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Haak et al.

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[54] **PRESSURE AND/OR FLOW COMPENSATING MEANS**

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[52] **U.S. Cl.**.....**60/52 VS**, 60/52 HE

[51] **Int. Cl.**.....**F15b 15/18**

[58] **Field of Search**.....60/52 VS, 52 HE, 52 SR

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[57] **ABSTRACT**

The control circuit of a high pressure work system powered by a variable displacement-positive displacement pump is provided with a compensating valve for controlling the pump displacement. The compensating valve is responsive to pressure in a control circuit and in the work circuit as well as to fluid flow in another circuit to alter the displacement of the work system pump.

4 Claims, 2 Drawing Figures

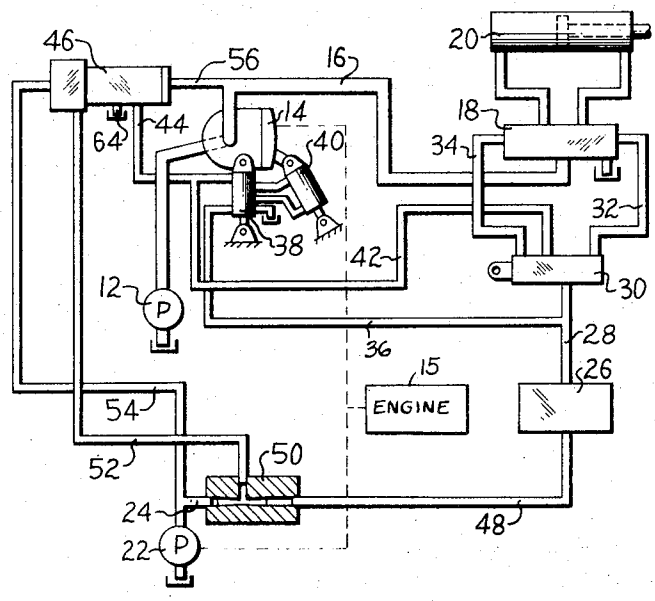


Fig. 1

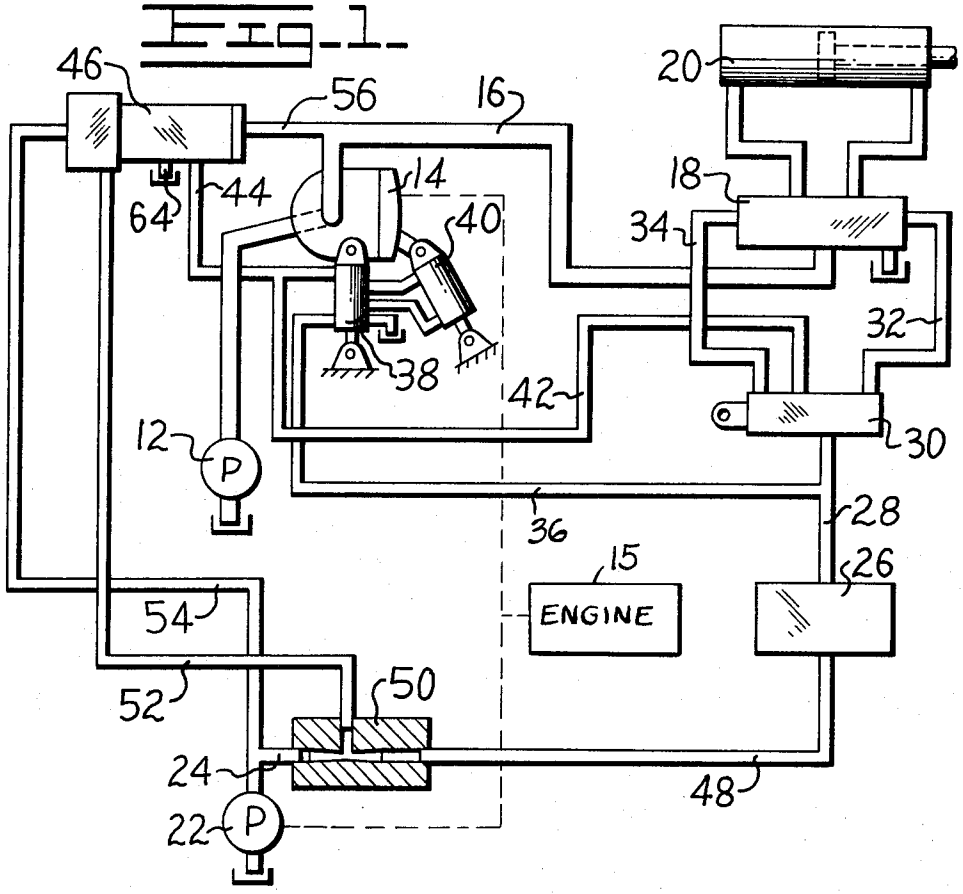
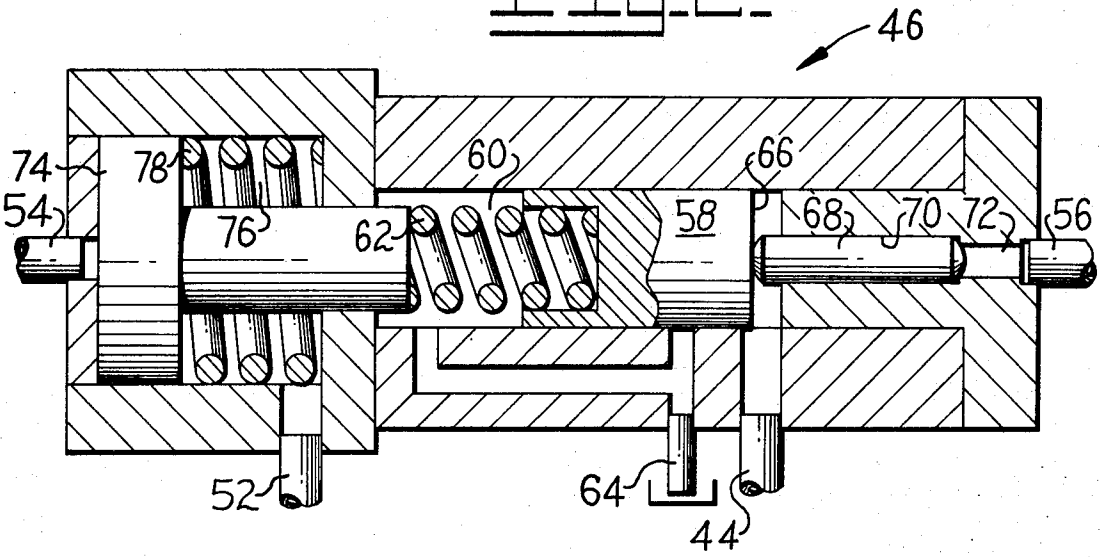


Fig. 2



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PRESSURE AND/OR FLOW COMPENSATING MEANS

BACKGROUND OF THE INVENTION

The present invention relates to valves and pertains more particularly to a valve capable of functioning to provide both flow compensation and pressure compensation in a hydraulic circuit.

Hydraulic power systems are extensively employed in lifting, loading, and other handling systems. Such hydraulic systems are provided with a prime mover, such as a gasoline or diesel engine driving a pump for delivery of pressurized fluid to the system. The hydraulic system is generally capable of delivering substantially the full horsepower of the prime mover to the work output of the system. One difficulty with such systems is that when engine speed is less than that for full horsepower, the hydraulic work system is still capable of demanding full maximum engine horsepower. This condition generally stalls the engine or at least hinders engine response considerably.

SUMMARY OF THE INVENTION

It is, therefore, the primary object of the present invention to provide a simple and reliable valve which is capable of adjusting the power demands of a hydraulic system to the output of the engine.

A further object of the present invention is the provision of a control valve for a hydraulic system which is responsive to engine speed and system pressure to adjust the system pump to prevent overloading the engine.

BRIEF DESCRIPTION OF THE DRAWINGS

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

FIG. 1 is a schematic illustration of a typical hydraulic power work output system embodying the present invention;

FIG. 2 is a side elevational view in section of a preferred embodiment of the present invention.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENT

Referring now to the drawings, and in particular to FIG. 1, there is illustrated a schematic illustration of a hydraulic power system in combination with a low pressure hydraulic control system embodying the present invention. The high pressure work output system comprises a supply pump **12** for supplying fluid to a high pressure variable displacement pump **14** which is driven by a suitable prime mover **15** which may be, for example, a diesel or gasoline powered engine. High pressure fluid from the variable displacement pump **14** is supplied by means of a conduit **16** to a directional control valve **18** which controls the direction of power output from the work system such as by means of a double-acting hydraulic motor **20**.

A low pressure hydraulic control circuit for controlling the valve **18** comprises a fixed displacement pump **22** directly drives by engine **15** and having a flow output which is directly proportional to vehicle engine speed. Fluid from the pump **22** is delivered by a conduit

24, a venturi **50** and a conduit **48** to an accumulator charging valve **26**, thence to a pilot control valve **30** which controls the directional control valve **18**. The pilot control valve **30** is generally manually controlled and supplies fluid by means of conduits **32** and **34** to the valve **18**.

A conduit **36** supplies fluid to a servo valve **38** which controls a fluid motor **40** for varying the displacement of pump **14**. A conduit **42** carries control fluid from pilot control valve **30** to another input side to servo valve **38** and control fluid pressure is communicated via conduit **44**, to a compensating valve **46** which will be described in detail below.

The flow of fluid in conduit **48** is monitored by means of venturi **50**, with conduits **52** and **54** supplying a differential pressure signal created across the venturi to the compensating valve **46**.

Referring now to FIG. 2, there is illustrated a compensating valve **46** which comprises a valve member **58** which acts as a variable setting relief valve, mounted for reciprocal movement within a substantially cylindrical chamber **60**. The valve member **58** is biased to the closed position by means of a spring **62** and thereby functions to cut off or block a flow of fluid from conduit **44** to an exhaust line **64** leading back to the reservoir. Face **66** of valve member **58** functions as a piston in response to pressurized fluid from conduit **44**. A second piston **68** is reciprocally mounted in a chamber **70** and cooperatively engages valve member **58** for urging the valve to the opened position in response to increased pressure in chamber **72** communicated by means of a conduit **56** from the high pressure work system. A third piston **74** is reciprocally mounted in a chamber **76** is operatively connected through spring **62** to valve member **58**. The spring **78** biases the piston **74** to one end of the chamber **76** to relieve pressure or force against spring **62**. Conduits **52** and **54** provide communication with chamber **76** on opposite sides of the piston **74**.

In order to understand the operation of the present invention, it is essential to understand the operation of a high pressure work system in which the present invention would be employed. Typically, such systems employ a variable stroke positive displacement pump such as **14** driven by the vehicle engine **15** which supplies pressurized fluid to a work output member such as a cylinder **20**. The horsepower output from the system will be a function of the load on the work cylinder **20** and the rate of movement of the output member from that cylinder. The rate of movement of the cylinder will be a function of the flow rate in the system which will be determined by the stroke of the pump **14** and its r.p.m. The pressure of the system will be normally the pressure required to move the output from the cylinder **20** against the workload. Thus, it can be seen that if the stroke of the pump **14** is at its maximum for a given r.p.m. setting of the prime mover, the flow rate through the circuit will be large and if a load is extremely large, then there will be a high demand of torque from the prime mover.

If the load on the system exceeds that which the engine can deliver at the given stroke, the engine will stall. The present invention eliminates this problem by the compensating valve **46** which senses the system pressure through line or conduit **56** and pilot signal

pressure through the line **44**. These pressures as they increase tend to move valve element **58** to the left and thereby relieve pressure in line **44** which controls servo valve **38** which in turn controls the fluid valve motor **40** for altering the displacement of pump **14**.

The r.p.m. of the engine which is also related to the horsepower output of the engine is compensated for by means of a flow sensing means which comprises a venturi **50** which senses the flow rate in a portion of a control circuit. This portion of the control circuit has a flow rate which is directly proportional to the prime mover speed since the fluid is supplied by means of a positive displacement pump **22** directly driven by the prime mover. When engine speed is high (rated), flow through the venturi is high and the differential pressure signal generated by the venturi is high. Piston **74** will be held to the right against spring **78** and the force on piston **58** imposed by spring **62** will be at a maximum. As a result, the pressure that can be developed in line **44** will be at a maximum and it will be possible to obtain maximum displacement (therefore, maximum flow) of pump **14**.

As engine speed decreases, flow rate and pressure differential across the venturi decrease tending to move the piston **74** to the left. This decreases the force that spring **62** imposes on valve **58** and decreases the maximum pressure level that can be obtained in line **44**. Since the pressure level in line **44** is directly proportional to displacement of pump **14**, limiting the maximum pressure in line **44** also limits the maximum displacement of pump **14**. Therefore, as engine speed and horsepower decrease, flow compensation of the system takes place automatically.

What is claimed is:

1. In combination a low pressure control circuit operatively connected for controlling a high pressure work output system, said work system including a high pressure variable displacement pump, said control circuit including a fixed displacement pump, prime mover means operatively connected to drive said pumps; a selectively actuable control valve in said control circuit, means operatively connecting the control valve to control said variable displacement pump whereby the amount of flow in the work output system is initially determined by said control valve and is controlled by varying the displacement of said variable displacement pump, and compensating valve means in said control circuit and responsive to a flow rate established by said fixed displacement pump in said control circuit and to fluid pressure in said pressure work system established by said variable displacement pump to control the output of said variable displacement pump.

2. In a low pressure control circuit for controlling a high pressure work output system, a selectively actuable control valve in said circuit, means connecting the control valve to a variable displacement pump whereby the amount of flow in the work output system is controlled, and compensating valve means responsive to a flow rate in said control circuit and to fluid pressure in said pressure work system to control the output of said pump, said compensating valve means comprising:

a valve member normally biased to a closed position;

first piston means operatively associated with said valve member and responsive to pressure in said work output system to urge said valve member to an open position; and,

second piston means operatively associated with said valve member and responsive to the rate of fluid flow in said control circuit to urge said valve member to the closed position.

3. In a low pressure control circuit for controlling a high pressure work output system, a selectively actuable control valve in said circuit, means connecting the control valve to a variable displacement pump whereby the amount of flow in the work output system is controlled, and compensating valve means responsive to a flow rate in said control circuit and to fluid pressure in said pressure work system to control the output of said pump, said compensating valve means comprising:

a cylindrical relief valve element confined in a cylindrical bore;

biasing means for differentially biasing said valve element to a closed position in response to a rate of flow of fluid in one portion of said circuit;

a first piston means responsive to fluid in one portion of said circuit to open said valve; and,

a second piston means responsive to fluid in a second portion of said circuit to assist the fluid in said one portion of said circuit to open said valve.

4. The valve of claim **3** comprising:

a third piston means;

said three piston means are arranged in separate concentric bores;

one of said piston means functions as a valve element;

spring means interposed between two of said piston means;

one of said piston means being responsive to rate of flow of fluid in said circuit to bias said valve piston;

another of said pistons being responsive to pressure in a portion of said circuit to bias said valve piston.

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