIGURE 8, causes the ports to communicate with the low pressure port 81 through auxiliary port 94, thereby preventing any precompression of the fluid in the cylinders by the pistons prior to communication with the low pressure discharge port.

Counter-clockwise motoring.—If the hydraulic unit operates under motoring conditions as described above with respect to FIGURE 8, except that high pressure fluid is ported to port 81, the cylinder block 25 is caused to rotate in a counter-clockwise direction, and the valve assemblies 100 and 101 assume the position shown in FIGURE 9.

In this position, auxiliary port 92 freely communicates with low pressure port 80, while auxiliary port 94 is blocked by check valve 115. In the lower crossover, auxiliary port 93 is blocked by check valve 165, while auxiliary port 95 freely communicates with high pressure inlet port 81.

The timing of the device in this mode is the same as that described above with respect to FIGURE 8, except that the direction of rotation of the cylinder block is reversed. That is, fluid is discharged from the cylinders as they pass over low pressure port 80 and upon the cessation of communication with this port a precompression of the fluid is effected due to the auxiliary port 93 being blocked at this time. After the cylinders pass piston top dead center in the lower crossover, the cylinder ports communicate with the open auxiliary port 95, thereby beginning the power stroke at this time. After the cylinders cease communication with high pressure port 81 the pistons continue with withdrawal of the cylinders as auxiliary port 94 is blocked thereby providing a preexpansion of the fluid in the cylinders at least to that of the low pressure port 80.

If the pressure in the cylinders at this time falls below that in the low pressure port 80, check valve 196 will open preventing any over-expansion. And as the cylinders pass top dead center they communicate with low pressure discharge port 80 through auxiliary port 92.

I claim:

1. A rotary hydraulic energy translating device adapted to operate either as a pump or a motor comprising: a cylinder block having a plurality of cylinders therein, pistons slidably mounted in said cylinder block, a valve means slidably engaging said cylinder block and having main inlet and outlet port means therein adapted to serially communicate with the cylinders, cam means for reciprocating said pistons upon relative rotation of said cylinder block and valve means; and means for assisting in the equalization in pressure between the cylinders and an approached port means including, said cam member being located relative to said port means so that the pistons move into or out of the cylinders during crossover from one port means to the other a sufficient distance so that the pressure in the cylinders increases or decreases toward that in the approached port, said valve means including a valve member with generally arcuate inlet and outlet main ports therein, said valve member having valve surface means slidably engaging said cylinder block; and overpressure relief valve assembly for providing communication between the cylinders and the approached high pressure one of said ports in either direction of relative rotation of said cylinder block including relief port means in said valve surface between said ports, a first check valve having an outlet connected to one of said main ports, a second check valve having its outlet connected to the other of said main ports, and shiftable valve means responsive to pressure in said ports for selectively communicating one of the check valves to said relief port means whereby the check valve associated with the high pressure one of the main ports will communicate with the relief port to relieve pressure in the cylinders approaching the high pressure port.

2. A rotary hydraulic energy translating device as defined in claim 1 wherein said overpressure relief valve assembly includes a valve bore in said valve member, said relief port communicating with said valve bore, said bore having two valve seats therein each engageable by one of the check valves, said shiftable valve including a seal member slidably mounted in said bore for selectively connecting said relief port to one of said check valves and

blocking communication between said relief port and the other of said check valves.

3. A rotary hydraulic fluid energy translating device adapted to operate as either a pump or a motor in either direction of rotation, the combination comprising: a valve member having generally arcuate main inlet and outlet ports therein spaced from each other and defining therebetween first and second crossover portions, a cylinder block slidably engaging said valve member and having a plurality of cylinders therein adapted to serially communicate with the main ports, pistons slidable in said cylinder block, an inclined cam member for reciprocating said pistons upon relative rotation of said cylinder block and valve member, said cam being positioned so that the axes of the pistons at bottom dead center intersect said valve member in said first crossover position, and the axes of the pistons at bottom dead center intersect said valve member in said second crossover portion; means for equalizing the pressure in the cylinders with the main ports approached thereby including two auxiliary ports in said first crossover each selectively communicable with one of the main ports, two auxiliary ports in the second crossover each selectively communicable with one of the main ports, a relief port in one of said crossover portions serially communicable with said cylinders, pressure relief valve means connected to said relief port for permitting communication between the cylinders and the approached high pressure one of said ports if the cylinder pressure is above that in the high pressure port, said relief valve means also selectively communicating the auxiliary ports in said one crossover portion with their associated main ports, and pressure responsive means for shifting said relief valve means to block the auxiliary port in the first crossover associated with the high pressure main port, to provide communication between the other auxiliary port in the first crossover portion and the low pressure one of said ports, and to connect said relief port through the relief valve to the high pressure port.

4. A rotary hydraulic energy translating device as defined in claim 3 including a second relief port in said second crossover portion, second relief valve means communicable with said relief valve port for providing communication between the cylinders and the approached high pressure one of said ports if the pressure in the cylinders exceeds that in the high pressure port, said second relief valve means also controlling the selective communication between said second crossover auxiliary ports and the associated main ports, and pressure responsive means for shifting said second relief valve in response to pressure in said main ports to block the auxiliary port in the second crossover associated with the low pressure one of the main ports, to provide communication between the other auxiliary port in the second crossover and the high pressure one of said main ports, and to connect the relief port to the high pressure port through the relief valve means.

5. A rotary hydraulic energy translating device as defined in claim 3 including a low pressure relief valve in each of said first and second crossover portions for porting fluid to the cylinders as they approach the low pressure one of said main ports as the pressure in the cylinders falls below that in the low pressure main port.

6. A rotary hydraulic energy translating device adapted to operate as either a pump or a motor in both directions of rotation comprising: a cylinder block having a plurality of cylinders therein, pistons slidably mounted in said cylinder block, valve means slidably engaging said cylinder block having main inlet and outlet port means therein adapted to serially communicate with the cylinders, said port means defining a first crossover portion and a second crossover portion communicable with said cylinders upon relative rotation of said cylinder block and said valve means, cam means for reciprocating said pistons upon said relative rotation, and means for assisting in the equaliza-

tion of pressure between the cylinders and an approached port means including, two auxiliary ports in said first crossover portion each adjacent one end of one of said inlet and outlet port means, valve means responsive to pressure in said inlet and outlet port means for connecting selectively the auxiliary ports with the adjacent main inlet and outlet port means so that the pistons move into or out of the cylinders a sufficient distance so that the pressure in the cylinders increases or decreases toward that in the approached port, relief port means in said first crossover portion substantially centrally between said auxiliary ports, high pressure relief valve means communicating with said relief port means for relieving pressure in the cylinders in excess of the pressure in the high pressure one of said main ports regardless of the direction of rotation of said cylinder block, and overexpansion relief valve means communicating with said relief port means for porting fluid to the cylinders as the pressure in the cylinders falls below the pressure in the low pressure one of the approached ports as the cylinders approach said low pressure port from said first crossover regardless of the direction of rotation of said cylinder block.

7. A rotary hydraulic energy translating device adapted to operate either as a pump or a motor comprising: a cylinder block having a plurality of cylinders therein, pistons slidably mounted in said cylinder block, a valve means slidably engaging said cylinder block and having main inlet and outlet port means therein adapted to serially communicate with the cylinders, cam means for reciprocating said pistons upon relative rotation of said cylinder block and valve means; and means for assisting in the equalization in pressure between the cylinders and an approached port means including, said cam member being located relative to said port means so that the pistons move into or out of the cylinders during crossover from one port means to the other a sufficient distance so that the pressure in the cylinders increases or decreases toward that in the approached port, said valve means including a valve member with generally arcuate inlet and outlet main ports therein, said valve member having valve surface means slidably engaging said cylinder block; an overpressure relief valve assembly for providing communication between the cylinders and the approached high pressure one of said ports in either direction of relative rotation of said cylinder block including relief port means in said valve surface between said ports, a first check valve having an outlet connected to one of said main ports, a second check valve having its outlet connected to the other of said main ports, shiftable valve means responsive to pressure in said ports for selectively communicating one of the check valves to said pressure port whereby the check valve associated with the high pressure one of the main ports will communicate with the relief port to relieve pressure in the cylinders approaching the high pressure port including an auxiliary timing port adjacent one end of each of the main ports for increasing the compression of fluid in the cylinders approaching the high pressure one of said main ports during pumping, both of said auxiliary ports being located in one of the crossovers between the main ports to accommodate reversible pumping, said one crossover being the crossover intersected by the piston axes at their top dead center positions, and means for blocking the auxiliary port adjacent the high pressure one of said ports and providing communication between the low pressure one of said main ports and the adjacent auxiliary port.

8. A rotary hydraulic energy translating device as defined in claim 7 wherein the means for connecting the auxiliary ports includes said first and second check valves.

9. A rotary hydraulic energy translating device as defined in claim 8 wherein said valve means includes a valve member having generally arcuate main inlet and outlet ports, the main ports being spaced and defining a crossover therebetween; said overpressure relief in-

cluding a valve bore in said valve member, spaced valve seats in said bore each selectively engageable with one of said check valves, said relief port communicating with said bore, a shiftable seal member slidable in said bore and selectively communicating the check valves with said relief port, said seal member having projecting portions normally engaging both of said check valves, said projections having a sufficient length so that only one of said check valves engages its associated seat whereby the check valves and seal shift as a unit, each of said check valves selectively connecting one of said auxiliary ports with the associated main port, and pressure responsive means for shifting said unit responsive to pressure in said main ports for simultaneously moving said seal to block communication between the relief port and the check valve connected with the low pressure main port, and to provide communication between the relief port and the check valve connected with the high pressure port, and for simultaneously moving said check valves to communicate one auxiliary port with the low pressure one of said main ports.

10. A rotary hydraulic fluid energy translating device adapted to operate as either a pump or a motor in either direction of rotation, the combination comprising: a valve member having generally arcuate main inlet and outlet ports therein spaced from each other and defining therebetween first and second crossover portions, a cylinder block slidably engaging said valve member and having a plurality of cylinders therein adapted to serially communicate with the main ports, pistons slidable in said cylinder block, an inclined cam member for reciprocating said pistons upon relative rotation of said cylinder block and valve member, said cam being positioned so that the axes of the pistons at bottom dead center intersect said valve member in said first crossover position, and the axes of the pistons at bottom dead center intersect said valve member in said second crossover portion; means for equalizing the pressure in the cylinders with the main ports approached thereby including two auxiliary ports in said first crossover each selectively communicable with one of the main ports, two auxiliary ports in the second crossover each selectively communicable with one of the main ports, a relief port in one of said crossover portions serially communicable with said cylinders, pressure relief valve means connected to said relief port for permitting communication between the cylinders and the approached high pressure one of said ports if the cylinder pressure is above that in the high pressure port, said relief valve means also selectively communicating the auxiliary ports in said one crossover portion with their associated main ports, pressure responsive means for shifting said relief valve means to block the auxiliary port in the first crossover associated with the high pressure main port, to provide communication between the other auxiliary port in the first crossover portion and the low pressure one of said ports, and to connect said relief port through the relief valve to the high pressure port, a second relief port in said second crossover portion, second relief valve means communicable with said relief valve port for providing communication between the cylinders and the approached high pressure one of said ports if the pressure in the cylinders exceeds that in the high pressure port, said second relief valve means also controlling the selective communication between said second crossover auxiliary ports and the associated main ports, pressure responsive means for shifting said second relief valve in response to pressure in said main ports to block the auxiliary port in the second crossover associated with the low pressure one of the main ports, to provide communication between the other auxiliary port in the second crossover and the high pressure one of said main ports, and to connect the relief port to the high pressure port through the relief valve means, each of said relief valve means including a bore in said valve member, spaced valve seats in said bore, check valves

engageable with each of said seats, resilient means biasing each of said valves toward its associated seat, a shiftable seal member slidable in said bore between said check valves and adapted to selectively communicate said relief port with one of the check valves, projections on said seal member normally engaging said check valves and spacing said valves apart a greater distance than said seats whereby when one of the check valves is closed the other will be open, and passage means in the valve member providing communication between the main ports and the bore so that the check valves and seal member shift as a unit when the high pressure shifts from one main port to the other.

11. A rotary hydraulic energy translating device as defined in claim 10 including low pressure relief valve means for porting fluid to the cylinders approaching the low pressure port as the pressure in the cylinders falls below the port pressure including low pressure check valves having an inlet communicating with a low pressure source, said check valves each having an outlet communicating with one of said crossover portions to provide flow to the cylinders if the proper pressure conditions exist.

12. A rotary hydraulic energy translating device adapted to operate as a pump or a motor comprising: a cylinder block having a plurality of cylinders therein, pistons slidably mounted in said cylinder block, valve means slidably engaging said cylinder block having main inlet and outlet port means therein adapted to serially communicate with the cylinders, said inlet and outlet port means being spaced defining a first crossover portion and a second crossover portion therebetween, cam means for reciprocating said pistons upon relative rotation of said cylinder block and said valve means, and means for assisting in equalization of pressure between the cylinders and an approached port means including at least one auxiliary port adjacent one end of one of said inlet and outlet port means, an overpressure relief valve for providing communication between the cylinders and the approached high pressure one of said ports including a movable valve member, said movable valve member also functioning to control communication between said auxiliary port and one of said port means.

13. A rotary hydraulic energy translating device comprising: a cylinder block having a plurality of cylinders therein, pistons slidably mounted in said cylinder block, valve means slidably engaging said cylinder block having main inlet and outlet ports therein adapted to serially communicate with the cylinders, said inlet and outlet ports being spaced defining a first crossover portion and

a second crossover portion, cam means for reciprocating said pistons upon relative rotation of said cylinder block and valve means, said cam means being positioned so that the axes of the pistons at bottom dead center intersect said first crossover portion and the axes of the pistons at top dead center intersect the second crossover portion, the point of intersection of the piston axes at bottom dead center not bisecting said first crossover portion so that the pistons when moving across said first crossover portion have a net inward or outward movement whereby the pressure of fluid in the cylinders approaches that in the approached port means, and the point of intersection of the piston axes at top dead center not bisecting the second crossover portion so that the pistons when moving across said second crossover portion have a net inward or outward movement whereby the pressure of fluid in the cylinders approaches that in the approached port means, said inlet and outlet port means being positioned with respect to said points of intersection so that the net inward or outward movement of the pistons during bottom dead center crossover is greater than the net inward or outward movement of the pistons during top dead center crossover.

14. A rotary hydraulic energy translating device as defined in claim 12 wherein said second crossover has a lesser arcuate length than said first crossover.

15. A rotary hydraulic energy translating device as defined in claim 12 further including means for shifting said points of intersection with respect to said first and second crossover portions so that precompression and pre-expansion of fluid in the cylinders is effected in either direction of rotation of said cylinder block, and means for maintaining the net inward or outward piston movement at bottom dead center crossover greater than the net inward or outward piston movement at top dead center crossover when the points of intersection shift with respect to said crossovers.

References Cited

UNITED STATES PATENTS

2,122,045	6/1938	Rose et al.	103—162
2,642,809	6/1953	Born et al.	103—162
2,649,741	8/1953	Henrichsen	103—162
2,661,695	12/1953	Ferris	103—162
2,963,983	12/1960	Wiggermann	103—162
3,179,060	4/1965	Lehrer	103—162

DONLEY J. STOCKING, *Primary Examiner.*

WILLIAM L. FREEH, *Examiner.*

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,382,813

May 14, 1968

George A. Schauer

It is certified that error appears in the above identified patent and that said Letters Patent are hereby corrected as shown below:

In the heading to the printed specification, line 4, "a corporation of Illinois" should read -- a corporation of Delaware --.

Signed and sealed this 10th day of March 1970.

(SEAL)

Attest:

Edward M. Fletcher, Jr.
Attesting Officer

WILLIAM E. SCHUYLER, JR.
Commissioner of Patents