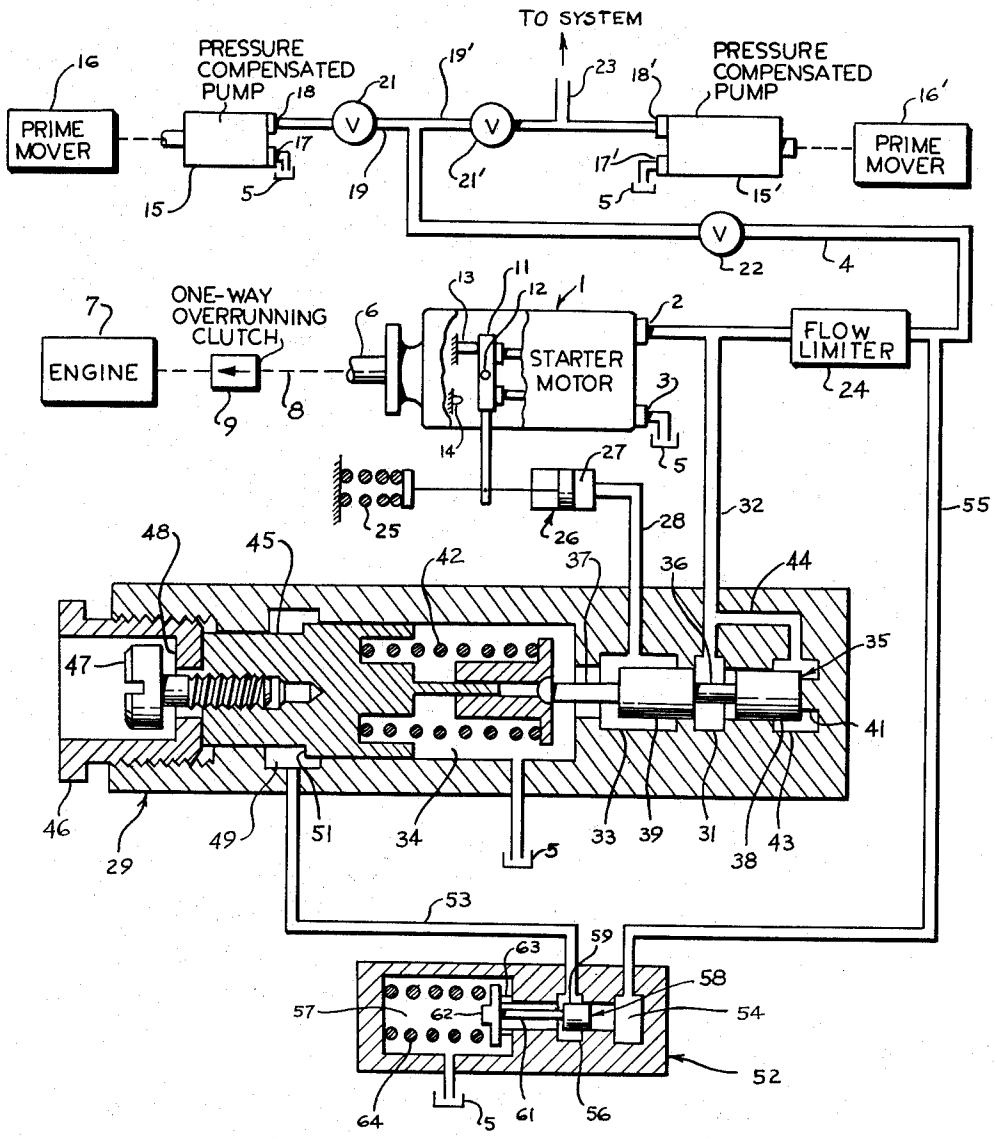
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HYDRAULIC SYSTEM

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HYDRAULIC SYSTEM

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This invention relates to hydraulic starting systems for engines.

The co-pending application of Melvin L. Kent and Edward V. Manning, Serial No. 119,170, filed June 23, 1961, discloses a hydraulic starting system utilizing a variable displacement starter motor which is supplied with motive fluid from a constant pressure source of limited flow capacity. By reason of the limited capacity of the pump in the source, or in the case where the source pump is larger than required, by reason of a flow limiter in the supply path to the motor, the maximum rate of supply to the motor is limited to a value equal to the demand of the motor when it is operating at maximum displacement and at a certain speed less than starter cut-out speed. This certain speed is approximately equal to the ignition speed of the engine. As the motor accelerates the engine from rest to ignition speed, its flow demand increases and this demand is satisfied by the increasing supply from the source. At ignition speed, the supply rate of the source is a maximum and, therefore, continued acceleration of the motor tends to imposed a flow demand on the source which exceeds its capability. The system of application Serial No. 119,170 includes a motor control that responds to the flow demand of the motor and reduces motor displacement during acceleration from ignition speed to cut-out speed so that the flow demand of the motor in this phase of the starting cycle is maintained substantially constant and equal to the maximum supply rate of the source.

In some installations in which these starting systems are used, the source includes a pair of supply pumps, one of which is normally used to supply motive fluid for starting the engine and the other of which normally supplies fluid under pressure to various auxiliary power operated devices. Normally, both pumps are of the pressure compensated type so each discharges fluid at a substantially constant pressure. However, since the demands of the auxiliary apparatus usually do not require the high pressure used during a normal starting cycle, the discharge pressure levels of the two pumps usually are different. Although it is desirable, from the standpoint of safety and reliability, to be able to use either pump as the source of motive fluid for the starter motor, this is not possible when using the system disclosed in application Serial No. 119,170 unless the discharge pressure levels of the two pumps are the same. This is so because motor operating pressure is used as the measure of motor flow demand and this pressure varies with supply pressure. Thus, if the supply pressure levels are different and the motor control is designed to maintain an operating pressure based on the higher supply pressure, selection of the pump having the lower pressure level will reduce operating pressure below the design value and cause the motor control to maintain the displacement control element of the motor in the minimum displacement position. As a result, the motor will not develop the required starting torque. On the other hand, if the operating pressure to be maintained by the motor control is based on the lower supply pressure, selection of the pump having the higher pressure level will raise operating pressure above the design value until the motor reaches the speed at which its displacement commences to be reduced. At this point the torque output of the motor is reduced to the design value and the motor is then prevented from utilizing the higher pressure supplied to it. In this case, when maximum starting capability is required, the motor is incapable of accelerating the engine to starter cut-out speed.

The object of the present invention is to provide a hydraulic starting system of the type disclosed in application Serial No. 119,170, which uses operating pressure as the measure of flow demand and which produces the same starting characteristics at two different supply pressures. As in the system of application Serial No. 119,170, the system of this invention includes motor control means that responds to the operating pressure of the motor for positioning the displacement control element of the motor in maximum displacement position when operating pressure is above a pedetermined pressure and for moving the displacement control element toward minimum displacement position as operating pressure tends to decrease below this predetermined pressure. However, the invention also provides regulating means that responds to supply pressure and serves to establish high and low values for the predetermined pressure in accordance with the selection of the high pressure and low pressure supply pumps, respectively. The differential between the high and low values of the predetermined pressure is equal to the differential between the pressure levels of the two supply pumps, and, therefore, the operation of the motor control is rendered independent of supply pressure.

In its preferred form, the motor control, like the one disclosed in application Serial No. 119,170, employs biasing means for urging the displacement control element toward minimum displacement position, a fluid pressure control motor for moving that element toward the maximum displacement position, and a control valve that responds to motor operating pressure and selectively pressurizes and vents the control motor accordingly as the operating pressure rises above and falls below a predetermined pressure. The valve is biased toward the venting position by a spring and is shifted toward the pressurizing position by a pressure motor that responds to motor operating pressure. In accordance with this form of the invention, the regulating means comprise a spring seat for the control valve spring that is movable between high spring-load and low spring-load positions, a fluid pressure motor for moving the spring seat to the high spring-load position, and a selector valve that responds to supply pressure and selectively vents and pressurizes the spring seat motor. When supply pressure is at the lower of the two values, the selector valve vents the spring seat motor and the normal preload in the spring determines the predetermined pressure at which the control valve pressurizes the control motor and thus commences to reduce motor displacement. On the other hand, when supply pressure is at the higher of the two values, the selector valve pressurizes the spring seat motor and causes it to move the seat to the high spring-load position and increase the bias exerted by the spring. The differential between the operating pressures required to move the control valve to the pressurizing position when the seat is in its low and high spring-load positions is equal to the differential between the two supply pressures, and, therefore, the displacement controlling action of the motor control is the same in both cases. As a result, motor operation is rendered independent of supply pressure.

The preferred embodiment of the invention is described herein with reference to the accompany drawing whose single FIGURE is a schematic diagram of the improved starting system.

As shown in the drawing, the starter system comprises a variable displacement starter motor 1 having high and low pressure ports 2 and 3, respectively, which are connected with supply conduit 4 and reservoir 5, and a drive shaft 6 which is coupled to the engine 7 through a drive

connection indicated at 8. The drive connection 8 includes a one-way overrunning clutch 9 arranged to transmit torque from motor 1 to engine 7 but not in the opposite direction. Motor 1 is of the rotary cylinder barrel longitudinally-reciprocating piston type and includes a cam plate 11 that is angularly adjustable about the axis of trunnion 12 between minimum and maximum displacement positions. A stop 13 defines the minimum displacement position, which may or may not be a zero displacement position, and a stop 14 defines the maximum displacement position.

Motive fluid for driving motor 1 is derived from a source that includes a pair of variable delivery discharge pressure compensated pumps 15 and 15' which are driven by prime movers 16 and 16'. The low pressure ports 17 and 17' of the two pumps are connected with reservoir 5 and the high pressure ports 18 and 18' are connected with supply conduit 4 through branch conduits 19 and 19' containing shut-off valves 21 and 21'. The two pumps may also be of the rotary cylinder barrel longitudinally-reciprocating piston type. A starter valve 22 interposed in supply conduit 4 controls the flow of motive fluid to starter motor 1. Pump 15 is the normal source of supply for starter motor 1 and is compensated to establish a higher full cut-off pressure than pump 15'. Pump 15', on the other hand, normally is used to supply fluid to various auxiliary power operated devices (not shown) with which it is connected by conduit 23 and branch conduit 19'.

In the illustrated embodiment, it is assumed that the maximum delivery rate of each of the pumps 15 and 15' is greater than the demand of motor 1 when the latter is driving engine 7 at ignition speed and cam plate 11 is in the maximum displacement position. Therefore, in this case, the rate of supply to motor 1 is limited to that demand by a flow limiter 24 interposed in supply conduit 4. The illustrated location of flow limiter 24 in the supply path to motor 1 is not critical and, as a matter of fact, it usually is most convenient to mount it in the casing of motor 1 in the passage connecting high pressure port 2 with the motor working chambers. Of course, if the maximum delivery rate of each pump is equal to the aforementioned demand of the motor at ignition speed, the flow limiter 24 can be eliminated.

Cam plate 11 is positioned by a motor control including a coil compression spring 25 that biases it in the counterclockwise direction about the axis of trunnion 12 toward the minimum displacement position, and a control motor 26 that is arranged to shift it in the opposite direction against the bias of the spring. Fluid is supplied to and exhausted from the working chamber 27 of motor 26 through conduit 28 under the control of valve 29. The control valve 29 comprises a housing containing an inlet chamber 31 which is connected with supply conduit 4 by conduit 32, an outlet chamber 33 which is connected with conduit 28, and an exhaust chamber 34 which is in continuous communication with reservoir 5. Communication between the outlet chamber 33 and the other two chambers is controlled by a sliding valve plunger 35 formed with two annular grooves 36 and 37 and a pair of valve lands 38 and 39. Valve plunger 35 is biased to the right into abutment with stop 41 by a coil compression spring 42 and in this position groove 37 interconnects chambers 33 and 34. The plunger is moved to the left, first to a lap position in which land 39 isolates chamber 33 from both of the chambers 31 and 34, and then to a supply position in which groove 36 interconnects chambers 31 and 33 by the fluid pressure in chamber 43 that acts upon the right end of the plunger. This chamber 43 is connected with supply conduit 4 at a point downstream of flow limiter 24 through passage 44 and conduit 32 and thus is subjected to the operating pressure of the motor.

The left end of the coil spring 42 of control valve 29 is seated on a reciprocable spring seat 45 which is shiftable between the illustrated low spring-load position, in which it abuts the inner end of sleeve 46, and a high spring-load position in which the head of bolt 47 engages the surface 48 on the inturned flange of sleeve 46. Since both the sleeve 46 and the bolt 47 are threaded, the longitudinal position of each member may be varied independently of the other to thereby afford independent adjustment of the high and low spring-load settings. Seat 45 is urged toward the low spring-load position by spring 42 itself, and is shifted to the high spring-load position by a fluid pressure motor comprising working chamber 49 and a piston defined by the annular shoulder 51 formed on the outer periphery of the seat. The working chamber 49 of the spring seat motor is selectively pressurized and vented by a selector valve 52 with which it is connected by conduit 53. This selector valve 52 comprises an inlet chamber 54 which is connected with supply conduit 4 at a point upstream of flow limiter 24 by conduit 55, an outlet chamber 56 which is connected with conduit 53, and an exhaust chamber 57 which is in continuous communication with reservoir 5. Outlet chamber 56 is selectively connected with the other two chambers by a reciprocable valve plunger 58 formed with a land 59 and an annular groove 61. The valve 58 is biased to the right to the vent position, in which its enlarged end portion 62 engages stops 63, by a coil compression spring 64, and is shifted in the opposite direction to the supply position by the fluid pressure in inlet chamber 54 that acts upon its right end.

Operation

For purposes of this discussion, it is assumed that pump 15 is compensated to a full cut-off pressure of 4,000 p.s.i. and that pump 15' is compensated to a full cut-off pressure of 3,000 p.s.i.

In a normal starting cycle, valves 21 and 22 are opened and valve 21' is closed. The fluid discharged by pump 15 flows to motor 1 through conduits 19 and 4 and the pressure in these conduits and in conduits 32 and 55 immediately rises. In response to this rise in pressure, valve plunger 58 of selector valve 52 is shifted to its pressurizing position so that fluid at supply pressure is transmitted to working chamber 49 via conduit 55, chambers 54 and 56, and conduit 53. The pressure force developed on annular shoulder 51 now moves seat 45 to its high spring-load position. Simultaneously, the fluid pressure in chamber 43 of control valve 29 shifts valve plunger 35 to its supply position and, therefore, the pressure in conduit 4 downstream of flow limiter 24 is transmitted to working chamber 27 of control motor 26 via conduit 32, chamber 31, plunger groove 36, chamber 33 and conduit 28. As a result, the control motor moves cam plate 11 in the clockwise direction about the axis of trunnion 12 against the bias of spring 25 to the maximum displacement position defined by stop 14. The fluid passing through motor 1 now causes it to develop torque and accelerate engine 7.

As motor 1 accelerates engine 7, its flow demand increases. Since pump 15 includes a discharge pressure compensator, it satisfies this demand and maintains supply pressure (i.e., the pressure upstream of flow limiter 24) substantially constant by increasing its rate of delivery. When engine 7 reaches the speed at which it is ignited, the flow demand of motor 1 equals the maximum rate of supply permitted by flow limiter 24. Therefore, as the engine commences to develop assisting torque and to accelerate from ignition speed, the demand of motor 1 exceeds the rate of supply and motor operating pressure (i.e., the pressure downstream of flow limiter 24) tends to decrease. Since chamber 43 is subjected to motor operating pressure, this change in pressure enables spring 42 to move valve plunger 35 to the right to the vent position in which working chamber 27 is vented to reservoir 5 through conduit 28, chamber 33, groove 37 and chamber 34. As a result, spring 25 moves the cam plate

in the displacement-decreasing direction. When the change in motor demand resulting from the increase in speed is offset by the reduction in motor displacement, motor operating pressure will be restored and valve plunger **35** will shift to the lap position in which control motor **26** is hydraulically locked. As engine **7** continues to accelerate, the motor control continues to effect a reduction in motor displacement in an effort to maintain its flow demand constant at the rate demanded at ignition speed. When the engine reaches starter cut-out speed, the control for motor **1** will have moved cam plate **11** to the minimum displacement position defined by stop **13**. At this time, valves **21** and **22** are closed and motor **1** comes to rest. Because of the presence of overrunning clutch **9**, the motor imposes no appreciable drag on the engine as the latter accelerates to normal operating speed.

When pump **15'** is used as the source of motive fluid for starting, valves **21'** and **22** are opened and valve **21** is closed. In this case, the 3,000 p.s.i. supply pressure is not sufficient to hold the valve plunger **58** in its supply position so spring **64** moves the plunger to the vent position in which working chamber **49** is connected with reservoir **5** through condiut **53**, chamber **56**, groove **61** and chamber **57**. Therefore, spring **42** now expands and moves seat **45** to the illustrated low spring-load position. Sleeve **46** and bolt **47** are so adjusted that the pressure required in chamber **43** to shift valve plunger **35** to the lap position when the seat **45** is in the low spring-load position is approximately 1,000 p.s.i. lower than the corresponding pressure required when the seat is in the high spring-load position. In other words, the differential between the operating pressures required to hold valve **29** in lap position is equal to the differential between the two supply pressures. Because of this, the operation of control valve **29** is insensitive to the change in supply pressure and the motor control will operate to maintain motor demand constant during acceleration from ignition speed to cut-out speed in the same manner as in the previous case when supply pressure was being maintained at 4,000 p.s.i.

In the preceding discussion, it is mentioned that the motor control commences to move cam plate **11** toward the minimum displacement position when engine **7** reaches ignition speed. While it is possible to operate the starter in this manner, it is preferred to design pumps **15** and **15'**, or in the case where the pumps are oversize, to design flow limiter **24**, to establish a maximum supply rate that enables motor **1** to operate at maximum displacement until the engine reaches a speed slightly higher (for example, 1,000 r.p.m. higher) than the recommended ignition speed. This procedure insures that the starter will be developing maximum torque when the engine is ignited without requiring the operator to initiate ignition at a precise speed.

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

What we claim is:

1. In combination, an engine and a hydraulic starting system for accelerating the engine to starter cut-out speed, the system comprising

(*a*) a variable displacement hydraulic motor connected with the engine for driving same and having a displacement control element movable between minimum and maximum displacement positions;

(*b*) a source of hydraulic fluid capable of delivering fluid at a selected one of two substantially constant supply pressures and at a rate that varies between a minimum value and a maximum value equal to the demand of the motor when the displacement control element is in maximum displacement position and the motor is operating at a speed less than starter cut-out speed;

(*c*) conduit means connected with the motor and the source for leading fluid from the source to and through the motor;

(*d*) motor control means connected with the displacement control element and responsive to the operating pressure of the motor for positioning said element in the maximum displacement position when operating pressure is above a predetermined pressure and for moving said element toward the minimum displacement position as operating pressure tends to decrease below said predetermined pressure, whereby the change in displacement of the motor maintains motor demand constant at said maximum delivery rate; and

(*e*) regulating means connected with the motor control means and responsive to the supply pressure for establishing a high value for said predetermined pressure when the source is delivering fluid at the higher of said two substantially constant supply pressures and for establishing a lower value for said predetermined pressure when the source is delivering fluid at the lower of said two substantially constant supply pressures.

2. The combination defined in claim 1 in which

(*a*) the motor control means comprises

(1) means biasing the displacement control element toward minimum displacement position,

(2) a fluid pressure control motor for moving the displacement control element toward maximum displacement position against the bias of the biasing means.

(3) a control valve having an inlet passage connected with the conduit means, an outlet passage connected with the control motor, an exhaust passage, and a valve member shiftable between first and second positions in which, respectively, the outlet passage is connected with the exhaust and inlet passages and having an intermediate position in which the outlet passage is isolated from both the inlet and exhaust passages,

(4) a spring biasing the valve member toward the first position, and

(5) means responsive to the operating pressure of the motor for shifting the valve member toward the second position against the bias of the spring; and

(*b*) the regulating means comprises

(1) a movable seat for the spring of the control valve which is shiftable in spring load-increasing and spring load-decreasing directions,

(2) a spring seat motor connected with the movable seat and which when energized shifts the seat in the load-increasing direction and when de-energized allows the spring to move the seat in the load-decreasing direction, and

(3) means responsive to supply pressure for energizing and de-energizing the spring seat motor, respectively, when supply pressure is at the higher and lower of said two substantially constant pressures.

3. In combination, an engine and a hydraulic starting system for accelerating the engine to starter cut-out speed, the system comprising

(*a*) a variable displacement hydraulic motor connected with the engine for driving same and having a displacement control element movable between minimum and maximum displacement positions;

(*b*) a source of hydraulic fluid capable of delivering fluid at a selected one of two substantially constant supply pressures and at a rate that varies between a minimum value and a maximum value equal to

the demand of the motor when the displacement control element is in maximum displacement position and the motor is operating at a speed less than starter cut-out speed;

(*c*) conduit means connected with the motor and the source for leading fluid from the source to and through the motor;

(*d*) means biasing the displacement control element toward minimum displacement position;

(*e*) a fluid pressure control motor for moving the displacement control element toward maximum displacement position against the bias of the biasing means;

(*f*) a control valve having an inlet passage connected with the conduit means, an outlet passage connected with the control motor, an exhaust passage, and a valve member shiftable between first and second positions in which, respectively, the outlet passage is connected with the exhaust and inlet passages and having an intermediate position in which the outlet passage is isolated from both the inlet and exhaust passages;

(*g*) a spring biasing the valve member toward the first position;

(*h*) means responsive to the operating pressure of the motor for shifting the valve member toward the second position against the bias of the spring;

(*i*) a movable seat for the spring which is shiftable between high spring-load and low spring-load positions;

(*j*) a fluid pressure spring seat motor connected with the movable seat and effective, when pressurized, to move the seat to the high spring-load position and, when vented, to permit the spring to move the seat to the low spring-load position; and

(*k*) a selector valve responsive to supply pressure for connecting the spring seat motor with the source when supply pressure is at the higher of said two substantially constant pressures and for venting the spring seat motor when supply pressure is at the lower of said two substantially constant pressures.

4. The combination defined in claim 3 including

(*a*) adjustable means associated with the movable seat for changing the low spring-load position; and

(*b*) means adjustable independently of the adjustable means for changing the high spring-load position of the movable seat independently of the low spring-load position.

No references cited.