Dec. 22, 1959 W. M. SHAFFER 2,917,897

HYDRAULIC DRIVE MECHANISM

Filed Feb. 4, 1955 5 Sheets-Sheet 1

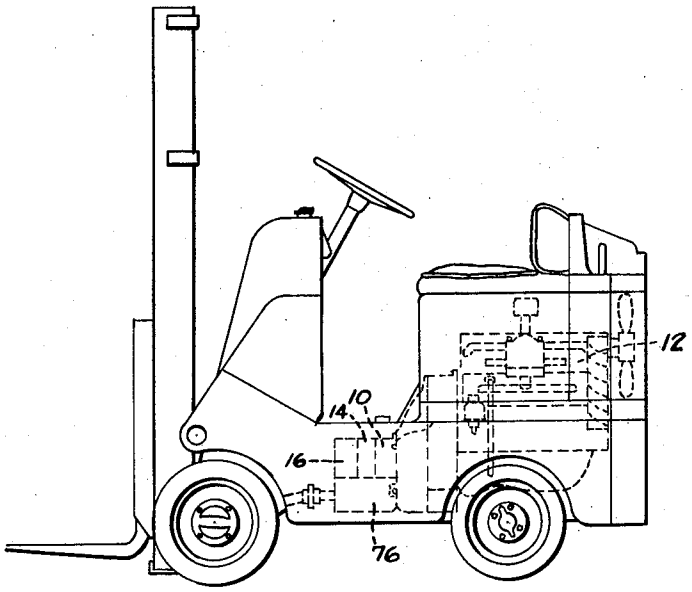


Fig. 1

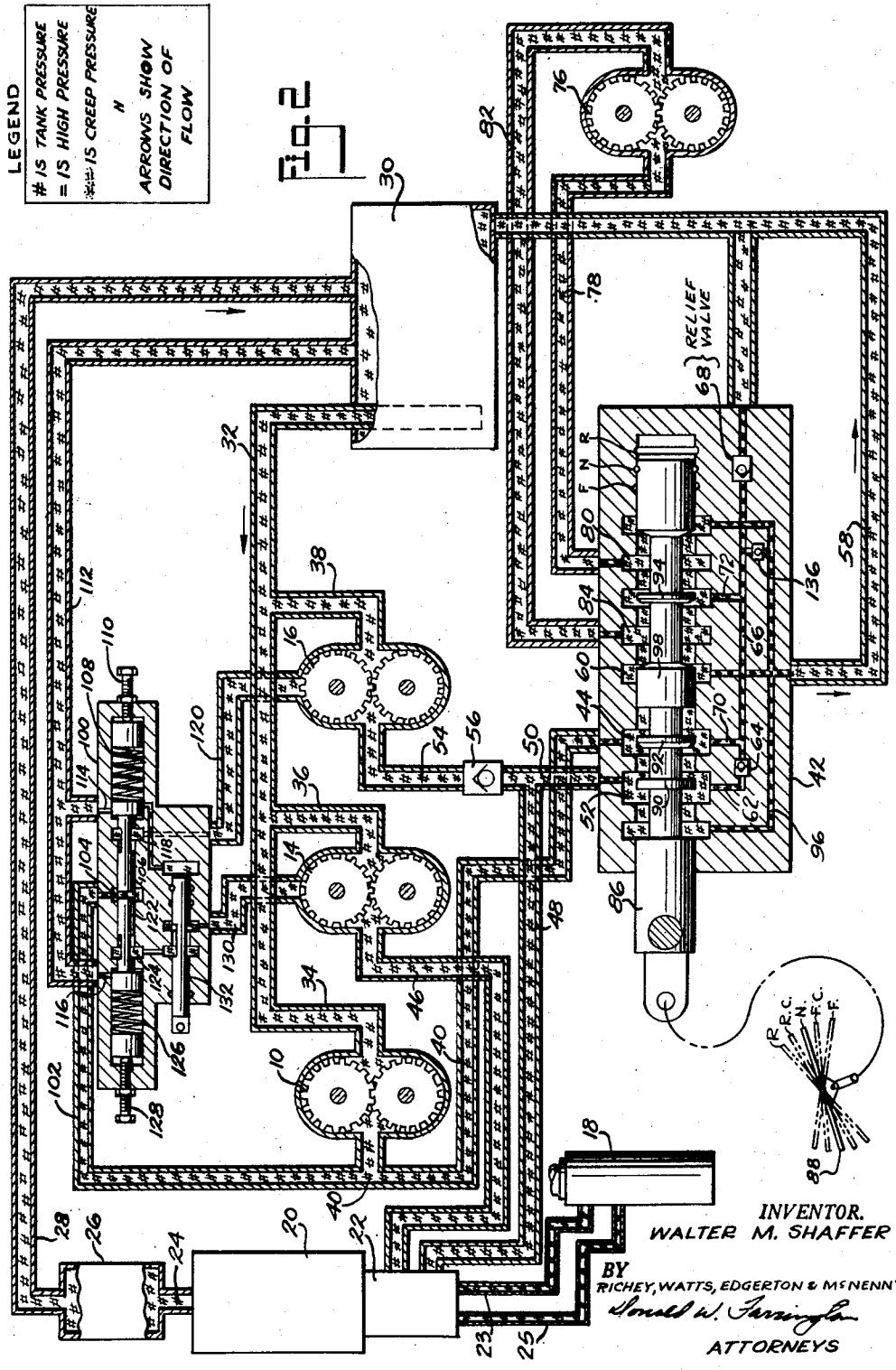
INVENTOR.
WALTER M. SHAFFER

BY
RICHEY, WATTS, EDGERTON & McNENNY
Harold W. Farrington
ATTORNEYS

Dec. 22, 1959 W. M. SHAFFER 2,917,897

HYDRAULIC DRIVE MECHANISM

Filed Feb. 4, 1955 5 Sheets-Sheet 2

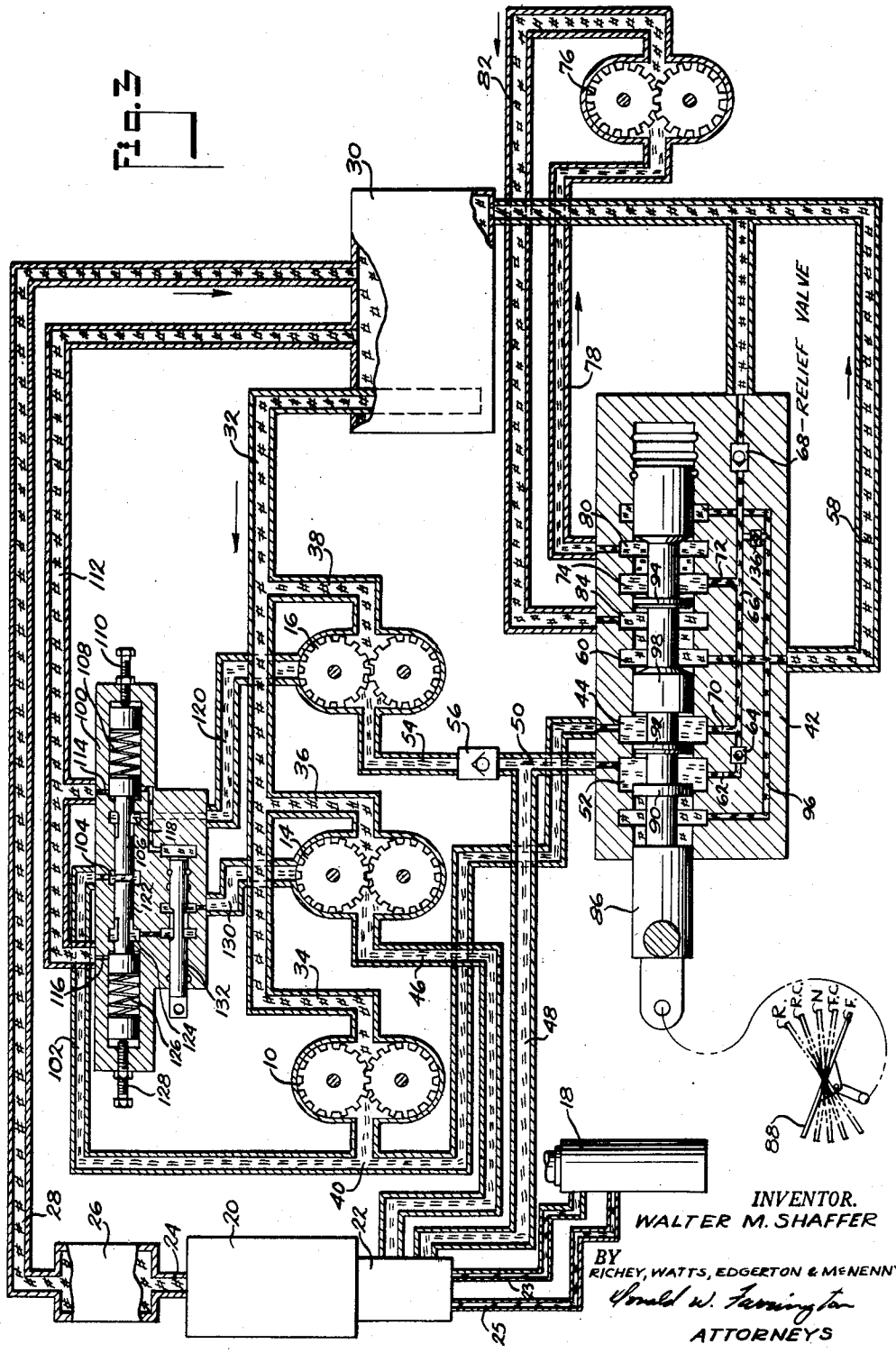


Dec. 22, 1959 W. M. SHAFFER 2,917,897

HYDRAULIC DRIVE MECHANISM

Filed Feb. 4, 1955 5 Sheets-Sheet 3

Fig. 3



INVENTOR.
WALTER M. SHAFFER

BY
RICHEY, WATTS, EDGERTON & McNENNY
Donald W. Farrington
ATTORNEYS

Dec. 22, 1959 W. M. SHAFFER 2,917,897

HYDRAULIC DRIVE MECHANISM

Filed Feb. 4, 1955 5 Sheets-Sheet 4

Fig. 4

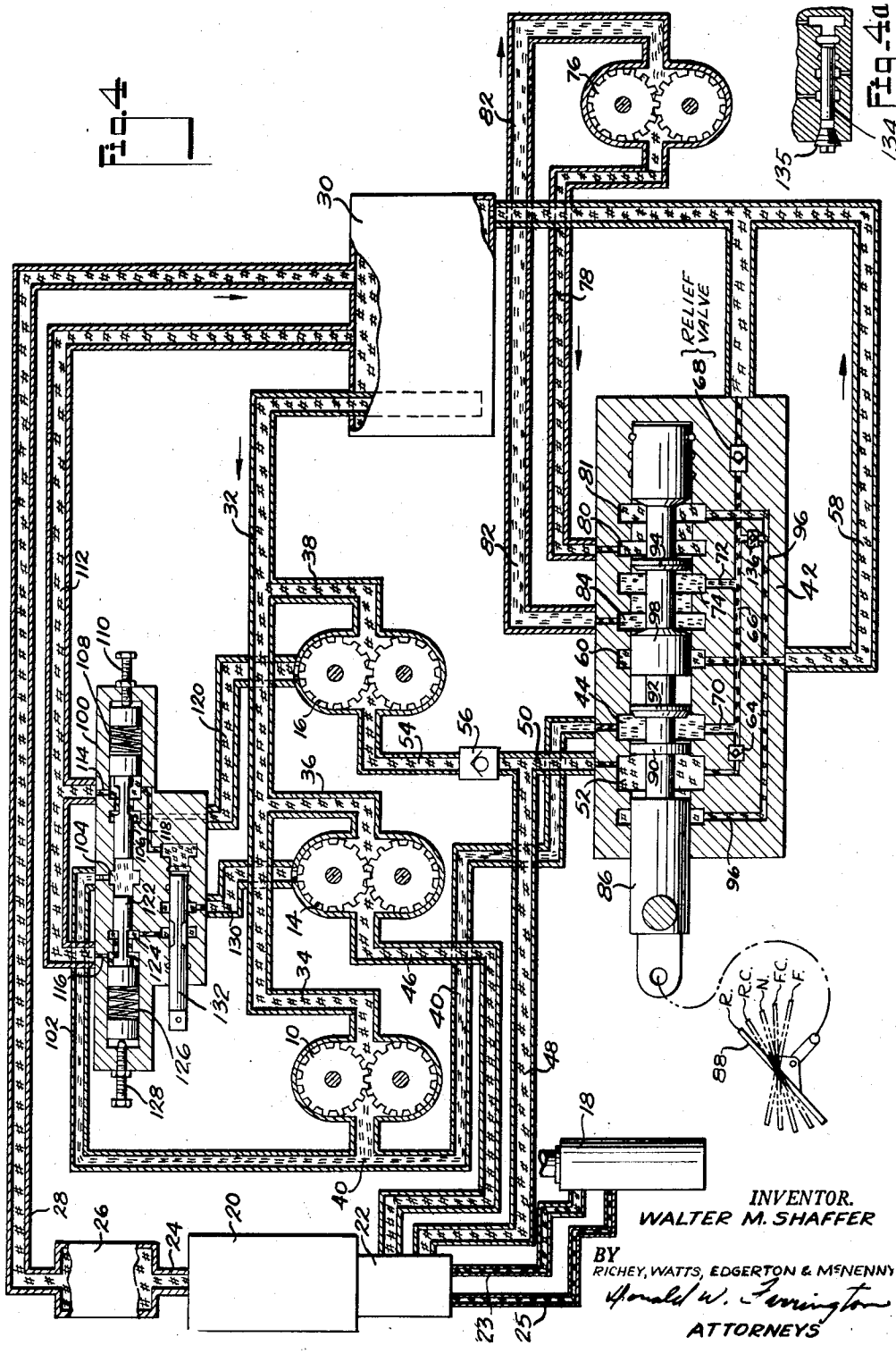


Fig. 4a

INVENTOR.
WALTER M. SHAFFER

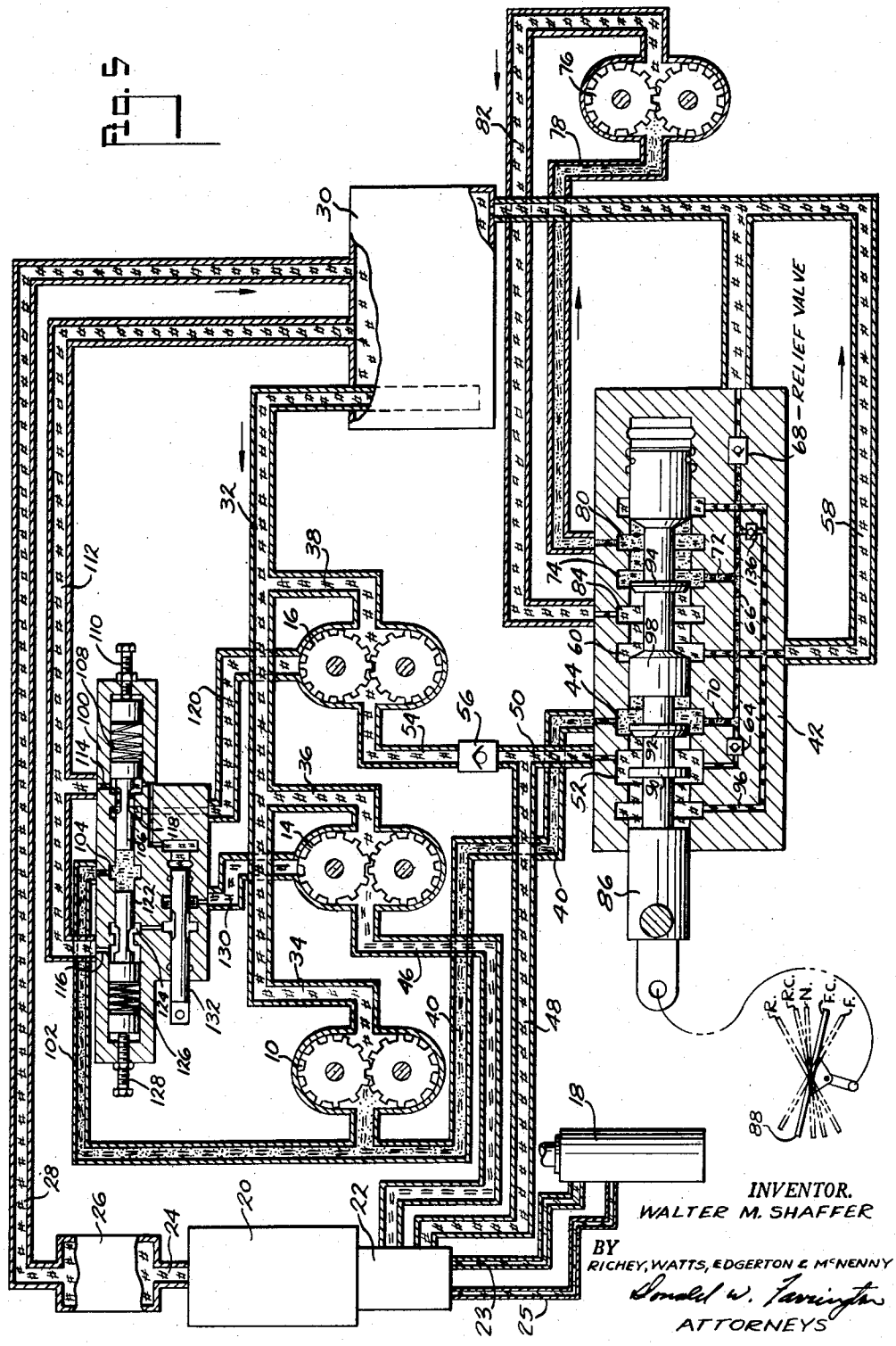
BY
RICHEY, WATTS, EDGERTON & McNENNY
Donald W. Pennington
ATTORNEYS

Dec. 22, 1959 W. M. SHAFFER 2,917,897

HYDRAULIC DRIVE MECHANISM

Filed Feb. 4, 1955 5 Sheets-Sheet 5

Fig. 5



INVENTOR.
WALTER M. SHAFFER
BY
RICHEY, WATTS, EDGERTON & McNENNY
Donald W. Farrington
ATTORNEYS

United States Patent Office 2,917,897 Patented Dec. 22, 1959

2,917,897

HYDRAULIC DRIVE MECHANISM

Walter M. Shaffer, Cleveland, Ohio

Application February 4, 1955, Serial No. 486,136

3 Claims. (Cl. 60—53)

This invention relates to lift trucks and more particularly to lift trucks having hydraulic drives as well as hydraulic lifting apparatus.

One of the objects of this invention is to provide a hydraulic drive system for a lift truck which replaces the usual drive system utilizing gears and friction clutches.

Another object of this invention is to provide a hydraulic drive for a lift truck which has directional, speed and creeping control obtained by valves in a fluid system.

A still further object of the invention is to provide a hydraulic drive for lift trucks in which automatic speed shifting based upon torque requirements at the motor output shaft is obtaned.

Another object of this invention is to provide a hydraulic drive for a lift truck which has a manual override for an automatic speed shifting system based on torque requirement at the motor output shaft.

Another important object of the invention is to provide a hydraulic drive system for a lift truck in which a plurality of pumps driven by the motor is used, the pumps being operated singly or in combination to obtain various speeds and output torques as required by the demand on the lift truck.

This invention contemplates the use of a pump of conventional design which operates to deliver oil under pressure whenever the lift truck engine is running. This pump supplies oil under pressure to a motor which is connected to the lift truck drive mechanism. The pump is sized to provide sufficient oil for low speed operation at a high torque requirement or for conditions where low speed is desired in maneuvering the vehicle. A second pump is automatically incorporated in the system when higher speed is desired requiring a larger supply of oil to the motor. When the motor speed is no longer required a second pump is unloaded and the first pump maintains the pressure in the system. A third pump is automatically loaded for supplying additional oil to the motor when high speed operation of the vehicle is required.

The combined displacement of the three pumps is equal to the displacement of the motor thereby producing a motor shaft speed equal to the lift truck engine speed in the high speed range.

The hydraulic lift system is connected to the pumps utilized for driving purposes thereby avoiding duplication and increasing the overall efficiency of the truck. Sufficient capacity is available for operating the lifting cylinder in the speed ranges other than the high speed range since there would be the capacity of at least one pump available for operating the lift and two pumps available for producing motion of the vehicle. By utilizing this pumping arrangement certain advantages of space requirements are obtained by locating the pumps at the rear of the engine and thereby eliminating the pump usually located at the fan end of the engine. This results in a shorter drive unit and frees space which may be used for additional counterweight or other purposes.

The control of the system is based generally on maintaining a predetermined pressure in the supply system. If the speed of the vehicle which is reflected in the displacement of the motor is greater than the displacement of the pump which is in operation, the supply pressure would be low and thus actuate a control valve which would cause another pump to begin operating and restore the supply pressure to normal. Since this control is automatic, the operator of the vehicle is relieved of the necessity of manual control of the number of pumps which are operating. However, manual override of the automatic control is provided when it is desired to operate the drive with one pump only even though the torque requirement is low.

Accordingly, it is an important object of this invention to provide a lift truck having a hydraulic drive and lift system wherein a plurality of pumps are available for operating both the lift and the drive system individually or simultaneously as required by the operating conditions.

A further important object of this invention is to provide a combination drive elevator actuating system for a lift truck which provides greater heat radiating capacity which results from circulation of the oil through a portion of the system which is not being operated to capacity.

Another object of this invention is to provide a power supply and control system for two separate functions on a lift truck, greater efficiency being achieved by making the peak capacity available to either system.

Other objects and advantages, more or less ancillary to the foregoing, and the manner in which all the various objects are realized will appear in the following description, which, considered in connection with the accompanying drawings, sets forth the preferred embodiment of the invention.

Referring to the drawings:

Fig. 1 is an elevational view of a lift truck utilizing the invention disclosed herein;

Fig. 2 is a schematic diagram of the hydraulic drive system in neutral condition;

Fig. 3 is a schematic diagram of the hydraulic drive system in forward condition;

Fig. 4 is a schematic diagram of the hydraulic drive system in the reversed condition,

Fig. 4*a* is a detailed showing of an alternative arrangement employing means for unloading one of the pumps in the pressure sensing valve; and

Fig. 5 is a schematic diagram of the hydraulic drive system in forward creep condition.

The preferred embodiment of my invention utilizes a conventional gear type hydraulic pump **10** which is mounted on the rear end of a lift truck engine **12**. Pump **10** is directly connected to the drive shaft of the engine **12** and rotates therewith to deliver hydraulic fluid under pressure whenever the engine is operating. A second pump **14** is connected in tandem with pump **10** and is of the unloading type, for example, and as well known in the pump art, the unloading may be accomplished by maintaining the wear plate behind one of the gears in the pumping position, and by utilizing an independent hydraulic pressure control, the wear plate may be moved toward and away from its gear to accomplish the loading and unloading thereof. A third pump **16** is driven in tandem with pumps **10** and **14** and has the control characteristics of pump **14**.

The hydraulic system is divided to accomplish certain requirements of the lift truck. The elevating apparatus utilizes at least one cylinder **18** for activating the mast carriage and accessories mounted thereon. A reservoir **20** is connected through a 4-way valve **22** to the cylinder **18** and functions as a sump for receiving the fluid from

the cylinders 18 and supplying the pumps 10, 14 and 16. A reservoir 20 is connected by a conduit 24 to a filter 26 which prevents the accumulation of foreign matter in the hydraulic system. A conduit 28 connects the filter 26 and the reservoir 20 to a second reservoir 30 which functions as the main reservoir for the drive system and the pumps 10, 14 and 16. A supply line 32 connects the reservoir 30 with the inlet connection 34 for pump 10, the inlet connection 36 for pump 14, and the inlet connection 38 for pump 16.

Oil discharged from pump 10 through conduit 40 to a main control valve 42 and is connected to port 44 therein. The oil discharge from the pump 14 passes through conduit 46 to the 4-way valve 22 which has a high pressure carryover which allows the passage of fluid therethrough to conduit 48 which is connected through line 50 to a port 52, in the valve 42. The oil discharged from pump 16 passes through conduit 54 to check valve 56 and conduit 50 to port 52. It can be readily seen that the capacity of pump 10 is available at port 44 and the combined capacities of pumps 14 and 16 is available at port 52 when the elevating apparatus is not being operated.

A return line 58 connecting to the reservoir 30 opens into a port 60 in the valve 42. A passage 62 from the port 52 connects through a check valve 64, a passage 66 and a relief valve 68 to the return line 58. A passage 70 connects the port 44 to the passage 66 between the check valve 64 and the relief valve 68. The relief valve 68 prevents over-pressure in the system and the check valve 64 prevents short-circuiting of the hydraulic fluid under certain operating conditions. A passage 72 connects the port 74 in the valve 42 to the passage 66 which is the low pressure manifold connecting ports 44, 52 and 74.

A reversible hydraulic motor 76 has its output shaft connected to the differential in the driving axle of the lift truck. The hydraulic fluid for operation in the motor 76 is conducted thereto through a conduit 78 which opens into port 80 in valve 42. A return conduit 82 connects the motor 76 to port 84 in valve 42. It is noted that the flow through conduits 78 and 82 may be reversed by operation of the control valve 42 so that conduit 78 may be either the inlet or the outlet to the motor 76, and conduit 82 may also be either the inlet or the outlet to the motor 76.

A plunger 86 is reciprocable to various positions in valve 42 to selectively direct a desired proportion of fluid from the pumps to motor 76 and to direct the remaining proportion of fluid to reservoir 30 through channels bypassing the motor. A plunger 86 is positioned by linkage which is attached to a foot pedal 88 which functions as the manual device operable by the driver of the vehicle to effect the desired change in speed or direction thereof.

In Figure 2 the plunger 86 is shown in the neutral position wherein lands 90 and 92 are located in the bypass ports 52 and 44 and away from the walls thereof to allow free fluid flow therein and a shoulder 91 of plunger 86 is disposed to prevent flow of fluid beyond a bypass port 93. In this position of the plunger, free fluid flow is allowed between ports 44, 52 and bypass 93 into a conduit 96, communicating with port 93, and thence to conduit 58 to the reservoir 30, thus making it impossible to build up any pressure to cause the hydraulic motor 76 to operate. Under the circumstances no fluid is pumped into or from ports 80, 84 and 74. In Figure 3 the plunger is shown in the forward drive position at which point the hydraulic fluid from the outlets of respective pumps 10, 14 and 16 is directed through line 40 to port 44 on the one hand, line 46, valve 22 and lines 48 and 50 to chamber 52 on another hand and through line 54 valve 56 and line 50 to port 52 on the third hand. Fluid applied at port 44 is restricted from flow into chamber 52 by land 92 and is discharged through line 70 and in sequence flows from line 70 into port 74, into port 80 along a reduced portion of plunger 86 between land 94 and a shoulder 95 of plunger 86 confining the fluid through conduit 78 and into motor 76. The fluid is discharged from motor 76 into line 82 and flows into port 84, along a reduced portion of plunger 86 into port 60 between lands 94 and 98 and back to reservoir 30 through conduit 58.

Automatic speed control of the vehicle is accomplished by a regulating valve 100 which is connected to the pump output line 40 by a connection 102. The fluid connection 102 senses the output pressure of the pump 10 and supplies this pressure to the port 104 in the valve 100. A plunger 106 is reciprocable in the valve 100 and biased in one direction by a compression spring 108. An adjusting screw 110 predetermines the force applied by the spring 108 to the plunger 106 and allows the valve 100 to be actuated at any desired pressure. The end of the plunger 106 at the port 104 has the flow pressure applied thereto which is balanced against the spring 108. Accordingly, when the pressure from the pump discharge reaches a high value the plunger 106 moves against the spring 108 and opens the valve.

A return line 112 connects the ports 114 and 116 in the valve 100 to the reservoir 30. The port 118 is connected by a line 120 to the pump 16 in such a manner that the oil under pressure utilized to actuate the pressure plate of pump 16 is vented upon opening of the valve 100. Accordingly, when the plunger 106 is moved by high pressure, oil flows through the line 120 to ports 118 and 114 and through the line 112 to the reservoir 30. At this condition the pump 16 ceases to discharge fluid and the output pressure falls to a predetermined normal level. Since only pumps 10 and 14 are operating, the available fluid for operating the motor 76 has been reduced with the result that the motor vehicle speed is also reduced in proportion.

If the torque requirement is greater at the motor than the intermediate speed range, the pressure transmitted to the port 104 would actuate the plunger 122 and valve 100 to connect port 116 to port 124. A compressed spring 126 provides a force opposite in direction to the hydraulic pressure and this force is adjustable by positioning the adjusting screw 128.

A manual override control valve section is located between the port 124 and the line 130 connecting the valve to the pump 14. A plunger 132 having two operating positions allows the operation of the automatic control in one position and in the other position closes the communication between port 124 and the line 130, thus preventing the unloading of pump 14. When the automatic control is in effect, pump 14 may be unloaded and the single pump 10 utilized to furnish the pressure required for driving the vehicle. Accordingly, the speed of the vehicle would be one-third that available when all of the pumps 10, 14 and 16 were loaded.

The check valve 56 prevents the return flow of hydraulic fluid through pump 16 when the pumps 10 and 14 are delivering fluid under pressure and pump 16 is unloaded. A check valve 64 in line 62 also prevents short-circuiting of the hydraulic fluid when pump 10 is the only pump operating to supply pressure to the motor 76. When the vehicle is rolling downgrade and the motor 76 has a displacement greater than the supply pump capacity, the deficiency of hydraulic fluid is made up through a valve 136 which automatically connects the reservoir 30 to the motor inlet under such condition.

When it is desired to reverse the vehicle the plunger 86 in valve 42 is moved to the right as shown in Fig. 4 by the operation of the pedal 88. In this condition the oil supplied through line 40 is connected through port 44, passage 70 to port 74. Due to the position of the plunger port 74 is open to port 84 and the oil is allowed to flow through the valve to line 82 which becomes the pressure supply line to the motor 76. The line 78 func-

tions as the return line from the motor 76 to port 80 in the valve 42. Port 80 is in communication with port 81. Due to the position of the plunger 86, port 81 being connected to passage 96 and return line 58 which conducts the oil from the motor 76 to the reservoir 30. The relief valve 68 protects the system from excessively high pressures by having a setting which is slightly lower than the maximum pressure which the engine could develop when only pump 10 is operating. When the plunger 86 is positioned to reverse the vehicle while the vehicle is still in motion, the relief valve 68 bypasses oil from line 70 to line 58 thus preventing stalling of the engine. With this feature the controls may be set for a change in direction of travel without first bringing the vehicle to a stop and further, the system may be used for breaking a vehicle by actuating the plunger 86 into a creeping position opposite to the motion of the truck.

In Fig. 4 the automatic control valve is shown with the plungers 106 and 122 in their actuated position which would result from high pump discharge pressures in line 40. In each instance the pressure on the pressure plates of the pumps is relieved to the reservoir 30 thus unloading the pumps to which they are connected. Line 120 connected to pump 16 communicates with port 118, port 114 and return line 112 to the reservoir 30. Pump 14 is connected through line 130 and manual control valve 132, port 124, port 116 to return line 112. With the valves set as shown in Fig. 4 the vehicle would be in reverse and in the low speed range since pumps 14 and 16 would not be supplying fluid for the operation of the motor 76 since these pumps are relieved as above described. Reverse flow of oil from pump 10 through the pumps 14 and 16 to the reservoir 30 is prevented by land 90 and check valve 64 in valve 42. Under these conditions fluid applied to valve 42 at port 44 is discharged from the port 44 into lines 70 and 72 into port 74 and therefrom flows into port 84 along a reduced portion of plunger 86 to be applied through line 82 to pump 76 in a reverse sense. In port 44, lands 90 and 92 restrain the fluid from flow along plunger 86 and in ports 74 and 84 lands 94 and 98 restrain the fluid from flow along the plunger except as to the space therebetween. The hydraulic system controls are connected to pump 14 by line 46. When the automatic speed control unloads both pumps 14 and 16 there is no pressure available in line 46 for actuation of the hydraulic system. When it is desired to operate the lifting mechanism which is controlled by the 4-way valve 22 the plunger 132 is served to close the passage of oil through line 130 from pump 14 which causes the pressure to build up on the pressure plates of pump 14. Under this condition pump 14 immediately begins to function and high pressure is available at the 4-way control valve 22. When the control valve 22 is set for actuating the lift cylinder 18 oil flows from line 46 through line 23 to the cylinder 18 and returns through line 25 to the hydraulic system reservoir 20. When the hydraulic system reservoir 20 overflows the oil is discharged through the filter 26 to the drive reservoir 30 thus cleaning the system fluid. Accordingly, the manual override control prevents the unloading of pump 14 thereby providing a continuous supply of fluid under pressure for actuation of the lift mechanism by the cylinder 18.

During the operation of a lift truck it is desirable to be able to move the vehicle slowly and for short distances which is commonly known as creeping. To accomplish the creeping operation it is necessary to slip a clutch in the conventional vehicle in order to get a small amount of movement and still maintain a reasonable degree of engine speed. Under some conditions a large part of the operation of the lift truck occurs in this manner with the result that maintenance of the clutch assemblies becomes an expensive item. The control system disclosed herein is particularly adapted for the creeping operation since the control is obtained by regulating the flow of fluid to the motor actuating the vehicle.

To obtain maximum efficiency of the truck it is necessary also to be able to operate the elevating mechanism while the vehicle is creeping. Since the controls and supply pumps are independent for the vehicle drive and the elevating mechanism actuators, the system is particularly flexible to accomplish this purpose. In Fig. 5 the plunger 86 of control valve 42 is shown in the position corresponding to the creep condition. This position of the plunger 86 is obtained by moving the control pedal 88 away from the neutral position toward the forward drive. The oil discharged from pump 10 through line 40 enters the port 44 which is connected through passages 70 and 72 to port 74. The port 74 is now open to port 80 which supplies oil through line 78 to the motor 76. Since the land 92 only partially restricts the flow of oil from port 44 to the return passage 96, the quantity of oil available to drive the motor 76 is reduced accordingly. Obviously by positioning the plunger 86 any part of the oil discharged from pump 10 may be directed to the motor 76 or to the return passage 96 and return line 58 to the reservoir 30.

In order to maintain operation of pump 14 for the supply of oil to the elevating apparatus during the creeping operation the automatic control valve may be prepositioned in such a manner that plunger 132 prevents the flow of oil through line 130 from the pressure plates of the pump 14. Under this condition pump 14 will continuously supply oil under pressure through line 46 to the 4-way valve 22 where it is available for actuation of hydraulic mechanism 18. The oil from pump 14 which passes through the high pressure carryover and line 48 will enter port 52 and be discharged through passage 96 and line 58 to the reservoir 30. Operation of the pump 14, therefore, does not affect the creep control characteristics since no part of the discharge therefrom is directed to the motor 76.

An alternate form of control for the pump 14 is shown in Fig. 4. A plug 134 which is inserted in the valve 100 closes the conduit 130, thus preventing the unloading of pump 14. A pipe, plug or cap 135 prevents the leakage of hydraulic fluid from the valve 100 after the plug 134 has been inserted. This plug 134 would be used when the type of operation of the vehicle would make it undesirable for pump 14 to unload. The operator would not have the means of control for changing the means of operation for pump 14. First speed of the vehicle would be obtained as previously described, however, the second speed would be obtained by the full forward or reverse position of pedal 88. Third speed would still be automatic in that pump 16 would still unload at the same pressure whether or not the plug is in position.

Although the foregoing is necessarily of a detailed character, in order that the invention may be completely set forth, it is to be understood that the specific terminology is not intended to be restrictive or confining, and that various rearrangements of parts and modifications of detail may be resorted to without departing from the scope or spirit of the invention as herein claimed.

What is claimed is:

1. A hydraulic pump-motor system comprising a first, second and third pump, a power unit to drive the pumps in unison, said first pump having an inlet and an outlet, said second pump having an inlet, an outlet and an unloading conduit, said third pump having an inlet, an outlet and an unloading conduit, a reservoir, a conduit connecting each of said pump inlets to the reservoir, pressure sensing valve means comprising first and second pressure responsive valves having pressure sensing chambers, said first pressure responsive valve being connected to the unloading conduit of said second pump, said second pressure responsive valve being connected to the unloading conduit of said third pump, a conduit connecting the outlet of said first pump to the pressure sensing chamber

of said first and second pressure responsive valves, a rotary hydraulic motor having an inlet and an outlet, a hydraulic lift cylinder, a four-way valve, a conduit connecting the outlet of said second pump to said four-way valve, a conduit connected to said third pump outlet, a check valve in said last-named conduit, a conduit leading from said four-way valve connected to the conduit having the check valve therein, a manually operable control valve, said control valve comprising a valve body having an axial bore therein, a plunger mounted for reciprocation in said bore, said valve body having a plurality of axially spaced annular ports opening into said bore, said plunger having a plurality of spaced lands to open and close adjacent ports with respect to fluid flow through the bore, a first port having conduit means connected to the outlet of said first pump, a second port adjacent said first port, means connecting the combined output from said conduit leading from said four-way valve and the output of said third pump to said second port, a third port connected to the inlet of said motor, a fourth port connected to the outlet of said motor, a fifth port between said third and fourth ports, said body having a passageway leading from said first and second ports to said fifth port and a land on the plunger in said fifth port to direct the fluid flow from said pumps into said fourth port to drive the motor in one direction or into said third port to drive the motor in an opposite direction, and conduit means connecting said third and fourth ports to the reservoir depending upon the directional drive to the motor.

2. A pump-motor system for hydraulically transmitting power from a plurality of pumps driven in unison to a rotary hydraulic motor and to a lift cylinder comprising a first pump having an inlet and an outlet, a pressure responsive valve, a manually operated control valve, said first pump outlet having two branches, a first branch leading to said pressure responsive valve and the other branch leading to said manually operated control valve, a second pump having an inlet and an outlet, said outlet being connected to a four-way valve for said lift cylinder, said second pump having an unloading outlet leading to said pressure responsive valve, a third pump having an outlet and an inlet, said last-named outlet having two branches, one branch leading directly to said manually operable control valve, said last-named branch having a check valve therein, and the other branch connecting said four-way valve to said one branch at the closed side of said check valve, a reservoir, said third pump having an unloading conduit leading to said pressure responsive valve, conduit means connecting said pressure responsive valve and said reservoir, conduit means connecting the inlets of said pumps and said reservoir, said rotary motor having an inlet and an outlet, said manually operable control valve comprising a valve body having an axial bore therein, a plunger reciprocable in said bore, said bore having a plurality of axially spaced ports therein surrounding the plunger, said plunger having a plurality of spaced lands adapted to open and close said bore with respect to adjacent ports, a first port connected to the said other branch of said first pump, a second port, said valve body having an open passageway therein leading from said first port to said second port, said second port being connected to the inlet of said rotary motor, a third port adjacent said second port, said third port being connected to the outlet of said motor, a fourth port adjacent said first port, said one branch of said third pump being connected to said fourth port, a passageway leading from said fourth port to said second port whereby under low torque at said motor said second and third pumps supplement the drive from the first pump, a land on the plunger between said second and third ports whereby fluid flow from the first and fourth ports is directed into either said second port or said third port to control the direction of rotation of said rotary motor, and conduit means connecting said second and third ports to the reservoir depending upon the direction of rotation of said motor.

3. A hydraulic pump-motor system for transmitting power from a plurality of pumps driven in unison to a motor, a reservoir, a first, second and third pump, each of said pumps having an inlet and an outlet, pressure responsive valve means for unloading said second and third pumps to the reservoir, said pressure responsive valve means including a pressure sensing chamber connected to the outlet of said first pump whereby said second and third pumps are unloaded in response to a predetermined high pressure in said sensing chamber and said pumps are loaded in response to a predetermined low pressure in said sensing chamber, manually operated control valve means to direct a portion of the output from said first pump to said motor and by-pass the balance of said output from said first pump to said reservoir, comprising a valve body having an axial bore, a plunger mounted for reciprocation in said bore, a plurality of spaced annular ports in said bore surrounding said plunger, a plurality of spaced lands on the plunger to open and close the bore with respect to adjacent ports, a first port connected to the outlet of said first pump, a second port adjacent the first port and connected to the outlet of said second and third pumps, a fourth port adjacent the second port and an opening thereinto, conduit means connecting said fourth port to the reservoir, a third port connected to the inlet of said motor, a fifth port adjacent said third port, said fifth port being connected to said motor outlet and to the reservoir, said valve body having a passageway therein connecting said first, second and third ports, said passageway having a branch leading to said reservoir, a relief valve in said branch, a land on said plunger between said third and fifth ports to direct flow from said first and second ports to said motor inlet and a land on said plunger between said first and second ports to direct a part of the flow from said first port to said second port and thence to the fourth port and the reservoir, and a land on the plunger between the second and fourth ports adapted to be moved with the plunger to close the fourth port with respect to the second port and thereby direct the output of all three pumps to the motor.

References Cited in the file of this patent

UNITED STATES PATENTS

1,716,881	Francis	June 11, 1929
1,982,711	Vickers	Dec. 4, 1934
2,074,618	Roeder	Mar. 23, 1937
2,276,895	Vosseler et al.	Mar. 17, 1942
2,320,600	Howell	June 1, 1943
2,507,357	Stoner	May 9, 1950
2,553,458	Jordan	May 15, 1951
2,598,865	Turner	June 3, 1952