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Wilson

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[54] **SPEED CONTROL MECHANISM**

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[52] **U.S. Cl.** **91/443,** 91/446, 91/461, 137/596.15, 137/625.66

[51] **Int. Cl.** **F15b 11/08,** F15b 13/042

[58] **Field of Search** 91/444, 443, 446, 468, 91/461, 304; 137/596, 596.15, 625.66

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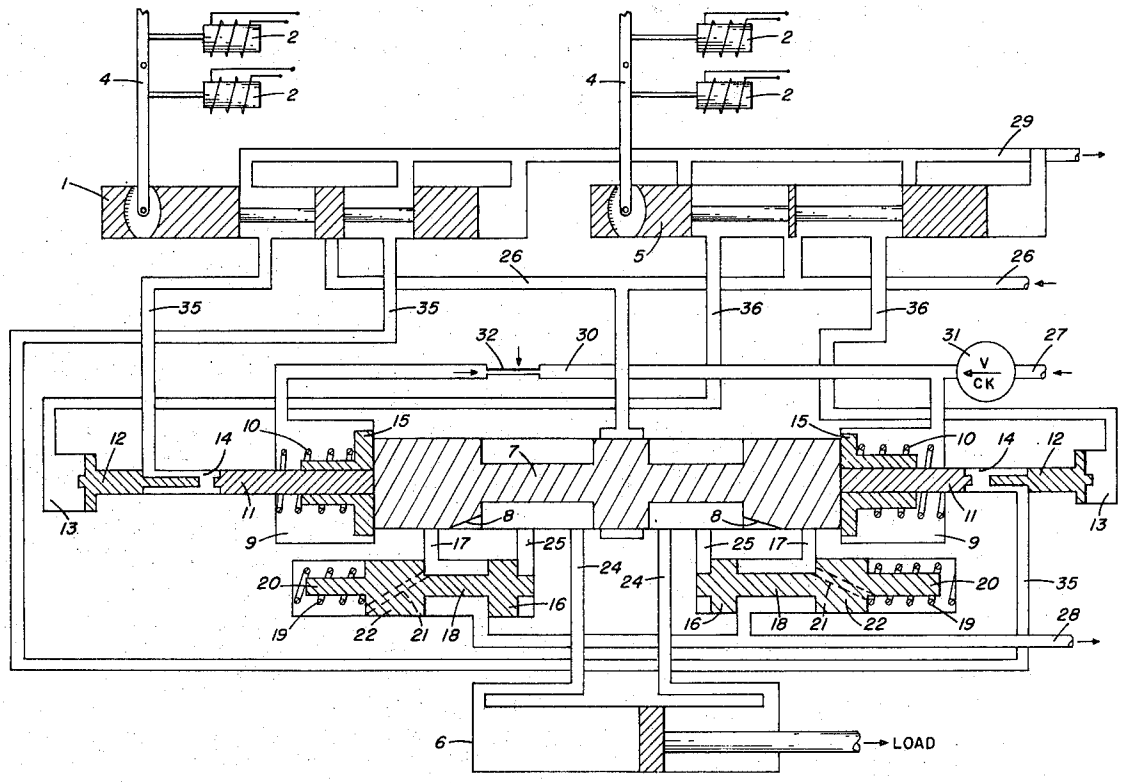
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[57] **ABSTRACT**

A speeed control mechanism having a main spool valve and metering valves controls the output to and return flow from an hydraulic fluid actuator. The main spool valve controls the direction and acceleration of the actuator, and the metering valves determine the maximum velocity of the actuator independent of the load. Adjustment features are provided to control the maximum displacement and the speed of the displacement of the main spool valve. Moreover, a novel adjustment feature is incorporated into the main spool valve to control the maximum volumetric flow rate through the metering valves.

4 Claims, 7 Drawing Figures



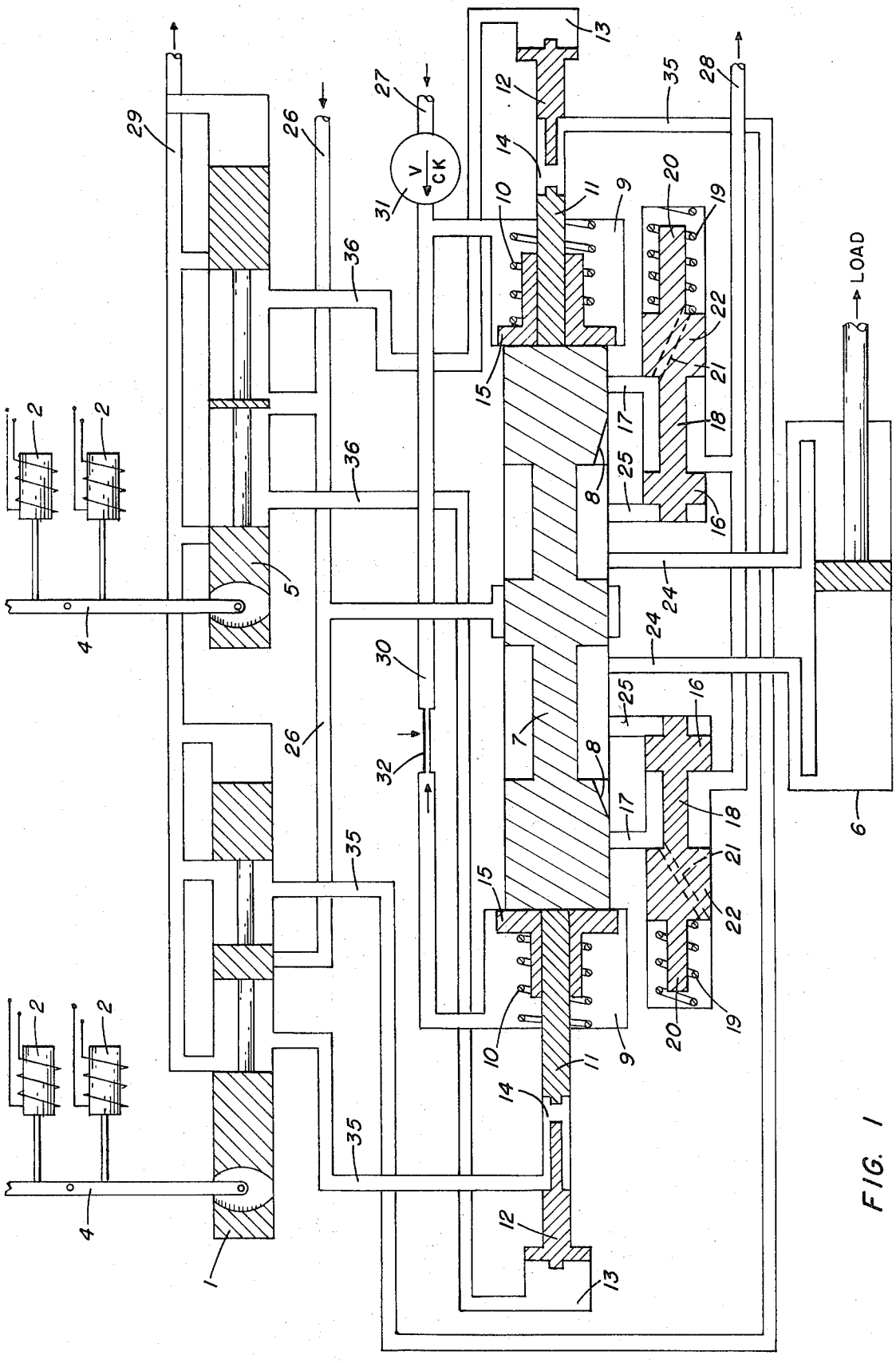


FIG. 1

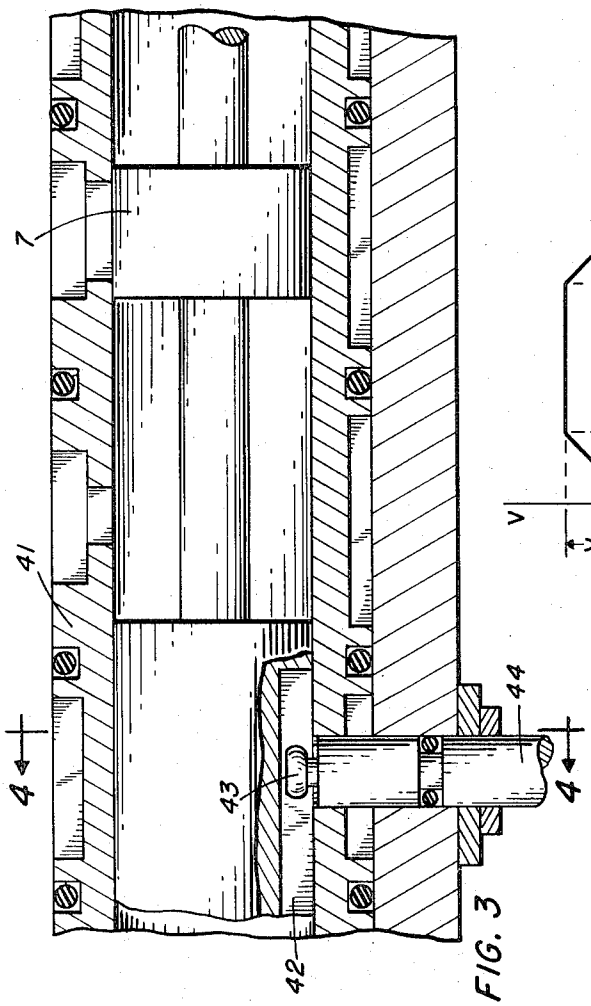


FIG. 3

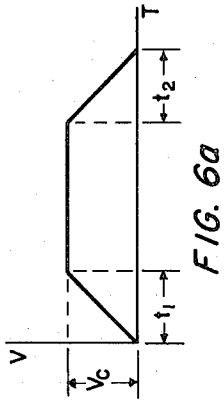


FIG. 6a

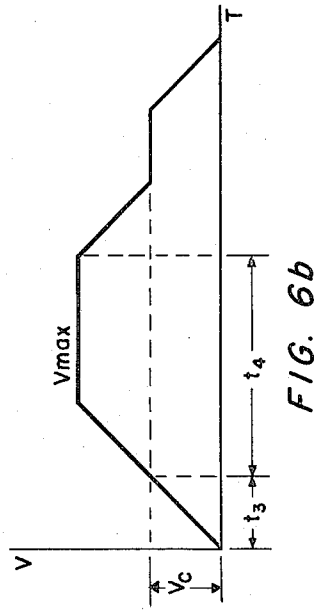


FIG. 6b

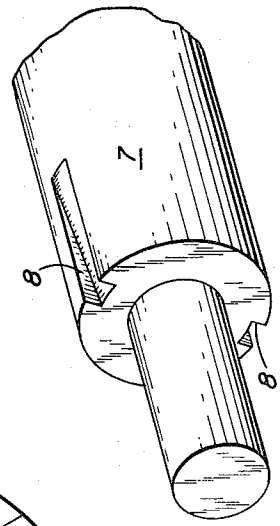


FIG. 5

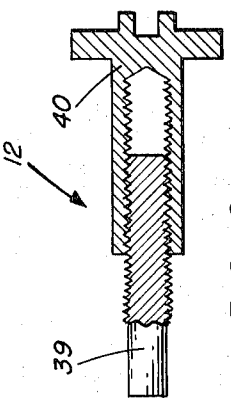


FIG. 2

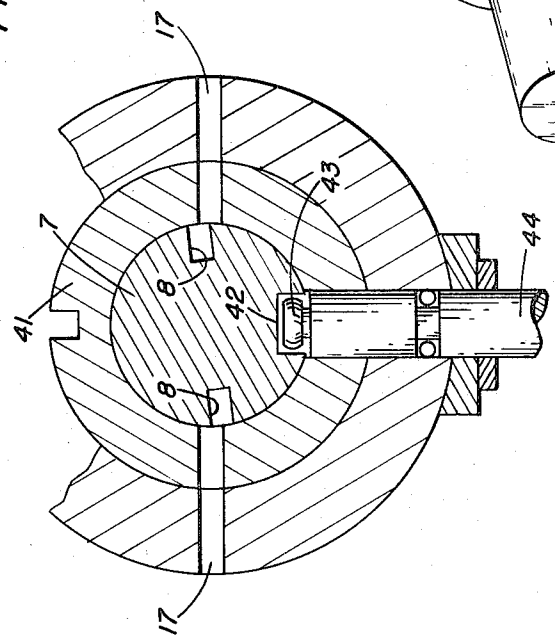


FIG. 4

SPEED CONTROL MECHANISM

STATEMENT OF GOVERNMENT INTEREST

The invention described herein may be manufactured and used by or for the Government of the United States of America for governmental purposes without the payment of any royalties thereon or therefor.

BACKGROUND OF THE INVENTION

Speed control mechanisms for linear and rotary hydraulic actuators or motors control the position and rate of actuator start, acceleration, velocity, deceleration, and stop by regulating the flow of hydraulic fluid to the actuator. The speed control mechanisms usually consist of two or more cam-actuated hydraulic flow control valves, interconnected to the actuator. Although these prior art devices function adequately, these multiple valves with their extensive interconnecting pipes and cam-actuated hardware are expensive, pose installation problems, add to hydraulic pressure drop within the system, and add unnecessary weight. A need has existed for a single unit, adjustable speed control mechanism that operates as effectively as multiple cam-actuated valves; yet is simpler to install, cheaper, and lighter.

Electrohydraulic, speed control devices for hydraulic actuators are known in the prior art. Individual integrated units for controlling the acceleration and the velocity of hydraulic actuators have previously been developed. However, none possess the adjustability features of the instant device which allows the cycle of the hydraulic actuator to be varied over a wide range. For example, a solenoid actuated speed control mechaism called the DeAccelatrol has been patented in the United States by Continental Machines, Inc., Savage, Minnesota, and is offered through its Continental Hydraulics Division. This device, however, does not possess the adjustment features of the instant invention which accurately control start, full speed constant velocity operation, and the mechanism response to variations in loads applied to the actuator.

SUMMARY OF THE INVENTION

The instant device is a single unit, adjustable, solenoid actuated speed control mechanism. It operates more effectively than multiple cam-actuated valves, offers simpler installation, and reduces weight and cost. In addition, the mechanism provides acceleration, constant velocity, and deceleration cycles which are insensitive to the variations in load on the actuator. Velocity, slow speed, acceleration and deceleration are all adjustable over a broad range. Basically, the invention achieves these results by controlling the rate at which hydraulic fluid under pressure is supplied to an actuator and the rate at which hydraulic fluid may leave the actuator and return to the sump.

OBJECTS OF THE INVENTION

An object of the present invention is the provision of a single-unit, hydraulic, speed control mechanism for more precise remote control of an hydraulic actuator.

Another object is to provide a speed control mechanism which is adjustable over a wide range of cycles.

A further object of the invention is the provision of a speed control mechanism for an hydraulic-actuator

that in case of an electrical failure will automatically decelerate and stop the hydraulic actuator.

Still another object is to provide a speed control mechanism which controls the maximum velocity of the hydraulic actuator independent of a load on the actuator.

Other objects, advantages, and novel features of the present invention will become apparent from the following detailed description of the invention when considered in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic diagram of one preferred embodiment of the invention which is shown in combination with an hydraulic actuator;

FIG. 2 is a cross section of the adjustable creep control piston of the instant invention;

FIG. 3 is a schematic diagram of a modified form of the invention illustrating an adjustment feature which may be employed to orient the main spool valve and thus control the maximum velocity of the load;

FIG. 4 is a sectional view of the structure illustrated in FIG. 3 taken along the line 4-4 thereof;

FIG. 5 is a fragmentary view of one of the lands on the main spool valve illustrating the throttling slots; and

FIG. 6 depicts some of the time-velocity cycles that may be achieved employing the instant device.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. 1 which illustrates a preferred embodiment of the invention shows a directional control solenoid valve 1, solenoids 2 and mechanical linkage 4. The directional control solenoid valve initiates actuator start and stop and determines the direction of the actuator movement. A spring (not shown) within the mechanical linkage 4 normally holds the directional control valve 1 as shown in the stop position. Energizing either directional control solenoid 2 shifts the valve 1 to the start position for the corresponding direction of actuator movement.

The creep control solenoid valve 5, solenoid 2, and mechanical linkage 4 inititates actuator acceleration and deceleration cycles and controls the duration of creep or slow speed of the hydraulic actuator. This solenoid valve is similar to the directional control solenoid valve. A spring within the mechanical linkage 4 normally holds the valve centered in the neutral position as shown. Energizing either creep control solenoid shifts the valve 5 to permit flow to one creep control piston chamber 13 and vent the other.

The main control valve spool 7 in conjunction with metering valves 18 regulates the direction and velocity of fluid flow to and from the speed control mechanism. This fluid flow, in turn, determines the direction and speed of the hydraulic actuator. The main control valve spool 7 consists of a center land and two outer lands. The outer lands have rectangular throttling slots 8.

The three-land, main, control-valve spool 7 is mounted within a central bore in a single unit valve body (not shown) between two centering spring chambers 9. Within the centering spring chambers are centering springs 10 and sleeves 15. Control pistons 11 react to pressurized fluid delivered to control piston chambers 14 from the directional control solenoid

valve 1 to displace the main valve spool 7. In the same central bore creep control pistons 12 are mounted and serve to limit the displacement of the main control valve 7. Creep control chambers 13 selectively receive pressurized fluid from creep control solenoid valve 5.

The metering valves 18 control fluid flow from a control valve return port 17, limiting the maximum discharge of the hydraulic actuator regardless of the load. As the actuator moves in one direction hydraulic fluid is returned through one of the actuator lines 24. The pressure of this fluid is sensed through sensing line 25 and acts on piston 16 of the metering valve. A pressure drop occurs across throttling slot 8 and is sensed on the rear side of land 22, having passed through passageway 21 indicated by the dotted lines. Thus, there is a pressure differential across metering valve 18 which will determine the maximum rate of volumetric flow (gallons per minute) able to leave actuator 6. This rate in turn determines the speed of the hydraulic actuator regardless of the load. Fluid from the metering valves is ported to the sump through line 28.

FIG. 2 illustrates in more detail the creep control pistons 12. As shown the creep control piston 12 consist of two parts, an adjustable stop 39 threadedly engaged into the piston head 40. By adjusting the stop 39 one may adjust the creep speed or slow speed of the hydraulic actuator by in effect limiting the displacement of the main control valve 7 and thereby adjusting the pressure drop across slots 8. It should be further noted that although the preferred embodiment is shown with creep control pistons 12, they are not necessary and merely serve to allow for additional control over the cycle of the hydraulic actuator.

FIG. 3 depicts a sectional view of a portion of the main control valve spool 7 adapted to a modified form of the invention. In this form of the invention the size of the pressure drop across throttling slot 8 is adjustable and thus the maximum velocity of the hydraulic actuator may be varied. In order to accomplish this speed control an actuator rod 44 having an off-set or eccentric cam 43 thereon is inserted into the valve unit. The eccentric cam 43 fits into a slot 42 in one of the outer lands on the main control valve spool 7. When eccentric cam 43 is rotated it causes the main spool valve 7 to also rotate thereby causing the orifice defined by throttling slot 8 and return port 17 to vary. In addition, a sleeve 41 is shown in FIG. 3 and it is contemplated that either the sleeve or spool may be rotated to achieve this regulatory function.

The actual forming of the orifice between the throttling slot 8 and return port 17 can be better seen in FIG. 4. As depicted in FIG. 4, as the actuator 44 is rotated the throttling slot 8 and return port 17 are moved into and out of alignment, thus causing an increase or decrease in orifice size.

FIG. 5 is a cut-away view depicting the preferred form of the throttling slots 8 in the outer land of the main control valve spool 7. As can be seen in the figure the throttling slots have a rectangular cross section.

The velocity-time graphs in FIG. 6 illustrate typical cycles of the speed control device, and may be employed to better visualize the operation of the device. The first graph, FIG. 6*a*, represents a cycle during which the creep control pistons 12 are not employed. In this cycle there is an acceleration portion represented during the time t_1 during which the main control spool valve 7 is displaced. To displace the main control valve 7, the directional control solenoid valve 1 is actuated. Moving the directional control solenoid valve allows fluid under pressure from line 26 supplied by a pump (not shown) to enter one of the control piston chambers 14 through a supply, exhaust line 35. The other supply, exhaust line 35 is vented to the sump through line 29.

Fluid under pressure enters one of the control piston chambers 14 and causes one of the control pistons 11 to force the main control valve 7 into the open position. As the main control valve 7 is moved by the control piston 11, one end will enter a centering spring chamber 9. Centering spring chamber 9 is constantly filled with fluid under pressure, which is supplied through line 27 and check valve 31. The fluid is forced from one centering spring chamber and moves through line 30 and variable restriction 32 into the other centering spring chamber. Adjustment of the variable resistance 32 determines the speed of displacement of the main control spool valve 7 and thus the acceleration during the time t_1 represented in the graph of FIG. 6a. This result is accomplished, since the orifice defined between throttling slot 8 and return port 17 increases as the main control spool valve 7 is displaced, allowing a higher and higher volumetric rate of flow from the actuator 6 to return line 28.

The main control spool valve 7 will continue its movement until it reaches the stop on creep control piston 12. The creep control piston will not move, since, as shown in FIG. 1, the creep control solenoid valve 5 is in a position supplying pressure to both creep control piston chambers 13. Therefore, the maximum constant velocity represented in the graph of FIG. 6a is in fact the creep speed V_c or slow speed of the device.

When it is desired to stop the device, directional control solenoid valve 1 is moved back to its neutral position as shown in FIG. 1. Control piston chambers 14 are then both vented to exhaust line 29 and the force of centering springs 10 will return the main control valve 7 to its neutral position as shown in FIG. 1. The speed of the return will again be determined by the variable resistance 32 and an equal deceleration rate will result during time t_2 as shown in the graph in FIG. 6*a*. Should solenoids 2 fail to function at any time during the cycle, springs (not shown) in mechanical linkage 4 will return the directional control solenoid valve 1 to its neutral position as shown in FIG. 1. Thus, the centering springs will act as a fail safe mechanism to return the main control spool valve 7 to its neutral or off position. It should further be noted that during the period of maximum displacement of the main control spool valve 7 the orifice defined by the throttling slot 8 and return port 17 is constant, therefore, there is a constant pressure drop across the orifice and the rate of the actuator is controlled regardless of the load, since the volumetric rate of exhaust flow will be held constant by metering valves 18.

The graph in FIG. 6*b* illustrates a cycle during which the creep control pistons are employed. During this cycle the creep control solenoid valve 5 is displaced to supply fluid under pressure to one creep control chamber 13 and vent the other to the exhaust line 29. The directional control solenoid valve is then actuated to supply pressure fluid to the control piston chamber 14 to move the main control spool 7 in the direction of the

vented creep control chamber **13.** As the main control valve **7** is displaced a control piston **11** will again make contact with a creep control piston stop **12**; however, in this instance because the creep control piston is not biased by fluid in the creep control pressure chamber **13,** the creep control piston **12** will also be displaced. Again the rate of displacement of the main control spool valve **7** is controlled by the rate of displacement of fluid through the variable restrictor **32.**

Because the main control spool valve may now be displaced to its fullest extent, the size of the orifice defined by throttling slot **8** and return port **17** is at its greatest; and the rate of movement of the load will be at its highest velocity Vmax as shown in FIG. **6**b. This constant velocity during time t_4 may be maintained as long as the operator wishes or may be predetermined using various electrical switching mechanisms (not shown).

To initiate the deceleration cycle, the creep control solenoid valve **5** is again actuated and moved back into its neutral position as shown in FIG. **1.** Fluid under pressure is again supplied to both creep control piston chambers **13** and creep control pistons **12** are moved to toward the center of the central bore. This movement will displace the main control spool valve **7** toward the center of the bore, thus decreasing the orifice defined between throttling slot **8** and return port **17.** This movement will effect a reduction in the rate of flow able to exhaust from the actuator **6** to the return line **28,** thereby reducing the velocity of the load. The actuator will remain in this creep speed or slow speed position and the load will move at the creep speed v_c for any predetermined period of time. During this time another mechanism may pick up the load from the hydraulic actuator **6** much like a second man on a relay team receives the baton, the first having slowed down to pass it off. To stop the device the directional control solenoid valve is again moved to its neutral position venting the piston control chambers **14.** Again the biasing of the centering spring **10** will move the main control valve **7** back to its neutral position.

As shown in FIG. **3** the orifice defined between throttling slot and return port **17** may be varied; therefore, the maximum velocity as represented in graphs of FIG. **6** may also be varied over a broad range. This allows the device to be used for various purposes and to be adapted to various cycles of operations.

Obviously many modifications and variations of the present invention are possible in the light of the above teachings. It is therefore to be understood that within the scope of the appended claims the invention may be practiced otherwise than as specifically described.

What is claimed is:

1. The combination of a speed control mechanism and hydraulic motor comprising:

an hydraulic motor;

a valve body having a bore therein;

a main spool valve having a central land and two outer lands thereon mounted in said bore;

a pressure fluid supply port communicating with said bore and normally closed by the central land on said main spool valve;

control ports communicating with said bore between said central and said outer lands and connected to said hydraulic motor;

return ports communicating with said bore and closed by said outer lands when said central land closes said pressure fluid supply port;

throttling slots in each outer land;

first actuator means for displacing said main spool valve from its position blocking said pressure fluid supply port and thereby allowing the pressure fluid supply port to communicate with a first control port to deliver fluid under pressure to said hydraulic motor, and allowing a second control port to communicate with a second return port through a throttling slot in one of said outer lands, said throttling slot and said second return port defining an orifice through which fluid returning from said hydraulic motor must pass;

cam means for rotating the main spool relative to said bore to adjust the size of the orifice defined by said throttling slot and said second return port when said main spool valve is displaced; and

pressure-responsive, metering valve means connected to each of said return ports and responsive to the pressure of fluid returning from said hydraulic motor and fluid passing through said orifice to control the maximum volumetric rate of flow through said return ports from said hydraulic motor.

2. The combination of claim **1,** further comprising pressure-actuated, creep speed control means for limiting the maximum displacement of said main spool valve and a second actuator means for controlling the creep speed control means.

3. The combination of claim **2,** wherein said creep speed control means comprises a first creep control piston mounted in said bore on one side of said main spool valve and a second creep control piston mounted in said bore on the other side of said main spool valve.

4. The combination of claim **1,** wherein said cam means comprises a cam slot in one of the outer lands on said main spool valve, an eccentric cam cooperatively engaging said cam slot, and an actuator rod connected to said cam whereby movement of said actuator rod will displace said cam and thereby cause the main spool valve to rotate in said bore.

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