

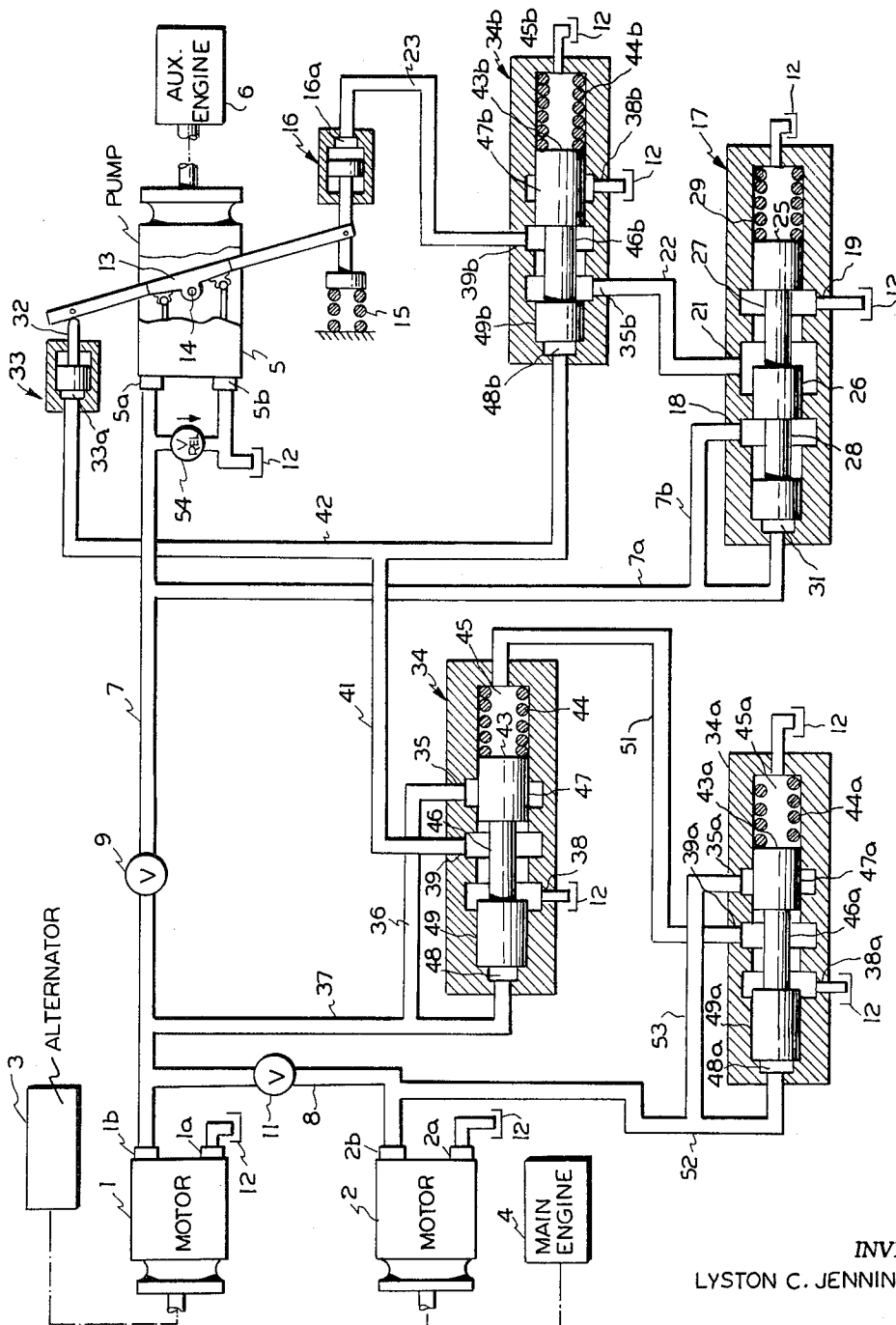
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HYDRAULIC SYSTEM

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HYDRAULIC SYSTEM

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This invention relates to hydraulic systems for supplying fluid under pressure from a variable delivery, discharge pressure compensated pump to a plurality of motors. As used herein, the term variable delivery, discharge pressure compensated pump means a pump whose delivery rate is decreased and increased automatically as pump discharge pressure rises above and falls below, respectively, a predetermined value.

In some systems of this kind, it is required that the output of the pump be delivered selectively to one or both of a pair of motors so that the first motor can be operated either by itself or in parallel with the second motor. Since the maximum capacity of the pump obviously must exceed the maximum flow demand of the first motor if it is to be capable of driving both motors at the desired speeds, and the pump will deliver fluid at the maximum rate until the discharge pressure rises to the setting of the compensator, it is apparent that some means must be provided to prevent overspeeding of the first motor when it is operating by itself. This means normally takes the form of a throttling device which limits the rate of supply of hydraulic fluid to the first motor. However, this solution to the overspeeding problem is undesirable first, because a large portion of the energy expended at the pump is wasted at the throttle, and second, because the presence of the throttle causes heating of the hydraulic oil and necessitates the use of a large heat sink or heat exchanger to provide adequate cooling for the system.

The object of this invention is to provide an improved supply system which is more efficient than, and does not require the excessive cooling capacity characteristic of, the prior system just mentioned. According to this invention, the system is provided with means for automatically changing the maximum delivery rate of the pump in accordance with the number of motors being supplied with motive fluid so that when only one motor is being operated the delivery rate is limited to a lower maximum value than when both motors are being operated. Since this approach to the overspeed problem in effect matches pump output to demand, it avoids the expenditure of energy at the pump which must subsequently be dissipated by throttling action. As a result, the improved system is more efficient and requires less cooling capacity than the prior system first mentioned.

In some installations in which the improved system can be used, the motor which is operated by itself drives a device, such as an alternator, whose operating speed must be closely regulated. Since the pump is a variable delivery unit, this requirement normally would necessitate the inclusion of a speed control valve that by-passes to tank that portion of the fluid supplied by the pump which is in excess of the amount needed to drive the motor at the desired speed. This by-pass action wastes energy. The preferred embodiment of the invention is designed for use in this type of installation, but, in lieu of a speed control by-pass valve, it incorporates means for auto-

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matically causing the pump to operate as a fixed delivery unit when the constant speed motor is operated by itself. In this scheme, control of motor speed is effected by regulating the speed of the prime mover that drives the pump, and, therefore, the energy losses that characterize the normal type of speed control are avoided. Since speed control also is required when both motors are running and the pump reverts to variable delivery operation, the first motor is equipped with a flow limiting or throttling device that limits the rate of flow to that motor to a value compatible with the regulated speed. Although energy is dissipated at this flow limiter, this loss is confined to those times when both motors are operating, and then only when the demand of the second motor is below its maximum.

With these concepts in mind, the preferred embodiment will now be described in detail with reference to the accompanying drawing whose single figure is a schematic diagram of an improved supply system suitable for use in a vehicle, such as an aircraft.

As shown in the drawing, the improved system is employed to supply motive fluid to the motors 1 and 2 which serve to drive alternator 3 and start main engine 4, respectively. Motive fluid for operating the motors is supplied by a variable displacement, discharge pressure compensated pump 5 which is driven by an auxiliary engine 6 and whose discharge port 5a is connected with the inlet port 1b of motor 1 by conduit 7 and with the inlet port 2b of motor 2 by conduits 7 and 8. These conduits 7 and 8 contain supply valves 9 and 11. The inlet port 5b of pump 5 and the discharge ports 1a and 2a of the motors are in continuous communication with a reservoir or tank 12.

Pump 5, which is of the rotary cylinder barrel, longitudinally reciprocating piston type, may actually be a pump-motor unit of the type described in U.S. Patent 3,116,595, granted January 7, 1964, and thus serve also to start auxiliary engine 6. However, in the interest of clarity, this unit is treated here solely as a pump and only the controls required for pumping operation are illustrated. The displacement of pump 5 is varied by a cam plate 13 which is angularly adjustable about the axis of trunnion 14 that is normal to and passes through the axis of rotation. The cam plate 13 is positioned by the discharge pressure compensator which includes a spring 15 that biases it toward its maximum displacement position, a control motor 16 for shifting it toward its zero displacement position against the bias of spring 15, and a pilot valve 17 which normally serves to pressurize and vent motor 16 in accordance with the pressure in conduit 7. Pilot valve 17 includes an inlet port 18 which is connected with conduit 7 through conduits 7a and 7b, an exhaust port 19 which is in continuous communication with tank 12, and a motor port 21 which is connected with the working chamber 16a of control motor 16 through conduits 22 and 23 and an isolation valve 34b, described below. Communication between the ports 18, 19 and 21 is controlled by a valve plunger 25 which has a vent position illustrated in the drawing, in which land 26 isolates inlet port 18 from motor port 21, and annular groove 27 connects the motor port with exhaust port 19, a supply position in which land 26 isolates motor port 21 from exhaust port 19 and annular groove 28 connects the motor port with inlet port 18, and an intermediate lap position in which land 26 isolates the

motor port from both of the other ports. A coil compression spring 29 biases valve plunger 25 to the left toward the vent position, and the plunger is shifted to the right against the opposing force of the spring by the fluid pressure in a chamber 31 which is in continuous communication with supply conduit 7 through conduit 7a.

The maximum displacement of pump 5 is determined by an adjustable stop 32 which is movable between the illustrated high maximum displacement setting and a lower maximum displacement setting by a fluid pressure motor 33 that is controlled by a selector valve 34. The lower maximum displacement setting is so chosen that pump 5 will be able to supply the demand of motor 1 when it is driving alternator 3 at the desired speed and auxiliary engine 6 is running at its governed speed, and the higher setting is so chosen that the pump can supply this demand as well as the maximum demand of motor 2. The selector valve 34 has an inlet port 35 which is connected with conduit 7 by conduits 36 and 37, an exhaust port 38 which communicates with tank 12, and a motor port 39 which is connected with the working chamber 33a of motor 33 by conduits 41 and 42, and is provided with a sliding valve plunger 43 which controls communication between these ports. This valve plunger is movable between the illustrated vent position toward which it is urged by spring 44 and by the fluid pressure in chamber 45 and in which motor port 39 is connected with exhaust port 38 by annular groove 46 and is isolated from inlet port 35 by land 47, and a supply position toward which it is shifted by the fluid pressure in chamber 48 and in which the annular groove 46 connects motor port 39 with inlet port 35 and land 49 isolates the motor port from exhaust port 38. Chamber 48 is connected with supply conduit 7 at a point downstream of supply valve 9 by conduit 37, so it is pressurized and vented whenever valve 9 is opened and closed, respectively. Chamber 45, on the other hand, is connected by conduit 51 with a second selector valve 34a which serves to pressurize and vent it according as supply valve 11 is opened and closed, respectively. Selector valves 34 and 34a are identical, except that the chamber 45a in valve 34a is continuously vented. The control chamber 48a of the second selector valve is connected with supply conduit 8 at a point downstream of valve 11 by conduit 52, and this conduit and conduit 53 deliver fluid to the inlet port 35a.

In addition to controlling motor 33, the selector valve 34 also controls the isolation valve 34b which was mentioned earlier and which is interposed between the conduits 22 and 23 leading from pilot valve 17 to control motor 16. Structurally, valve 34b is identical to selector valve 34a, but the functions assigned to two of its ports are different. In valve 34b, the port 35b is a control port which may be selectively pressurized and vented, and in the normal position of valve plunger 43b the annular groove 46b interconnects this port and motor port 39b and land 47b isolates the motor port from exhaust port 38b. The control chamber 48b of valve 34b is connected with conduit 42 so this chamber and the working chamber 33a of motor 33 are simultaneously pressurized and vented by selector valve 34.

In the illustrated embodiment, the motors 1 and 2, like pump 5, are of the rotary cylinder barrel, longitudinally reciprocating piston type. Motor 2, which serves as the starter for main engine 4, usually is a variable displacement unit and may take the form of the motor-pump unit described in co-pending application S.N. 119,170, filed June 23, 1961, now Patent No. 3,170,450, issued February 23, 1965. Since the motor controls form no part of the present invention, they are not illustrated. Motor 1, on the other hand, is a fixed displacement unit and, as mentioned earlier, it is provided with a flow limiter or flow control valve that prevents overspeeding at times when motor 2 is operating and its flow demand is below the maximum.

Operation

When the system is put in operation, auxiliary engine 6 runs at a substantially constant speed determined by the setting of its governor (not shown) and causes pump 5 to draw oil from tank 12 and to discharge it through port 5a to the supply conduit 7. If the supply valves 9 and 11 are closed, the pressure in those portions of the supply conduit downstream of the valves will have decayed to tank pressure and, therefore, springs 44 and 44a will have shifted valve plungers 43 and 43a, respectively, to their illustrated vent positions. Because of this, working chamber 33a and valve chamber 48b are vented to tank 12 through conduits 42 and 41, motor port 39, plunger groove 46, and exhaust port 38. Therefore, spring 44b will hold valve plunger 43b of isolation valve 34b in its illustrated position and stop 32 will be free to move to its higher maximum displacement setting under the action of compensator spring 15.

The pressure in supply conduit 7 is transmitted to the chamber 31 of pilot valve 17 through conduit 7a where it develops a force tending to move valve plunger 25 to the right against the bias of spring 29. Since supply valve 9 is closed, the pressure in conduit 7 rises rapidly to a value above the desired maximum which is to be maintained by the compensator, and valve plunger 25 moves to its supply position in which plunger groove 28 interconnects inlet port 18 and motor port 21. Oil under pressure is now transmitted to working chamber 16a through conduits 7a and 7b, inlet port 18, annular groove 28, motor port 21, conduit 22, control port 35b, plunger groove 46b, motor port 39b, and conduit 23. As a result, control motor 16 moves cam plate 13 in the clockwise direction about the axis of trunnion 14 toward its zero displacement position. This movement of the cam plate 14 reduces the displacement of pump 5, and, when the rate of discharge from the pump equals the rate of leakage from the system, the system pressure will have been restored to the desired maximum and spring 29 will shift valve plunger 25 to its lap position. Control motor 16 is now hydraulically locked and holds cam plate 14 in the reduced displacement position.

When supply valve 9 is opened, the oil discharged by pump 5 flows to and through motor 1 thereby causing it to drive alternator 3. This imposition of a flow demand on pump 5 produces a momentary decrease in pressure in conduit 7 which allows spring 29 to shift pilot valve plunger 25 back to its vent position. This relieves the hydraulic lock at control motor 16 so compensator spring 15 now commences to move cam plate 13 in the displacement-increasing direction. Since the pressure in conduit 7 downstream of supply valve 9 rises upon opening of valve 9, and this pressure is transmitted to the chamber 48 in selector valve 34, the plunger 43 of this valve shifts to its supply position and causes oil under pressure to be delivered to working chamber 33a and valve chamber 48b via conduits 37 and 36, inlet port 35, plunger groove 46, motor port 39, and conduits 41 and 42. As a result, motor 33 moves cam plate stop 32 to its lower maximum displacement position and isolation valve 34b shifts to the right to a position in which land 49b isolates control motor 16 from the pilot valve 17 and plunger groove 46b vents the motor 16 to tank 12. Compensator spring 15 now moves cam plate 14 to, and holds it in, the lower maximum displacement position, so pump 5 operates as a fixed displacement unit and discharges oil at the lower maximum rate. Since auxiliary engine 6 runs at a governed speed, it follows that the speed at which motor 1 drives alternator 3 will be substantially constant. Although pilot valve 17 continues to respond to the discharge pressure of the pump 5, it is ineffective to vary pump displacement. Under these conditions, the high pressure relief valve 54 protects the system against overload.

In order to start main engine 4, the operator opens supply valve 11 so that a portion of the oil in conduit 7

is diverted to and through motor 2. When valve 11 opens, the pressure in conduit 8, and in the communicating chamber 48a of valve 34a, rises and valve plunger 43a shifts to its supply position in which annular groove 46a interconnects inlet port 35a and motor port 39a. Oil under pressure is now delivered to the chamber 45 of selector valve 34 along a path including conduits 52 and 53, inlet port 35a, groove 46a, motor port 39a and conduit 51. Since the opposite ends of valve plunger 43 have equal cross-sectional areas and are subjected to substantially equal pressures, spring 44 shifts that plunger back to its illustrated vent position. Therefore, cam stop 32 can now return to its higher maximum displacement position and spring 44b shifts valve plunger 43b back to its illustrated position. The additional flow demand imposed by motor 2 results in a momentary drop in pressure in conduit 7 so if the plunger 25 of pilot valve 17 is not already in its vent position it moves to that position and thus maintains open a vent path from control motor 16 to tank 12. As a result, spring 15 moves cam plate 14 in the counterclockwise direction and increases the displacement of pump 5 from the lower maximum value toward the higher maximum value. When the output of the pump satisfies the demand of both motors and system pressure is restored to the desired level, valve plunger 25 shifts to its lap position and again hydraulically locks control motor 16. During the starting cycle of main engine 4, the flow demand of motor 2 increases progressively to a maximum, and this increasing demand causes the pump compensator to move cam plate 13 progressively toward its higher maximum displacement position. Therefore, the total demand is satisfied and system pressure is maintained substantially constant. The flow limiter incorporated in motor 1 limits to the required rate the amount of oil delivered to this motor and consequently the speed of alternator 3 is maintained within the desired limits.

After main engine 4 has been started, valve 11 is closed and motor 2 stops. The pressure in that portion of supply conduit downstream of valve 11 now decays to tank pressure with the result that spring 44a shifts valve plunger 43a to its vent position wherein it connects chamber 45 of selector valve 34 with tank 12. The shifting force developed by the system pressure in the chamber 48 of valve 34 now shifts valve plunger 43 back to its supply position, thus causing pump 5 to operate again as a fixed displacement unit and deliver fluid at the lower maximum rate.

When the supply valve 9 is closed, motor 1 and alternator 3 come to rest and the pressure in conduit 7 downstream of the valve decreases. The spring 44 in selector valve 34 now moves plunger 43 to its vent position. As a result, pump 5 reverts to variable displacement operation and is permitted to deliver oil at the higher maximum rate. The pump now may be used to supply additional loads (not shown) connected with conduit 7 at a point upstream of valve 9.

As stated previously, the drawing and description relate only to the preferred embodiment of the invention. Since changes can be made in the structure of this embodiment without departing from the inventive concept, the following claims should provide the sole measure of the scope of the invention.

What I claim is:

1. In combination

- (a) a variable delivery discharge pressure compensated hydraulic pump including a delivery control element movable in delivery-increasing and delivery-decreasing directions;
- (b) a pair of hydraulic motors;
- (c) circuit means interconnecting the pump and motors so that the fluid discharged by the pump may be delivered to one motor or to both motors in parallel; and
- (d) pump delivery-limiting means automatically effective upon operation of one motor to prevent move-

ment of the delivery control element in the delivery-increasing direction beyond a first position, and effective upon operation of both motors to permit movement of the delivery control element in the delivery-increasing direction to a position beyond said first position.

2. The combination defined in claim 1 which includes means for causing the pump to operate as a fixed delivery pump when only one motor is operating.

3. In combination

- (a) a variable delivery, discharge pressure compensated hydraulic pump having a discharge port and a delivery control element movable in delivery-increasing and delivery-decreasing directions;
- (b) a pair of hydraulic motors;
- (c) first conduit means, including a first supply valve, for selectively connecting one motor with and isolating it from the pump discharge port;
- (d) second conduit means, including a second supply valve, for selectively connecting the second motor with and isolating it from the first conduit means at a point downstream of the first supply valve; and
- (e) control means responsive to the pressures in the first and second conduit means downstream of the supply valves therein for preventing movement of the delivery control element in the delivery-increasing direction beyond a first position when the pressures in the first and second conduit means are high and low, respectively, and for permitting movement of the delivery control element in the delivery-increasing direction to a position beyond said first position when the pressures in both conduit means are high.

4. The combination defined in claim 3 including means operated by the control means for causing the pump to operate as a fixed delivery pump when the control means is preventing movement of the delivery control element beyond said first position.

5. In combination

- (a) a variable delivery hydraulic pump having a discharge port and a discharge pressure compensator;
- (b) a pair of hydraulic motors;
- (c) first conduit means, including a first supply valve, for selectively connecting one motor with and isolating it from the pump discharge port;
- (d) second conduit means, including a second supply valve, for selectively connecting the second motor with and isolating it from the first conduit means at a point downstream of the first supply valve;
- (e) adjustable stop means movable between first and second positions for limiting the maximum delivery rate of the pump to a first value and a second lower value, respectively;
- (f) an actuating motor for shifting said stop means from the first to the second position;
- (g) a selector valve connected with the actuating motor and shiftable between a first position in which it vents the motor and a second position in which it pressurizes the motor with fluid discharged by the pump; and
- (h) control means responsive to the pressures in the first and second conduit means downstream of the supply valves therein for positioning the selector valve in the first position when the pressures in the conduit means are equal and for positioning the selector valve in its second position when the pressure in the first conduit means is higher than the pressure in the second conduit means.

6. The combination defined in claim 5 including means controlled by the selector valve for causing the pump to operate as a fixed delivery pump when the actuating motor is pressurized and for causing the pump to operate as a variable delivery, discharge pressure compensated pump when the actuating motor is vented.

7. The combination defined in claim 5 in which the control means (h) comprises

- (a) spring means biasing the selector valve toward its first position;
- (b) first pressure responsive means connected with the first conduit means downstream of the first supply valve and arranged to shift the selector valve toward its second position; 5
- (c) second pressure responsive means arranged to shift the selector valve toward its first position; and
- (d) a control valve responsive to the pressure in the second conduit means downstream of the second supply valve for selectively venting the second pres- 10

sure responsive means and connecting it with the second conduit means.

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