

[54] **HYDRAULIC CIRCUITRY FOR AN EXCAVATOR**

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Related U.S. Application Data

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[58] Field of Search **60/421, 428, 484, 486; 91/414; 214/138 R**

[56] **References Cited**

UNITED STATES PATENTS

3,146,593 9/1964 Stacey..... 60/421

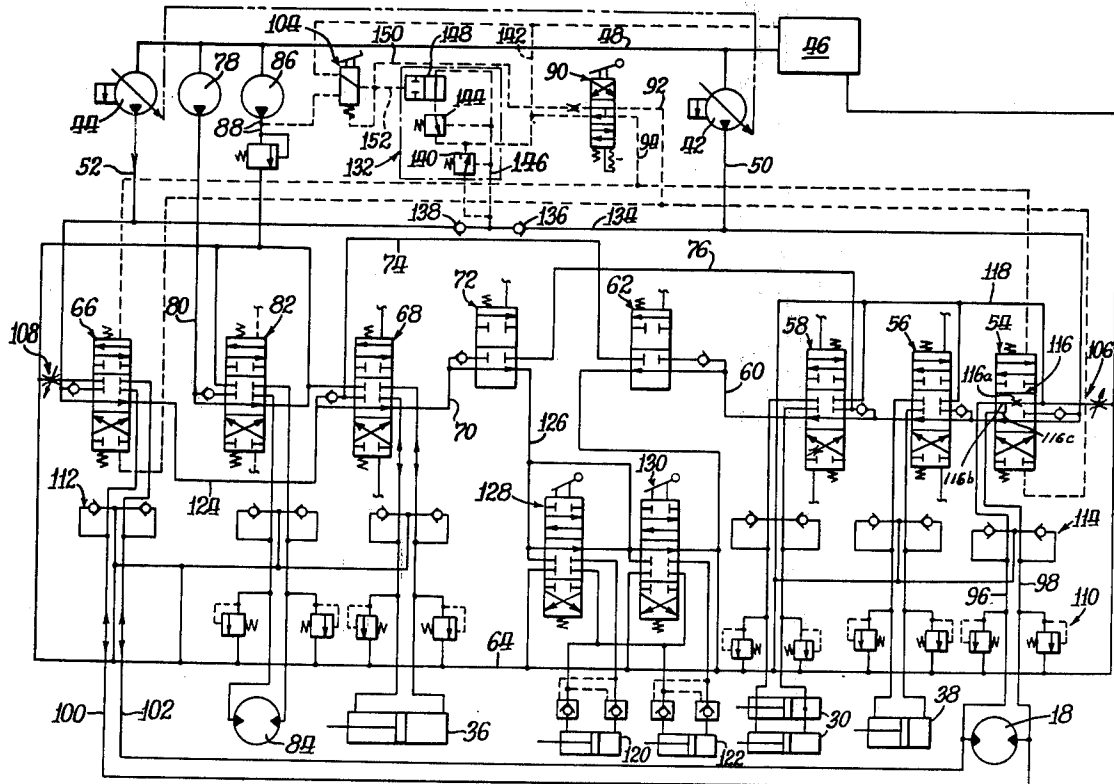
3,208,221 9/1965 Schuetz..... 91/414 X
3,686,862 8/1972 Grider et al. 91/414 X
3,720,059 3/1973 Schurawski et al. 60/421

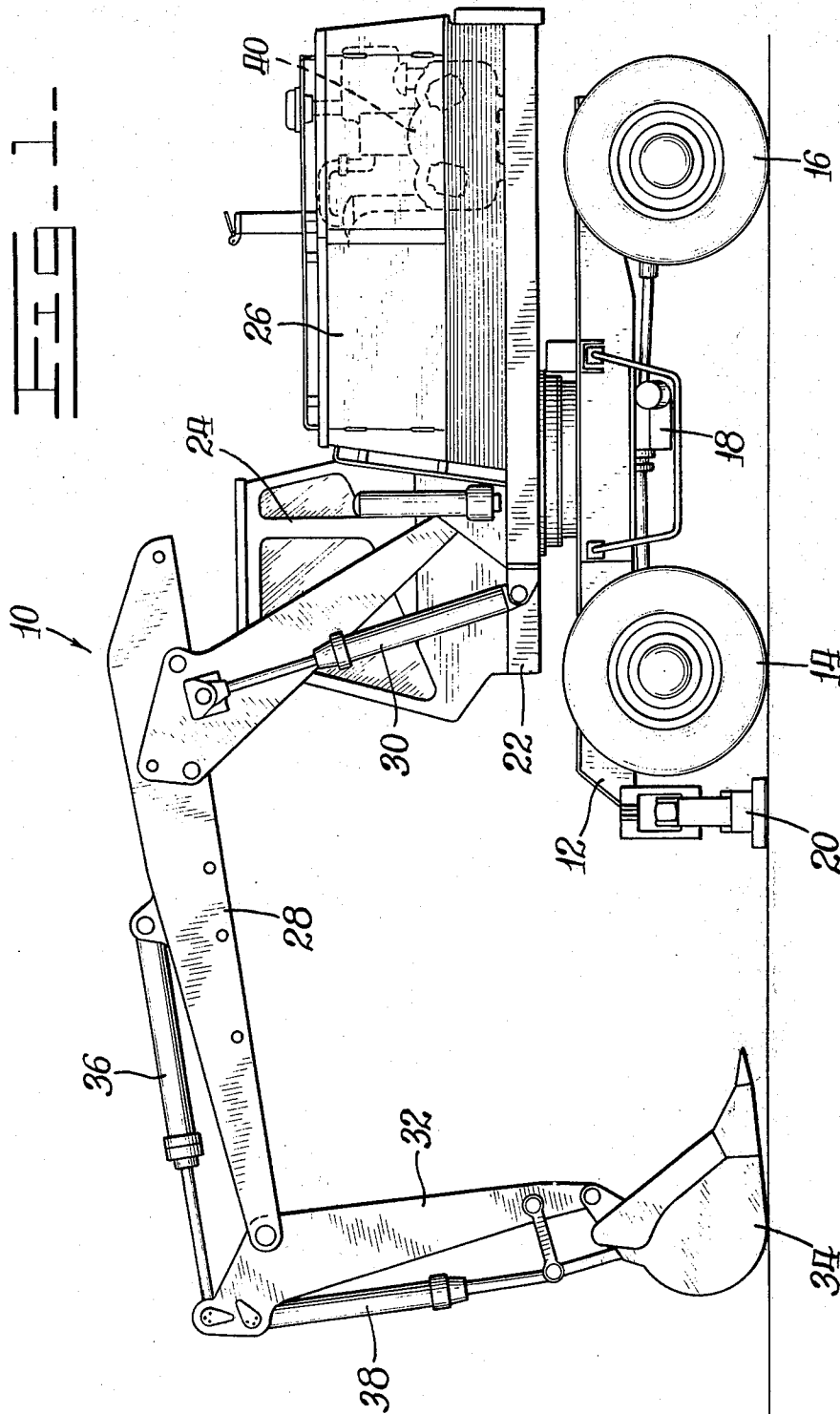
Primary Examiner—Edgar W. Geoghegan
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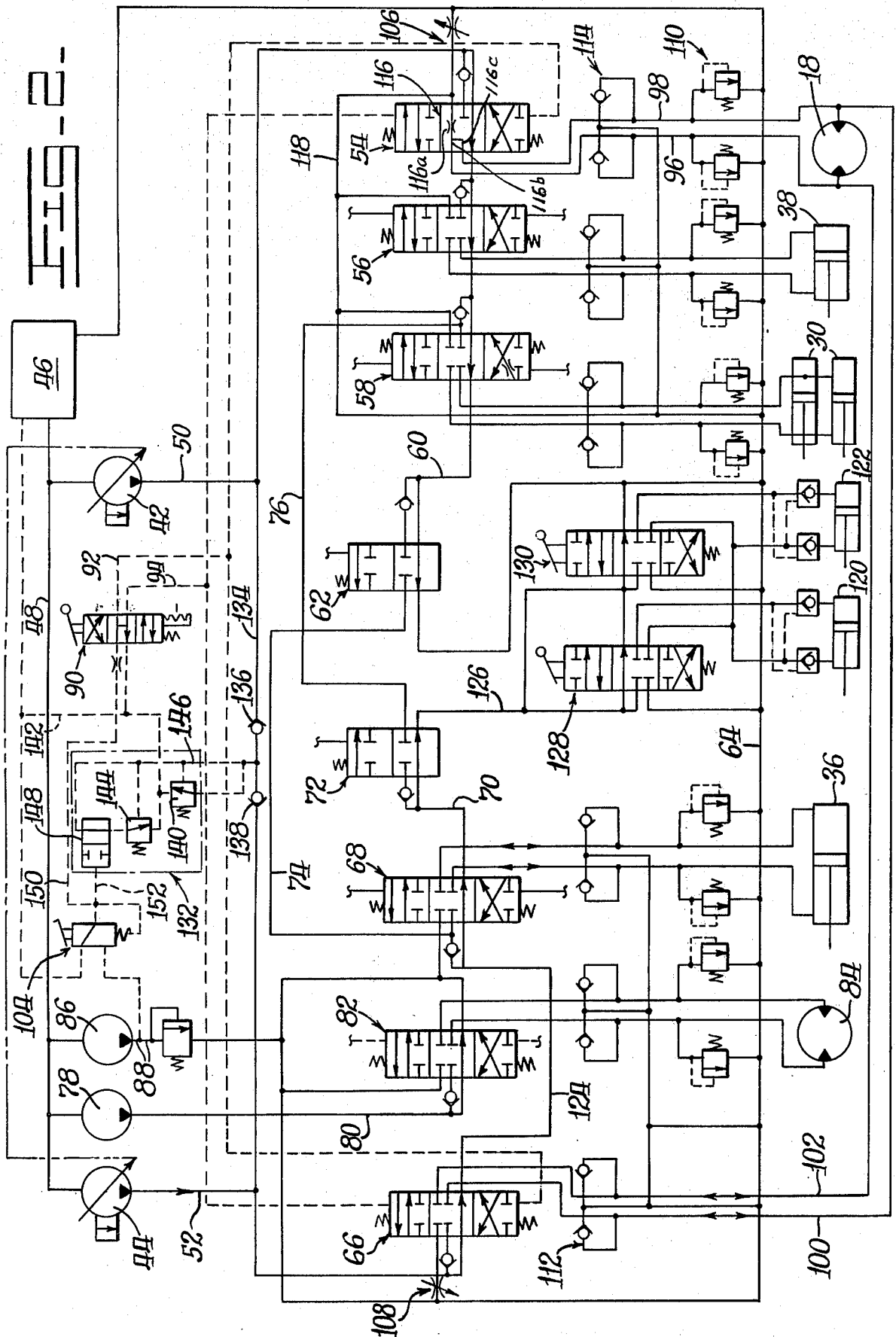
[57] **ABSTRACT**

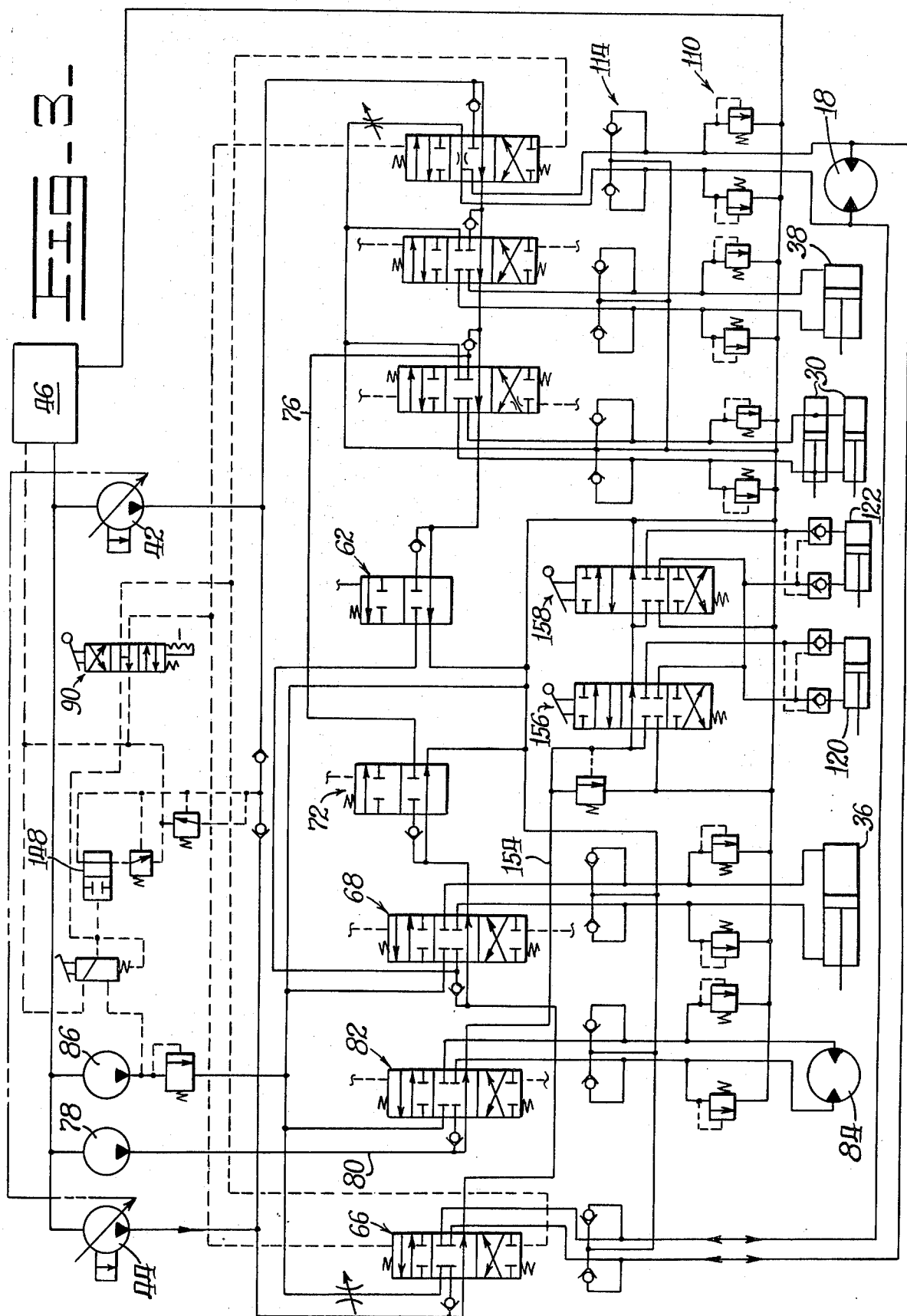
A hydraulic circuit is provided for powering a wheel-type excavator with a plurality of pumps supplying pressurized fluid for the circuit. The circuit is arranged so that fluid is provided from a plurality of pumps in a sequence manner to permit maximum utilization of pump volume for translation of the machine. A unique sequence of the distributor valves also provides positive independent control of the various implement motors of the machine, with means provided so that fluid may also be combined to operate selected implement motors for faster operation and maximum efficiency of the machine.

16 Claims, 3 Drawing Figures









HYDRAULIC CIRCUITRY FOR AN EXCAVATOR

REFERENCE TO RELATED APPLICATION

The present invention is a continuation-in-part of our copending application Ser. No. 207,027 filed Dec. 13, 1971, entitled Hydraulic Circuitry for an Excavator.

BACKGROUND OF THE INVENTION

The present invention is directed to a hydraulic power and control system and pertains more particularly to a hydraulic system for powering the translation motor and the implement motors of a hydraulic excavator.

Hydraulically operated material-handling machines such as backhoes and hydraulic excavators have been known for some time. Such machines are extremely versatile and hence widely used in building and construction industries.

These machines normally utilize an internal combustion engine to drive one or more pumps which supply pressurized fluid to operate separate hydraulic motors for propelling the machine as well as operating the material-handling implements. Considerable room exists for improvement in the efficiency and safety of existing prior art machines.

The prior art is exemplified by such U.S. Pat. Nos. as: 3,720,059 issued Mar. 3, 1973 to Schurawski et al; 3,428,647 issued Nov. 18, 1969 to Gerber et al; 3,476,274 issued Nov. 4, 1969 to W. J. Witwer; 3,208,221 issued Sept. 28, 1965 to J. R. Schuetz; 3,172,552 issued Mar. 9, 1965 to R. Metailler; and 2,768,499 issued Oct. 30, 1956 to J. S. Pilch.

A number of foreign patents are as follows;

British: No. 1,044,851 dated Oct. 5, 1966 to Auxitra; 1,057,752 dated Feb. 8, 1967 to Ornstein-Koppel, and No. 1,108,232 dated Apr. 3, 1968 to Tractem.

French: No. 1,259 to De Venissieux, dated Mar. 13, 1961; No. 1,301,228 to Weitz, dated July 2, 1962; No. 1,343,512 to Ornstein-Koppel dated Oct. 14, 1963, and No. 1,330,457 to Tevs, dated May 13, 1903.

The problem of efficiency with such machines is the result of the constant shifting between the various implement motors, the need for simultaneous positive operation of a number of motors, and the need for repetitive high-speed operation of one or more motors. Such requirements impose unusual requirements on the construction and arrangement of valves and circuits.

In order to obtain positive displacement of a motor, it must have a fluid supply unaffected by the operation of another motor. This, ideally, means a separate pump for each motor. Such an arrangement, however, could be extremely expensive and would result in a substantial waste of pump capacity during times when some of the motors are not being used. This type arrangement having separate pumps is disclosed in U.S. Pat. No. 3,172,552 issued Mar. 9, 1965 to R. Metailler.

In order to make maximum use of pump capacity, it is desirable to combine pump flow to motors which can use the increased capacity, while other motors are idle. In addition to making maximum utilization of the pump capacity, this type arrangement results in faster operation of certain machine functions. Such arrangements not only increase the efficient use of the pump capacities, but also speed operation of the particular machine function and thereby increase the efficiency of overall material-handling operation.

When the vehicle is propelled by a hydraulic motor or motors, the efficient use of the fluid capacity of the machine becomes quite important. When the vehicle is mounted on crawler tracks, low-speed power and maneuverability is of prime importance. When the vehicle is mounted on rubber tires, low-speed control as well as high-velocity translation are of prime importance. These different systems require different means of controlling the fluid to the translation motors.

One proposal to increase use of pump flow is to connect the distributor valve in parallel. This arrangement, however, has the disadvantage of lack of positive control when two motors requiring different pressures are simultaneously operated. If, for example, one motor meets increased resistance, it will slow down or stop, while the other motor with less resistance will speed up. Such a parallel arrangement is disclosed in U.S. Pat. No. 3,208,221 issued Sept. 28, 1965 to Schuetz; and U.S. Pat. No. 3,720,059 issued Mar. 3, 1973 to Schurawski et al.

Other proposals have included series-connected distributors with the pumps combined upstream of a series of distributors. This arrangement permits use of combined flow at a particular motor but cuts off flow to downstream motors.

Still another proposal provides separate pumps and distributors with means to divert flow from one pump to combine with the flow from another for operation of one or more motors. Such an arrangement leaves a portion of the hydraulic circuit inoperative, and adds another control lever which the operator must consciously manipulate in order to use the maximum power and speed of the machine.

SUMMARY OF THE INVENTION

It is the primary object of the present invention to overcome the above problems of the prior art.

Another object of the present invention is to provide a more efficient and versatile hydraulic control system for material-handling machines and the like.

A further object of the present invention is to provide a hydraulic control system that is arranged for more efficient utilization of available fluid power than has heretofore been known.

Still another object of the present invention is to provide a hydraulic system for powering hydraulic excavators that automatically obtains the maximum efficient use of the hydraulic power for propelling the vehicle and for movements of the material-handling apparatus from the power available.

In accordance with the present invention, a hydraulic power and control system for an excavator comprises a plurality of pumps supplying fluid independently to separate series of distributor valves. The distributors are arranged to supply fluid from the pumps to selected fluid motors for powering the translation motor and the implements of the apparatus. The distributors and motors are arranged in an order of priority with means for combining pump flow for operation of selected implement motors and in sequence for operation of the translation motor.

BRIEF DESCRIPTION OF THE DRAWINGS

The above and other objects and advantages of the present invention will become apparent to those of skill in the art from the following description when read in conjunction with the accompanying drawings wherein:

FIG. 1 is an elevational view of a hydraulic excavator incorporating the present invention;

FIG. 2 is a schematic layout of a circuit incorporating a preferred embodiment of the present invention;

FIG. 3 is a schematic layout of an alternate embodiment of the present invention.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENT

Referring now to the drawings, with particular reference to FIG. 1, there is illustrated a wheel-type hydraulic excavator generally designated by the numeral 10 and comprising a body or undercarriage 12 supported on suitable rotatable front and rear wheels 14 and 16 respectively driven by a suitable hydraulic translation motor 18. Outriggers 20 are secured to the front of the frame and extendable by suitable hydraulic motors (not shown) into engagement with the ground to stabilize the vehicle during operation. A suitable rotatable platform 22 is supported from 360° rotation from the body or undercarriage 12 and includes an operator compartment 24 and an engine compartment 26. A boom 28 is pivotally mounted on the rotatable platform 22 and manipulated about its pivotal point by means of a pair of hydraulic motors 30. An arm or jib 32 is pivotally carried by the boom 28 and pivotally carries on the end thereof a bucket 34 with motors 36 and 38 operatively connected for manipulating the arm and bucket respectively. Fluid for manipulating and controlling these motors is supplied by a circuit to be described below and pressurized by means of pumps driven by an engine 40.

Referring now to FIG. 2, there is illustrated a preferred embodiment of the present invention comprising a unique arrangement of distributor valves and motor connections which provide for an optimum utilization of system power and fluid flow. In accordance with this preferred arrangement, the circuit comprises a pair of large variable displacement pumps 42 and 44 controlled by suitable horsepower controlling means (not shown) which is responsive to govern the horsepower output of the pumps in conformance with the output of the engine. The pumps 42 and 44 draw fluid from a tank or reservoir 46 by way of line or conduit 48 and supply the fluid under pressure by way of supply lines 50 or conduits 52 to respective first and second series of distributor valves for directing fluid to and controlling the various motors for carrying out the functions of the machine.

The first variable displacement pump 42 supplies fluid to the first series of valves comprising a first distributor valve 54 for directing fluid to the translation motor 18, a second distributor valve 56 for directing fluid to bucket motor 38, and a third distributor valve 58 for directing fluid to the boom motors 30. The fluid, after flowing through these open-centered valves without diversion for operation of a motor, passes along return passage 60 through a cross-over valve 62 and to return passage or conduit 64 which leads to tank 46. This arrangement of distributors is an interrupted series arrangement in that actuation of a distributor interrupts the supply of fluid to the downstream distributors. Thus, an upstream distributor is a preferred distributor taking precedence over the downstream distributors.

The second variable displacement pump 44 supplies fluid along conduit 52 to a first distributor valve 66 of the second series, operative for directing fluid from that pump to translation motor 18, and then to a sec-

ond distributor valve 68 for directing fluid to the jib or stick motor 36. After flowing through the above-described open-centered control valves, the fluid flows along a return passageway or conduit 70 and through the open center of a second cross-over valve 72 and into return passage 64. Means are provided comprising the cross-over valves 62 and 72 and conduits 74, 76 for combining the fluid when not used from either of the above series of valves with the fluid to the last distributor in each of the other of the above series. This combining function is accomplished by an interconnection of the valves so that the cross-over valves operate with the respective distributor to which the combined fluid is directed.

For example, cross-over valve 72 shifts automatically with distributor valve 58, so that fluid in line 70 is directed across valve 72 and by way of line 76 to the inlet side of valve 58. Thus fluid from both pumps 42 and 44 may be supplied to distributor 58 which is operative to direct the fluid to the boom lift cylinders 30.

The cross-over valve 62 is operative to shift with distributor valve 68, so that fluid from line 60 is directed across valve 62 and by way of line 74 to the inlet side of valve 68. The valve 72 may be made to shift as valve 68 is shifted in one or both directions.

The circuit includes a third main supply pump 78 which is a lower-volume pump than the previous pumps and may, for example, be a suitable fixed displacement pump for supplying fluid by way of a supply line 80 to distributor 82 for controlling the supply of fluid to the slewing or platform rotation motor 84. This separate slewing motor and circuit for the platform adds to the versatility as well as safety of the machine. The versatility of the machine is enhanced by the fact that fluid is available for rotating the platform into position at the same time that the vehicle may be in translation along the ground with the drive motor operating at full speed from fluid supplied by both pumps 42 and 44. The safety of the machine is enhanced by the fact that attempts to slew or swing the platform while the vehicle is in translation cannot rob the drive motor of fluid necessary to prevent the vehicle being thrown into a sudden uncontrolled change of speed.

A pilot supply pump 86 supplies fluid for a pilot control circuit by way of a line 88 for operation of pilot valves for controlling or operating the main control valves as well as fluid for actuating and releasing brakes for the slew and for the wheels.

The translation drive motor is powered by the separate variable displacement pumps 42 and 44 in sequence for translatory movement of the vehicle. More particularly, the first pump 42 supplies fluid to the motor for low speed range up to the full capacity of that pump, then the second pump 44 is automatically brought in for higher speed range. Translatory movement of the vehicle is accomplished by movement of pilot-operated directional control valve 90 which directs fluid along either one of pilot lines 92 or 94 for shifting of motor control distributor valves 54 and 66. Valve 54 is responsive to shift first for directing fluid along motor control lines 96 or 98 to motor 18. Shifting of valve 66 for directing fluid along either one of motor control lines 100 or 102 to drive motor 18 occurs after valve 54 has been almost fully shifted to full open. The direction of drive of the vehicle will depend on which direction the drive control valves 54 and 66 are shifted. The control valves 54 and 66 are pilot-operated and are shifted in either direction by means of a pilot directional valve 90 and the degree of shifting of the valves

is controlled by a pilot speed control valve 104. Pilot valve 90 controls direction of the pilot signal and pilot valve 104 controls the strength of the signal.

The drive system of the present circuit is arranged to provide a low range and a high range operative from the same control such as a foot pedal connected to operate the speed control valve 104. This dual range is provided by an arrangement wherein the output from a first pump is directed to the hydraulic drive motor for the low speed range and the output from a second pump is added or combined with that from the first pump for the high speed range.

This speed range is obtained by providing one of valves 54 and 66 with a fairly low pilot pressure shifting range and providing the other valve with a higher pilot pressure shifting range. In the preferred embodiment, valve 54 is set to operate over the range of 35 to 155 psi. That is, the valve begins shifting at 35 psi and shifts over its range of movement up to 155 psi, the valve being fully open at the 155 psi. This range of pilot pressures is controlled by the valve 104 which is operative to modulate the pressure over at least those ranges.

The pilot pressure for shifting of valve 66 begins at 145 psi and ends at approximately 218 psi. In other words, this valve 66 shifts over the range of 145 to 218 psi. This provides an overlap of approximately 10 psi with the valve 54 to ensure a smooth transition from the range of single pump 42 to the range of dual pump 44. This shifting of the valves occurs automatically as the foot pedal operating valve 104 is moved through its range of movement.

Each of the distributor valves 54 and 56 has associated therewith pressure-compensating flow-control valves 106 and 108 respectively. These valves are provided in the exhaust line of each of the valves and operate to prevent the motor 18 from overspeeding. This overspeed is prevented by restricting the flow back to the return line 64, which returns fluid back to the tank. The problem of overspeed could occur when the excavator vehicle is traveling down an incline or hill and the inertial force of the machine tends to drive the propulsion motor faster than it would normally go.

Each of the motor control supply lines is also protected by relief valve 110 and the motor itself is protected by a pair of anticavitation valves 112 and 114.

The first translation motor distributor valve 54 is also provided with restricted open communication means indicated generally at 116 in the form, for example, of a restricted passageway 116a or the like communicating the motor control lines 96 and 98 with a return line or passageway 118 which is in communication with the return line 64. This communication means 116 is operative when the spool 54 is in the neutral position to communicate both lines 96 and 98 by way of suitable passages 116b and 116c with the restriction 116a and return line 64. The function of this restricted communication means 116 is of course to prevent locking of the motor 18 when the valve 54 is returned to neutral position. Under normal conditions, for example, the pressure relief valves 110 are set high enough to cause an immediate braking and locking of the motor 18 when the valve 54 is shifted to the neutral position as shown. Thus, if the vehicle is in motion and the valve 54 is returned suddenly to the neutral position, the wheels will lock and the vehicle will come to a sudden stop. With the present arrangement, however, fluid is permitted to pass to the return lines and the reservoir and thus the

vehicle will not come to an immediate and sudden stop when the valve is shifted into neutral position.

The outriggers 20 shown in FIG. 1 are operated by a pair of hydraulic motors 120 and 122 which are supplied by fluid from pump 44 downstream of the distributor valve 68 which supplies fluid to the jib or stick. As can be seen from FIG. 1, fluid passing through valve 66 passes along a passageway or conduit means 124 by-passing distributor valve 82 and entering the inlet to supply fluid for distributor valve 68. When valve 68 is in neutral position, the fluid flows across the distributor valve 68 and along the passageway 70, diverter valve 72 and if that valve is in neutral as shown, the fluid passes along passageway or conduit means 126, which branches to supply fluid simultaneously to a pair of distributor valves 128 and 130, which are respectively operative to supply fluid for operation of the outrigger motors 120 and 122. The outrigger valves and motors are of course downstream of the distributor valves 66 and 68 and thus do not affect the operation of the motors 18 and 36 from the pump 44.

A valve indicated generally at 132 provides a dual pressure relief setting for pumps 42 and 44.

As can be seen from FIG. 2, the output lines 50 and 52 from pumps 42 and 44 are communicated by way of suitable conduit means 134 and a pair of check valves 136 and 138 with the dual pressure relief valve 132. The dual pressure relief valve assembly 132 includes a first relief valve 140 which is operative to unload at pressures in excess of 4500 psi and communicate with the conduit means 134 and the return line 142. A second relief valve 144 is in selective communication by way of line 146 with this communication being controlled by means of a selectively actuable valve 148 with the output line 134. This valve 144 is responsive to a pressure level lower than that of the first valve means 140 and on the order of preferably approximately 4000 psi. The valve 144 is in selective communication by means of a selector valve 148 which is operative in response to operation of the speed control valve 104. As will be appreciated, the pilot control signal from valve 104 is communicated along conduit means 150 and communicates said signal by means of a line 152 with the valve 148 to operate same by shifting it to the locking position to the right of the position shown.

When the implements of the vehicle are being actuated and the vehicle itself is not being driven, low pressure valve 144 is in communication, as shown, to provide a 4000 psi setting for the system. During travel of the vehicle when speed control valve 104 is actuated, valve 148 is shifted to the right to the blocking position, thus blocking flow to relief valve 144 and rendering it inoperative. However, relief valve 140 is still in effect and by being directly connected to the output line 134 of pumps 42 and 44 and to tank permits unloading of the drive motor 18 only at pressures in excess of 4500 psi.

Turning now to FIG. 3, there is illustrated an alternate embodiment wherein identical parts and identical arrangements are identified by the same numeral. The primary distinction of this embodiment over that of the FIG. 2 embodiment is that the outrigger motors 120 and 122 are supplied with fluid from the slewing pump 78. In this embodiment, fluid supplied to valve 82 and not diverted thereby to the operation of motor 84 passes therethrough and along a passage or conduit means 154 where it is supplied to a pair of distributor valves 156 and 158 which are operative respectively to

supply fluid to outrigger motors **120** and **122**. This arrangement whereby the outriggers are last in line for the fluid does not create any problems since the outriggers are normally extended into place before any operation of the implements is begun. Once the outriggers are extended, they are left in place as long as operation of the machine takes place.

While specific embodiments have been utilized to illustrate the invention, it is to be understood that the invention is to be limited only by the scope of the appended claims.

We claim:

1. A hydraulic circuit for a hydraulic excavator comprising:

- a first and a second variable displacement pump for supplying pressurized fluid to said circuit;
- a first series of series-connected distributor valves operatively connected for directing fluid from said first pump to a translation motor and to a first pair of implement control motors;
- a second series of series-connected distributor valves operatively connected for directing fluid from said second pump to said translation motor and at least another implement control motor;
- pilot control means for shifting said distributor valves for said translation motor in sequence;
- said pilot control means including a source of pressurized pilot control fluid;
- drive control means including pilot valve means operative to direct said pilot fluid for shifting said distributor valves controlling said translation motor and operative to vary the pressure of said pilot pressure over a wide range of pressures; and,
- the first of said distributor valves being operative to shift progressively over a first range of pilot fluid pressures and the second of said distributor valves being operative to progressively shift over a second range of pilot fluid pressures so that the first pump is operative to supply fluid for low speed operation and fluid from said second pump is added to said first pump fluid for high speed operation.

2. The hydraulic system of claim 1 wherein said drive control means comprises a first distributor valve for directing fluid from a first of said pumps to said translation motor;

- a second distributor valve operative to direct fluid from a second of said pumps to said translation motor;

pilot control means for operating said distributor valves;

said pilot control means including a valve operative to provide a variable pilot pressure signal to said distributor valves;

said first distributor valve being responsive to a first range of pressures of said pilot signal; and

said second distributor valve being responsive to a second range of pressures of said pilot signal.

3. The hydraulic system of claim 2 wherein said drive control means includes a directional control valve for selecting the direction of shifting of said distributor valves.

4. The hydraulic system of claim 3 wherein said first distributor valve includes means for restrictively communicating fluid from both sides of said translation motor to the sump when said distributor valve is in a neutral position.

5. The hydraulic circuit of claim 1 including dual-pressure relief valve means having a first relief pressure

when said translation motor is in operation, and a second relief pressure when said implement motors are in operation.

6. The hydraulic circuit of claim 4 wherein said circuit includes diverter valve means at the end of each of said series of distributors operative to direct fluid from downstream of each of said series to the last distributor in the other of said series.

7. The hydraulic circuit of claim 6 wherein said circuit includes a pair of distributor valves downstream from said diverter valves for directing fluid for operation of a pair of outrigger jacks.

8. The hydraulic circuit of claim 6 wherein said circuit includes a third pump and a distributor for separately supplying fluid for operation of a slewing motor for said excavator, and a pair of distributor valves downstream of said slewing distributor for directing fluid to a pair of outrigger jacks.

9. The hydraulic circuit of claim 6 wherein said circuit includes a dual relief valve assembly having a first relief valve operative when said translation motor is in operation, and a second relief valve operative when said translation motor is not in operation.

10. The hydraulic circuit of claim 9 wherein said relief valve assembly includes a shuttle valve that is responsive to said pilot valve means to bypass said second relief valve and said second relief valve has a lower relief setting than said first relief valve.

11. The hydraulic circuit of claim 9 wherein said first range of pilot fluid pressures is from 35 to 155 psi, and said second range of pilot fluid pressure is from 145 to 218 psi.

12. A hydraulic system for powering the translation motor and implements of an excavator having a boom, a stick, and a bucket supported on a rotatable platform, said system comprising:

- a first and a second variable displacement pump driven by an internal combustion engine;

- a single translation motor for propelling the vehicle;
- a slew drive motor for controlling the movement of the platform;

a boom control motor;

a stick control motor;

a bucket control motor;

a first series of interrupted series-connected distributors connecting said first pump first to the translation motor and lastly to said stick control motor; and

a second series of interrupted series-connected distributors connecting the second pump first to the translation motor, then to said bucket control motor, and lastly to said boom control motor;

the first distributor in each of said series being operated by single pilot control means operative to shift the first of said first distributors over a first range of pilot control pressure and to shift the second of said first distributors over a second range of pilot control pressure so that said translation motor operates from said first pump over said first range and from both pumps over said second range; and

the last distributor in at least one of said series being operative to combine fluid available from the other series for operation of its respective control motor.

13. The invention of claim 12 comprising a third pump and a distributor valve operative to direct fluid from said third pump to said slew control motor independent of said translation motor.

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14. The invention of claim 13 wherein the slew control circuit is completely independent of the other circuits of said systems.

15. The hydraulic system of claim 13 including dual-pressure relief valve means having a first relief pressure when said translation motor is operating, and having a second relief pressure when said implement control

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motors are in operation.

16. The hydraulic system of claim 15 wherein said first range of pilot control pressure is from 35 to 155 psi and said second range of pilot control pressure is from 155 to 218 psi.

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