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[54] SPEED CONTROL SYSTEM FOR HYDRAULIC MOTOR

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[52] U.S. Cl **60/393,** 60/468, 60/384

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

[58] **Field of Search**.....60/53 WW, 53 R

[56] **References Cited**

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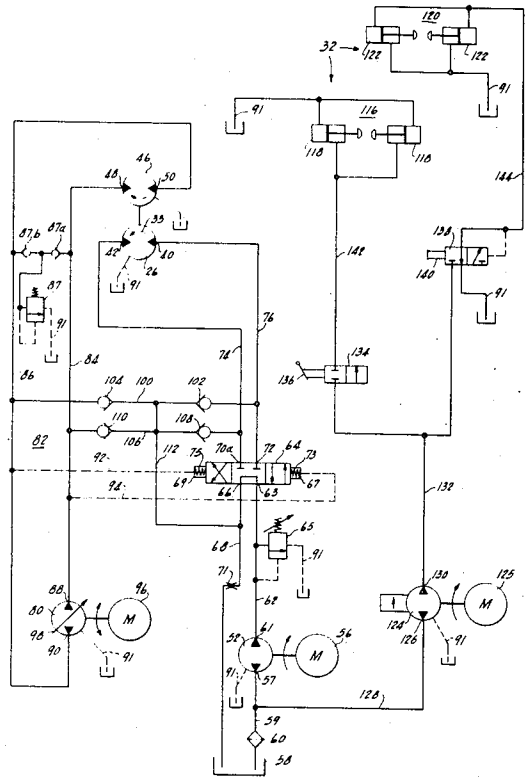
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[57] **ABSTRACT**

Several embodiments of a speed control system for a hydraulic hoist motor are disclosed. The speed control means comprise a main hydraulic pump for driving the hoist motor; regulating means to vary the direction of flow and volume of fluid supplied to the hoist motor to thereby regulate motor direction and speed, respectively; and selectively operable operating means to actuate the regulating means. The operating means comprise a speed sensing hydraulic motor responsive to hoist motor speed. The operating means further comprise a manually or remotely controlled over-center pump for supplying a predetermined volume of fluid to the speed sensing motor through a control circuit to obtain a predetermined operating speed for the hoist motor. The fluid pressure in the control circuit theoretically is constant for any predetermined hoist motor speed but actually varies in proportion to changes in hoist motor speed (caused by load changes) to effect constant motor speed. In one embodiment, the main pump is a constant volume pump and it s output (and therefore hoist motor speed) is regulated by a pilot operated multi-position directional valve responsive to fluid pressure in the control circuit. In another embodiment, the main pump is a variable volume pump having its output (therefore hoist motor speed) regulated by a servo-stem device movable by a hydraulic actuator which, in turn, is responsive to fluid pressure in the control circuit.

16 Claims, 8 Drawing Figures



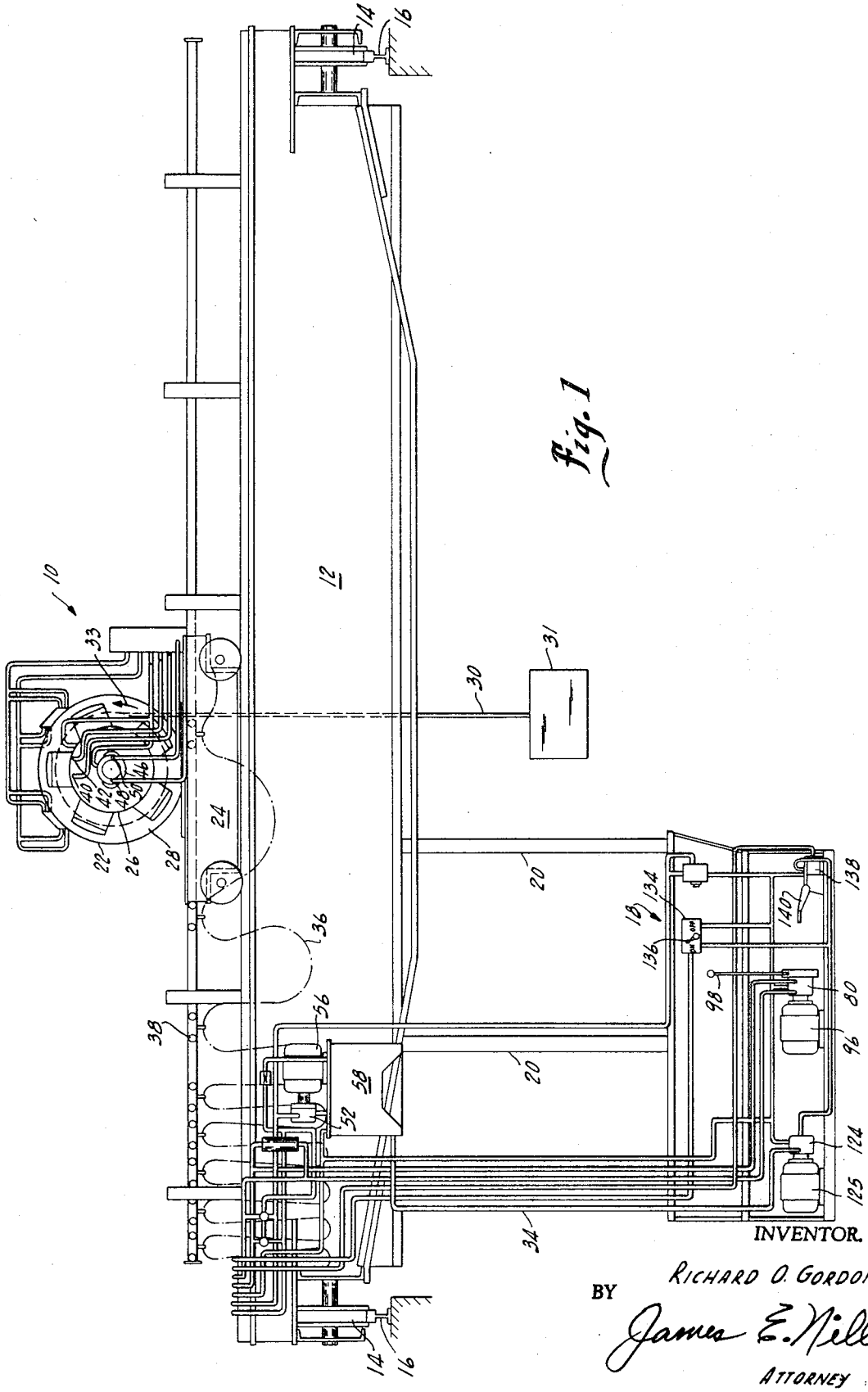


Fig. 1

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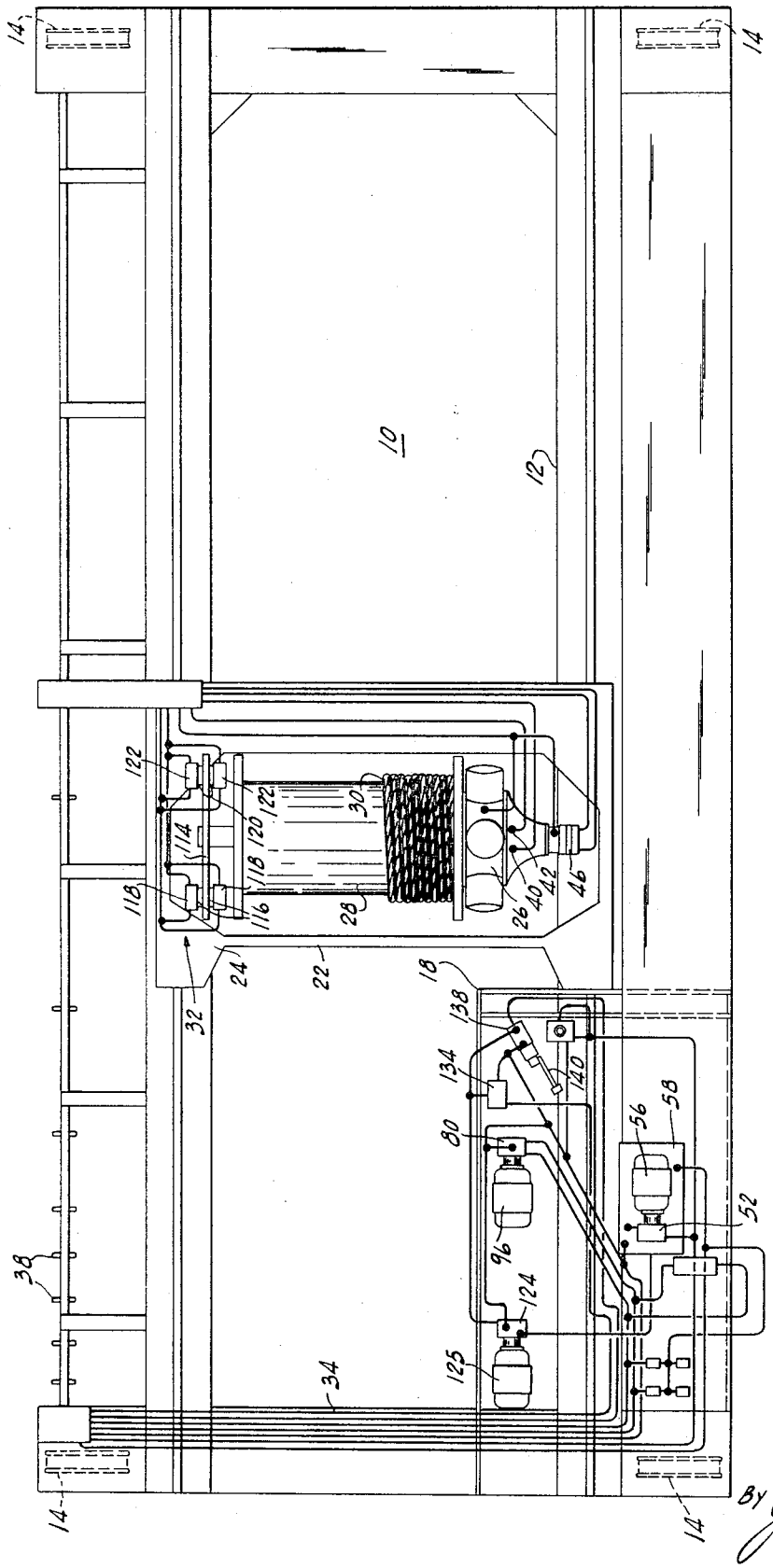


Fig. 2

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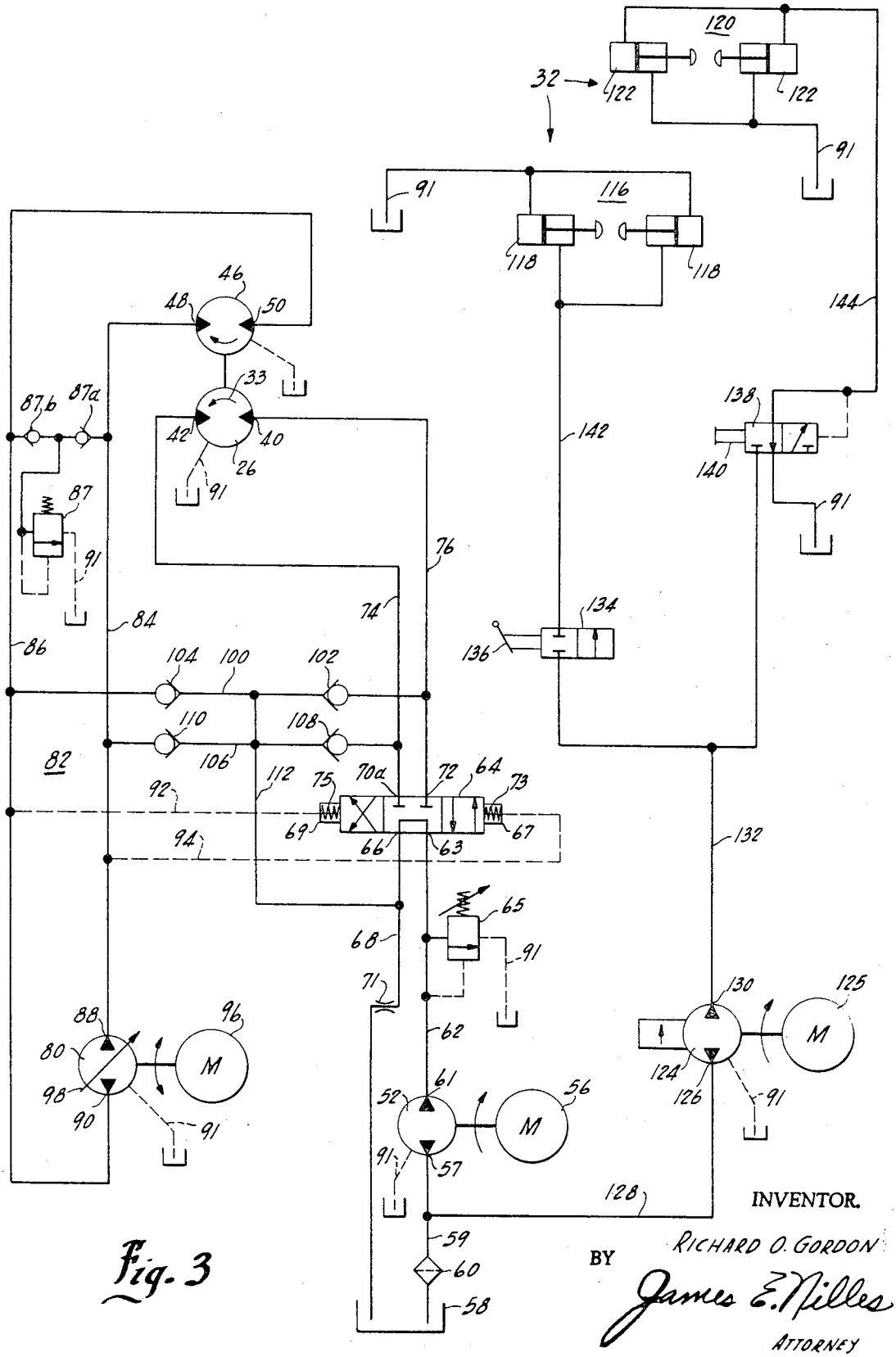


Fig. 3

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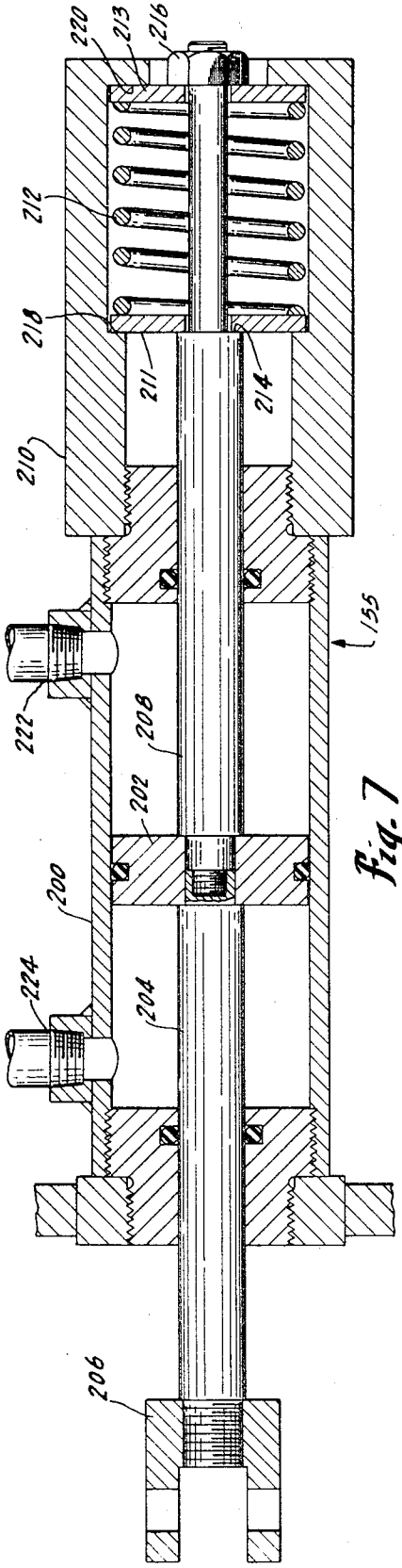


Fig. 7

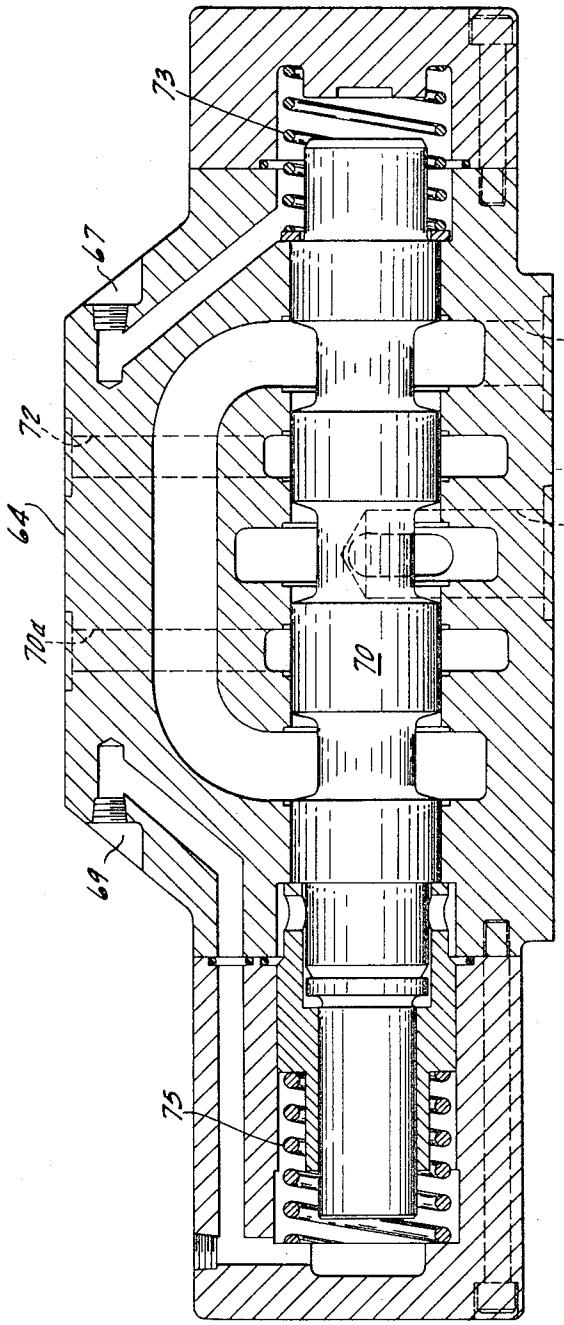


Fig. 4

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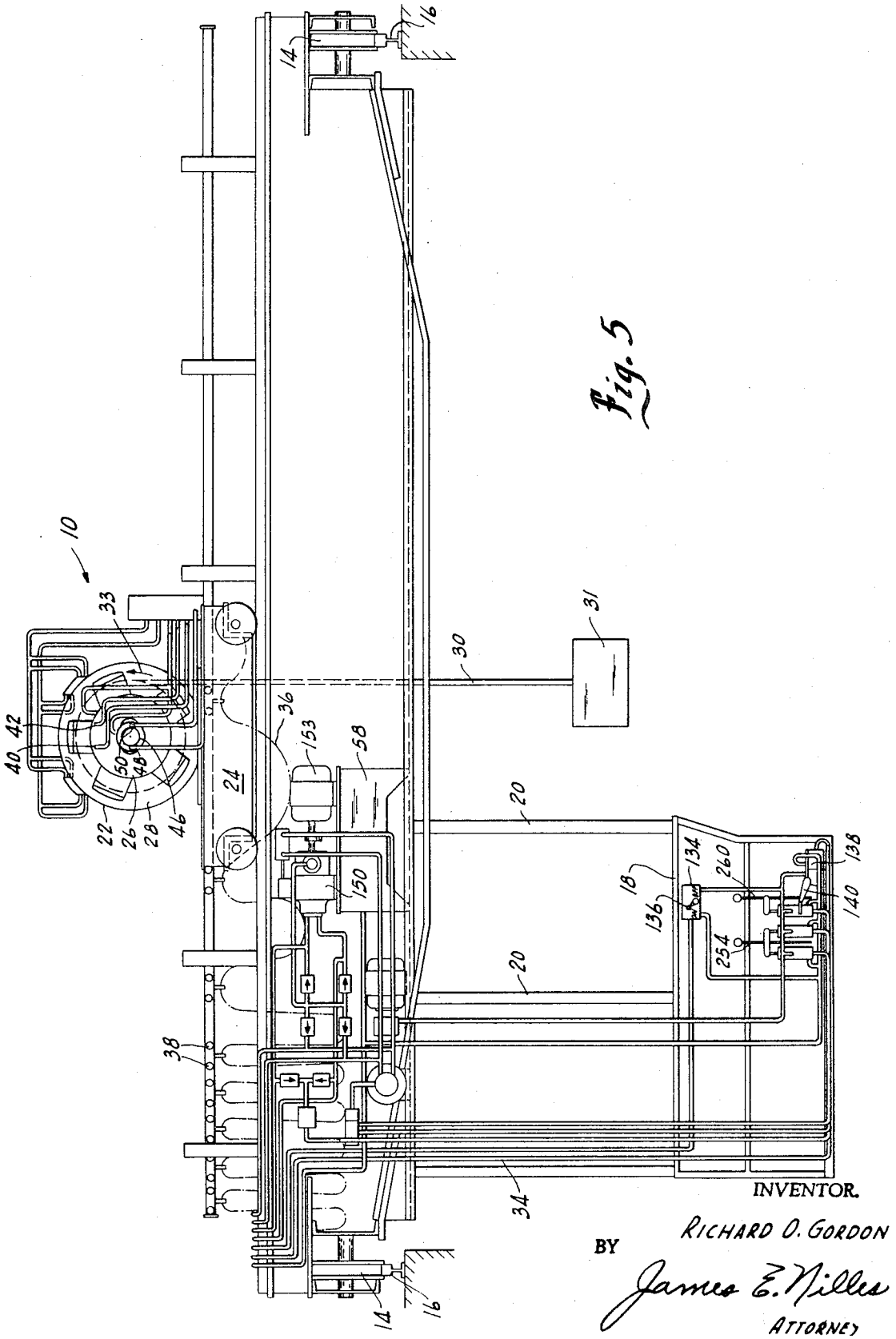


Fig. 5

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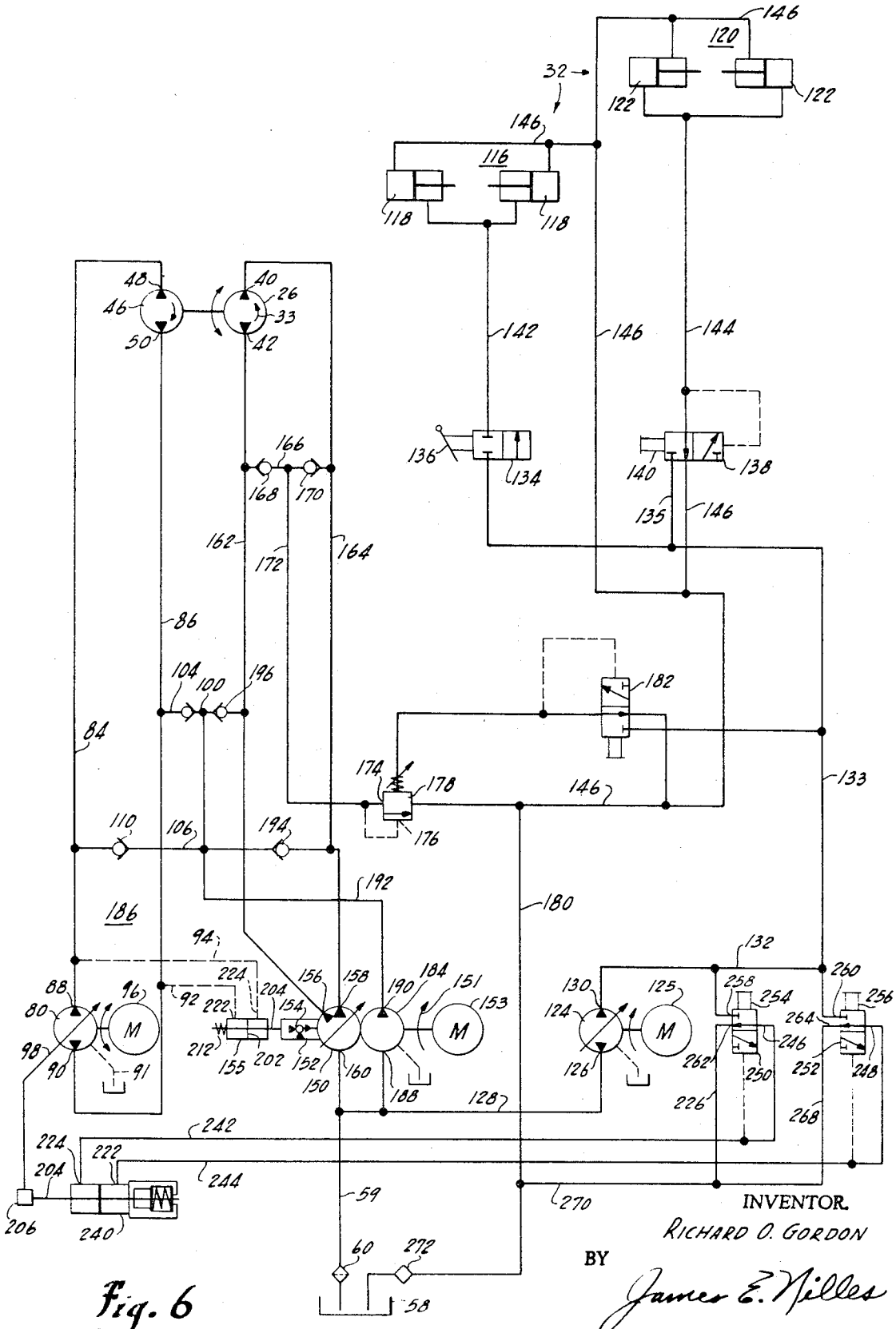


Fig. 6

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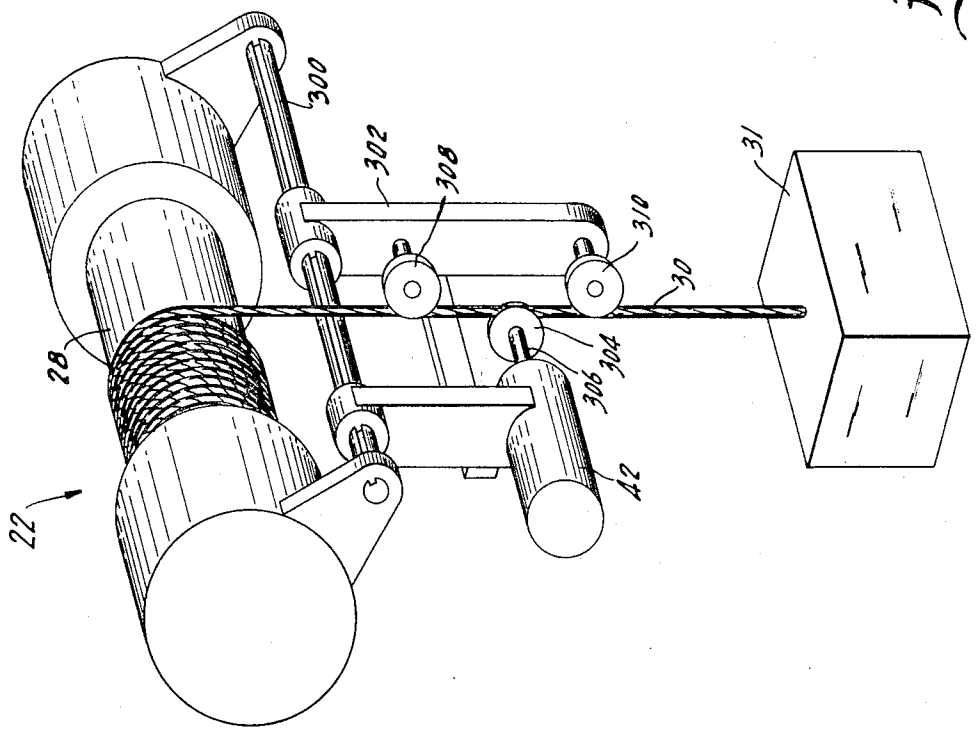


Fig. 8

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SPEED CONTROL SYSTEM FOR HYDRAULIC MOTOR

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates generally to hydraulic control systems for hydraulic motors. In particular it relates to control systems which regulate motor direction and speed and maintain some predetermined constant speed despite variations in load imposed on the motor. The invention is disclosed as applied to hydraulic hoist motors.

2. Description of the Prior Art

Some winches, hoists and similar apparatus, for example, are driven by a hydraulic hoist motor which is supplied with hydraulic fluid from a hydraulic pump driven by an electric motor. It is sometimes desirable that the hoist motor operate at a constant predetermined speed or remain stationary despite different loads imposed thereon. This can be accomplished by controlling the volume of hydraulic fluid (assuming constant pressure) supplied to the hoist motor or by controlled braking of the hoist. While prior art systems of various types using either or both of these principles are available to effect constant speed regulation of hydraulic motors and are suitable for their intended purposes, it is desirable to provide new and improved hydraulic control means or systems which have many advantages over prior art systems.

SUMMARY OF THE INVENTION

Speed control system or means in accordance with the invention are used to operate a main hydraulic motor (such as a hoist motor) in a desired direction of rotation and to maintain motor speed constant at some predetermined level despite variations in the load on the motor. The control means comprises a main hydraulic pump for driving the main motor; regulating means actuable to vary the flow direction and volume of fluid supplied to the main motor to regulate motor rotation direction and speed; and manual or remotely controlled operating means to actuate the regulating means. The operating means comprises a speed sensing hydraulic motor responsive to main motor speed and manually or remotely controlled over-center pump means to supply a predetermined volume of fluid to the speed sensing motor through a control circuit to establish a predetermined operating speed for the main motor. Fluid pressure in the control circuit theoretically is constant for a predetermined main motor speed but actually varies in proportion to changes in main motor speed. Pressure in the control circuit is sensed and used to actuate the regulating means to maintain main motor speed constant despite variations in load imposed thereon. Pressure in the control circuit is also used to establish the direction of motor rotation.

In one embodiment, the regulating means comprise a constant volume main pump having its output to the main motor regulated by a pilot operated multiposition directional valve which is responsive to fluid pressure in the control circuit. In a second embodiment, the regulating means comprise a variable volume main pump having its output and direction of fluid flow to the main motor regulated by a movable servo-stem device on the main pump. The servo-stem device is moved by a hydraulic actuator which is responsive to fluid pressure in the control circuit.

The over-center pump means comprises a variable volume over-center pump driven by a constant speed electric motor and having a multiposition movable control member (manually controlled or remotely controlled by a hydraulic actuator) for controlling direction and volume of fluid flow therefrom. The aforesaid hydraulic actuator is operated by two "joy-stick" type manually controlled valves. If preferred in specific applications, the over-center pump means may comprise a constant volume over-center pump driven by a constant speed electric motor and having a multiposition variable volume control valve (with a movable control member) for controlling direction and volume of fluid flow to the speed sensing motor. The control member of the control valve could be manually movable or else remotely controlled by the spring-centered piston of a hydraulic actuator as described hereinbefore. Or, if preferred, the over-center pump means could comprise a constant volume pump driven by a stepping motor (i.e., one remotely controllable to effect a predetermined number of revolutions of the constant volume pump in either direction of rotation).

The speed sensing motor of the operating means may be arranged to respond directly to the speed of rotation of the main motor to provide constant motor speed under changing load conditions. Or, in the case of a hoist driven by the main motor, the speed sensing motor may respond directly to the speed of the hoist line (i.e., winch speed) to maintain this speed constant despite changing load conditions by means of regulating the speed of the main motor.

Another advantage of the present invention is that the main motor may be stopped and held in a predetermined position, despite the application of a load thereon and despite variations in this load, without reliance on conventional shoe-or disc-type brakes.

Furthermore, the speed control mans in accordance with the present invention are infinitely variable (as distinguished from incrementally variable) from zero to maximum speed.

Finally, while the present invention is disclosed herein as being applied to hydraulic motors for hoists, it could be used to regulate hydraulic motors used for vehicle propulsion, steering systems, braking systems or other purposes.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an elevational view of an overhead crane having a main hydraulic hoist motor which employs speed control means in accordance with one embodiment of the present invention;

FIG. 2 is a top plan view of the crane shown in FIG. 1;

FIG. 3 is a schematic view of the speed control means for the crane shown in FIGS. 1 and 2;

FIG. 4 is an enlarged cross-sectional view of a multiposition valve shown in FIG. 3;

FIG. 5 is an elevational view of an overhead crane having a main hydraulic hoist motor which employs speed control means in accordance with a second embodiment of the present invention;

FIG. 6 is a schematic view of the speed control means for the crane shown in FIG. 4;

FIG. 7 is an enlarged cross-sectional view of an actuator shown in FIG. 6;

FIG. 8 is an isometric view of a modification to the type of hoists shown in FIGS. 1 and 4 and shows means

for connecting a speed sensing motor to respond directly to hoist line speed.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIGS. **1, 2** and **5** of the drawings, the numeral **10** generally designates an overhead crane with which hydraulic motor speed control means in accordance with the present invention are advantageously employed. Crane **10** comprises a bridge member **12** having wheels **14** at both ends which adapt it for movement along rails **16** on which it is supported. A crane operator's control station or cab **18** is suspended from bridge member **12** by rigid supporting means **20.** A hoist **22** is mounted on bridge member **12** and is adapted for movement therealong by means of a wheeled carriage **24.** Hoist **22** comprises a main hydraulic motor **26,** a hoist drum **28** rotatable in either direction by the main motor, a hoist line or rope **30** wrapped on the hoist drum and supporting load **31,** and brake means **32,** shown in FIG. **2,** for the hoist drum. Motor **26** is, for example, a conventional high torque, low speed, radial piston-type rotary motor. Various operating controls and components, hereinafter described, for hoist **22** are located on bridge member **12** and on cab **18** within reach of the crane operator and are connected to the hoist by means of rigid hydraulic fluid lines, generally designated **34,** by means of a festoon system, generally designated **36,** comprising flexible fluid lines supported on bridge member **12** by slidably movable festoon supports **38.** It is to be understood that hoist **22** is intended to raise and lower load **31** at a constant predetermined speed. Therefore, hoist motor **26** is operable in forward (line wrapping) direction (arrow **33** in FIG. **5**) to hoist load **31** and in reverse (line unwrapping) direction to lower the load. Motor **26** is provided with hydraulic fluid ports **40** and **42** which are connected to hydraulic fluid lines hereinafter described. In accordance with the invention, motor **26** is understood to be adapted by means of suitable gearing to drive a small speed-sensing hydraulic motor **46** in the opposite direction as, but at a speed proportional to, that of motor **26.** Speed-sensing motor **46** is, for example, a conventional piston-type motor. Motor **46** is provided with hydraulic fluid ports **48** and **50** which are connected to hydraulic fluid lines hereinafter described.

The speed control means for the hoist **22** shown in FIGS. **1** and **2** is disclosed in FIG. **3** and that for the hoist **22** shown in FIG. **5** is disclosed in FIG. **6.** However, in the drawings, which show different embodiments and modifications thereof, the same reference numerals are used to designate identical components.

As hereinafter more particularly described, the speed control means for each embodiment of the invention comprises a main hydraulic pump means for driving hoist motor **26;** regulating means to vary the direction of flow and volume of fluid supplied from the main pump means to the hoist motor to thereby regulate motor direction and speed; and selectively operable operating means, including speed sensing motor **46,** to actuate or control the regulating means.

First Embodiment

FIG. **3** is a schematic diagram of a speed control means in accordance with a first embodiment of the invention. The control means comprises a main hydraulic pump **52** for driving hoist motor **26.** Pump **52** is, for example, a conventional, fixed piston, constant speed, constant volume pump. Pump **52** is driven in one direction by a conventional constant speed electric motor **56.** A fluid intake port **57** for pump **52** is supplied with hydraulic fluid from a reservoir **58** by a hydraulic fluid line **59** having a filter **60.** Pump **52,** and other components hereinafter described, are provided with a drain, to which they are connected by a drain line.

A fluid output or pressure port **61** of pump **52** is connected through a fluid line **62** to a fluid intake port **63** of a conventional pilot-operated, multi-position directional valve **64,** hereinafter described. A fluid return port **66** of directional valve **64** is connected by a fluid return line **68** to reservoir **58.** Line **68** contains a back-pressure orifice **71.**

A conventional adjustable pressure relief valve **65** is connected to fluid line **62** to limit the maximum fluid pressure supplied from pump **52** to motor **26** and thereby limit the maximum torque motor **26** can exert when lifting load **31.** Valve **65** is adjustable to accommodate loads of various weight. Valve **65** is normally closed but opens at a predetermined pressure level to relieve pressure in line **62** and divert fluid flow through its drain line **91** to reservoir **58.**

Directional valve **64,** shown in detail in FIG. **4,** is part of the regulating means which varies the direction of flow and volume of fluid supplied to hoist motor **26** from main pump **52** to thereby regulate motor direction and speed, respectively. Directional valve **64** comprises fluid ports **70***a* and **72** which are connected by fluid lines **74** and **76,** respectively, to the fluid ports **42** and **40,** respectively, of hoist motor **26.** Direction valve **64** is a three-position valve which is controlled by application of pressurized hydraulic fluid to its pilot ports **67** and **69** to move a valve spool **70,** which is movably maintained in neutral position by biasing springs. When pressure at pilot ports **67** and **69** is equal, movable spool **70** in valve **64** assumes a neutral position (shown in FIGS. **3** and **4**) wherein pressure line **62** from pump **52** is connected directly to return line **68.** When pressure at pilot port **67** is greater than at pilot port **69,** valve spool **70** moves from neutral to the left (with respect to FIGS. **3** and **4**) to hoist position wherein ports **40** and **42** of motor **26** are connected to pressure line **62** and return line **68,** respectively, and motor **26** is pressurized for rotation in the hoist direction. When pressure at pilot port **69** is greater than at port **67,** valve spool **70** moves from neutral to the right (with respect to FIGS. **3** and **4**) to lowering position wherein ports **40** and **42** of motor **26** are connected to return line **68** and pressure line **62,** respectively, and motor **26** is pressurized for rotation in the lowering direction. It is to be understood that the amount of fluid pressure applied at the pilot ports of directional valve **64** determines the extent of movement of valve spool **70** and, therefore, regulates the volume of fluid able to pass through the directional valve and thus control motor speed as well as direction.

Selectively operable operating means in the form of a rotary servo-system are provided to actuate or operate directional valve **64** of the regulating means to control the direction of rotation and speed of hoist motor **26.** Such operating means comprises speed sensing motor **46,** hereinbefore referred to, an over-center pump **80** and an associated control circuit **82.** The fluid ports **48** and **50** of speed sensing motor **46** are con-

nected by fluid lines **84** and **86**, respectively, to fluid ports **88** and **90**, respectively, of over-center pump **80**. Fluid lines **84** and **86** are connected by hydraulic fluid pilot lines **92** and **94**, respectively, to pilot ports **69** and **67**, respectively, of directional valve **64**.

A conventional pressure relief valve **87** is connected to fluid lines **84** and **86** through check valves **87a** and **87b**, respectively, poled as shown in FIG. **3**, to limit fluid pressure therein to a predetermined upper level.

An anticavitation and fluid supply system is provided for control circuit **82** and comprises a fluid line **100** connected between fluid lines **76** and **86** and containing check valves **102** and **104**; a fluid line **106** connected between fluid lines **74** and **84** and containing check valves **108** and **110**; and a fluid line **112** connecting return line **68** to both lines **100** and **106** at points between their respective check valves. Operation of the anticavitation and fluid supply system is hereinafter described. Pump **80** is, for example, a variable volume piston-type pump which is driven by a conventional constant speed motor **96**. Furthermore, pump **80** is provided with a manual control lever **98** which is selectively movable in either direction from a neutral position (wherein no fluid pressure is supplied from its ports **88** and **90**) toward either a hoist position (wherein fluid pressure is provided at port **88** of pump **80** to operate directional valve **64** to hoist position) or a lower position (wherein fluid pressure is provided at port **90** of pump **80** to operate direction valve **64** to lower position). The degree of movement of control lever **98** of pump **80** in either direction regulates the volume of fluid supplied by pump **80** to speed sensing motor **46** through control circuit **82** and also to directional valve **64**.

Speed sensing motor **46**, over-center pump **80**, directional valve **64** and control circuit **82** cooperate to provide directional and speed control for hoist motor **26** in the following manner. For example, at a predetermined speed of rotation of hoist motor **26** (i.e., 200 r.p.m.), speed sensing motor **46**, which is driven thereby (in the opposite direction), is only able to accommodate passage of a predetermined volume of hydraulic fluid (i.e., 2 gallons per minute) from overcenter pump **80**. If the volume of fluid flow from pump **80** is increased by appropriate movement of its control lever **98**, fluid pressure builds up in either line **84** or line **86** (depending on direction of control lever movement) because speed sensing motor **46** cannot immediately accommodate this increase. As a result, spool **70** of directional valve **64** is moved in an appropriate direction to thereby change the volume, for example, of the fluid flow between main pump **52** and hoist motor **26**. This, in turn, has the effect of causing a change in the speed of rotation of hoist motor **26** and, consequently, the speed of sensing motor **46**. In effect, then sensing motor **46** is brought up to a speed level where it can accommodate the increased volume of fluid supplied thereto from pump **80** and fluid pressure in either line **84** or **86** decreases. A more detailed description of the manner in which the direction of rotation and speed of hoist motor **26** are controlled is provided hereinafter.

The brake means **32** for hoist **22** are shown in FIG. **2** as comprising a disc brake **114** on hoist **22**; a locking brake comprising a first set of brake shoes **116** which are understood to be spring applied and hydraulically released by pressurization of a first set of hydraulic brake cylinders **118**; and an operating brake comprising second set of brake shoes **120** which are understood to be spring released and hydraulically applied by pressurization of a second set of hydraulic brake cylinders **122**.

As FIG. **3** shows, the cylinders **118** and **122** are supplied with pressurized fluid from a conventional pressure-compensated piston type pump **124**, driven by a conventional constant speed electric motor **125**. Pump **124** has its inlet port **126** connected by a fluid line **128** to fluid line **59** and, thus, to the fluid in reservoir **58**. The outlet or pressure port **130** of pump **124** is connected by a fluid line **132** to an on-off valve **134**, having a manual control lever **136**, which controls pressurization of the first set of cylinders **118**. The outlet or pressure port **130** of pump **124** is also connected by fluid line **132** to a brake valve **138**, having a pedal type control lever **140**, which controls pressurization of the second set of cylinders **122**. Valves **134** and **138** are connected to the cylinders **118** and **122**, respectively, by fluid lines **142** and **144**, respectively. The cylinders **118** and **122** and valve **138** are provided with drain lines **91**.

Valve **134**, shown in "brake applied" position, is manually movable to "brake release" position wherein it pressurizes cylinder **118** which then acts against biasing spring means to release the brake shoes **116** from disc brake **114**.

Valve **138**, shown in "brake released" position is pedal-operated to "brake applied" position wherein it pressurizes cylinder **122** which then acts against biasing spring means to apply the brake shoes **120** against disc brake **114**.

The embodiment of the invention disclosed in FIGS. **1**, **2** and **3** operates as follows:

Assume motor **26** of hoist **22** is stationary and is to be operated to hoist load **31** at some predetermined desired constant rate of speed. Further assume that brake means **32** are released and that drum **28** is free to rotate. Also assume that main pump **52** and over-center pump **80** are in operation but that control lever **98** of the over-center pump and, therefore, directional valve **64** are in neutral position. With these assumptions, hoist motor **26** and, therefore, speed sensing motor **46** are stationary. All references to direction of movement hereinafter made in connection with the description of operation of the first embodiment of the invention are to be understood to be with respect to FIG. **3**.

To hoist load **31**, manual control lever **98** of over-center pump **80** is moved leftward from neutral position toward hoist position to pressurize line **84**. The extent of leftward movement of lever **98** determines the volume of fluid flow from over-center pump **80** and, thus establishes the predetermined rate of speed for hoist **22**. Since hoist motor **26** and speed sensing motor **46** are stationary, fluid pressure builds up in the lines **84** and **94** and pressure at pilot port **67** of directional valve **64** exceeds that at pilot port **69**. Consequently, valve spool **70** of directional valve **64** moves leftward from neutral toward hoist position for a distance proportional to the pilot pressure applied thereto, and this pressure level is determined by how far lever **98** is moved. Fluid then begins to flow from pressure port **61** of main pump **52**, through line **62**, through ports **63** and **72** of directional valve **64**, line **76**, ports **40** and **42** of hoist motor **26**, return line **74**, through ports **70a** and **66** of directional valve **64**, and return line **68** to reser-

voir 58. Such fluid flow causes rotation of hoist motor 26 in the hoist direction and also corresponding rotation of speed sensing motor 46. The speed of rotation of motors 26 and 46 increases until the predetermined speed of rotation is reached. As the predetermined rotational speed is approached, fluid pressure in lines 84 and 94 decreases from its initial level because speed sensing motor 46 is now rotating and accelerating. This decrease in pressure effects corresponding rightward movement of valve spool 70 in directional valve 64 and a proportional decrease in flow of fluid to hoist motor 26 from main pump 52. At the predetermined rotational speed of motor 26 fluid flow through directional valve 64 stabilizes and remains constant because speed sensing motor 46 is rotating at a speed sufficient to accommodate the predetermined volume of fluid supplied thereto by over-center pump 80. In stabilized condition, a pressure differential exists between lines 84 and 86 (and consequently between pilot ports 67 and 69) which maintains spool 70 of directional valve 64 at some point in hoist position.

As load 31 is hoisted at a constant predetermined speed by hoist 22, cable 30 is wrapped on hoist drum 28. However, as one layer of cable starts to build upon another, the apparent effect as regards load 31 is that of an increase in drum diameter and, consequently, hoist motor 26 and speed sensing motor 48 begin to slow down under the increased load. As speed sensing motor 46 slows down, the volume of fluid it can accommodate decreases and pressure in the lines 84 and 94 increases. This causes an increase in pressure at pilot port 67 of directional valve 64, and valve spool 70 moves farther leftward to increase fluid flow from main pump 52 to hoist motor 26 to thereby increase the speed of the latter until it (and speed sensing motor 46) return to the predetermined speed. When this occurs, the situation again stabilizes until further readjustment is required.

If manual control lever 98 of over-center pump 80 is moved partway (but not fully) toward neutral position, a new slower predetermined rate of hoist speed is established but operation is the same as hereinbefore described.

If manual control lever 98 is returned to neutral position, fluid pressure in the lines 84 and 86 equalizes and directional valve 64 returns to neutral position, and the motors 26 and 46 slow down and stop. Load 31 will remain suspended, even though brake means 32 are not applied, because return flow of fluid from both ports 40 and 42 of motor 26 is blocked by directional valve 64. Since hoist motor 26 is a piston-type motor, very little fluid leakage occurs therefrom. However, since some leakage does occur, fluid pressure in line 74 tends to decrease and fluid pressure in line 76 increases so load 31 would normally tend to descend at a very slow rate. In accordance with the present invention, however, pressure increase in line 76 and pressure loss in line 74 is compensated for by supplying a small amount of pressurized fluid thereto from pressure port 61 of pump 52 (which is still in operation), through line 62, valve ports 63 and 66 of directional valve 64, line 112 and check valve 108. Simultaneously, speed sensing motor 46 tends to rotate and transfer fluid from its port 50 to port 48 thereby causing a pressure increase in lines 84 and 94 to shift valve spool 70 leftward to admit fluid from its port 63 to its port 72 to thereby rotate main motor 26 in the forward or hoist direction. Spool 70 will remain in this shifted position to continue to supply a volume of fluid flow to motor 26 equal to the leakage from main motor 26. In this manner, slow descent or slippage of load 31 is prevented even though manual control lever 98 of overcenter pump 80 is in neutral position and brake means 32 are not applied.

To lower load 31 at some predetermined speed, manual control lever 98 of over-center pump 80 is moved rightward from neutral position toward the lowering position to pressurize line 86 at a predetermined pressure level. Fluid pressure builds up in the lines 86 and 92 and pressure at pilot port 69 of directional valve 64 exceeds that at pilot port 67. Consequently, valve spool 70 moves from neutral rightward toward lowering position for a distance proportional to the pilot pressure applied thereto. Again, the extent of movement of lever 98 determines the volume of fluid flow from over-center pump 80 and thus establishes a predetermined rate of speed for hoist 22 in the lowering direction. Fluid then begins to flow from pressure port 61 of main pump 52, through line 62, through ports 63 and 70a of directional valve 64, line 74, through port 42 to port 40 of hoist motor 26, line 76, through ports 72 and 66 of directional valve 64, and through return line 68 to reservoir 58. Such fluid flow causes rotation of hoist motor 26 in the lowering direction and also causes corresponding (but opposite) rotation of speed sensing motor 46. The speed of rotation of motors 26 and 46 increases until the predetermined speed of rotation is reached. As the predetermined rotational speed is approached, fluid pressure in lines 86 and 92 decreases from its initial level because speed sensing motor 46 is now rotating and accellerating. This decrease in pressure effects a corresponding leftward movement of valve spool 70 in directional valve 64 and a proportional decrease in flow of fluid to hoist motor 26 from main pump 52. Fluid flow through directional valve 64 would tend to stabilize and remain constant when speed sensing motor 46 is rotating at a speed sufficient to accommodate the predetermined volume of fluid supplied thereto by over-center pump 80, except for the fact that hoist motor 26 is operating in the lowering direction and the weight of load 31 tends to increase hoist motor speed. Therefore, an overspeed condition arises wherein speed sensing motor 46 begins to operate faster than over-center pump 80 is supplying fluid thereto. When the overspeed condition occurs, fluid pressure in lines 86 and 92 decreases toward, to or below the fluid pressure level in the lines 84 and 94. As a result, pressure at pilot port 69 decreases toward, to or below that at pilot port 67 and valve spool 70 moves from rightward position toward or to neutral position, or even beyond that, to some degree of leftward position. As spool 70 moves toward neutral position, the volume of fluid applied to port 42 of hoist motor 26 decreases and the hoist motor slows down. This effects corresponding slowdown of rotation of speed sensing motor 46. However, if this slowdown is still insufficient to maintain a constant lowering speed of hoist motor 26, valve spool 70 is moved even further leftward to neutral position to thereby entirely cut off fluid flow to port 42 of motor 26. Again, if this does not effect sufficiently rapid reduction in speed, valve spool 70 moves from neutral position to some degree of leftward position whereby fluid is directed by directional valve 64 to port 40 of hoist motor 26 instead of to port 42, to oppose the rotation of hoist motor 26. Such repositioning

of valve spool **70** in response to fluid pressure conditions in lines **86** and **94** and lines **84** and **92** is quite rapid and therefore effects constant speed control of hoist motor **26.**

Second Embodiment

FIG. **6** is a schematic diagram of a speed control means in accordance with a second embodiment of the invention. The control means comprises a main hydraulic pump **150,** driven in the direction of arrow **151** by a conventional constant speed electric motor **153,** for driving hoist motor **26.** Pump **150** is, for example, a conventional variable volume piston type pump having a servo-stem speed control valve **152** thereon having a servo-stem **154** which is linearly movable in either direction from a neutral position, shown in FIG. **6,** by means of a hydraulic actuator **155** to effect pressurization of either pressure port **156** or **158** of the pump as hereinafter explained. The extent of linear movement of stem **154** in a given direction controls the amount of fluid volume at a given fluid port. A fluid intake port **160** on pump **150** is supplied with hydraulic fluid from reservoir **58** by a hydraulic fluid line **59** having a filter **60.**

The fluid ports **156** and **158** of main pump **150** are connected through hydraulic fluid lines **162** and **164,** respectively, to the fluid ports **42** and **40,** respectively, of main motor **26.** The lines **162** and **164** are connected by a fluid line **166** containing conventional check valves **168** and **179,** poled as shown in FIG. **6.** A point in line **166** between these check valves is connected by fluid line **172** to an intake port **174** of a conventional adjustable pressure relief valve **176.** A pressure relief port **178** of relief valve **176** is connected by a drain line **180** to reservoir **58.** Pressure relief valve **176** limits the maximum fluid pressure supplied from pump **150** to motor **26** and thereby limits the maximum torque motor **26** can exert when lifting load **31.** Valve **176** is adjustable by means of a remotely controlled load control valve **182,** hereinafter described, to accommodate loads of various weight. Valve **176** is normally closed but opens at a predetermined pressure level to relieve pressure in line **172** and divert fluid flow through line **180** to reservoir **58.**

Electric motor **153** also drives a conventional hydraulic pump **184** which supplies fluid to charge, when necessary, both the hydraulic circuit for operating hoist motor **26** and a hydraulic control circuit **186,** hereinafter described. An intake port **188** of charging pump **184** is connected to reservoir **58** by line **59.** A pressure port **190** of charging pump **184** is connected through conventional check valves **194** and **196,** poled as shown in FIG. **6,** to the fluid lines **164** and **162,** respectively. Pressure port **190** of charging pump **184** is also connected through fluid line **192** to control circuit **186,** as hereinafter described.

Servo-stem speed control valve **152** on main pump **150** and its hydraulic actuator **155** are part of the regulating means which varies the direction of flow and volume of fluid supplied to hoist motor **26** from main pump **156** to thereby regulate motor direction and speed respectively. Assume, for example, that rightward or leftward linear movement (with respect to FIG. **6**), of servo-stem **154** of speed control valve **152** on main pump **150** effected by actuator **155** causes pressurization of the ports **40** and **42,** respectively, of hoist motor **26** and, therefore, effects operation of the hoist motor in the hoist or lowering direction, respectively.

Actuator **155,** shown in detail in FIG. **7,** comprises a hydraulic cylinder **200** having a spring centered piston **202** disposed therein. Piston **202** is connected to a piston rod **204** which extends from one end of cylinder **200** and is connectable by a yoke **206** to servo-stem **154.** Piston **202** is also connected to another piston rod **208** which extends from the other end of cylinder **200** into a spring housing **210** attached to the other end of cylinder **200.** A compression spring **212** is disposed in housing **210** between a pair of washers or discs **211** and **213** which are slideably mounted on piston rod **208.** Piston rod **208** is provided with a pair of spaced apart abutments thereon, in the form of a shoulder **214** and a stop nut **216.** Linear travel of the discs **211** and **213** with respect to housing **210** is limited by shoulder **218** and **220,** respectively, integrally formed in spring housing **210.** Hydraulic fluid ports **222** and **224** are provided in cylinder **200** on opposite sides of piston **202.**

When fluid pressure at the ports **222** and **224** is equal, piston **202** is maintained in a neutral or spring-centered position shown in FIGS. **6** and **7** by the action of spring **212** and maintains servo-stem **154** in a corresponding neutral position whereby there is no fluid flow from the ports **156** and **158** of main pump **150** to operate hoist motor **26.** However, when pressurized fluid is supplied to either port **222** or **224** of actuator cylinder **200,** piston **202** moves in the appropriate direction against the bias of spring **212** and effects corresponding movement of servo-stem **154.** More specifically, pressurization of port **224,** for example, causes rightward movement (with respect to FIG. **7**) of piston **202** and its associated rods **204** and **208.** As rod **208** moves rightward, its shoulder **214** bears against disc **211** which, in turn compresses spring **212.** When pressure at port **224** is relieved, spring **212** causes piston **202** to return toward neutral position. Pressurization of port **222** of actuator **155** causes leftward movement (with respect to FIG. **7**) of piston **202** and the rods **204** and **208.** As rod **208** moves leftward, stop nut **216** bears against disc **213** which in turn, compresses spring **212.** Actuator **155** is designed, for example, so that 100 p.s.i. of fluid pressure at the ports **222** or **224** is required to initiate movement of piston **202** and each additional 100 p.s.i. of fluid pressure effects an additional quarter inch of travel. It is to be noted, that actuator **155** is disposed in one direction in FIG. **6** and is disposed in the opposite direction in FIG. **7.**

Selectively operable operating means or a rotary servo-system are provided to actuate or operate actuator **155** of the regulating means to control the direction of rotation and speed of hoist motor **26.** Such operating means comprises speed sensing motor **46,** hereinbefore described, over-center pump **80,** hereinbefore described, and an associated control circuit **186.** The fluid ports **48** and **50** of speed sensing motor **46** are connected by fluid lines **84** and **86,** respectively, to fluid ports **88** and **90,** respectively, of over-center pump **80.** Fluid lines **84** and **86** are connected by hydraulic fluid pilot lines **94** and **92,** respectively, to pilot ports and **224** and **222,** respectively, of actuator **155.**

An anticavitation and fluid supply system is provided for control circuit **186** and comprises a fluid line **100** connected between fluid lines **162** and **86** and containing check valves **104** and **196;** a fluid line **106** connected between fluid lines **164** and **84** and containing check valves **110** and **194;** and fluid line **192** connected

to both lines **100** and **106** at points between their respective check valves. Operation of the anticavitation and fluid supply system is hereinafter described. Pump **80** has a drain line **91**. Pump **80** is, for example, a variable volume piston-type pump which is driven by a conventional constant speed motor **96**. Furthermore, pump **80** is provided with control lever **98** which is selectively movable by an actuator **240** (operable by control valves **250** and **252**) from a neutral position (wherein no fluid pressure is supplied from its ports **88** and **90**) toward either a hoist position (wherein fluid pressure is provided at port **88** of pump **80** to operate actuator **155** toward hoist position) or a lower position (wherein fluid pressure is provided at port **90** of pump **80** to operate actuator **155** toward lower position). The degree of movement of control lever **98** of pump **80** in either direction regulates the volume of fluid supplied by pump **80** to speed sensing motor **46** through control circuit **186** and also to actuator **155**.

Actuator **240** for control lever **98** is understood to be identical in construction and mode of operation to actuator **155** hereinbefore described in detail in connection with FIG. **7**. Piston rod **204** of actuator **240** is connected by a yoke **206** to control lever **98** of pump **80** and effects varying degrees of movement thereof in either direction. Pilot ports **224** and **222** of actuator **240** are connected by fluid line **242** and **244**, respectively, to the fluid output or pressure ports **246** and **248**, respectively, of control valves **250** and **252**, respectively. Control valves **250** and **252** are two-position, variable pressure regulating valves having manual control levers **254** and **256**, respectively. The fluid input ports **258** and **260** of the control valves **250** and **252**, respectively, are connected through fluid line **132** to pressure port **130** of pump **124**, which pump also supplies fluid for the brake means **32**. Fluid return ports **262** and **264** of the control valves **250** and **252**, respectively, are connected through fluid return lines **226** and **268**, respectively, to a fluid return line **270** which is connected through a filter **272** to reservoir **58**.

Both control valves **250** and **252** are shown in "off" position in FIG. **6**, and therefore, actuator **240** assumes the neutral or spring-centered position shown in FIGS. **6** and **7**. Control valve **252** controls hoisting operation of hoist motor **26** and control valve **250** controls its lowering operation.

Speed sensing motor **46**, over-center pump **80**, control circuit **186** actuator **240** for the overcenter pump, servo-stem speed control valve **152**, actuator **155** for the control valve **152**, and control valves **250** and **252** cooperate to provide directional and speed control for hoist motor **26** in the following manner. For example, at a predetermined speed of rotation of hoist motor **26** (i.e., 200 r.p.m.), speed sensing motor **46**, which is driven thereby (in the opposite direction), is only able to accommodate passage of a predetermined volume of hydraulic fluid (i.e., 2 gallons per minute) from over-center pump **80**. If the volume of fluid flow from pump **80** is increased by appropriate movement of its control lever **98** by actuator **240** in response to operation of the control valve **250** and **252**, fluid pressure builds up in either line **84** or line **86** (depending on direction of control lever movement) because speed sensing motor **46** cannot immediately accommodate this increase. As a result, actuator **155** causes servo-stem **152** to move in an appropriate direction to thereby change the volume, for example, of the fluid flow between main pump **150** and hoist motor **26**. This, in turn, has the effect of causing a change in the speed of rotation of hoist motor **26** and, consequently, the speed of sensing motor **46**. In effect, then, sensing motor **46** is brought up to a speed level where it can accommodate the increased volume of fluid supplied thereto from pump **80** and fluid pressure in either line **84** or **86** decreases. A more detailed description of the manner in which the direction of rotation and speed of hoise motor **26** are controlled is provided hereinafter.

The brake means **32** for hoist **22** are shown in FIG. **2** as comprising a disc brake **114** on hoist **22**; a locking brake comprising a first set of brake shoes **116** which are understood to be spring applied and hydraulically released by pressurization of a first set of hydraulic brake cylinders **118**; and an operating brake comprising a second set of brake shoes **120** which are understood to be spring released and hydraulically applied by pressurization of a second set of hydraulic brake cylinders **122**.

As FIG. **6** shows, the cylinders **118** and **122** are supplied with pressurized fluid from a conventional pressure-compensated piston type pump **124**, driven by a conventional constant speed electric motor **125**. Pump **124** has its inlet port **126** connected by a fluid line **128** to fluid line **59**, and thus, to the fluid in reservoir **58**. The outlet or pressure port **130** of pump **124** is connected by fluid lines **132** and **133** to an on-off valve **134**, having a manual control lever **136**, which controls pressurization of the first set of cylinders **118**. The outlet or pressure port **130** of pump **124** is also connected by fluid lines **132**, **133** and **135** to brake valve **138**, having a pedal type control lever **140**, which controls pressurization of the second set of cylinders **122**. Valves **134** and **138** are connected to the cylinders **118** and **122**, respectively, by fluid lines **142** and **144**, respectively. The cylinders **118** and **122** and valve **138** are provided with fluid return lines, generally designated **146**, which are connected through drain line **180** to reservoir **58**.

Valve **134**, shown in "brake applied" position, is manually movable to "brake release" position wherein it pressurizes cylinders **118** which then act against biasing spring means to release the brake shoes **116** from disc brake **114**.

Valve **138**, shown in "brake released" position is pedal-operated to "brake applied" position wherein it pressurizes cylinders **122** which then act against biasing spring means to apply the brake shoes **120** against disc brake **114**.

The embodiment of the invention discloses in FIGS. **5** and **6** operates as follows.

Assume motor **26** of hoist **22** is stationary and is to be operated to hoist load **31** at some predetermined desired constant rate of speed. Further, assume that brake means **32** are released and that drum **28** is free to rotate. Also assume that main pump **52** and overcenter pump **80** are in operation but that control lever **98** of the over-center pump and, therefore servo-stem speed control valve **152** are in neutral position. With these assumptions, hoist motor **26** and, therefore, speed sensing motor **46** are stationary. All references to direction of movement hereinafter made in connection with the description of operation of the second embodiment of the invention are to be understood to be with respect to FIG. **6**.

To hoist load 31, control lever 98 of over-center pump 80 is moved from neutral position toward hoist position to pressurize line 84 by activator 240 and control valve 250. The extent of movement of lever 98 determines the volume of fluid flow from over-center pump 80 and, thus establishes the predetermined rate of speed for hoist 22. Since hoist motor 26 and speed sensing motor 46 are stationary, fluid pressure builds up on the lines 84 and 94 and pressure at port 224 of actuator 155 exceeds that at port 222. Consequently, piston 202 moves leftward from neutral toward hoist position for a distance proportional to the pilot pressure applied thereto and this pressure level is determined by how far lever 98 is moved. Leftward movement of piston 202 effects corresponding movement of servo-stem control member 154 of valve 152. Fluid then begins to flow from pressure port 158 of main pump 150, through line 164 to port 40, from port 40 to 42 of hoist motor 26, and through return line 162 to port 156 of pump 150. Such fluid flow causes rotation of hoist motor 26 in the hoist direction and also corresponding rotation (but in the opposite direction) of speed sensing motor 46. The speed of rotation of motors 26 and 46 increases until the predetermined speed of rotation is reached. As the predetermined rotational speed is approached, fluid pressure in lines 84 and 94 decreases from its initial level because speed sensing motor 46 is now rotating and accellerating. This decrease in pressure effects corresponding rightward movement of piston 202 in actuator 155 and servo-stem 154 in valves 152 and a proportional decrease in flow of fluid to hoist motor 26 from main pump 150. At the predetermined rotational speed of motor 26, fluid flow from main pump 150 stabilizes and remains constant when speed sensing motor 46 is rotating at a speed sufficient to handle the predetermined volume of fluid supplied thereto by over-center pump 80. In stabilized condition, a pressure differential exists between lines 84 and 86 (and consequently between ports 222 and 224 of actuator 155) which maintains servo-stem valve 154 in hoist position.

As load 31 is raised at a constant predetermined speed by hoist 22, cable 30 is wrapped on hoist drum 28. However, as one layer of cable starts to build upon another, the apparent effect as regards load 31 is that of an increase in drum diameter and, consequently, hoist motor 26 and speed sensing motor 46 begin to slow down under the increased load. As speed sensing motor 46 slows down, the volume of fluid it can accommodate decreases and pressure in the lines 84 and 94 increases. This causes an increase in pressure at port 224 of actuator 155, and piston 202 moves farther leftward to increase fluid flow from main pump 150 to hoist motor 26 to thereby increase the speed of the latter until it (and speed sensing motor 46) return to the predetermined speed. When this occurs, the situation again stabilizes until further readjustment is required.

If control lever 98 of over-center pump 80 is moved partway (but not fully) toward neutral position, a new slower predetermined rate of hoist speed is established but operation is the same as hereinbefore described.

If control lever 98 is returned to neutral position, fluid pressure in lines 84 and 86 equalizes as the motors 26 and 46 slow down and stop and servo-stem 154 of servo-stem valve 154 returns to neutral position. Under this condition, load 31 will remain suspended, even though brake means 32 are not applied, because return flow of fluid from both ports 40 and 42 of motor 26 is blocked by main pump 150. Since hoist motor 26 is a piston-type motor, very little fluid leakage occurs therefrom. However, since some leakage does occur, fluid pressure in line 162 tends to decrease (and pressure in line 164 tends to increase) and load 31 would normally tend to descend at a very slow rate. In accordance with the present invention, however, pressure loss in lines 162 compensated for by supplying a small amount of pressurized fluid thereto from pressure port 190 of charging pump 184 (which is still in operation), through line 192 and check valve 196. Simultaneously, speed sensing motor 46 tends to rotate and transfer fluid from its port 50 to port 48 thereby causing a pressure increase in lines 84 and 94 to shaft piston 202 in actuator 155 leftward to shift servo-stem 154 leftward to establish fluid flow fron port 158 of pump 150 through line 164 to port 40 of motor 20 thereby rotate main motor 26 in the forward or hoist direction. Servo-stem 154 will remain in this shifted position to continue to supply a volume of fluid flow to motor 26 equal to the leakage from main motor 26. In this manner, then, slow descent of slippage of load 31 is prevented even though control lever 98 of over-center pump 80 is in neutral position.

To lower load 31 at some predetermined speed, control lever 98 of over-center pump 80 is moved from neutral position toward lower position by actuator 240 and contral valve 250 to pressurize line 86 at a predetermined pressure level. Fluid pressuire builds up in the lines 86 and 92 and pressure at port 222 of actuator 155 exceeds that at port 224. Consequently, piston 202 moves from neutral rightward toward lowering position for a distance proportional to the pilot pressure applied thereto. Again, the extent of movement of lever 98 determines the volume of fluid flow from over-center pump 80 and thus establishes a predetermined rate of speed for hoist 22 in the lowering direction. Fluid then begins to flow from pressure port 156 of main pump 150, through line 162, to port 42, from port 42 to 40 to hoist motor 26 and through line 164 to port 158 of pump 150. Such fluid flow causes rotation of hoist motor 26 in the lowering direction and also causes corresponding (but opposite) rotation of speed sensing motor 46. The speed of rotation of motors 26 and 46 increases until the predetermined speed of rotation is reached. As the predetermined rotational speed is approached, fluid pressure in lines 86 and 92 decreases from its initial level because speed sensing motor 46 is now rotating and accellerating. This decrease in pressure effects a corresponding leftward movement of piston 202 in actuator 155 and a proportional decrease in flow of fluid to hoist motor 26 from main pump 150. Fluid flow would tend to stabilize and remain constant when speed sensing motor 46 is rotating at a speed sufficient to accommodate the predetermined volume of fluid supplied thereto by over-center pump 80, except for the fact that hoist motor 26 is operating in the lowering direction and the weight of load 31 tends to increase hoist motor speed. Therefore, an overspeed condition arises wherein speed sensing motor 46 operates faster than overcenter pump 80 can supply fluid thereto at the predetermined setting of pump 80. When the overspeed condition occurs, fluid pressure in lines 86 and 92 decreases toward, to or below the fluid pressure level in the lines 84 and 94. As a result, pressure at port 222 of actuator 155 decreases toward, to or

below that at port 224 and piston 202 moves from rightward position toward or to neutral position, or even beyond that to some degree of leftwards position. As piston 202 and servo-stem control 154 move from some degree of rightward position toward neutral position, the volume of fluid applied to port 42 of hoist motor 26 decreases and the hoist motor slows down. This effects corresponding slowdown of rotation of speed sensing motor 46. However, if this slowdown is still insufficient to maintain a constant lowering speed of hoist motor 26, piston 202 of actuator 155 is moved even further leftward to neutral position to thereby entirely cut off fluid flow to port 42 of motor 26. Again, if this does not effect sufficiently rapid reduction in speed, piston 202 of actuator 155 and servo-stem control 154 move from neutral position to some degree of leftward position whereby fluid is supplied by pump 150 to port 40 of hoist motor 26 instead of port 42 to oppose the rotation of hoist motor 26. Such repositioning of servo-stem control 154 in response to fluid pressure conditions in lines 86 and 94 is quite rapid and therefore effects constant speed control of hoist motor 26.

In the embodiments of the invention hereinbefore described, speed sensing motor 42 is connected directly to hoist motor 26 and is directly responsive to the speed of rotation of the hoist motor to provide constant motor speed regulation. In such an arrangement, as hereinbefore noted, hoist motor 26 tends to slow down as layers of cable 30 build up on hoist drum 28 and tends to speed up as the layers of cable decrease. Therefore, speed adjustment is required in connection with each layer of cable. The net result is to maintain constant speed on hoist motor 26 but the speed of cable 30 then varies in accordance with the number of wraps on drum 28. In accordance with another aspect of the invention, shown in FIG. 8, speed sensing motor 42 may be arranged to respond directly to the speed of hoist line 30 instead of indirectly thereto through hoist motor 26 to provide constant line speed instead of constant motor speed. As FIG. 8 shows, hoist 22 comprises means for supporting speed sensing motor 42 so that it can be driven directly by cable 30 and also track the cable as it winds on and unwinds from hoist drum 28. Such means comprise a grooved shaft 300 supported in fixed position parallel to drum 28. A carriage 302 is mounted on shaft 300 and is slidably movable in either direction therealong so as to be able to track cable 30 as the cable moves along the axis of drum 28. Speed sensing motor 42 is rigidly mounted on carriage 302 and is movable therewith. Speed sensing motor 42 is provided with a pulley 304 on its shaft 306 which engages and is driven by cable 30. Driving and tracking pressure between cable 30 and pulley 304 is maintained by two other pulleys 308 and 310 which are mounted on carriage 302 and bias cable 30 against pulley 304 on speed sensing motor 42. In operation, linear movement of cable 30 effects rotation of pulley 304 and, therefore, corresponding operation of speed sensing motor 42. As cable 30 moves back and forth with respect to the axis of drum 28, carriage 304 is moved in a corresponding manner along shaft 306 because cable 30 is entrapped in the grooves in the pulleys 304, 308 and 310.

Resume

A speed control system in accordance with the invention accomplishes these main functions, as follows. It controls direction of rotation of the hydraulic motor. It controls speed of rotation in the desired direction and maintains it at some predetermined level despite variation in load imposed thereon. It maintains the motor stationary in some desired position despite the load imposed thereon and despite variations in the size of that load, even though conventional brakes are not applied to the motor or to equipment driven by the motor.

I claim:

1. A control system for a hydraulic motor comprises:
   a. fluid supply means, including a main pump, for supplying pressurized hydraulic fluid to said motor to effect its direction and speed of rotation;
   b. regulating means for said fluid supply means to vary the volume of fluid supplied from said fluid supply means to said motor and thereby control motor direction and speed, said regulating means comprising first multi-position valve means including a first multi-position control member therefor;
   c. and operating means for said regulating means comprising:
      i. a speed sensing hydraulic motor responsive to the direction and speed of said motor; and
      ii. pump means for supplying a predetermined volume of fluid to said speed sensing motor;
      said regulating means being responsive to the pressure of said fluid being supplied to said speed sensing motor to vary the volume of fluid supplied from said fluid supply means to said motor to regulate the direction and speed of said motor.

2. A control system according to claim 1 wherein said pump means comprises an over-center pump and second multi-position valve means including a second multi-position control member therefor.

3. A control system according to claim 2 wherein said main pump is a constant volume pump; wherein said first multi-position valve means is a directional valve wherein a movable valve spool serves as said first multi-position control member; and wherein said valve spool responds to the pressure of the fluid being supplied to said speed sensing motor.

4. A control system according to claim 2 wherein said main pump is a variable volume pump; wherein said first multi-position valve means is a multi-position speed control valve on said main pump wherein a servo-stem serves as said first multi-position control member; and including actuator means responsive to the pressure of the fluid being supplied to said speed sensing motor to move said servo-stem.

5. A control system according to claim 2 wherein said over-center pump operates at constant speed and said second multi-position valve means controls the volume of fluid delivered by said over-center pump.

6. A control system according to claim 5 wherein said second multi-position control member is manually operated.

7. A control system according to claim 5 wherein said second multi-position control member is remotely operated by actuator means.

8. A control system according to claim 3 wherein said directional valve is a three-position variable orifice type valve for regulating motor speed and direction of rotation; and wherein said second multi-position valve is a three-position variable orifice type valve for said over-center pump for effecting changes in motor speed and direction of rotation.

9. A control system according to claim 4 wherein said multi-position speed control valve on said main pump is a three-position variable orifice type valve for regulating motor speed and direction of rotation; and wherein said second multi-position valve is a three-position vairable orifice type valve for said over-center pump for effecting changes in motor speed and direction of rotation.

10. A control system according to claim 1 wherein said motor is a hoist motor and wherein said speed sensing motor is directly responsive to the speed and direction of rotation of said hoist motor to effect constant speed of said hoist motor despite variations in load imposed thereon.

11. A control system according to claim 1 wherein said motor is a hoist motor and wherein said speed sensing motor is directly responsive to the speed and direction of movement of a hoist line to effect speed regulation of said hoist motor to maintain hoist line speed constant despite variation in load imposed on said hoist motor.

12. A control system for a hydraulic motor comprising:

a. a main pump for supplying pressurized hydraulic fluid to said motor to effect its direction and speed of rotation;

b. regulating means for said main pump to vary the direction and volume of fluid supplied from said pump to said motor and thereby control motor direction and speed of rotation, said regulating means comprising first multi-position valve means and a first multi-position control member therefor;

c. and operating means for said first multi-position valve means comprising:

i. a speed sensing hydraulic motor responsive to the direction and speed of rotation of said motor, said speed sensing motor capable of accommodating a predetermined volume of fluid at a predetermined speed of rotation;

ii. pump means for supplying a predetermined volume of fluid to said speed sensing motor, said pump means comprising an over-center pump and second multi-position valve means and a second multi-position control member therefor; and

iii. a control circuit connected between said speed sensing motor and said over-center pump and to said regulating means; said regulating means being responsive to fluid pressure conditions in said control circuit to vary the direction and volume of fluid supplied to said motor to thereby control motor direction and speed of rotation.

13. A control system according to claim 12 wherein said speed sensing motor and said over-center pump each have two ports; wherein said control circuit comprises two fluid lines, each line being connected between one speed sensing motor port and one over-center pump port; and wherein said two fluid lines are connected to said first multi-position valve means of said regulating means to effect movement of said first multi-position control member therefor.

14. A control system according to claim 13 including a pair of fluid lines whereby fluid is supplied to said motor from said main pump and further including means for supplying additional fluid to said pair of fluid lines and to said two fluid lines in said control circuit to make up for fluid leakage in all of said lines when said motor and said speed sensing motor are being maintained in stationary condition and a load is imposed on said motor.

15. A control system according to claim 14 wherein said means for supplying additional fluid comprises said main pump.

16. A control system according to claim 14 wherein said means for supplying additional fluid comprises an auxiliary pump.

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