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Allen

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[54] **AUTOMATIC CONTROL SYSTEM FOR USE WITH A HYDRAULIC DRIVE SYSTEM**

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[52] U.S. Cl. **60/420**, 60/476, 180/6.48, 104/244.1, 404/84, 91/412, 91/414

[51] Int. Cl. **F15b 11/22**

[58] Field of Search .... 60/395, 420, 476, 60/97 E; 180/6.48; 104/244.1; 404/84

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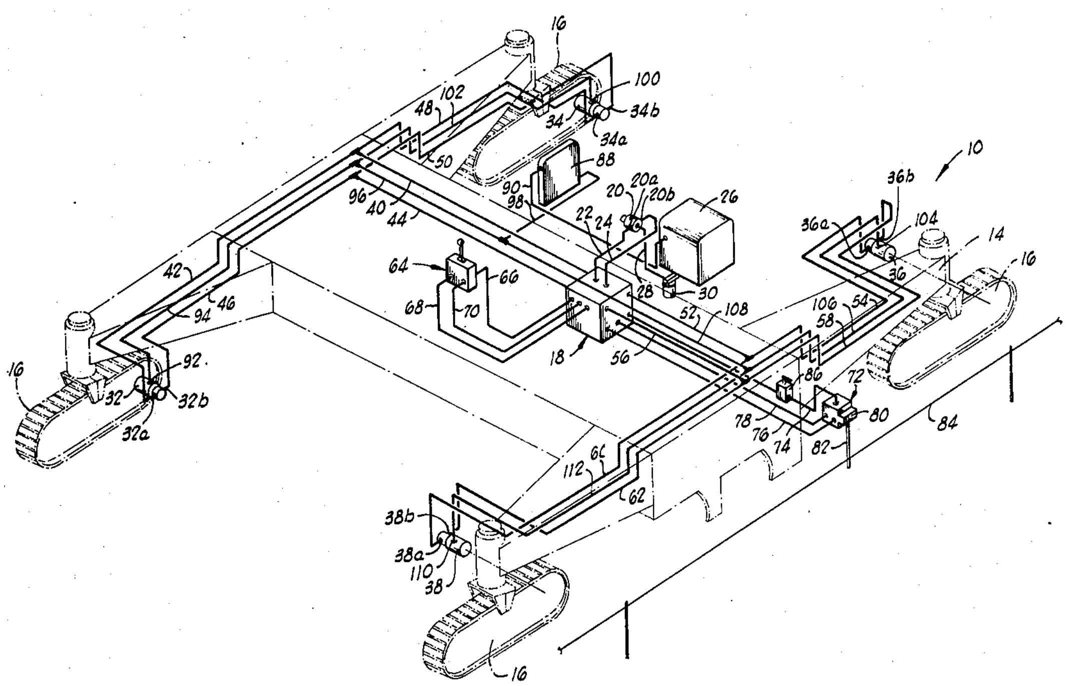
[57] **ABSTRACT**

An automatic control system for use with a hydraulic drive system which includes a source of pressurized hydraulic fluid and first and second hydraulic drive motors deriving their power from said source of pressurized hydraulic fluid, comprising a differential valve in fluid communication with the source of pressurized hydraulic fluid and a flow divider in fluid communication with the differential valve and with a flow compensator. The flow compensator is in fluid communication with the first and second hydraulic drive means. A compensator valve member is carried by the flow compensator for controlling the relative rates of flow of hydraulic fluid to the first and second hydraulic drive motors in response to load variations encountered by the hydraulic drive means. A control unit is in fluid communication with the source of pressurized hydraulic fluid and with the flow divider and includes a control valve member for controlling hydraulic fluid signals being sent therefrom to the flow divider. An actuator is operatively connected to the control valve member for actuating the control valve member in response to stimulus from a source external to the automatic control system. A flow divider valve member is carried by the flow divider for controlling the relative rates of flow of hydraulic fluid through the flow divider in response to the hydraulic fluid signals from the control unit.

17 Claims, 10 Drawing Figures



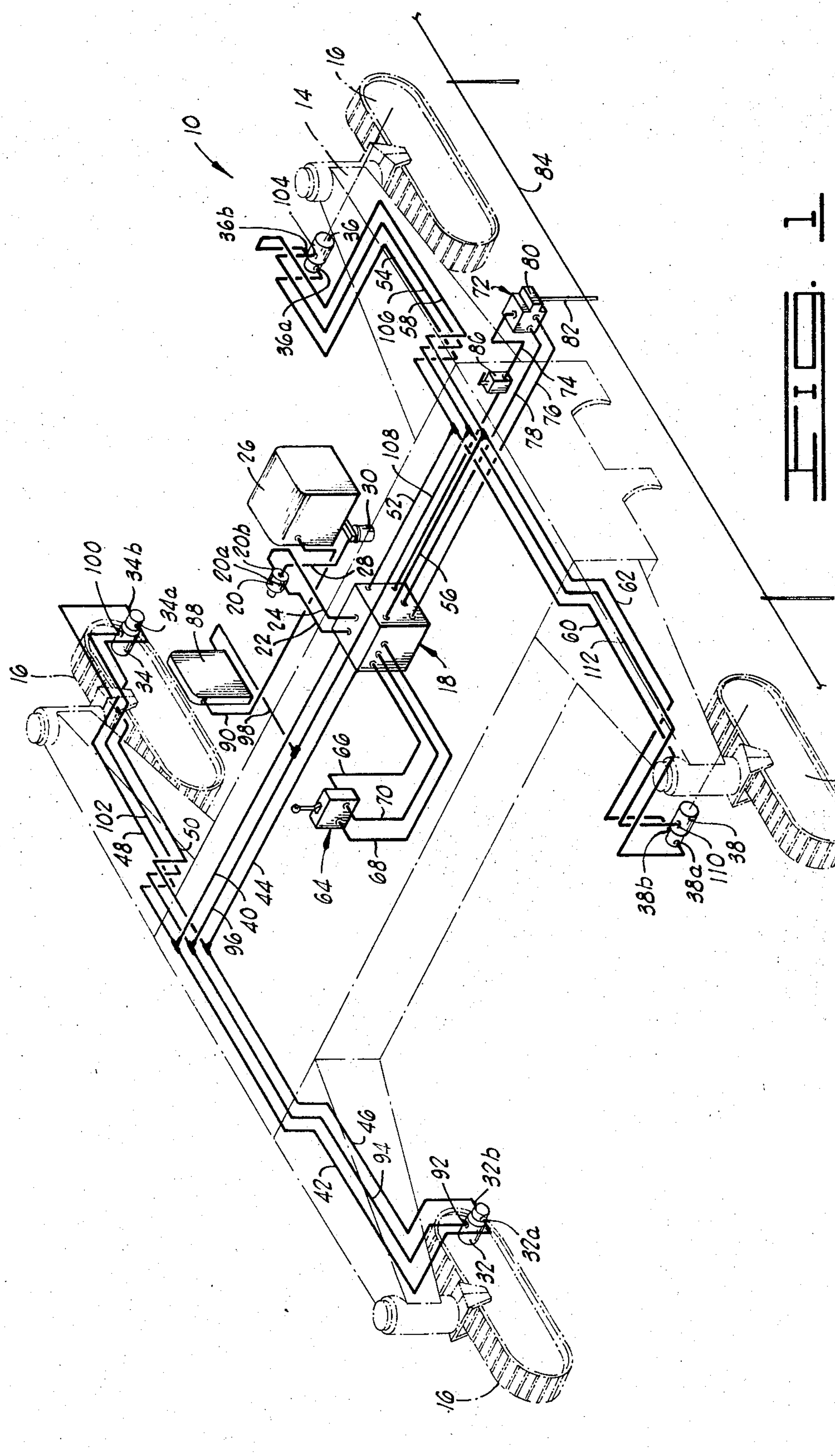

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FIG. 1

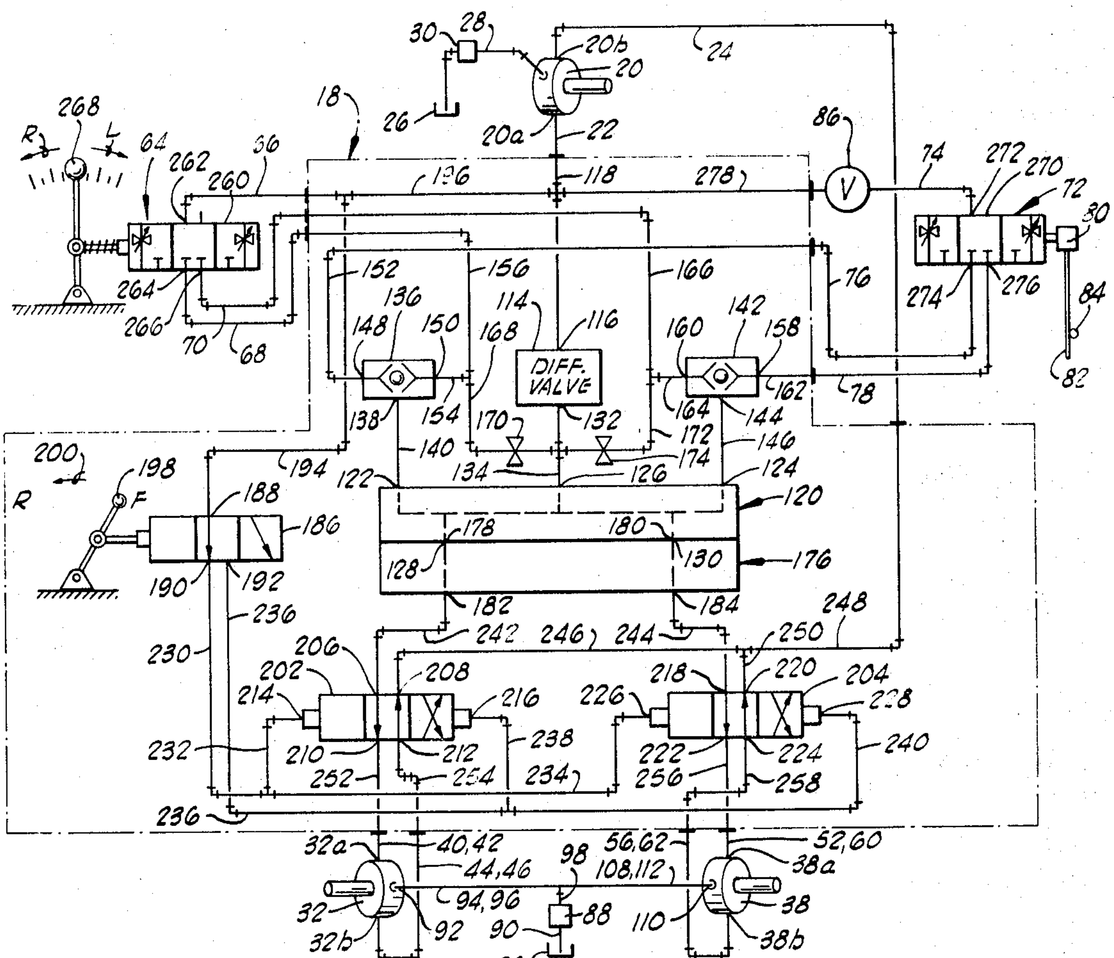

FIG. 2

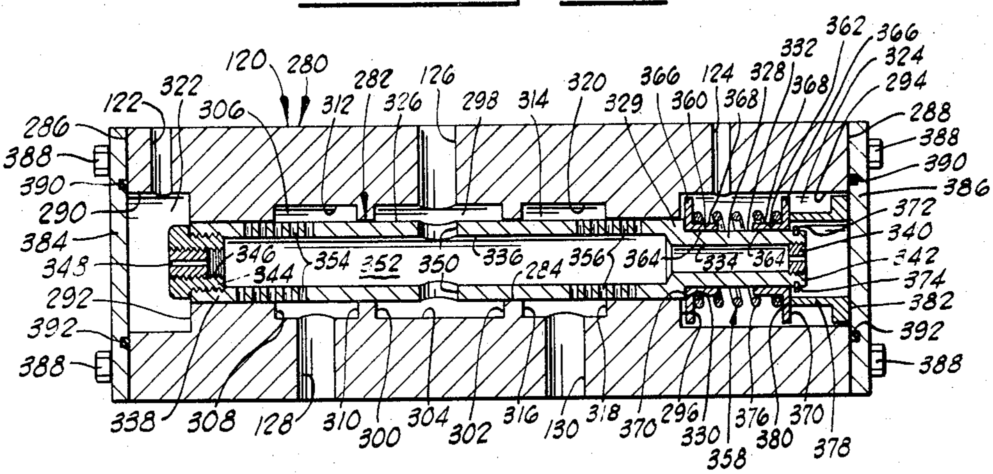

FIG. 3

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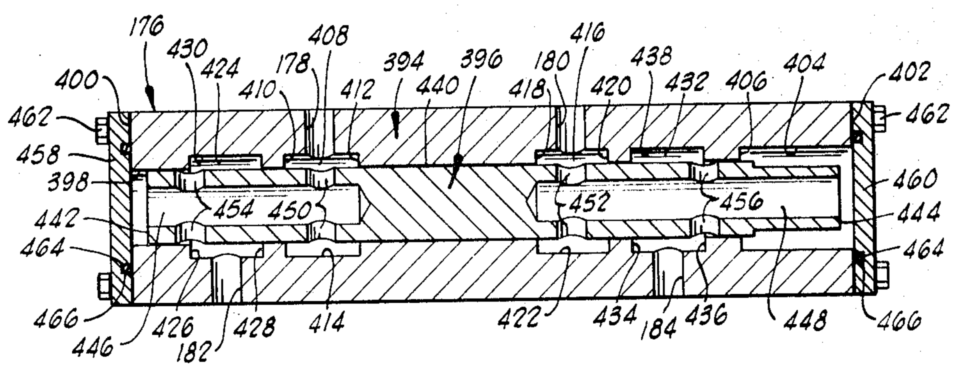

FIG. 4

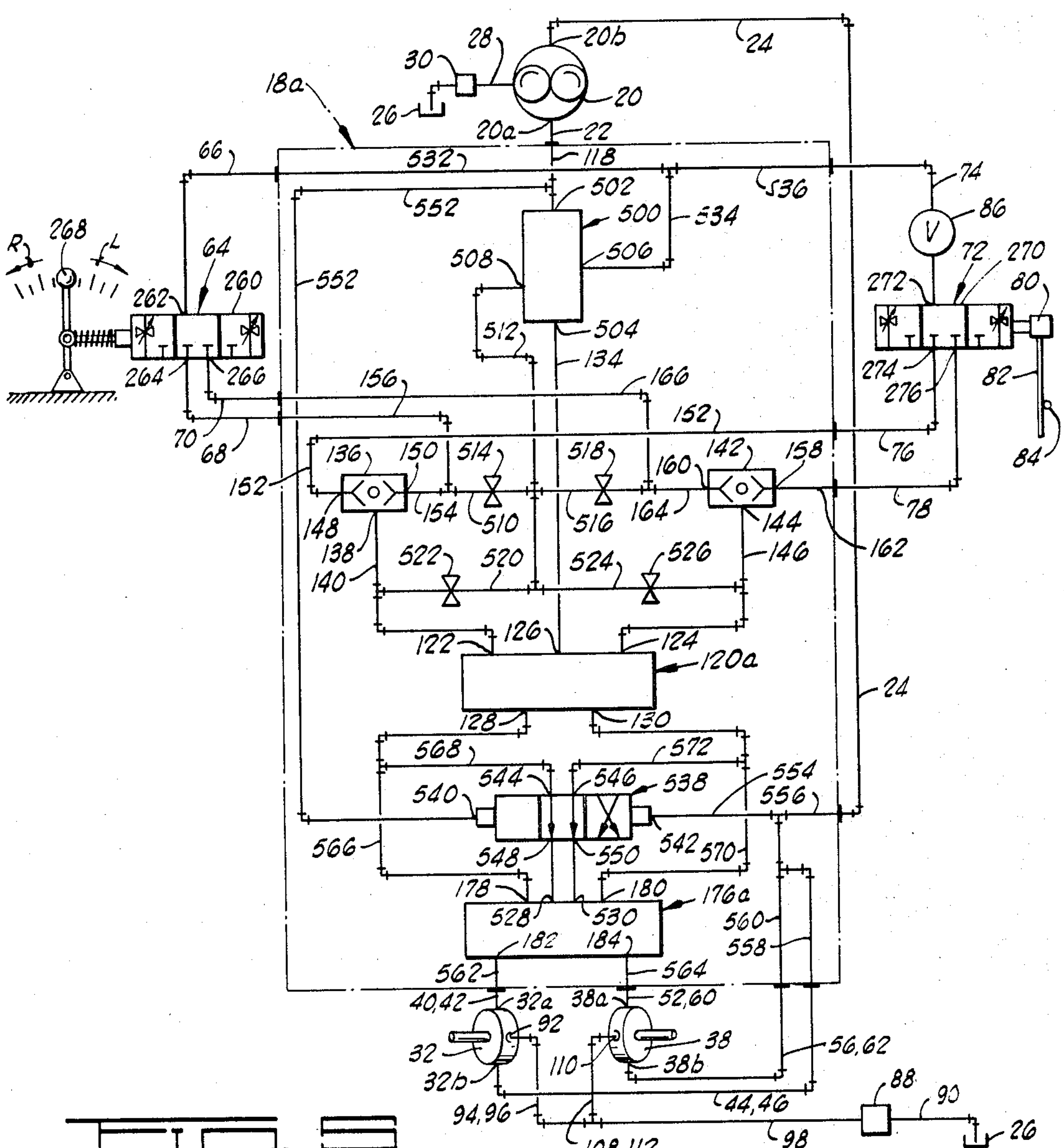

FIG. 5

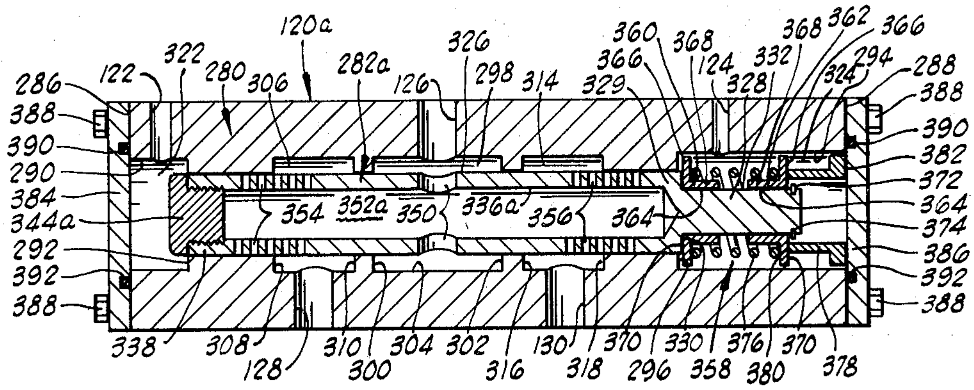

FIG. 6

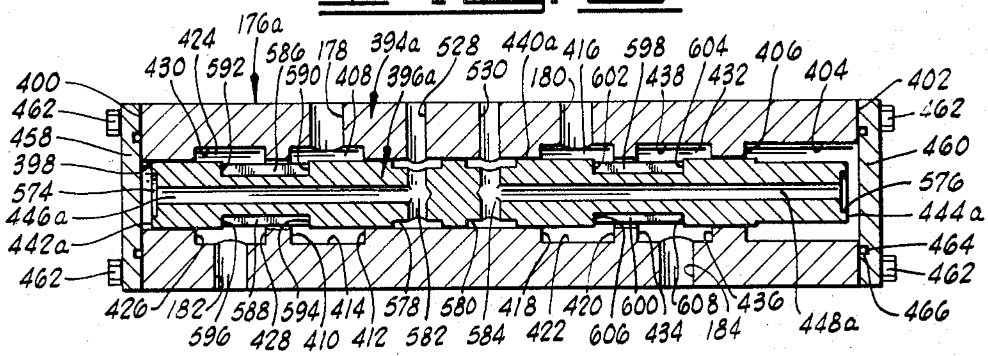

FIG. 7

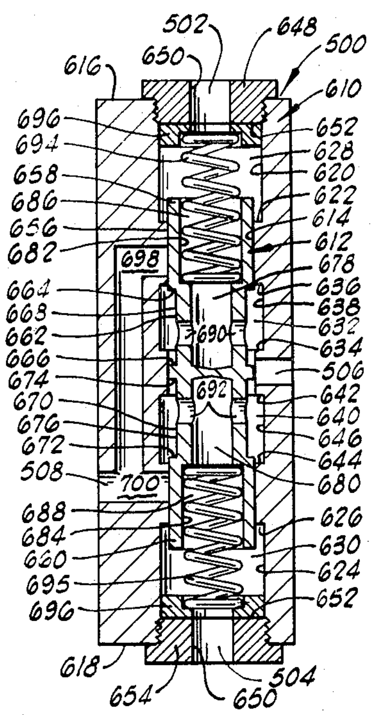

FIG. 8

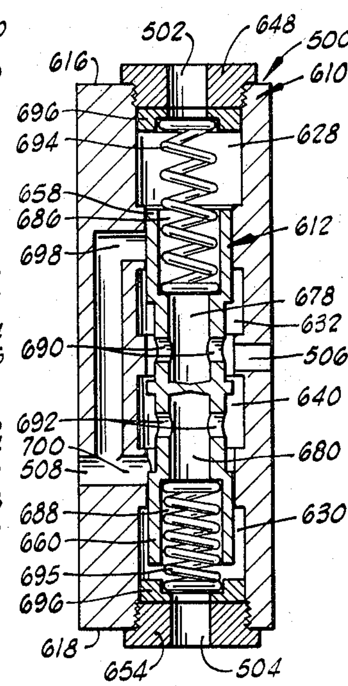

FIG. 9

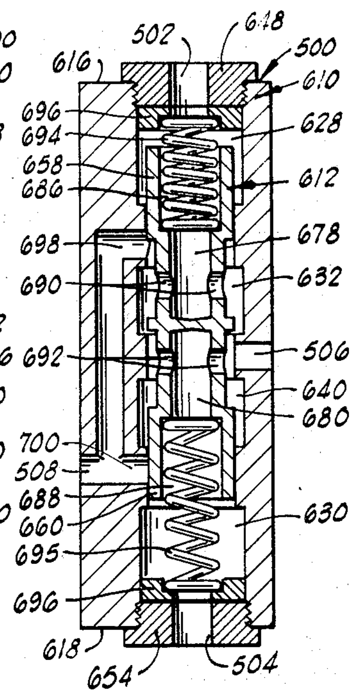

FIG. 10

AUTOMATIC CONTROL SYSTEM FOR USE WITH A HYDRAULIC DRIVE SYSTEM

BACKGROUND OF THE INVENTION

1. Field of the Invention

The invention relates generally to automatic control systems for use with hydraulic drive systems, and more specifically, but not by way of limitation, to automatic control systems for use with heavy construction machinery such as paving machines or the like.

2. Brief Description of the Prior Art

Known examples of the prior art teach various systems for automatically controlling hydraulic drive systems. Certain of these systems employ complex and expensive mechanically and electrically actuated valving to provide a desired degree of control. Many prior art systems have not been entirely satisfactory due to the expense involved in both the manufacture and maintenance of various component parts thereof. Others of the prior art systems have not been entirely satisfactory due to delay in response of the drive system to the control signals sent thereto by the automatic control system, and due to wander or oscillation inherent in the automatic control system.

SUMMARY OF THE INVENTION

The present invention contemplates an improved automatic control system for use with a hydraulic drive system of the type which includes a source of pressurized hydraulic fluid and first and second hydraulic drive means deriving their power from said source of pressurized hydraulic fluid. The automatic control system comprises differential valve means, having an inlet in fluid communication with the source of pressurized hydraulic fluid and having an outlet, for providing a hydraulic pressure differential between the inlet and outlet thereof. A flow divider having first and second signal inlets, a power inlet, and first and second power outlets communicates with the outlet of the differential valve means via the power inlet thereof. A flow compensator having first and second power inlets and first and second power outlets, is connected to the flow divider via the first and second power inlets thereof in fluid communication with the first and second power outlets of the flow divider, respectively. The flow compensator is further in fluid communication with the first and second hydraulic drive means by means of the first and second power outlets, respectively, thereof. Compensator valve means is carried by said flow compensator between the first power inlet and the first power outlet, and between the second power inlet and the second power outlet thereof for controlling the relative rates of flow of hydraulic fluid to said first and second hydraulic drive means from said flow compensator in response to load variations encountered by said first and second hydraulic drive means. A first control unit having an inlet in fluid communication with the source of pressurized hydraulic fluid and having first and second signal outlets is in fluid communication with the first and second signal inlets of the flow divider by means of the first and second signal outlets thereof, respectively. First control valve means is carried by the first control unit between the inlet thereof and the first and second signal outlets thereof for controlling the relative rates of flow of hydraulic fluid through the first and second signal outlets of the first control unit to the first and second signal inlets of the flow divider, respectively. First actuation means is operatively connected to the first control valve means for actuating the first control valve means in response to stimulus from a source external to the automatic control system. Flow divider valve means is carried by the flow divider between the power inlet thereof and the first and second power outlets thereof, and in fluid communication with the first and second signal inlets thereof for controlling the relative rates of flow of hydraulic fluid from the power inlet to the first power outlet and from the power inlet to the second power outlet in response to the relative flow rates of hydraulic fluid from the first control unit into the first and second signal inlets of the flow divider.

An object of the present invention is to provide an improved automatic control system for use with a hydraulic drive system which will automatically compensate for deviation of the hydraulic drive system from a desired path.

Another object of the present invention is to provide an improved automatic control system for use with a hydraulic drive system which will automatically compensate for variations in the loads encountered by the individual hydraulic drive motors of the hydraulic drive system.

Yet another object of the present invention is to provide an improved automatic control system for use with a hydraulic drive system which is all-hydraulic in operation.

A still further object of the present invention is to provide an automatic control system for use with a hydraulic drive system employing components which are inexpensive to manufacture and maintain.

Other objects and advantages of the present invention will be evident from the following detailed description when read in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a substantially diagrammatical perspective view illustrating the apparatus of the present invention installed on a hydraulically driven self-propelled vehicle.

FIG. 2 is a schematic diagram illustrating one form of the automatic control system of the present invention.

FIG. 3 is a cross-sectional view of one form of flow divider for use with the present invention illustrating the details of construction thereof.

FIG. 4 is a cross-sectional view of one form of flow compensator for use with the present invention illustrating the details of construction thereof.

FIG. 5 is a schematic diagram illustrating another form of the automatic control system of the present invention.

FIG. 6 is a cross-sectional view of another form of flow divider for use with the present invention illustrating the details of construction thereof.

FIG. 7 is a cross-sectional view of another form of flow compensator for use with the present invention illustrating the details of construction thereof.

FIG. 8 is a cross-sectional view of one form of bidirectional differential valve for use with the present invention illustrating the details of construction thereof.

FIG. 9 is a cross-sectional view of the valve of FIG. 8 showing the disposition of the valve member when the pump is driven in the forward direction.

FIG. 10 is a cross-sectional view of the valve of FIG 8 showing the disposition of the valve member when the pump is driven in the reverse direction.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the drawings, and to FIG. 1 in particular, the apparatus of the present invention is generally designated by the reference character **10.** The apparatus **10** includes a self-propelled vehicle **12** comprising a frame **14** and four hydraulic motor-driven track units **16** positioned at each of the four corners of the frame **14** and supporting the frame **14** on the ground.

While the vehicle **12** is described herein to include four hydraulic motor-driven track units **16,** i6 will be readily understood that the present invention is equally well suited for use with vehicles having two drive units disposed respectively on opposite sides of the vehicle.

Mounted on and supported by the frame **14** is an automatic control assembly **18.** Details of the construction of the automatic control assembly **18** will be described in greater detail hereinafter.

A track drive pump **20** is mounted on the frame **14** and is connected to the automatic control assembly **18** by means of a pair of hydraulic conduits **22** and **24.** The track drive pump **20** is driven by a power take-off which is driven by a suitable engine such as an internal combustion engine of the piston or turbine type (not shown). A hydraulic fluid reservoir **26** is mounted on the frame **14** and is in fluid communication with the track drive pump **20** by means of a hydraulic conduit **28.** A hydraulic fluid filter **30** is interposed in the conduit **28** between the track drive pump **20** and the hydraulic fluid reservoir **26.** In certain applications, alternate or additional filters may be desirable.

Hydraulic track drive motors **32, 34, 36** and **38** are mounted respectively on and drivingly connected to the hydraulic motor-driven track units **16.** The first port **32*a*** of the track drive motor **32** is in fluid communication with the automatic control assembly **18** via hydraulic conduits **40** and **42.** The second port **32*b*** of the track drive motor **32** is in fluid communication with the automatic control assembly via hydraulic conduits **44** and **46.** The first port **34*a*** of track drive motor **34** is in fluid communication with the automatic control assembly **18** via hydraulic conduits **40** and **48.** The second port **34*b*** of track drive motor **34** is in fluid communication with the automatic control assembly **18** via hydraulic conduits **50** and **44.**

It should be noted that it may be desirable in certain circumstances to include a conventional flow divider at the junction of conduits **42** and **48** to divide the flow of hydraulic fluid therethrough from conduit **40.** Similarly, it may also be desirable to include a second conventional flow divider at the junction of conduits **60** and **54** to divide the flow of hydraulic fluid therethrough from conduit **52.** In addition, a conventional flow divider may also be included at the junctions of conduits **46** and **50** to divide the flow of hydraulic fluid therethrough from conduit **44.** Similarly, it may also be desirable to include a conventional flow divider at the junction of conduits **62** and **58** to divide the flow of hydraulic fluid therethrough from conduit **56.** Such flow dividers may suitably be of the 50—50 type which is well known to those skilled in the art.

The first port **36*a*** of track drive motor **36** is in fluid communication with the automatic control assembly **18** via hydraulic conduits **52** and **54.** The second port **36*b*** of track drive motor **36** is in fluid communication with the automatic control assembly **18** via hydraulic conduits **56** and **58.** The first port **38*a*** of track drive motor **38** is in fluid communication with the automatic control assembly **18** via hydraulic conduits **52** and **60.** The second port **38b** of track drive motor **38** is in fluid communication with the automatic control assembly **18** via hydraulic conduits **56** and **62.**

A hydraulic manual steering control assembly **64** is mounted on the frame **14** and is in fluid communication with the automatic control assembly **18** via hydraulic supply conduit **66** and hydraulic signal conduits **68** and **70.**

A sensor control assembly **72** is mounted on and to one side of the frame **14.** The sensor control assembly **72** is in fluid communication with the automatic control assembly **18** via hydraulic supply conduit **74** and hydraulic signal conduits **76** and **78.** The sensor control assembly **72** includes a sensor unit **80** operatively connected thereto by means which will be described in greater detail hereinafter. Each sensor unit **80** is provided with a tracer **82** which engages a string line or grade line **84** which is supported above the surface of the ground to provide a suitable reference datum to facilitate the automatic steering of the apparatus **10.** A shut-off valve **86** is installed in the hydraulic supply conduit **74** intermediate the sensor control assembly **72** and the automatic control assembly **18** to provide means for manually deactivating the sensor control assembly **72** when desired.

The sensor control assembly **72** is preferably mounted on the vehicle **12** proximate the forward end thereof when the vehicle **12** is moving in the forward direction. Conversely, the sensor control assembly **72** is preferably mounted on the vehicle **12** proximate the rear end thereof when the vehicle **12** is moving in the reverse direction.

A heat exchanger **88** is mounted on the frame **14** and is in fluid communication with the hydraulic fluid reservoir **26** via conduit **90.** The bypass outlet **92** of the track drive motor **32** is in fluid communication with the heat exchanger **88** via hydraulic conduits **94, 96** and **98.** The bypass outlet **100** of the track drive motor **34** is in fluid communication with the heat exchanger **88** via hydraulic conduits **102, 96** and **98.** The bypass outlet **104** of the track drive motor **36** is in fluid communication with the heat exchanger **88** via hydraulic conduits **106, 108** and **98.** The bypass outlet **110** of the track drive motor **38** is in fluid communication with the heat exchanger **88** via hydraulic conduits **112, 108** and **98.** The track drive motors **32, 34, 36** and **38** are adapted, by means of their internal valving, to direct excess fluid volume through the respective bypass outlets **92, 100, 104** and **110** through the associated hydraulic conduits to the heat exchanger **88.** At the heat exchanger **88** the excess hydraulic fluid is cooled and directed from the heat exchanger **88** through hydraulic conduit **90** to the hydraulic fluid reservoir **26** for re-use in the hydraulic drive system of the apparatus **10.**

It will be readily apparent to those skilled in the art that the apparatus **10** may be propelled over the ground by means of the four hydraulic motor-driven track units **16** which are driven respectively by the track drive motors **32, 34, 36** and **38,** which track drive motors derive their power from the flow of hydraulic fluid emanating from the track drive pump 20 and conveyed thereto through the previously described hydraulic conduits and the automatic control assembly 18. Control of the relative speed and power outputs of the track drive motors 32, 34, 36 and 38 is achieved by the automatic control assembly 18 interposed between the track drive pump 20 and the track drive motors. Preferred embodiments of the automatic control assembly 18 will be described in detail hereinafter.

DESCRIPTION OF THE EMBODIMENT OF FIGS. 2, 3 AND 4

FIG. 2 generally illustrates, in schematic form, the automatic control assembly 18 in fluid communication with the hydraulic manual steering control assembly 64, the sensor control assembly 72, the shut-off valve 86, and the track drive motors 32 and 38 as described in detail above. The track drive motors 34 and 36 are not illustrated in FIG. 2, it being believed that it will be readily apparent to one skilled in the art that the operation of the motors 34 and 36 is substantially identical to the operation of motors 32 and 38 which will be described in detail herein. FIG. 2 also illustrates the track drive pump 20 in fluid communication with the automatic control assembly 18 as described above.

The automatic control assembly 18 includes a differential valve 114. The inlet 116 of the differential valve 114 is in fluid communication with the first port 20a of the track drive pump 20 via hydraulic conduits 118 and 22.

The automatic control assembly 18 also includes a flow divider 120 having first and second signal inlets 122 and 124, a power inlet 126 and first and second power outlets 128 and 130. The power inlet 126 of the flow divider 120 is in fluid communication with the outlet 132 of the differential valve 114 via conduit 134.

A shuttle valve 136 having an outlet 138 is in fluid communication with the first signal inlet 122 of the flow divider 120 via conduit 40. A second shuttle valve 142 having an outlet 144 is in fluid communication with the second signal inlet 124 of the flow divider 120 via conduitl 146.

The shuttle valve 136 also includes a first inlet 148 and a second inlet 150. The first inlet 148 is in fluid communication with the sensor control assembly 72 via conduits 152 and 76. The second inlet 150 is in fluid communication with the hydraulic manual steering control assembly 64 via conduits 154, 156 and 68.

The second shuttle valve 142 includes a first inlet 158 and a second inlet 160. The first inlet 158 is in fluid communication with the sensor control assembly 72 via conduits 162 and 78. The second inlet 160 is in fluid communication with the hydraulic manual steering control assembly 62 via conduits 164, 166 and 70.

The second inlet 150 of the shuttle valve 136 is also in fluid communication with the power inlet 126 of the flow divider 120 via conduits 154, 168 and 134. An orifice 170 is interposed in conduit 168 intermediate conduits 154 and 134. The second inlet 160 of the shuttle valve 142 is also in fluid communication with the power inlet 126 of the flow divider 120 via conduits 164, 172 and 134. An orifice 174 is interposed in the conduit 172 intermediate the conduits 164 and 134.

The automatic control assembly 18 further includes a flow compensator 176. The flow compensator 176 includes first and second power inlets 178 and 180, and first and second power outlets 182 and 184. The first and second power inlets 178 and 180 are in fluid communication with the first and second power outlets 128 and 130 of the flow divider 120, respectively.

Included in the automatic control assembly 18 is a forward-reverse control valve 186. The valve 186 includes an inlet port 188, a foward outlet port 190 and a reverse outlet port 192. The inlet port 188 is in fluid communication with the first port 20a of the track drive pump 20 via conduits 194, 196, 118 and 22. The valve 186 is illustrated in the forward mode with the inlet port 188 and the forward outlet port 190 in fluid communication with one another. It will be readily apparent that the valve 186 may be placed in the reverse mode by placing the inlet port 188 in fluid communication with the reverse outlet port 192 by moving the control lever 198 in the direction indicated by the arrow 200 to the reverse position.

The automatic control assembly 18 further includes a pair of forward-reverse spool valves 202 and 204. The spool valve 202 includes power ports 206, 208, 210 and 212. The spool valve 202 also includes a pair of control ports 214 and 216.

The spool valve 204 includes power ports 218, 220, 222 and 224. The spool valve 204 also includes a pair of control ports 226 and 228.

The forward outlet port 190 of the forward-reverse control valve 186 is in fluid communication with the control port 214 of spool valve 202 via conduits 230 and 232. The forward outlet port 190 is also in fluid communication with the control port 226 of spool valve 204 via conduits 230 and 234. The reverse outlet port 192 of the forward-reverse control valve 186 is in fluid communication with the control port 216 of spool valve 202 via conduits 236 and 238. The reverse outlet pport 192 is also in fluid communication with the control port 228 of spool valve 204 via conduits 236 and 240.

The first power outlet 182 of the flow compensator 176 is in fluid communication with the power port 206 of spool valve 202 via conduit 242. The second power outlet 184 of the flow compensator 176 is in fluid communication with power port 218 of the spool valve 204 via conduit 244.

The power port 208 of spool valve 202 is in fluid communication with the second port 20b of the track drive pump 20 via conduits 246, 248 and 24. The power port 220 of spool valve 204 is in fluid communication with the second port 20b of the track drive pump 20 via conduits 250, 248 and 24. The power port 210 of spool valve 202 is in fluid communication with the track drive motor 32 via conduits 252, 40 and 42. The power port 212 of spool valve 202 is in fluid communication with the track drive motor 32 via onduits 254, 44 and 46. The power port 222 of spool valve 204 is in fluid communication with the track drive motor 28 via conduits 256, 52 and 60. The power port 224 of spool valve 204 is in fluid communication with the track drive motor 38 via conduits 258, 56 and 62.

The hydraulic manual steering control assembly 64 preferably includes a conventional hydraulic differential flow control valve 260 having an inlet port 262 and a pair of signal outlet ports 264 and 266. The valve 260 may be suitably actuated by a manually operated control lever or the like as shown at 268. The inlet port 262 of the contorl valve 260 is in fluid communication with the automatic control assembly 18 and the first port

20a of the track drive pump 20 via conduits 66, 196, 118 and 22. The signal outlet 264 is in fluid communication with conduit 68 and the signal outlet 266 is in fluid communication with the conduit 70.

The sensor control assembly 72 includes a conventional hydraulic differential flow control valve 270 having an inlet port 272 and a pair of signal outlet ports 274 and 276. The signal outlet port 274 is in fluid communication with conduit 76 and the signal outlet port 276 is in fluid communication with conduit 78. The inlet port 272 is in fluid communication with the automatic control assembly 18 and the track drive pump 20 via conduits 74, 278, 118 and 22. As described above, a shut-off valve 86 is installed in the conduit 74 intermediate the inlet port 272 of the sensor control assembly 72 and the automatic control assembly 18. The hydraulic differential flow control valve 270 is of conventional construction and its operation will be readily apparent to those skilled in the art.

The sensor unit 80 is operatively connected to the hydraulic differential flow control valve 270 of the sensor control assembly 72 in order to actuate the valve 270 in response to variation in the position of the sensor unit 80 relative to the string line or grade line 84. The sensor unit 80 may be directly connected to the valve 270 by mechanical means or may be connected electrically to the valve 270 to provide actuation thereof. Such sensor units are well known in the art and need not be described in detail herein.

The valves 260 and 270 are commonly referred to as selector or directional control valves. The valve 260 is preferably of the low gain type while the valve 270 is preferably of the high gain type. In each of the valves 260 and 270 the cross-sectional flow area to one or the other signal outlet ports is directly proportional to the input signal from the control lever 268 or the sensor unit 80, respectively.

As shown in FIG. 3, the flow divider 120 comprises a valve body 280 and a valve member 282 slidably disposed therein. The valve body 280 includes a bore 284 extending therethrough and intersecting the opposite end portions 286 and 288 thereof. A counterbore 290 is formed in the bore 284 intersecting the end portion 286 and forming an annular wall 292 in the valve body 280. A counterbore 294 is formed in he bore 284 intersecting the opposite end portion 288 of the valve body 280 and forming an annular wall 296 therein.

A central annular chamber 298 is formed in the medial portion of the valve body 280 and includes opposite annular walls 3oo and 302 interconnected by a cylindrically shaped annular surface 304. The annular chamber 298 is coaxial with the bore 284.

A second annular chamber 306 is formed in the valve body 280 coaxial with the bore 284 and intermediate the counterbore 280 and the central annular chamber 298. The annular chamber 306 includes opposite annular walls 308 and 310 interconnected by a cylindrically shaped annular surface 312.

A third annular chamber 314 is formed in the valve body 280 coaxial with the bore 284 therethrough and intermediate the central annular chamber 298 and the counterbore 294. The annular chamber 314 includes opposite annular walls 216 and 218 interconnected by a cylindrically shaped annular surface 320.

The first signal inlet 122 of the flow divider 120 communicates with an annular chamber 322 formed by the counterbore 290. The second signal inlet 124 of the flow divider 120 communicates with an annular chamber 324 formed by the counterbore 294. The power inlet 126 of the flow divider 120 communicates with the central annular chamber 298. The first and second power outltes 128 and 130 of the flow divider 120 communicate with the annular chambers 306 and 314, respectively.

The valve member 282 is slidably dispose in the bore 284 of the valve body 280. The substantially cylindrically shaped outer periphery 326 of the valve member 282 has a diameter sized to provide a substantially fluid-tight seal between the outer periphery 326 and the walls of the bore 284. A cylindrically shaped extension 328 is formed on one end portion 329 of the valve member 282 and extends into the annular chamber 324. The extension 328 is coaxial with the outer periphery 326 of the valve member 282. An annular shoulder 330 interconnects the cylindrical surface 332 of the extension 328 and the cylindrically shaped outer periphery 326.

A bore is formed in the cylindrically shaped extension 328 coaxial with the cylindrical surface 332. A counterbore 336 is formed in the valve member 282 intersecting the end portion 338 thereof and communicating with the bore 334 in the extension 328. A removable orifice 340 is threadedly secured in the end portion 342 of the extension 328. A plug 344 having an aperture 346 formed therein is threadedly secured in the counterbore 336 at the end portion 338 of the valve member 282. A removable orifice 348 is threadedly secured in the aperture 346.

Two ports 350 are formed in the valve member 282 providing communication between the chamber 352 formed therein by the counterbore 336 and the central annular chamber 298 in the valve body 280.

A plurality of ports 354 are formed in the valve member 282 proximate to the end protion 338 thereof. The ports 354 provide fluid communication between the chamber 352 and the cylindrically shaped outer periphery 326 of the valve member 282. Similarly, a plurality of ports 356 are formed in the valve member 282 proximate to the end portion 329 thereof. The ports 356 provide fluid communication between the chamber 352 and the cylindrically shaped outer periphery 326 of the valve member 282. Preferably, the valve member 282 includes ten ports 354 and ten ports 356. The diameters of the ports 354 and 356 are preferably identical. As shown in FIG. 3, the ports 354 are divided with five ports on one side of the valve member 282 and five ports on the opposite side diiametrically opposed thereto. Similarly, the ports 356 are divided with five ports on one side of the valve member 282 and with five ports on the other side diametrically opposed thereto. The ports 354 and 356 are drilled or otherwise formed in the valve member 282 on centers substantially equal to the diameter of the ports. It should also be noted that the centers of the five ports 354 and 356 on one side of the valve member 282 are preferably staggered approximately one-half the diameter of a port from the centers of the ports 354 and 356, respectively, formed in the opposite side of the valve member 282. The staggered relation of the ports 354 and 356 provides smooth transition as the valve member 282 moves left or right within the valve body 280. It should be noted that the arrangement and/or diameter of the ports 354 and 356 may be varied to achieve other than linear flow response to movement of the valve member

282 within the valve body 280. It should further be noted that other forms of ports such as long slots may be substituted for the previously described plurality of ports 354 and 356.

As shown in FIG. 3, the valve member 282 is positioned in its center of medial position within the valve body 280. In this position, it will be observed that five ports 354 communicate between the chamber 352 of the valve member 282 and the annular chamber 306 of the valve body 280. Similarly, five ports 356 provide communication between the chamber 352 of the valve member 282 and the annular chamber 314 of the valve body 380. The remaining ports 354 and 356 are blocked by the cylindrical walls of the bore 284 and therefore provide no communication between the chamber 352 and the chambers 306 and 314.

As will be readily apparent to those skilled in the art, as the valve member 282 moves either left or right of the central or medial position the cross-sectional area of the ports 354 and 356 providing fluid communication between the chamber 352 and the cylindrically shaped outer periphery 326 of the valve member 282 always remains equal to the total cross-sectional area of 10 ports.

The valve member 282 is urged into its central or medial position relative to the valve body 280 as shown in FIG. 3, by means of a spring assembly 358 disposed within the chamber 324 formed by the counterbore 294 in the valve body 280. The spring assembly comprises two identical annular spring seats 360 and 362. Each spring seat 360 and 362 has a cylindrically shaped aperture 364 formed therein and is L-shaped in cross-section with an outwardly extending flange portion 366 and a cylindrically shaped portion 368. An annular end wall 370 is formed on each flange portion 366.

The spring seat 360 is slidably disposed on the cylindrically shaped extension 328 with the extension 328 extending through the aperture 364 formed therein. The end wall 370 thereof abuts the annular shoulder 300 of the valve member 282 and also abuts the annular wall 296 of the valve body 280 when the valve member 282 is positioned in its central or medial position. The annular spring seat 362 is slidably disposed on the cylindrically shaped extension 328 with the extension 328 extending through the aperture 364 therein and with the end wall 370 thereof facing away from the annular spring seat 360. The annular spring seats 360 and 362 are retained on the cylindrically shaped extension 328 by means of a snap ring 372 disposed in an annular groove 374 formed in the cylindrical surface 332 of the extension 328.

The spring assembly 358 further includes a coil spring 376 disposed about the cylindrically shaped extension 328. The coil spring 376 is supported at each end thereof by the respective cylindrically shaped portion 368 of the respective annular spring seat 360 and 362. An annular sleeve 378 having an L-shaped cross-section and having a first end face 380 and a second end face 382 is disposed within the counterbore 294 with the first end face 380 thereof abutting the end wall 370 of the annular spring seat 362.

End plates 384 and 386 are secured respectively to the opposite end portions 286 and 288 of the valve body 280 by suitable means such as a plurality of threaded cap screws 388. A suitable fluid-tight seal is provided between the end plates 384 and 386 and the valve body 280 by means of O-rings 390 disposed in annular grooves 392 formed in the end plates 384 and 386.

The second end face 382 of the annular sleeve 378 abuts the end plate 386. Movement of the valve member 282 within the valve body 280 is yieldably resisted by the urging of the coil spring 376. Movement of the valve member 282 to the right, as viewed in FIG. 3, is resisted by the urging of the coil spring 376 acting through the end plate 386, the annular sleeve 378, the spring seat 362, the coil spring 376, the spring seat 360, and the annular shoulder 330 of the valve member 282. Movement of the valve member 282 to the left, as viewed in FIG. 3, is resisted by the urging of the coil spring 376 acting through the annular wall 296 of the valve body 280, the spring seat 360, the coil spring 376, the spring seat 362, the snap ring 372, and the cylindrically shaped extension 328 of the valve member 282.

As shown in FIG. 4, the flow compensator 176 comprises a valve body 394 and a valve member 396 slidably disposed therein. The valve body 394 includes a bore 398 extending therethrough and intersecting the opposite end portions 400 and 402 thereof. A counterbore 404 is formed in the bore 398 intersecting the end portion 402 and forming an annular wall 406 in the valve body 394.

A first annular chamber 408 is formed in the valve body 394 and includes opposite annular walls 410 and 412 interconnected by a cylindrically shaped annular surface 414. The annular chamber 408 is coaxial with the bore 398.

A second annular chamber 416 is formed in the valve body 394 coaxial with the bore 398. The annular chamber 416 includes opposite and annular walls 418 and 420 interconnected by a cylindrically shaped annular surface 422.

The first power inlet 178 of the flow compensator 176 communicates with the first annular chamber 408. The second power inlet 180 of the flow compensator 176 communicates with the second annular chamber 416.

A third annular chamber 424 is formed in the valve body 394 coaxial with the bore 398 therethrough and intermediate the first annular chamber 408 and the end portion 400 of the valve body 394. The annular chamber 424 includes opposite annular walls 426 and 428 interconnected by a cylindrically shaped annular surface 430.

A fourth annular chamber 432 is formed in the valve body 394 coaxial with the bore 398 therethrough and intermediate the second annular chamber 416 and the end portion 402 of the valve body 394. The annular chamber 432 includes opposite annular walls 434 and 436 interconnected by a cylindrically shaped annular surface 438.

The first power outlet 182 of the flow compensator 176 communicates with the third annular chamber 424; and the second power outlet 184 of the flow compensator 176 communicates with the fourth annular chamber 432.

The valve member 396 is slidably disposed in the bore 398 of the valve body 394. The cylindrically shaped outer periphery 440 of the valve member 396 has a diameter sized to provide a substantially fluid-tight seal between the outer periphery 440 and the walls of the bore 398. The valve member 396 includes opposite end faces 442 and 444, each lying in a plane normal to the longitudinal axis of the valve member

396. A cylindrically shaped chamber **446** is formed in one end of the valve member **396** with one end thereof intersecting the end face **442.** A second cylindrically shaped chamber **448** is formed in the opposite end of the valve member **396** with one end thereof intersecting the end face **444.** It should be noted that the chambers **446** and **448** do not communicate with each other within the valve member **396.**

Two ports **450** are formed in the valve member **396** providing communication between the chamber **446** formed therein and the first annular chamber **408** formed in the valve body **394.** Two ports **452** are formed in the valve member **396** providing communication between the chamber **448** formed therein and the second annular chamber **416** formed in the valve body **394.** Two ports **454** are formed in the valve member **936** providing communication between the chmaber **446** formed therein and the third annular chamber **424** formed in he valve body **394.** Two ports **456** are formed in the valve member **396** providing communication between the chamber **448** formed therein and the fourth annular chamber **432** formed in the valve body **394.**

As shown in FIG. **4,** the member **396** is positioned in its center or medial position within the valve body **394.** It will be observed that the centerline of the ports **454** and the centerline of the ports **456** are positioned substantially in the planes of the annular walls **426** and **436** of the third and fourth annular chambers **424** and **432,** respectively.

End plates **458** and **460** are secured respectively to the opposite end portions **400** and **402** of the valve body **394** by suitable means such as a plurality of threaded cap screws **462.** A suitable fluid-tight seal is provided between the end plates **458** and **460** and the valve body **394** by means of O-rings **464** disposed in annular grooves **466** formed in the end plates **458** and **460.**

The valve member **396** is of such length that when it is displaced to the extreme left, as viewed in FIG. **4,** with the end face **442** thereof abutting the end plate **458,** the cross-sectional area of fluid communication between the chamber **446** of the valve member **396** and the third annular chamber **424** of the valve body **394** via the ports **454** is closed off entirely. At the same time, the communication between the chamber **448** and the valve member **396** with the fourth annular chamber **442** of the valve body **394** via the ports **456** is at a maximum. Alternately, when the valve member **396** is displaced to the extreme right, as viewed in FIG. **4,** with the end face **44** thereof abutting the end plate **460,** the cross-sectional area of fluid communication between the chamber **448** of the valve member **396** with the fourth annular chamber **432** of the valve body **394** is closed off entirely. At the same time, communication between the chamber **446** of the valve member **396** with the third annular chamber **424** of the valve body **394** via the ports **454** is at a maximum. It should be noted that regardless of the relative position of the valve member **396** within the valve body **394** the communication between the chamber **446** of the valve member **396** and the first annular chamber **408** of the valve body **394** remains unchanged. Similarly, the communication between the chamber **448** of the valve member **396** and the second annular chamber **416** of the valve body **394** via the ports **452** remains unchanged regardless of the position of the valve member **396** relative to the valve body **394.**

It should be noted that the previously described flow divider **120** and flow compensator **176** may be advantageously housed in a single unitary valve body. Such a valve body would include all of the features described for the valve bodies **280** and **394** of the flow divider **120** and flow compensator **176,** respectively, and would place the first and second power outlets **128** and **130** of the flow divider **120** in fluid communication with the first and second power inlets **178** and **180,** respectively, of the flow compensator **176** within the combined valve body. Furthermore, the end plates **458** and **384** may be advantageously combined into one end plate as may end plates **460** and **386.** In either configuration the functions of the flow divider **120** and the flow compensator **176** would be identical.

OPERATION OF THE EMBODIMENT OF FIGS. 2, 3 AND 4

In operation, the track drive pump **20** is driven by the engine and power take-off (not shown) to provide a source of pressurized hydraulic fluid. The pressurized hydraulic fluid provided by the pump **20** may be in either a low pressure range of from 600 to 1,500 psi or in a high pressure range of from 1,500 to 3,500 psi. While these pressure ranges are disclosed as oreferable for the operation of the present invention, it is not intended that the present invention be limited thereby.

The pressurized hydraulic fluid is directed from the first port **20*a*** of the pump **20** through conduits **22** and **118** to the inlet **116** of the differential valve **114.** The hydraulic fluid passes through the differential valve **114** and exits from the outlet **132** thereof. The differential valve **114** provides a pressure differential of approximately **75** psi between the high pressure inlet **116** in the lower pressure outlet **132.** The hydraulic fluid is then directed from the outlet **132** through conduit **134** to the power inlet **126** of the flow divider **120.**

Pressurized hydraulic fluid is also directed from the pump **20** via conduits **22, 118, 278** and **74** to the inlet port **272** of the sensor control assembly **72.** In order for the pressurized hydraulic fluid to reach the inlet port **272** of the sensor control assembly **72** it is necessary for the shut-off valve **86,** installed in conduit **74,** to be in its open position. If the shut-off valve **86** is in the closed position the sensor control assembly **72** is deactivated and performs no function in the operation of the present invention. The pressurized hydraulic fluid flows from the inlet port **272** through the differential flow control valve **270** of the sensor control assembly **72** and exits therefrom through the signal outlet port **274** and **276.** The hydraulic fluid emanating from the signal outlet port **274** flows to the first inlet **148** of shuttle valve **136** via conduits **76** and **152.** The hydraulic fluid emanating from the signal outlet port **276** flows to the first inlet **158** of shuttle valve **142** via conduits **78** and **162.**

Pressurized hydraulic fluid also flows from the pump **20** to the inlet port **262** of the hydraulic differential flow control valve **260** of the hydraulic manual steering control assembly **64** via conduits **22, 118, 196** and **66.** The hydraulic fluid entering the inlet port **262** passes through the hydraulic differential flow control valve **260** and exits therefrom through signal outlets **264** and **266.** A portion of the hydraulic fluid emanating from the signal outlet **264** flows to the second inlet **150** of the shuttle valve **136** via conduits **68, 156** and **154.** The remaining portion of the hydraulic fluid emanating from the signal outlet **264** flows to the power inlet **126** of the flow divider **120** via conduits **68, 156, 168** and **134.** It should be noted that the last-mentioned hydraulic fluid must also pass through orifice **170** in conduit **168.** The passage of the hydraulic fluid through the orifice **170** provides the pressure drop required in order for the hydraulic fluid to reach the lowered pressure of the hydraulic fluid emanating from the outlet **132** of the differential valve **114.**

A portion of the hydraulic fluid emanating from the signal outlet **266** flows to the second inlet **160** of the shuttle valve **142** via conduits **70, 166** and **164.** The remaining portion of the hydraulic fluid emanating from the signal outlet **266** flows to the power inlet **126** of the flow divider **120** via conduits **70, 166, 172** and **134.** It should be noted that this hydraulic fluid must also pass through orifice **174** in the conduit **172** which provides the necessary pressure drop for the hydraulic fluid to reach the lower pressure of the hydraulic fluid emanating from the outlet **132** of the differential valve **114.**

Pressurized hydraulic fluid entering either of the inlets **148** or **150** of the shuttle valve **136** exits therefrom through outlet **138** and flows to the first signal inlet **122** of the flow divider **120** via conduit **140.** Pressurized hydraulic fluid entering either of the inlets **158** or **160** of the shuttle valve **142** exits therefrom through outlet **144** and flows to the second signal inlet **124** of the flow divider **120** via conduit **146.**

The hydraulic fluid entering the power inlet **126** of the flow divider **120** passes through the power inlet **126** into the chamber **352** in the valve member **282** via the central annular chamber **298** of the valve body **280** and the ports **350** formed in the valve member **282.** The hydraulic fluid entering the first signal inlet **122** of the flow divider **120** flows into the annular chamber **322** of the valve body **280** through the first signal inlet **122,** and then flows from the annular chamber **322** into the chamber **352** of the valve member **282** through the orifice **348.** The hydraulic fluid entering the second signal inlet **124** of the flow divider **120** flows into the annular chamber **324** of the valve body **280** through the second signal inlet **124,** and then flows from the annular chamber **324** into the chamber **352** of the valve member **282** through the orifice **340** and the bore **334** of the extension **328.**

It should be noted that the hydraulic fluid in the annular chambers **322** and **324** is at a pressure approximately 75 psi greater than the hydraulic fluid contained within the chamber **352** of the valve member **282.** As the hydraulic fluid enters the annular chambers **322** and **324** flows into the chamber **352** through the respective orifices **348** and **340** the pressure drops to that of the hydraulic fluid within the chamber **352** of the valve member **282.**

The hydraulic fluid in the chamber **352** of the valve member **282** flows therefrom through the ports **354** and **356** into the second and third annular chambers **306** and **314** of the valve body **280,** respectively. The hydraulic fluid entering the second annular chamber **306** through the ports **354** exits therefrom through the first power outlet **128;** and the hydraulic fluid entering the third annular chamber **314** through the ports **356** exits therefrom through the second power outlet **130.** The hydraulic fluid exiting from the first power outlet **128** of the flow divider **120** flows therefrom into the first power inlet **178** of the flow compensator **176.** The hydraulic fluid exiting from the second power outlet **150** of the flow divider **120** flows therefrom into the second power inlet **180** of the flow compensator **176.**

Orifices **170, 174, 340** and **348** can be sized to provide various steering rates in response to manual and sensor signal inputs.

The hydraulic fluid entering the first power inlet **178** of the flow compensator **176** flows therefrom into the chamber **446** of the valve member **396** through the first annular chamber **408** of the valve body **394** and the ports **450** of the valve member **396.** The hydraulic fluid entering the second power inlet **180** of the flow compensator **176** flows therefrom into the chamber **448** formed in the valve member **396** through the second annular chamber **416** of the valve body **394** and the ports **452** formed in the valve member **396.** The hydraulic fluid in the chamber **446** of the valve member **396** flows therefrom through the ports **454** formed in the valve member **396** and the third annular chamber **424** of the valve body **394** and exits from the flow compensator **176** through the first power outlet **182.** The hydraulic fluid in the chamber **448** of the valve member **396** flows therefrom through the ports **456** in the valve member **396** and the fourth annular chamber **432** of the valve body **394** and exits from the flow compensator **176** through the second power outlet **184.**

The hydraulic fluid exiting from the first power outlet **182** of the flow compensator **176** flows therefrom to the power port **206** of the forward-reverse spool valve **202** via conduit **242.** The hydraulic fluid exiting from the second power outlet **182** of the flow compensator **176** flows therefrom to the power port **218** of forward-reverse spool valve **204** via conduit **244.** When the forward-reverse spool valves **202** and **204** are in their forward positions, the hydraulic fluid entering power port **206** of spool valve **202** exits therefrom through power port **210,** and the hydraulic fluid entering the power port **218** of spool valve **204** exits therefrom through power port **222.**

The hydraulic fluid exiting from power port **210** of spool valve **202** flows therefrom through conduits **252, 40** and **42** to track drive motor **32** thereby driving the motor **32** in a forward direction. Most of the hydraulic fluid passing through the track drive motor **32** is routed therefrom back to the power port **212** of the spool valve **202** via conduits **44, 46** and **254.** Excess hydraulic fluid volume bypassed by the internal valving of the track drive motor **32** is directed from bypass outlet **92** through conduits **94, 96** and **98** to heat exchanger **88** and from heat exchanger **88** through conduit **90** to reservoir **26.**

Hydraulic fluid exiting from power port **222** of spool valve **204** flows therefrom through conduits **256, 52** and **60** to track drive motor **38** thereby driving the motor **38** in a forward direction. Most of the hydraulic fluid passing through track drive motor **38** is routed therefrom back to power port **224** of spool valve **204** via conduits **56, 62** and **258.** Excess hydraulic fluid volume bypassed by the internal valving of track drive motor **38** is directed from bypass outlet **110** through conduits **108, 112** and **98** to heat exchanger **88** and from heat exchanger **88** through conduit **90** to reservoir **26.**

Hydraulic fluid entering power port **212** of spool valve **202,** when spool valve **202** is in the forward position, exits therefrom through power port **208** and flows back to track drive pump **20** via conduits **246, 248** and

24. Similarly, when spool valve **204** is in the forward position hydraulic fluid entering power port **224** exits therefrom through power port **220** and flows back to the track drive pump **20** via conduits **250, 248** and **24.**

To place the forward-reverse spool valves **202** and **204** in the proper forward or reverse positions, hydraulic fluid is directed from the first port **20a** of the pump **20** through the forward-reverse control valve **186** to the appropriate control ports on the spool valves **202** and **204.** Pressurized hydraulic fluid flows to the inlet port **188** of the forward-reverse control valve **186** from the pump **20** via conduits **22, 118, 196** and **194.** When the forward-reverse control valve **186** is in the forward position, as shown in FIG. **2,** the hydraulic fluid entering the inlet port **188** flows through the valve **186** and exits from the forward outlet port **190.** The hydraulic fluid exiting from the forward outlet port **190** communicates with the control ports **214** and **226** of the spool valves **202** and **204,** respectively. Fluid communication between the forward outlet port **190** and control port **214** of spool valve **202** is accomplished via conduits **230** and **232;** and fluid communication between the forward outlet port **190** and the control port **226** of spool valve **204** is accomplished via conduits **230** and **234.**

When the forward-reverse control valve **186** is placed in the reverse position by moving the control lever **198** in the direction of the arrow **200,** the pressurized hydraulic fluid entering inlet port **188** of the valve **186** exits therefrom through reverse outlet port **192.** The reverse outlet port **192** is in fluid communication with control ports **216** and **228** of spool valves **202** and **204,** respectively. The pressurized hydraulic fluid emanating from reverse outlet port **192** communicates with control port **216** via conduits **236** and **238** thereby placing spool valve **202** in the reverse position. The hydraulic fluid emanating from reverse outlet port **192** also communicates with control port **228** of spool valve **204** via conduits **236** and **240** thereby placing spool valve **204** in the reverse position.

At this point it should be noted that, while a single forward-reverse control valve **186** is disclosed for simultaneously controlling the forward-reverse spool valves **202** and **204,** it will be understood that it may be preferable to control the spool valves **202** and **204** individually by separate forward-reverse control valves similar to the previously described valve **186.** In addition, the spool valves **202** and **204** may be individually controlled by separate mechanical means such as manually operated levers directly connected to the valves **202** and **204,** respectively. Such individual control of the spool valves **202** and **204** will permit one spool valve to be placed in the reverse mode while the other spool valve is simultaneously placed in the forward mode thereby permitting the vehicle **12** to make spot turns to the left or the right about a vertical axis.

When forward-reverse spool valve **202** is in the reverse position, hydraulic fluid entering the valve **202** through power port **206** exits the valve **202** through power port **212** and flows to the track drive motor **32** via conduits **254, 44** and **46** thereby driving the track drive motor **32** in the reverse direction. Most of the hydraulic fluid passing through track drive motor **32** flows therefrom through conduits **40, 42** and **252** to power port **210** of spool valve **202.** The hydraulic fluid entering power port **210** passes through spool valve **202** and exits from power port **208** to return to the second port **20b** of the pump **20** via conduits **246, 248** and **24.**

When forward-reverse spool valve **204** is in the reverse position, hydraulic fluid entering port **218** of the spool valve **204** exits therefrom through power port **224** and flows through conduits **258, 56** and **62** to track drive motor **38** thereby driving track drive motor **38** in the reverse direction. Most of the hydraulic fluid passing track drive motor **38** flows therefrom through conduits **52, 60** and **256** to power port **222** of spool valve **204.** Hydraulic fluid entering power port **222** passes through spool valve **204** and exits from power port **220** to return to the second port **20b** of the pump **20** via conduits **250, 248** and **24.**

When track drive motors **32** and **38** are driven in the reverse direction, excess hydraulic fluid volume bypassed by the internal valving of the track drive motors **32** and **38** is directed from bypass outlets **92** and **110,** respectively, back to the reservoir **26** as previously described in detail for the forward operation of the track drive motors **32** and **38.**

It will be readily apparent to those skilled in the art that the valving described for providing the reversing of the flow of hydraulic fluid to the track drive motors **32** and **38** may be actuated in any number of well-known ways. One such way would be by the application of direct mechanical force to the valves **202** and **204** either in substitution for the hydraulic actuation means previously described, or as a back-up actuation system in addition to the hydraulic actuation system. Another suitable actuation means would be the utilization of electric solenoids for actuation of the valves **202** and **204.**

Automatic control of the output of track drive motors **32** and **38** is accomplished by properly setting the sensor control assembly **72** such that the tracer **82** will properly engage the string-line or grade-line **84** to provide actuation of the hydraulic differential flow control valve **270.** When the sensor control assembly **72** is properly adjusted on the apparatus **10,** as illustrated in FIG. **1,** the shut-off valve **86** is placed in the open position to permit the flow of pressurized hydraulic fluid from the pump **20** through conduits **22, 118, 278** and **74** to the inlet port **272** of the hydraulic differential flow control valve **270.**

The forward-reverse control valve **186** is placed in the forward position thereby causing the spool valves **202** and **204** to be placed in the forward position when the pump **20** is driven by the engine through the power take-off, as shown in FIG. **2.** When the apparatus **10** is in proper alignment with the string line **84,** as sensed by the sensor control assembly **72,** pressurized hydraulic fluid from the pump **20** passes through the appropriate conduits through the differential valve **114** and into the flow divider **120** where the hydraulic fluid entering power inlet **126** thereof is equally divided and emanates from the flow divider **120** through the first and second power outlets **128** and **130** in streams having substantially equal flow rates. The hydraulic fluid emanating from the power outlets **128** and **130** enters the flow compensator **176** through the first and second power inlets **178** and **180** thereof, respectively. Assuming that the loads encountered by track drive motors **32** and **38** are equal, the separate streams of hydraulic fluid entering the first and second power inlets **178** and **180** exit the first and second power outlets **182** and **184,** respectively, of the flow compensator **176** at equal flow rates. The hydraulic fluid emanating from the first power outlet **182** flows through appropriate conduits and the spool valve **202** to the track drive motor **32** thereby driving the motor **32** in a forward direction. Similarly, the hydraulic fluid emanating from the second power outlet **184** flows through appropriate conduits and the spool valve **204** to the track drive motor **38** thereby driving the motor **38** in the forward direction.

Since the rates of flow of hydraulic fluid exiting from the first and second power outlets **182** and **184** of the flow compensator **176** are equal, the respective track drive motors **32** and **38** are therefore driven at the same speed. It will, therefore, be readily apparent that the apparatus **10** will then be driven alongside the string line **84** on a path parallel thereto.

If for some reason the apparatus **10** should deviate from the desired path parallel to the string line **84**, the sensor control assembly **72** will sense the movement of the apparatus **10** away from or into the string line **84** and, through the sensor unit **80** and the tracer **82**, the hydraulic differential flow control valve **270** will be actuated to cause automatic correction of the path of the apparatus **10** to bring it back to the desired path parallel to the string line **84**.

If the apparatus **10** deviates from the desired path toward the string line **84** the deviation is sensed by the sensor control assembly **72** which causes the hydraulic differential flow control valve **270** to be actuated by the sensor unit **80** causing hydraulic fluid to emanate from the signal outlet port **276** while there is no flow of hydraulic fluid emanating from the signal outlet port **274**. The hydraulic fluid exiting from the signal outlet port **276** flows through conduits **78** and **162** into the first inlet **158** and out the outlet **144** of shuttle valve **142**. The hydraulic fluid exiting from the outlet **144** flows through conduit **146** into the second signal inlet **124** of the flow divider **120**.

As best shown in FIG. **3**, the hydraulic fluid entering the second signal inlet **124** of the flow divider **120** flows into the annular chamber **324** formed therein and exerts hydraulic pressure on the valve member **282** thereby urging the valve member **282** to the left within the valve body **280**, as viewed in FIG. **3**. The hydraulic fluid within the annular chamber **324** flows therefrom through orifice **340** into chamber **352** in the valve member **282**. As the hydraulic fluid flows through the orifice **340**, the pressure thereof drops to that of the hydraulic fluid within the chamber **352**.

Since hydraulic fluid is entering the second signal inlet **124** while there is no flow of hydraulic fluid entering the first signal inlet **122**, the hydraulic pressure urging the valve member **282** to the left is greater than the hydraulic pressure urging the valve member **282** to the right. Due to this pressure differential being exerted on the valve member **282**, the valve member **282** is displaced to the left an amount proportional to the differential in these two hydraulic pressures thereby overcoming, to some extent, the urging of spring assembly **358**.

When the valve member **282** moves from the center position within the valve body **280** to the left, it will be readily apparent that the cross-sectional area of fluid communication between the chamber **352** and the third annular chamber **314** afforded by the ports **356** is increased an amount proportional to the displacement of the valve member **282** to the left. Similarly, it will also be readily apparent that the cross-sectional area of fluid communication between the chamber **352** and the second annular chamber **306** afforded by the ports **354** is decreased in an amount proportional to the displacement of the valve member **282** to the left within the valve body **280**.

As a result of the previously described displacement of the valve member **282** to the left within the valve body **280**, in response to the differential hydraulic signal received from the sensor control assembly **72**, the rate of flow of hydraulic fluid emanating from the third annular chamber **314** through the second power outlet **130** is proportionally greater than the rate of flow of the hydraulic fluid emanating from the second annular chamber **306** through the first power outlet **128**. The stream of hydraulic fluid emanating from the second power outlet **130** flows through the flow compensator **176** and the spool valve **204** in the previously described conduits to the track drive motor **38** while the stream of hydraulic fluid emanating from the first power outlet **128** flows through the flow compensator **176** and the spool valve **202** and the previously described conduits to the track drive motor **32**. Since the rate of flow of the stream of hydraulic fluid entering track drive motor **38** is proportionally greater than the rate of flow of the stream of hydraulic fluid entering track drive motor **32**, the speed of the track drive motor **38** is increased proportionally over the speed of the track drive motor **32** thereby causing the apparatus **10** to swing away from the string line **84** back toward its proper line of direction parallel to the string line **84**.

As the apparatus **10** approaches proper alignment with the string line **84**, the sensor control assembly **72** senses the approach of the apparatus **10** toward proper alignment and causes the hydraulic differential flow control valve **270** to be gradually actuated back into its neutral position so that hydraulic fluid ceases to emanate from the signal outlet port **276** thereof when the apparatus **10** is again in proper alignment with the string line **84**. When hydraulic fluid flow from the control valve **270** ceases, the hydraulic pressures acting on the opposite ends of the valve member **282** of the flow divider **120** are equal thus permitting the valve member **282** to be properly centered within the valve body **280** by the spring assembly **358**. When the valve member **282** is in the center position the rates of flow of hydraulic fluid emanating from the first and second power outlets **128** and **130** of the flow divider **120** are substantially equal and the resulting speeds of the track drive motors **32** and **38** are also substantially equal.

If the apparatus **10** deviates from the desired path away from the string line **84**, the deviation is sensed by the sensor control assembly **72** which causes the hydraulic differential control valve **270** to be actuated by the sensor unit **80** causing hydraulic fluid to emanate from the signal outlet port **274** while there is no flow of hydraulic fluid emanating from the signal outlet port **276**. The hydraulic fluid exiting from the signal outlet port **274** flows into the first signal inlet **122** of the flow divider **120** via the previously described conduits and shuttle valve **136**.

As described above, the hydraulic fluid entering the second signal inlet **124** of the flow divider **120** flows into the annular chamber **324** and exerts hydraulic pressure on the valve member **282** thereby urging the valve member **282** to the left within the valve body **280**, as viewed in FIG. **3**. The hydraulic fluid within the annular chamber **324** flows therefrom through orifice **340** into chamber **352** in the valve member **282.** As the hydraulic fluid flows through the orifice **340** the pressure thereof drops to that of the hydraulic fluid within the chamber **352.**

Since hydraulic fluid is entering the first signal inlet **122** while there is no flow fo the hydraulic fluid entering the second signal inlet **124,** the hydraulic pressure urging the valve member **282** to the right is greater than the hydraulic pressure urging the valve member **282** to the left. Due to this pressure differential being exerted on the valve member **282,** the valve member **282** is displaced to the right in an amount proportional to the differential in these two hydraulic pressures thereby overcoming, to some extent, the urging of spring assembly **358.**

When the valve member **282** moves from the center position within the valve body **280** to the right, it will be readily apparent that the cross-sectional area of fluid communication between the chamber **352** and the third annular chamber **314** afforded by the ports **356** is decreased an amount proportional to the displacement of the valve member **282** to the right. Similarly, it will be also readily apparent that the cross-sectional area of fluid communication between the chamber **352** and the second annular chamber **306** afforded by the ports **352** is increased an amount proportional to the displacement of the valve member **282** to the right within the valve body **280.**

As a result of the previously described displacement of the valve member **282** to the right within the valve body **280,** in response to the differential hydraulic signals received from the sensor control assembly **72,** the rate of flow of hydraulic fluid emanating from the second annular chamber **306** through the first power outlet **128** is proportionally greater than the rate of flow of the hydraulic fluid emanating from the third annular chamber **314** through the second power outlet **130.** As described above, the stream of hydraulic fluid emanating from the second power outlet **130** flows through the flow compensator **176** and the spool valve **204** to the track drive motor **38** while the stream of hydraulic fluid emanating from the first power outlet **128** flows through the flow compensator **176** and the spool valve **202** to the track drive motor **32.** Since the rate of flow of the stream of hydraulic fluid entering track drive motor **32** is proportionally greater than the rate of flow of the stream of hydraulic fluid entering track drive motor **38,** the speed of the track drive motor **32** is increased proportionally over the speed of track drive motor **38** thereby causing the apparatus **10** to swing back toward the string line **84** and its proper line of direction parallel to the string line **84.**

As the apparatus **10** approaches proper alignment with the string line **84,** the sensor control assembly **72** senses the approach of the apparatus **10** toward proper alignment and causes the hydraulic differential flow control valve **270** to be gradually actuated back into its neutral position so that hydraulic fluid ceases to emanate from the signal outlet port **274** thereof when the apparatus **10** is again in proper alignment with the string line **84.** When hydraulic fluid flow from the control valve **270** ceases, the hydraulic pressures acting on the opposite ends of the valve member **282** in the flow divider **120** are equal thus permitting the valve member **282** to be properly centered within the valve body **280** by the spring assembly **358.** When the valve member **282** is in the center position the rates of flow of hydraulic fluid emanating from the first and second power outlets **128** and **130** of the flow divider **120** are substantially equal and the resulting speeds of the track drive motors **32** and **38** are also substantially equal.

It will be readily apparent to those skilled in the art that the operation of the sensor control assembly **72** and the flow divider **120** during the forward movement of the apparatus **10** applies equally when the apparatus **10** is operated in the reverse direction. The apparatus **10** is placed in condition to operate in the reverse direction by placing the forward-reverse control valve **186** in the reverse position thereby causing the spool valves **202** and **204** to be placed in the reverse position when the pump **20** is driven by the engine through the power take-off. Automatic correction of the path of the apparatus **10** when deviating from the proper path parallel to string line **84** is accomplished just as described above for forward operation of the apparatus **10.**

When the apparatus **10** is moving in the forward direction and is following the proper path parallel to the string line **84,** it is not uncommon for the track drive motors **32** and **38** driving the apparatus **10** through the respective hydraulically driven track units **16** to encounter different loads thereby requiring greater power from one track drive motor than from the other in order for the apparatus **10** to continue along the proper path. The apparatus of the present invention provides for the automatic adjustment of the power outlets of the track drive motors through the action of the flow compensator **176.**

As previously described above, when the apparatus **10** is properly following a path parallel to the string line **84** the flow rates of the streams of hydraulic fluid emanating from the first and second power outlets **128** and **130** of the flow divider **120** are equal. When the track drive motors **32** and **38** are each encountering the same amount of load, the streams of hydraulic fluid entering the first and second power inlets **178** and **180** of the flow compensator **176** from the first and second power outlets **128** and **130** of the flow divider **120,** are also equal. The stream of hydraulic fluid entering the first power inlet **178** flows into the chamber **446** formed in the valve member **396** through the ports **450.**

Similarly, the stream of hydraulic fluid entering the second power inlet **180** flows into the chamber **448** formed in the valve member **396** through ports **452.** The hydraulic fluid in the chamber **446** flows through ports **454** into the third annular chamber **424** and out therefrom through the first power outlet **182** of the flow compensator **176.** The hydraulic fluid in the chamber **448** flows therefrom through the ports **456** into the fourth annular chamber **432** and out therefrom through the second power outlet **184** of the flow compensator **176.**

The stream of hydraulic fluid emanating from the first power outlet **182** flows through the spool valve **202** to the track drive motor **32** through the previously described conduits. The hydraulic stream emanating from the second power outlet **184** flows therefrom through the spool valve **204** to the track drive motor **38** through the previously described interconnecting conduits.

If the track drive motor **32,** for example, encounters a greater load than the track drive motor **38** the rate of flow of hydraulic fluid into the track drive motor **32** is reduced thus increasing the pressure of the hydraulic fluid in the conduits interconnecting the first power outlet 182 and the track drive motor 32 over the hydraulic pressure of the hydraulic fluid in the conduits interconnecting the track drive motor 38 and the second power outlet 184. The increase in hydraulic pressure at the first power outlet 182 over the hydraulic pressure at the second power outlet 184 is communicated through the ports 454 and 456 into the respective chambers 446 and 448. Since the hydraulic pressure in the chamber 446 is greater than the hydraulic pressure in the chamber 448, the valve member 396 is displaced to the right within the valve body 394 in an amount proportional to the pressure differential between the hydraulic fluid in the chamber 446 and the hydraulic fluid in the chamber 448.

As the valve member 396 is displaced to the right within the valve body 394, as viewed in FIG. 4, the cross-sectional area of fluid communication between the chamber 446 and the third annular chamber 424 afforded by the ports 454 is increased while the cross-sectional area of fluid communication between the chamber 448 and the fourth annular chamber 432 afforded by the ports 456 is decreased. The maximum displacement of the valve member 396 to the right within the valve body 394 is mechanically limited by the abutment of the end face 444 of the valve member 396 with the end plate 460. It should be noted that when the end face 444 abuts the end plate 460, fluid communication between the chamber 448 and the fourth annular chamber 432 through the ports 456 is preferably completely eliminated.

It will be readily apparent that displacement of the valve member 396 to the right within the valve body 394 will decrease the cross-sectional area of fluid communication between the second power inlet 180 and the second power outlet 184 and simultaneously increase the cross-sectional area of fluid communication between the first power inlet 178 and the first power outlet 182. In effect, a dummy load is introduced between the second power inlet 180 and the second power outlet 184 which results in substantially equal hydraulic pressure drops between the first power inlet 178 and the first power outlet 182, and between the second power inlet 180 and the second power outlet 184. Thus, the hydraulic fluid streams emanating from the first and second power outlets 128 and 130 of the flow divider 120 will encounter substantially equal loads regardless of the loads encountered by the track drive motors 32 and 38.

On the other hand, if the track drive motor 38 encounters a greater load than that encountered by the track drive motor 32, the pressure of the hydraulic fluid communicating between the chamber 448 and the track drive motor 38 will be increased proportionally over the pressure of the hydraulic fluid communicating between the chamber 446 and the track drive motor 32. This differential in hydraulic pressure causes the displacement of the valve member 396 to the left within the valve body 394 in an amount proportional to the pressure differential between the hydraulic fluid in the chamber 448 and the hydraulic fluid in the chamber 446.

As the valve member 396 is displaced to the left in the valve body 394, as viewed in FIG. 4, the cross-sectional area of fluid communication between the chamber 448 and the fourth annular chamber 432 afforded by the ports 456 is increased while the cross-sectional area of fluid communication between the chamber 446 and the third annular chamber 424 is decreased. The maximum displacement of the valve member 396 to the left within the valve body 394 is mechanically limited by abutment of the end face 442 of the valve member 396 with the end plate 458. It should be noted that when the end face 442 abuts the end plate 458, fluid communication between the chamber 446 and the third annular chamber 424 through the ports 454 is preferably completely eliminated.

It will be readily apparent that the displacement of the valve member 396 to the left within the valve body 394 will increase the cross-sectional area of fluid communication between the second power inlet 180 and the second power outlet 184 and simultaneously decrease the cross-sectional area of fluid communication between the first power inlet 178 and the first power outlet 182. In effect, as noted above, a dummy load is introduced between the first power inlet 178 and the first power outlet 182 which results again in substantially equal hydraulic pressure drops between the first power inlet 178 and the first power outlet 182, and between the second power inlet 180 and the second power outlet 184. Thus, again, the hydraulic fluid streams emanating from the first and second power outlets 128 and 130 of the flow divider 120 will continue to encounter substantially equal loads regardless of the loads encountered by the track drive motors 32 and 38.

It should also be noted that the operation of the flow compensator 176 when the apparatus 10 is in proper condition to move in the reverse direction is identical to that previously described. The displacement of the valve member 396 in either direction within the valve body 394 in response to load variations encountered by the track drive motors 32 and 38 is proportional to the differential between the respective loads encountered thereby. It will be readily apparent that as the differential between the loads encountered by the track drive motors continually increases or decreases, the resulting displacement of the valve member 396 within the valve body 394 continuously controls the hydraulic pressure drops between the first power inlet 178 and first power outlet 182, and between the second power inlet 180 and second power outlet 184, thus maintaining these hydraulic pressure drops substantially equal regardless of the loads encountered by the track drive motors 32 and 38.

It should also be noted that provision has been made for manually overriding the automatic control of the apparatus 10 by the automatic control assembly 18 by means of the hydraulic manual steering control assembly 64. The hydraulic manual steering control assembly 64 also provides the primary means of steering the apparatus 10 when the shut-off valve 86 is placed in the closed position thereby deactivating the previously described sensor control assembly 74.

If the shut-off valve 86 is in the closed position with the sensor control assembly 72 deactivated, and the operator desires to steer the forwardly moving apparatus 10 to the right, the operator moves the control lever 268 in the direction R as shown in FIG. 2. The movement of the control lever 268 in the direction R causes the hydraulic fluid entering the hydraulic differential flow control valve 260 through the inlet port 262 to emanate from the signal outlet 266 in a stream having a greater flow rate than the stream of hydraulic fluid emanating from the signal outlet 264. The differential in the flow rates of the hydraulic fluid streams emanating from the signal outlets **266** and **264** is proportional to the magnitude of the movement of the control lever **268** by the operator. The stream of hydraulic fluid emanating from the signal outlet **266** flows through the previously described conduits to the shuttle valve **142** and from the shuttle valve **142** through the conduit **146** to the annular chamber **324** of the flow divider **120.** The stream of hydraulic fluid emanating from the signal outlet **264** flows through the previously described conduits to shuttle valve **136** and from shuttle valve **136** through conduit **140** to the annular chamber **322** of the flow divider **120.**

As described in detail above for the automatic operation of the automatic control assembly **18,** the difference in the hydraulic pressures acting on the opposite ends of the valve member **282** causes the displacement thereof to the left, as viewed in FIG. 3, within the valve body **280** and results in a proportionally greater flow of hydraulic fluid to the track drive motor **38** than the flow rate of hydraulic fluid to the track drive motor **32** thereby increasing the speed of the track drive motor **38** over the speed of the track drive motor **32** and causing the apparatus **10** to swing to the right.

It will be readily apparent to those skilled in the art that the movement by the operator of the control lever **268** in the direction L, as shown in FIG. 2, will cause displacement of the valve member **282** to the right within the valve body **280** of the flow divider **120,** thereby causing the track drive motor **32** to operate at a speed greater than that of the track drive motor **38** thereby causing the apparatus **10** to swing to the left.

The operation of the hydraulic manual steering control assembly **64** when used to override the control signals of the sensor control assembly **72** is identical to that described in detail above. Assuming the shut-off valve **86** is in the open position and the sensor control assembly **72** is activated, it will be readily apparent to those skilled in the art that if the flow rate of hydraulic fluid emanating from the signal outlet **264** of the hydraulic steering control assembly **74** is greater than the flow rate of the hydraulic fluid emanating from the signal outlet **274** of the sensor control assembly **72,** the shuttle valve **136** will check the flow of hydraulic fluid entering the first inlet **148** thereof thereby allowing the hydraulic fluid entering the second inlet **150** to pass into the check valve and out through the outlet **138** to flow to the flow divider **120** through conduit **140.** Similarly, if the flow rate of hydraulic fluid emanating from the signal outlet **266** of the hydraulic manual steering control assembly **64** is greater than the flow rate of hydraulic fluid emanating from the signal outlet port **276** of the sensor control assembly **72,** the shuttle valve **142** will act to close the first inlet **158** thereof and open the second inlet **160** thereof, thereby passing hydraulic fluid into the check valve **142** to exit from the outlet **144** thereof and flow to the flow divider **120** through conduit **146.**

It should be noted that when the sensor control assembly **72** is deactivated by placing the shut-off valve **86** in the closed position, the shuttle valves **136** and **142** will operate to close the respective inlets **148** and **158** thereof. It should also be noted that when the sensor control assembly **72** is activated by placing the shut-off valve **86** in the open position and when the hydraulic manual steering control assembly **64** is in the neutral position, the shuttle valves **136** and **142** will operate to close the respective inlets **150** and **160** thereof.

DESCRIPTION OF THE EMBODIMENT OF FIGS. 5, 6, 7 AND 8

FIG. 5 generally illustrates, in schematic form, a slightly modified automatic control assembly **18*a*** in fluid communication with the hydraulic manual steering control assembly **64,** the sensor control assembly **72,** the track drive pump **20,** the shut-off valve **86,** and the track drive motors **32** and **38** as previously described in detail. The track drive motors **34** and **36** are not illustrated in FIG. 5 for the same reason as explained in the discussion of the automatic control assembly **18** above.

The automatic control assembly **18*a*** comprises a bi-directional differential valve **500.** The bi-directional differential valve **500** includes a first inlet-outlet port **502,** a second inlet-outlet port **504,** a high pressure outlet port **506** and a low pressure return port **508.** The first inlet-outlet port **502** is in fluid communication with the first port **20*a*** of the track drive pump **20** via conduits **118** and **22.**

The automatic control assembly **18*a*** also includes a slightly modified flow divider **120*a*** having first and second signal inlets **122** and **124,** a power inlet **126** and first and second power outlets **128** and **130.** The power inlet **126** of the flow divider **120*a*** is in fluid communication with the second inlet-outlet port **504** of the differential valve **500** via conduit **134.**

A shuttle valve **136** having an outlet **138** is in fluid communication with the first signal inlet **122** of the flow divider **120*a*** via conduit **140.** A second shuttle valve **142** having an outlet **144** is in fluid communication with the second signal inlet **124** of the flow divider **120*a*** via conduit **146.**

The shuttle valve **136** further includes a first inlet **148** and a second inlet **150.** The first inlet **148** is in fluid communication with the sensor control assembly **72** via conduits **152** and **76.** The second inlet **150** is in fluid communication with the hydraulic manual steering control assembly **64** via conduits **154, 156** and **68.**

The second shuttle valve **142** further includes a first inlet **158** and a second inlet **160.** The first inlet **158** is in fluid communication with the sensor control assembly **72** via conduits **162** and **78.** The second inlet **160** is in fluid communication with the hydraulic manual steering control assembly **64** via conduits **164, 166** and **70.**

The second inlet **150** of the shuttle valve **136** is also in fluid communication with the low pressure return port **508** of the bi-directional differential valve **500** via conduits **154, 510** and **512** with an orifice **514** interposed in conduit **510** intermediate conduits **154** and **512.** The second inlet **160** of the shuttle valve **142** is also in fluid communication with the low pressure return port **508** of the bi-directional differential valve **500** via conduits **164, 516** and **512** with an orifice **518** interposed in conduit **516** intermediate conduits **164** and **512.**

The outlet **138** of shuttle valve **136** is in fluid communication with the low pressure return port **508** of the bi-directional differential valve **500** via conduits **140, 520** and **512** with an orifice **522** interposed in conduit **520** intermediate conduits **140** and **512.** The outlet **144** of shuttle valve **142** is also in fluid communication with the low pressure return port **508** of the bi-directional differential valve **500** via conduits **146, 524** and **512** with an orifice **526** interposed in conduit **524** intermediate conduits **146** and **512.**

The automatic control assembly **18a** further includes a slightly modified flow compensator **176a.** The flow compensator **176a** includes first and second power inlets **178** and **180,** and first and second power outlets **182** and **184.** The flow compensator **176a** further includes first and second reverse-compensation inlets **528** and **530.**

The hydraulic manual steering control assembly **64** preferably includes a conventional hydraulic differential flow control valve **260,** described above, having an inlet port **262** and a pair of signal outlet ports **264** and **266.** The valve **260** may be suitably actuated by a manually operated control lever or the like as shown at **268.** The inlet port **262** of the control valve **260** is in fluid communication with the high pressure outlet port **506** of the bi-directional differential valve **500** via conduits **66, 532** and **534.** The signal outlet **264** is in fluid communication with the conduit **68** and the signal outlet **266** is in fluid communication with the conduit **70.**

The sensor control assembly **72** includes a conventional hydraulic differential flow control valve **270,** described above, having an inlet port **272** and a pair of signal outlet ports **274** and **276.** The signal outlet port **274** is in fluid communication with conduit **76** and the signal outlet port **276** is in fluid communication with conduit **78.** The inlet port **272** is in fluid communication with the high pressure outlet port **506** of the bi-directional differential valve **500** via conduits **74, 536** and **534.** As described above, a shut-off valve **86** is installed in the conduit **74** intermediate the inlet port **272** of the sensor control assembly **72** and the automatic control assembly **18a.** The hydraulic differential flow control valve **270** is of conventional construction and its operation will be readily apparent to those skilled in the art.

The sensor unit **80** is operatively connected to the hydraulic differential flow control valve **270** of the sensor control assembly **72** for actuation of the valve **272** in response to variation in the position of the sensor unit **80** relative to the string line **84.** The sensor unit **80** may be directly connected to the valve **272** by mechanical means or may be connected electrically to the valve **272** to provide actuation thereof. Such sensor units are well-known in the art and need not be described in detail herein.

The automatic control assembly **18a** further includes a hydraulically actuated selector spool valve **538** having first and second control ports **540** and **542,** first and second inlets **544** and **546,** and first and second outlets **548** and **550.** The first control port **540** is in fluid communication with the first inlet-outlet port **502** of the bi-directional differential valve **500** via conduits **552** and **118.** The second control port **542** is in fluid communication with the second port **20b** of the track drive pump **20** via conduits **554, 556** and **24.** The second control port **542** is also in fluid communication with the track drive motor **32** via conduits **554, 558, 44** and **46.** The second control port **542** is further in fluid communication with the track drive motor **38** via conduits **554, 560, 56** and **62.**

The first port **32a** of the track drive motor **32** is in fluid communication with the first power outlet **182** of the flow compensator **176a** via conduits **40, 42** and **562.** The first port **38a** of the track drive motor **38** is in fluid communication with the second power outlet **184** of the flow compensator **176a** via conduits **52, 60** and **564.**

The first power outlet **128** of the flow divider **120a** is in fluid communication with the first power inlet **178** of the flow compensator **176a** via conduit **566.** The first inlet **544** of the selector valve **538** is in fluid communication with the first power outlet **128** of the flow divider **120a** via conduits **568** and **566.** The second power outlet **130** of the flow divider **120a** is in fluid communication with the second power inlet **180** of the flow compensator **176a** via conduit **570.** The second inlet **546** of the selector valve **538** is in fluid communication with the second power outlet **130** of the flow divider **120a** via conduits **572** and **570.**

As shown in FIG. **6,** the flow divider **120a** comprises a valve body **280** and a slightly modified valve member **282a** slidingly disposed therein. The valve body **280** is identical to the valve body **280** described above for the flow divider **120** and therefore will not be described in detail again. The valve member **282a** is substantially identical to the previously described valve member **282** of the flow divider **120** and the same reference characters will be used to designate those portions of the valve member **282** which are unchanged therefrom.

The cylindrically shaped outer periphery **326** of the valve member **282a** has a diameter sized to provide a substantially fluid-tight seal between the outer periphery **326** and the walls of the bore **284** through the valve body **280.** A cylindrically shaped extension **328** is formed on one end portion **329** of the valve member **282a** and extends into the annular chamber **324** of the valve body **280.** The extension **328** is coaxial with the outer periphery **326** of the valve member **282a.** An annular shoulder **330** interconnects the cylindrical surface **332** of the extension **328** and the cylindrically shaped outer periphery **326.**

A bore **336a** is formed in the valve member **282a** intersecting the end portion **338** thereof. The bore **336a** extends only partially through the valve member **282a** thereby forming a chamber **252a** therein. A slightly modified plug **344a** having no aperture formed therein is threadedly secured in the bore **336a** at the end portion **338** of the valve member **282a.**

Two ports **350** are formed in the valve member **282a** providing communication between the chamber **352a** formed therein and the central annular chamber **298** of the valve body **280.**

A plurality of ports **354** are formed in the valve member **282a** proximate to the end portion **338** thereof. The ports **354** provide fluid communication between the chamber **352a** and the outer periphery **326** of the valve member **282a.** Similarly, a plurality of ports **356** are formed in the valve member **282a** proximate to the end portion **329** thereof. The ports **356** provide fluid communication between the chamber **352a** and the outer periphery **326** of the valve member **282a.** Preferably, the valve member **282a** includes **10** ports **354** and **10** ports **356.** The arrangement of the ports **354** and **356** in the valve member **282a** is identical to the arrangement previously described for the valve member **282** and will not be described in detial again. It should be noted that the valve member **282a** is positioned in its center or medial position within the valve body **280** as shown in FIG. **6.**

As will be readily apparent to those skilled in the art, as the valve member **282a** moves either left or right of the central or medial position, the cross-sectional area of ports **354** and **356** providing fluid communication between the chamber **352***a* and the outer periphery **326** of the valve member **282***a* always remains equal to the total cross-sectional area of **10** ports.

The valve member **282***a* is urged into its central or medial position relative to the valve body **280,** as shown in FIG. **6,** by means of the spring assembly **358** described in detail above. The spring assembly **358** is retained on the cylindrically shaped extension **328** by means of a snap ring **372** disposed in an annular groove **374** formed in the cylindrical surface **332** of the extension **328.** The previously described annular sleeve **378** is disposed within the annular chamber **324** of the valve body **280** with the first end face **380** thereof abutting the spring assembly **358** and with the second end face **382** thereof abutting the end plate **386.**

FIG. **7** illustrates a slightly modified flow compensator **176***a* comprising a slightly modified valve body **394***a* and a slightly modified valve member **396***a* slidably disposed therein. Since the slightly modified flow compensator **176***a* is identical in many respects to the previously described flow compensator **176,** like elements will be identified by the same reference characters used previously.

The slightly modified valve body **394***a* differs from the previously described valve body **394** only in the addition of the first and second reverse compensation inlets **528** and **530** which provide fluid communication between the exterior of the valve body **394***a* and the bore **398** extending through the valve body **394***a.*

The valve member **396***a* is slidably disposed in the bore **398** of the valve body **394***a.* The cylindrically shaped outer periphery **440***a* of the valve member **396***a* has a diameter sized to provide a substantially fluid-tight seal between the outer periphery **440***a* and the walls of the bore **398.** The valve member **396***a* includes opposite end faces **442***a* and **444***a*, each lying in a plane normal to the longitudinal axis of the valve member **396***a*. A cylindrically shaped chamber **446***a* is formed in one end of the valve member **396***a* with one end thereof intersecting end face **442***a*. A counterbore **574** is formed in the chamber **446***a* intersecting the end face **442***a*. A second cylindrically shaped chamber **448***a* is formed in the opposite end of the valve member **396***a* with one end thereof intersecting the end face **444***a*. A counterbore **576** is formed in the chamber **448***a* intersecting the end face **444***a*. It should be noted that the chambers **446***a* and **448***a* do not communicate with each other within the valve member **396***a*.

First and second circumferential grooves **578** and **580** are formed in the medial portion of the valve member **396***a*. The circumferential grooves **578** and **580** communicate respectively with the first and second reverse compensation inlets **528** and **530** of the valve body **394***a*. Each groove **578** and **580** is of sufficient width to provide full fluid communication with the respective inlets **528** and **530** throughout the full range of displacements of the valve member **396***a* within the valve body **394***a*.

A first transverse bore **582** extends transversely through the valve member **396***a* communicating at each end thereof with the first circumferential groove **578.** The bore **582** intersects the previously described bore **446***a* providing fluid communication between the first reverse compensation inlet **528** of the valve body **294***a* and the end face **442***a* of the valve member **396***a*.

A second transverse bore **584** extends transversely through the valve member **396***a* with its opposite ends intersecting the second circumferential groove **580.** The bore **582** intersects the cylindrical chamber **448***a* thereby providing fluid communication between the second reverse compensation inlet **530** of the valve body **394***a* and the end face **444***a* of the valve member **396***a*.

First and second longitudinal grooves **586** and **588** are formed in the cylindrically shaped outer periphery **440***a* of the valve member **396***a*. The grooves **586** and **588** provide fluid communication between the first annular chamber **408** and the third annular chamber **424** of the valve body **394***a*. The longitudinal groove **586** includes a first end wall **590** and a second end wall **592.** The second longitudinal groove **588** includes a first end wall **594** and a second end wall **596.** While the grooves **586** and **588** are preferred other forms of passages, such as an annular groove or the like, may be substituted therefor.

Third and fourth longitudinal grooves **598** and **600** are formed in the cylindrically shaped outer periphery **440***a* of the valve member **396***a*. The longitudinal grooves **598** and **600** provide fluid communication between the second annular chamber **416** and the fourth annular chamber **432** of the valve body **394***a*. The longitudinal groove **598** includes a first end wall **602** and a second end wall **604.** The longitudinal groove **600** includes a first end wall **606** and a second end wall **608.** While the grooves **598** and **600** are preferred, other forms of passages, such as an annular groove or the like, may be substituted therefor.

The valve member **396***a* is of such length that when it is displaced to the left, as viewed in FIG. **7,** with the end face **442***a* thereof abutting the end plate **458,** the cross-sectional area of fluid communication between the annular chambers **408** and **424** of the valve body **394***a* is closed off entirely by the respective end walls **590** and **594** of the grooves **586** and **588.** Alternately, when the valve member **396***a* is displaced to the extreme right, as viewed in FIG. **7,** with the end face **444***a* thereof abutting the end plate **460,** the cross-sectional area of fluid communication between the annular chambers **416** and **432** of the valve body **394***a* is closed off entirely by the respective end walls **606** and **604** of the grooves **598** and **600.** It should be noted that regardless of the relative position of the valve member **396***a* within the valve body **394***a* the cross-sectional area of fluid communication between the grooves **586** and **588** and the annular chamber **424** will preferably be equal to or greater than the cross-sectional area of fluid communication between the grooves **586** and **588** and the annular chamber **408.** Similarly, the cross-sectional area of fluid communication between the grooves **598** and **600** and the annular chamber **432** will preferably be equal to or greater than the cross-sectional area of fluid communication between the grooves **598** and **600** and the annular chamber **416.**

It should be noted that the previously described flow divider **120***a* and the flow compensator **176***a* may be advantageously housed in a single unitary valve body. Such a valve body would include all of the features described for the valve bodies **280** and **394***a* of the flow divider **120***a* and flow compensator **176***a*, respectively, and would place the first and second power outlets **128** and **130** of the flow divider **120***a* in fluid communication with the first and second power inlets **178** and **180,** respectively, of the flow compensator **176***a* within the combined valve body. Furthermore, the end plates **458** and **384** may be advantageously combined into one end plate as may the end plates **460** and **386.** In either of the described configurations the functions of the flow divider **120***a* and the flow compensator **176***a* would be identical.

As shown in FIG. **8,** the bi-directional differential valve **500** comprises a valve body **610** and a valve member **612** slidably disposed therein. The valve body **610** includes a bore **614** extending therethrough and intersecting the opposite end of portions **616** and **618** thereof.

A counterbore **620** is formed in the bore **614** intersecting the end portion **616** and forming an annular wall **622** therein. A second counterbore **624** is formed in the bore **614** intersecting the opposite end portion **618** of the valve body **610** and forming an annular wall **626** therein.

The counterbore **620** forms a first annular chamber **628** within the valve body **610** adjacent the end portion **616** thereof. The counterbore **624** forms a second annular chamber **630** within the valve body **610** adjacent the end portion **618** thereof.

A third annular chamber **632** is formed in the valve body **610** coaxial with the bore **614** and includes opposite annular walls **634** and **636** interconnected by a cylindrically shaped annular surface **638.** A fourth annular chamber **640** is formed in the valve body **610** coaxial with the bore **614** therethrough and includes opposite annular walls **642** and **644** interconnected by a cylindrically shaped annular surface **646.**

The previously mentioned high pressure outlet port **506** communicates with the bore **614** intermediate the third and fourth annular chambers **632** and **640** thereby providing fluid communication between the bore **614** and the exterior of the valve body **610.** A threaded plug **648** is threadedly secured in the counterbore **620.** The threaded plug **648** includes an aperture **650** formed therein coaxial with the bore **614** and providing the previously described first inlet-outlet port **502** of the valve **500.** The plug **648** further includes an inwardly facing annular end face **652** formed thereon. A second threaded plug **654,** identical to the threaded plug **648,** is threadedly secured in the counterbore **624** of the valve body **610.** The aperture **650** formed in the plug **654** provides the previously mentioned second inlet-outlet port **504** of the valve **500.** The plug **654** also includes an inwardly facing annular end face **652** formed thereon.

The valve member **612** includes a cylindrically shaped outer periphery **656** and opposite end portions **658** and **660.** A first circumferential groove **662** is formed in the outer periphery **656** of the valve member **612** and includes opposite annular walls **664** and **666** interconnected by a cylindrically shaped circumferential surface **668.** A second circumferential groove **670** is formed in the outer periphery **656** of the valve member **612** and includes opposite annular walls **672** and **674** interconnected by a cylindrically shaped circumferential surface **676.**

The valve member **612** further includes opposite blind bores **678** and **680** formed therein coaxial with the axis of the valve member **612.** A first counterbore **682** is formed in the blind bore **678** and a second counterbore **684** is formed in the blind bore **680.** The bore **678** and the counterbore **682** formed therein form a chamber **686** in the valve member **612.** Similarly, the bore **680** and the counterbore **684** formed therein form a chamber **688** in the valve member **612.**

A pair of ports **690** provide fluid communication between the bore **678** and the cylindrically shaped circumferential surface **668** of the valve member **612.** A pair of ports **692** provide fluid communication between the bore **680** and the cylindrically shaped circumferential surface **676** of the valve member **612.**

The valve member **612** is yieldably urged into its normal, centered position, as shown in FIG. **8,** by means of a pair of coil compression springs **694** and **695** disposed respectively in the counterbores **682** and **684** of the valve member **612.** Each coil spring **694** and **695** is secured at the end thereof opposite the valve member **612** in an annular spring seat **696** having an L-shaped cross-section. The spring seats **696** bear respectively against the end faces **652** of the threaded plugs **648** and **654.**

The previously described low pressure return port **508** is in fluid communication with the bore **614** through the valve body **610** via passageways **698** and **700** formed in the valve body **610.** The passageway **698** communicates with the bore **614** intermediate the first annular chamber **628** and the third annular chamber **632** formed in the valve body **610.** The passageway **700** communicates with the bore **614** intermediate the second annular chamber **630** and the fourth annular chamber **640** formed in the valve body **610.**

Operation of the Embodiment of FIGS. 5, 6, 7 and 8

In operation, the track drive pump **20** is driven by the engine and power take-off (not shown) to provide a source of pressurized hydraulic fluid. The pressurized hydraulic fluid provided by the pump **20** may be in either a low pressure range of from 600 to 1,500 psi or in a high pressure range of from 1,500 to 3,500 psi. While these pressure ranges are disclosed as preferable for the operation of this embodiment of the present invention, it is not intended that the present invention be limited thereby.

Forward and reverse operation of the apparatus **10** is provided by alternately driving the track drive pump **20** in what will be called the forward direction or, alternately, driving the pump **20** in the reverse direction. This particular mode of operation is contrasted with the previously described mode of operation wherein the track drive motors **32** and **38** were reversed by means of actuation of forward-reverse spool valves which served to switch the point of introduction of the stream of pressurized hydraulic fluid in the respective track drive motors **32** and **38.**

Referring now to FIG. **5,** it will first be assumed that the apparatus **10** is to be driven in the forward direction which is accomplished by driving the pump **20** in a forward direction. The pressurized hydraulic fluid is directed from the first port **20***a* of the pump **20** through conduits **22** and **118** to the first inlet-outlet port **502** of the bi-directional differential valve **500.** The pressurized hydraulic fluid entering the port **502** forces the valve member **612** downwardly against the resistance of the lower coil spring **695** as best shown in FIG. **9.** When the valve member **612** is displaced downwardly to a point where the annular wall **666** thereof is slightly below the annular wall **642** of the valve body **610,** hydraulic fluid will flow from the port **502** through the first annular chamber **628** of the valve body **610,** the chamber **686** in the valve member **612**, through the ports **690** into the third annular chamber **632** of the valve body **610**, from the chamber **632** into the chamber **640** through the interconnecting portion of the bore **614**, from the chamber **640** through the ports **692** in the valve member **612** into the chamber **688**, from the chamber **688** of the valve member **612** into the second annular chamber **630** of the valve body **610**, and from the chamber **630** out through the second inlet-outlet port **504** of the valve **500**. As the hydraulic fluid passes from the third annular chamber **632** into the fourth annular chamber **640** of the valve body **610**, the hydraulic pressure thereof is reduced due to the constriction in the path of flow caused by the proximity of the annular wall **666** to the annular wall **642**. The amount of this hydraulic pressure drop is controlled by the spring rate of the lower coil spring **695** which spring resists the downward displacement of the valve member **612** within the valve body **610**. The spring rate of the lower spring **695** is such that the pressure drop is preferably approximately 75 psi.

It will also be readily apparent that a portion of the hydraulic fluid in the fourth annular chamber **640** will flow outwardly from the bore **614** through the low pressure return port **508**. It will also be readily apparent that a portion of the hydraulic fluid flowing from the third annular chamber **632** will pass outwardly from the bore **614** in the valve body **610** through the high pressure outlet port **506**.

It should be noted, therefore, that when the pump **20** is operating in the forward direction, the pressure of the hydraulic fluid exiting from the high pressure outlet port **506** is substantially equal to the pressure of the hydraulic fluid entering the first inlet-outlet port **502**, and the hydraulic pressure of the streams of hydraulic fluid exiting from the second inlet-outlet port **504** and returning to the low pressure return port **508** are each preferably approximately 75 psi less than the hydraulic pressure of the stream of hydraulic fluid exiting from the high pressure outlet port **506**.

The stream of hydraulic fluid emanating from the high pressure outlet port **506** of the bi-directional differential valve **500** flows in part to the inlet port **272** of the sensor assembly **72** via conduits **534, 536** and **74**. In order for the pressurized hydraulic fluid to reach the inlet port **272** of the sensor control assembly **72** it is necessary for the shut-off valve **86**, in the conduit **74**, to be in its open position. If the shut-off valve **86** is in the closed position the sensor control assembly **72** is deactivated and performs no function in the operation of the present invention. The remainder of the hydraulic fluid emanating from the high pressure outlet port **506** flows to the inlet port **262** of the hydraulic manual steering control assembly **64** via conduits **534, 532** and **66.**

The operation of the sensor control assembly **72** in the hydraulic manual steering control assembly **64** and the operation of the slightly modified flow divider **120a** is substantially identical to the operation previously described for the system illustrated in FIG. **2** and, therefore, will not be described again. It should be noted, however, that hydraulic fluid flowing from the outlet **138** of shuttle valve **136** flows through conduits **140, 520** and **512** and orifice **522** to return to the low pressure outlet port **508** of the bi-directional differential valve **500.** The hydraulic fluid emanating from the outlet **138** of the shuttle valve **136** communicates the hydraulic pressure thereof through the first signal inlet **122** of the flow divider **120a** to act upon the end portion **338** of the valve member **282a** thereby urging the valve member **282a** to the right as viewed in FIG. **6.**

Similarly, the hydraulic fluid flowing from the outlet **144** of the shuttle valve **142** flows through conduits **146, 524** and **512** and the orifice **526** to the low pressure outlet port **508** of the bi-directional differential valve **500.** The hydraulic pressure of the fluid emanating from the outlet **144** is communicated through conduit **146** and the second signal inlet **124** of the flow divider **120a** to act upon the end portion **329** of the valve member **282a** thereby urging the valve member **282a** to the left as viewed in FIG. **6.**

Pressurized hydraulic fluid flowing from the first port **20a** of the track drive pump **20** is communicated with the first control port **540** of the hydraulically actuated selector spool valve **538** via conduits **22, 118** and **552.** The introduction of pressurized hydraulic fluid into the first control port **540** places the first inlet **544** and the first outlet **548** in fluid communication while placing the second inlet **546** and the second outlet **550** in fluid communication.

The stream of hydraulic fluid emanating from the first power outlet **128** of the flow divider **120a** flows through conduit **566** into the first power inlet **178** of the flow compensator **176a.** The stream of hydraulic fluid emanating from the second power utlet **130** of the flow divider **120a** flows through conduit **570** into the second power inlet **180** of the flow compensator **176a.** The hydraulic pressure of the stream of hydraulic fluid emanating from the first power outlet **128** of the flow divider **120a** is communicated to the first reverse compensation inlet **528** of the flow compensator **176a** through conduits **566** and **568** and selector valve **538.** The hydraulic pressure of the stream of hydraulic fluid emanating from the second power outlet **130** of the flow divider **120a** is communicated to the second reverse compensation inlet **530** of the flow compensator **176a** through conduits **570** and **572** and selector valve **538.**

The stream of hydraulic fluid entering the first power inlet **178** of the flow compensator **176a** flows therethrough to emanate from the first power outlet **182** via the first annular chamber **408** of the valve body **394a**, the first and second longitudinal grooves **586** and **588** of the valve member **396a**, and the third annular chamber **424** of the valve body **394a.** The hydraulic fluid entering the second power inlet **180** of the flow divider **176a** flows therethrough to emanate from the second power outlet **184** via the second annular chamber **416** of the valve body **394a**, the third and fourth longitudinal grooves **598** and **600** of the valve member **396a**, and the fourth annular chamber **432** of the valve body **394a.**

The hydraulic fluid emanating from the first power outlet **182** of the flow compensator **176a** flows to the first port **32a** of track drive motor **32** via conduits **562, 40** and **42.** The hydraulic fluid emanating from the second power outlet **184** of the flow compensator **176a** flows to the first port **38a** of the track drive motor **38** via conduits **564, 52** and **60.** The track drive motors **32** and **38** are thereby driven in the forward direction. Most of the hydraulic fluid passing through the track drive motor **32** and out the second port **32b** is routed therefrom back to the second port **20b** of the track drive pump **20** via conduits **44, 46, 558, 556** and **24.**

Excess hydraulic fluid bypassed by the internal valving of the track drive motor 32 is directed from bypass outlet 92 through conduits 94, 96 and 98 to heat exchanger 88 and from heat exchanger 88 through conduit 90 to reservoir 26. Most of the hydraulic fluid passing through track drive motor 38 and out the second port 38b is routed therefrom back to the track drive pump 20 via conduits 56, 62, 560, 556 and 24. Excess hydraulic fluid volume bypassed by the internal valving of track drive motor 38 is directed from bypass outlet 110 through conduits 108, 112 and 98 to heat exchanger 88 and from heat exchanger 88 to conduit 90 to reservoir 26.

The operation of the flow compensator 176a, when the apparatus 10 is operated in the forward direction, is substantially identical to the operation previously described for the flow compensator 176. If the track drive motor 32 encounters a greater load than that encountered by the track drive motor 38, the hydraulic pressure in the conduits leading to the track drive motor 32 will increase proportionally over the hydraulic pressure of the hydraulic fluid in the conduits leading to the track drive motor 38. This increased hydraulic pressure is communicated through the previously described conduits to the first reverse compensation inlet 528 of the flow compensator 176a. This hydraulic pressure is communicated therefrom to the end face 442a of the valve member 396a via the first circumferential groove 578 of the valve member 396a, the first transverse bore 582, and through the cylindrically shaped chamber 446a of the valve member 396a. The proportionally lower hydraulic pressure of the hydraulic fluid supplying the track drive motor 38 is communicated through the previously described conduits to the second reverse compensation inlet 530 of the flow compensator 176a. This lower hydraulic pressure is communicated to the end face 444a of the valve member 396a via the second circumferential groove 580, the second transverse bore 584, and the cylindrically shaped chamber 448a of the valve member 396a.

The differential in the hydraulic pressures acting on the opposite end faces 442a and 444a of the valve member 396a causes a resulting displacement of the valve member 396a to the right, as viewed in FIG. 7, within the valve body 394a thereby resulting in a proportional increase in cross-sectional area of fluid communication between the grooves 586 and 588 of the valve member 396a and the first annular chamber 408 of the valve body 394a while, simultaneously, reducing the cross-sectional area of fluid communication between the grooves 598 and 600 of the valve member 396a and the second annular chamber 416 of the valve body 394a. The maximum displacement of the valve member 396a to the right within the valve body 394a is mechanically limited by the abutment of the end face 444a of the valve member 396a with the end plate 460.

It should be noted that when the end face 444a abuts the end plate 460, fluid communication between the second annular chamber 416 and the fourth annular chamber 432 through the grooves 598 and 600 is greatly reduced but is not entirely eliminated.

It will be readily apparent that displacement of the valve member 396a to the right within the valve body 394a will decrease the cross-sectional area of fluid communication between the second power inlet 180 and the second power outlet 184 and simultaneously increase the cross-sectional area of fluid communication between the first power inlet 178 and the first power outlet 182. In effect, a dummy load is introduced between the second power inlet 180 and the second power outlet 183 which results in substantially equal hydraulic pressure drops between the first power inlet 178 and first power outlet 182, and between the second power inlet 180 and the second power outlet 184. Thus, the hydraulic fluid streams emanating from the first and second power outlets 128 and 130 of the flow divider 120a will encounter substantially equal loads regardless of the loads encountered by the track drive motors 32 and 38.

On the other hand, if the track drive motor 38 encounters a greater load than that encountered by track drive motor 32, the hydraulic pressure of the hydraulic fluid supplying the track drive motor 38 will proportionally increase over the hydraulic pressure of the hydraulic fluid supplying the track drive motor 32. This differential in hydraulic pressure will be communicated through the first and second reverse compensation inlets 528 and 530 of the flow compensator 176a thereby causing a proportional displacement of the valve member 396a to the left within the valve body 394a, as viewed in FIG. 7. Maximum displacement of the valve member 396a to the left within the valve body 394a is mechanically limited by the abutment of the end face 442a of the valve member 396a with the end plate 458.

As the valve member 396a moves to the left within the valve body 394a, the cross-sectional area of fluid communication between the second annular chamber 416 and the fourth annular chamber 432 of the valve body 394a afforded by the grooves 598 and 600 of the valve body 396a increases while the cross-sectional area of fluid communication between the first annular chamber 408 the third annular chamber 424 of the valve body 394a afforded by the grooves 586 and 5588 of the valve member 396a decreases. In effect, a dummy load is introduced between the first power inlet 178 and the first power outlet 182 which results in substantially equal pressure drops across the flow compensator 176a between the first power inlet 178 and the first power outlet 182, and between the second power inlet 180 and the second power outlet 184. Thus, the hydraulic fluid streams emanating from the first and second power outlets 128 and 130 of the flow divider 120a will encounter substantially equal loads as noted above.

It should be noted that when the valve member 396a is displaced the maximum amount to the left with the end face 442a thereof abutting the end plate 458 the cross-sectional area of fluid communication between the first annular chamber 408 and the third annular chamber 424 is entirely closed off.

To operate the present invention in the reverse direction thereby driving the apparatus 10 in a reverse direction, the track drive pump 20 is driven in the reverse direction by the engine through the power take-off (not shown). When the pump 20 is operated in the reverse direction, high pressure pressurized hydraulic fluid is directed from the port 20b to the track drive motors 32 and 38 via conduits 24, 556, 558, 44 and 46, and conduits 24, 556, 560, 56 and 62, respectively. The pump 20 also provides pressurized hydraulic fluid to the second control port 542 of the hydraulically actuated selector spool valve 538 via conduits 24, 556, 554. The track drive motors 32 and 38 are driven in the reverse direction, and the selector valve 538 is actuated to place the first inlet **544** in fluid communication with the second outlet **550**, and to place the second inlet **546** in fluid communication with the first outlet **548**.

The above noted actuation of the selector valve **538** prevents the valve member **396*a*** of the flow compensator **176*a*** from being improperly displaced within the valve body **394*a*** in the event one of the track drive motors encounters a greater load than the other track drive motor.

If, for example, track drive motor **32** would encounter a greater load than track drive motor **38** the hydraulic pressure drop across the track drive motor **32** from the pump **20** to the first power outlet **182** of the flow compensator valve **176*a*** would be greater than the hydraulic pressure drop across the track drive motor **38** from the pump **20** to the second power outlet **184** of the flow compensator **176*a***. Assuming the valve member **396*a*** to be in its medial position within the valve body **394*a***, it will be readily apparent that the pressure of the hydraulic fluid emanating from the first power inlet **178** of the flow compensator **176*a*** will be less than the hydraulic pressure of the hydraulic fluid emanating from the second power inlet **180** of the flow compensator **176*a***.

The higher hydraulic pressure of the hydraulic fluid emanating from the second power inlet **180** is communicated to the end face **442*a*** of the valve member **396*a*** via conduits **570** and **572**, selector valve **538**, the first reverse compensation inlet **528**, the first circumferential groove **578**, the first transverse bore **582**, and the cylindrically shaped chamber **446*a***. Similarly, the lower hydraulic pressure of the stream of hydraulic fluid emanating from the first power inlet **178** of the flow compensator **176*a*** is communicated to the end face **444*a*** of the valve member **396*a*** via conduits **566** and **568**, the selector valve **538**, the second reverse compensation inlet **530**, the second circumferential groove **580**, the second transverse bore **584**, and the cylindrically shaped chamber **448*a***.

This pressure differential acting upon the opposite end faces **442*a*** and **444*a*** of the valve member **396*a*** causes the valve member **396*a*** to move to the right within the valve body **394*a*** thereby tending to cause some restriction to the flow of hydraulic fluid from the track drive motor **38** through the flow compensator **176*a*** and emanating from the second power inlet **180** thereof.

Since the hydraulic pressures of the streams of hydraulic fluid emanating from the track drive motors **32** and **38** and flowing to the flow compensator **176*a*** are relatively low in comparison to the hydraulic pressure of the hydraulic fluid supplied to the track drive motors **32** and **38**, the pressure differential between the two streams of hydraulic fluid entering the flow compensator **176*a*** is of very low magnitude. It will be readily apparent, however, that if the selector valve **538** were to remain in the forward position with the first inlet **544** communicating with the first outlet **548** and the second inlet **546** communicating with the second outlet **550** the valve member **396*a*** would move to the left instead of to the right thereby shutting off the lower pressure hydraulic fluid stream thus incapacitating the system.

Similarly, if the track drive motor **38** encounters greater load than the track drive motor **32** the valve member **396*a*** will be displaced to the left within the valve body **394*a***. This actuation of the valve member **396*a*** will be readily apparent to those skilled in the art in view of the detailed discussion above and, therefore, will not be described in detail again.

The stream of hydraulic fluid emanating from the first power inlet **178** of the flow compensator **176*a*** flows through conduit **566** and enters the first power outlet **128** of the flow divider **120*a***. The stream of hydraulic fluid emanating from the second power inlet **180** of the flow compensator **176** flows through conduit **570** and enters the second power outlet **130** of the flow divider **120*a***. The stream of hydraulic fluid flowing into the first power outlet **128*a*** flows into the chamber **352*a*** of the valve member **280*a*** via the second annular chamber **306** and the ports **354**. The stream of hydraulic fluid entering the second power outlet **130** flows into the chamber **352*a*** via the third annular chamber **314** and the ports **356**.

The cross-sectional area of fluid communication between the second annular chamber **306** and the chamber **352*a*** is controlled by the displacement of the valve member **282*a*** within the valve body **280** as is the cross-sectional area of fluid communication between the third annular chamber **314** and the chamber **352*a***. The displacement of the valve member **282*a*** within the valve body **280** is controlled by the sensor control assembly **72**, the hydraulic manual steering control assembly **64**, or a combination of the two as described in detail above.

It will be readily apparent that an increase in the hydraulic pressure communicated to the end portion **329** of the valve member **282*a*** over the hydraulic pressure communicated to the end portion **338** of the valve member **282*a*** will cause the valve member **282*a*** to be displaced to the left in the valve body **280** thereby causing an increase in cross-sectional area available for the return flow of hydraulic fluid emanating from the track drive motor **38** while proportionally decreasing the cross-sectional area available for the return flow of hydraulic fluid emanating from the track drive motor **32**. The increase in flow rate of hydraulic fluid flowing through the track drive motor **38** relative to the flow rate of hydraulic fluid through the track drive motor **32** will cause the apparatus **10** to swing away from the string line **84** either in response to a signal from the sensor control assembly **72** or from the hydraulic manual steering control assembly **64**.

It will be readily apparent that movement of the valve member **282*a*** to the right will provide a proportional increase in speed of the track drive motor **32** over the track drive motor **38** thereby causing the apparatus **10** to swing toward the string line **84** in response to either a signal from the sensor control assemby **72** or the hydraulic manual steering control assembly **64**.

The hydraulic fluid within the chamber **352*a*** exits therefrom through ports **350** and the central annular chamber **298** to exit from the flow divider **120a** through the power inlet **126**. The hydraulic fluid flows from the power inlet **126** through the conduit **134** and enters the bi-directional differential valve **500** through the second inlet-outlet port **504** and causes the valve member **612** thereof to displace upwardly against the urging of spring **694** within the bore **614** therethrough as illustrated in FIG. **10**. Hydraulic fluid having approximately the same hydraulic pressure as the fluid entering the second inlet-outlet port **504** flows through the chamber **688** of the valve member **612** and flows therefrom through ports **692** to exit from the valve body **610** through the high pressure outlet port **506**.

A portion of the hydraulic fluid entering the second inlet-outlet port **504** passes around the annular wall **674** of the valve member **612** into the third annular chamber **632** of the valve body **610** and flows therefrom through passageway **698** to exit from the valve body **610** through the low pressure outlet port **508.** The pressure of the hydraulic fluid exiting from the low pressure outlet port **508** is approximately 75 psi less than the pressure of hydraulic fluid entering the second inlet-outlet port **504.**

A portion of the hydraulic fluid entering the third annular chamber **632** passes therefrom through the ports **690** into the chamber **686** of the valve member **612,** and flows therefrom through the first annular chamber **628** and out the first inlet-outlet port **502** of the valve body **610.** The hydraulic pressure of the hydraulic fluid exiting through the first inlet-outlet port **502** is approximately 75 psi less than the pressure of the hydraulic fluid entering the second inlet-outlet port **504.**

It will readily be seen that the novel design of the bi-directional differential valve **500** provides a pressure differential between the high pressure outlet port **506** and the low pressure outlet port **508** thereof of approximately 75 psi regardless of the direction in which the track drive pump **20** is pumping the hydraulic fluid in the system. This novel bi-directional differential valve **500** provides the necessary pressure differential essential for the proper operation of the sensor control assembly **72** and the hydraulic manual steering control assembly **64** regardless of the direction in which the track drive pump **20** is being driven.

It should be noted that the automatic control system of the present invention is well suited for controlling the flow of hydraulic fluid to various other hydraulic drive elements such as hydraulic power cylinders or the like. Such an automatic control system will find application as an automatic grade control system for use with highway construction equipment where it is necessary to accurately control the position of a grader blade or pavement slip-forming apparatus or the like relative to a predetermined reference datum such as a grade line or string line.

From the foregoing detailed description of the various embodiments of the automatic control system for use with a hydraulic drive system, it can be readily seen that the present invention provides an improved system which will automatically compensate for deviation of the driven vehicle from the desired path and variations in load encountered by the hydraulic drive motors propelling the vehicle. It may be further readily seen that the system constructed in accordance with the present invention provides an all-hydraulically controlled system capable of operating on pressurized hydraulic fluid received from a single drive pump. It should, however, be noted that either or both of the control assemblies **72** and **64** may be driven by one or more separate pumps driven independently of the track drive pump **20.**

Changes may be made in the construction and arrangement of parts or elements of the various embodiments as disclosed herein without departing from the spirit and scope of the present invention.

What is claimed is:

1. An automatic control system for use with a hydraulic drive system of the type which includes a source of pressurized hydraulic fluid and first and second hydraulic drive means deriving their power from said source of pressurized hydraulic fluid, comprising:

differential valve means, having an inlet in fluid communication with said source of pressurized hydraulic fluid and having an outlet, for providing a hydraulic pressure differential between the inlet and outlet thereof;

a flow divider having first and second signal inlets, a power inlet, and first and second power outlets, the power inlet thereof being in fluid communication with the outlet of said differential valve means and the first and second power outlets thereof being in fluid communication with said first and second hydraulic drive means, respectively;

a first control unit having an inlet in fluid communication with said source of pressurized hydraulic fluid and having first and second signal outlets in fluid communication with the first and second signal inlets of said flow divider, respectively;

first control valve means carried by said first control unit between the inlet thereof and the first and second signal outlets thereof for controlling the relative rates of flow of hydraulic fluid through the first and second signal outlets of said first control unit to the first and second signal inlets of said flow divider, respectively;

first actuation means operatively connected to said first control valve means for actuating said first control valve means in response to stimulus from a source external to said automatic control system;

flow divider valve means carried by said flow divider between the power inlet thereof and the first and second power outlets thereof, and in fluid communication with the first and second signal inlets thereof for controlling the relative rates of flow of hydraulic fluid from the power inlet to the first power outlet and from the power inlet to the second power outlet in response to the relative flow rates of hydraulic fluid from said first control unit.

2. An automatic control system as defined in claim 1 wherein said first actuation means is characterized further to include:

sensor means operatively connected to said first control valve means for actuating said first control valve means;

tracer means operatively connected to said sensor means and engageable with a suitable reference datum for actuating said sensor means in response to stimulus imparted thereto by said reference datum; and

wherein said sensor means actuates said first control valve means in response to the actuation of said sensor means by said tracer means.

3. An automatic control system as defined in claim 1 wherein said first actuation means is characterized further to include:

lever means operatively connected to said first control valve means for actuating said first control valve means in response to the application of manual force to said lever means.

4. An automatic control system as defined in claim 1 characterized further to include:

a second control unit having an inlet in fluid communication with said source of pressurized hydraulic fluid and having first and second signal outlets in fluid communication with the first and second signal inlets of said flow divider, respectively;

second control valve means carried by said second control unit between the inlet thereof and the first and second signal outlets thereof for controlling the relative rates of flow of hydraulic fluid through the first and second signal outlets of said control unit to the first and second signal inlets of said flow divider, respectively; and

lever means operatively connected to said second control valve means for actuating said second control valve means in response to the application of manual force to said lever means.

5. An automatic control system as defined in claim **4** wherein said first actuation means is characterized further to include:

sensor means operatively connected to said first control valve means for actuating said first control valve means; and

tracer means operatively connected to said sensor means and engageable with a suitable reference datum for actuating said sensor means in response to stimulus imparted thereto by said reference datum; and

wherein said sensor means actuates said first control valve means in response to the actuation of said sensor means by said tracer means.

6. An automatic control system as defined in claim **1** characterized further to include:

valve means interposed between the first and second power outlets of said flow divider and said first and second hydraulic drive means for reversing the direction of operation of said first and second hydraulic drive means.

7. An automatic control system for use with a reversible hydraulic drive system of the type which includes a reversible pump having a first port and a second port for providing a source of pressurized hydraulic fluid, and first and second reversible hydraulic drive motors deriving their power from said pressurized hydraulic fluid, each of said reversible hydraulic drive motors having a first port and a second port with the second ports thereof in fluid communication with the second port of said reversible pump, comprising:

differential valve means, having a first port in fluid communication with the first port of said reversible pump, a second port, a high-pressure outlet port, and a low-pressure return port, for providing a hydraulic pressure differential between the high-pressure outlet and the low-pressure return thereof regardless of the direction in which said reversible pump is pumping hydraulic fluid;

a flow divider having first and second signal inlets, and first, second and third power ports formed therein, the first power port thereof being in fluid communication with the second port of said differential valve means;

a flow compensator having first, second, third and fourth power ports formed therein, the first and second power ports thereof being in fluid communication with the second and third power ports of said flow divider, respectively, the third power port thereof being in fluid communication with the first port of said first reversible hydraulic drive motor, and the fourth power port thereof being in fluid communication with the first port of said second reversible hydraulic drive motor, and having first and second selector ports formed therein;

selector valve means having first and second inlet ports formed therein in fluid communication with the first and second power ports of said flow compensator, respectively, having first and second outlet ports formed therein in fluid communication with the first and second selector ports to said flow compensator, respectively, and having first and second actuation ports formed therein in fluid communication with the first and second ports of said reversible pump, respectively, for placing the first power port and the first selector port of said flow compensator in fluid communication and placing the second power port and the second selector port of said flow compensator in fluid communication when said reversible pump is pumping hydraulic fluid out through the first port thereof and, alternately, for placing the first power port and the second selector port of said flow compensator in fluid communication and placing the second power port and the first selector port of said flow compensator in fluid communication when said reversible pump is pumping hydraulic fluid out through the second port thereof;

compensator valve means carried by said flow compensator between the first and third power ports thereof, and between the second and fourth power ports thereof for controlling the relative rates of flow of hydraulic fluid through said first and second reversible hydraulic drive motors from said flow compensator in response to load variations encountered by said first and second reversible hydraulic drive motors when said reversible pump is pumping hydraulic fluid out through the first port thereof;

a first control unit having an inlet in fluid communication with the high-pressure outlet of said differential valve means and having first and second signal outlets in fluid communication with the first and second signal inlets of said flow divider, respectively;

first control valve means carried by said first control unit between the inlet thereof and the first and second signal outlets thereof for controlling the relative rates of flow of hydraulic fluid through the first and second signal outlets of said first control unit to the first and second signal inlets of said flow divider, respectively;

first actuation means operatively connected to said first control valve means for actuating said first control valve means in response to stimulus from a source external to said automatic control system; and

flow divider valve means carried by said flow divider between the first power port and the second and third power ports thereof, and in fluid communication with the first and second signal inlets thereof for controlling the relative rates of flow of hydraulic fluid between the first power port and the second power port and between the first power port and the third power port in response to the relative pressure of hydraulic fluid acting on said flow divider valve means through the first and second signal inlets of said flow divider.

8. An automatic control system as defined in claim **7** wherein said first actuation means is characterized further to include:

sensor means operatively connected to said first control valve means for actuating said first control valve means; and

tracer means operatively connected to said sensor means and engageable with a suitable reference datum for actuating said sensor means in response to stimulus imparted thereto by said reference datum; and

wherein said sensor means actuates said first control valve means in response to the actuation of said sensor means by said tracer means.

9. An automatic control system as defined in claim **7** wherein said first actuation means is characterized further to include:

lever means operatively connected to said first control valve means for actuating said first control valve means in response to the application of manual force to said lever means.

10. An automatic control system as defined in claim **7** characterized further to include:

a second control unit having an inlet in fluid communication with the high-pressure outlet of said differential valve means and having first and second signal outlets in fluid communication with the first and second signal inlets of said flow divider, respectively;

second control valve means carried by said second control unit between the inlet thereof and the first and second signal outlets thereof for controlling the relative rates of flow of hydraulic fluid through the first and second signal outlets of said control unit to the first and second signal inlets of said flow divider, respectively; and

lever means operatively connected to said second control valve means for actuating said second control valve means in response to the application of manual force to said lever means.

11. An automatic control system as defined in claim **10** wherein said first actuation means is characterized further to include:

sensor means operatively connected to said first control valve means for actuating said first control valve means; and

tracer means operatively connected to said sensor means and engageable with a suitable reference datum for actuating said sensor means in response to stimulus imparted thereto by said reference datum; and

wherein said sensor means actuates said first control valve means in response to the actuation of said sensor means by said tracer means.

12. An automatic control system for use with a hydraulic drive system of the type which includes a source of pressurized hydraulic fluid and first and second hydraulic drive means deriving their power from said source of pressurized hydraulic fluid, comprising:

differential valve means, having an inlet in fluid communication with said source of pressurized hydraulic fluid and having an outlet, for providing a hydraulic pressure differential between the inlet and outlet thereof;

a flow divider having first and second signal inlets, a power inlet, and first and second power outlets, the power inlet thereof being in fluid communication with the outlet of said differential valve means;

a flow compensator, having first and second power inlets and first and second power outlets, the first and second power inlets thereof being in fluid communication with the first and second power outlets of said flow divider, respectively, and the first and second power outlets thereof being in fluid communication with said first and second hydraulic drive means, respectively;

compensator valve means carried by said flow compensator between the first power inlet and the first power outlet, and between the second power inlet and the second power outlet thereof for controlling the relative rates of flow of hydraulic fluid to said first and second hydraulic drive means from said flow compensator in response to load variations encountered by said first and second hydraulic drive means;

a first control unit having an inlet in fluid communication with said source of pressurized hydraulic fluid and having first and second signal outlets in fluid communication with the first and second signal inlets of said flow divider, respectively;

first control valve means carried by said first control unit between the inlet thereof and the first and second signal outlets thereof for controlling the relative rates of flow of hydraulic fluid through the first and second signal outlets of said first control unit to the first and second signal inlets of said flow divider, respectively;

first actuation means operatively connected to said first control valve means for actuating said first control valve means in response to stimulus from a source external to said automatic control system;

flow divider valve means carried by said flow divider between the power inlet thereof and the first and second power outlets thereof, and in fluid communication with the first and second signal inlets thereof for controlling the relative rates of flow of hydraulic fluid from the power inlet to the first power outlet and from the power inlet to the second power outlet in response to the relative flow rates of hydraulic fluid from said first control unit communicated to the first and second signal inlets of said flow divider.

13. An automatic control system as defined in claim **12** wherein said first actuation means is characterized further to include:

sensor means operatively connected to said first control valve means for actuating said first control valve means; and

tracer means operatively connected to said sensor means and engageable with a suitable reference datum for actuating said sensor means in response to stimulus imparted thereto by said reference datum; and

wherein said sensor means actuates said first control valve means in response to the actuation of said sensor means by said tracer means.

14. An automatic control system as defined in claim **12** wherein said first actuation means is characterized further to include:

lever means operatively connected to said first control valve means for actuating said first control valve means in response to the application of manual force to said lever means.

15. An automatic control system as defined in claim **12** characterized further to include:

a second control unit having an inlet in fluid communication with said source of pressurized hydraulic fluid and having first and second signal outlets in fluid communication with the first and second signal inlets of said flow divider, respectively;

second control valve means carried by said second control unit between the inlet thereof and the first and second signal outlets thereof for controlling the relative rates of flow of hydraulic fluid through the first and second signal outlets of said control unit to the first and second signal inlets of said flow divider, respectively; and

lever means operatively connected to said second control valve means for actuating said second control valve means in response to the application of manual force to said lever means.

16. An automatic control system as defined in claim **15** wherein said first actuation means is characterized further to include:

sensor means operatively connected to said first control valve means for actuating said first control valve means; and

tracer means operatively connected to said sensor means and engageable with a suitable reference datum for actuating said sensor means in response to stimulus imparted thereto by said reference datum; and

wherein said sensor means actuates said first control valve means in response to the actuation of said sensor means by said tracer means.

17. An automatic control system as defined in claim **12** characterized further to include:

valve means interposed between the first and second power outlets of said flow compensator and said first and second hydraulic drive means for reversing the direction of operation of said first and second hydraulic drive means.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,774,401 Dated November 27, 1973

Inventor(s) Thomas E. Allen

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

In Column 5, line 40, "40" should be --140--.

In Column 5, line 55, "62" should be --64--.

In Column 7, line 45, "he" should be --the--.

In Column 8, line 5, "outltes" should be --outlets--.

In Column 8, line 8, "dispose" should be --disposed--.

In Column 9, line 6, "of" should be --or--.

In Column 11, line 17, "936" should be --396--.

In Column 11, lines 17 and 18, "chmaber" should be --chamber-

In Column 11, line 19, "he" should be --the--.

In Column 11, line 51, "44" should be --444--.

In Column 12, line 52, "port" should be --ports--.

In Column 19, line 6, "fo" should be --of--.

In Column 22, line 56, "74" should be --72--.

In Column 23, line 41, "74" should be --64--.

In Column 26, line 62, "detial" should be --detail--.

In Column 32, line 28, "utlet" should be --outlet--.

In Column 40, line 6, "to" should be --of--.

Signed and sealed this 23rd day of April 1974.

(SEAL)
Attest:

EDWARD M. FLETCHER, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patent