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(54) ACTUATOR SYSTEM FOR DUAL PATH TRANSMISSION

(71) We, SUNDSTRAND CORPORATION, a corporation organised and existing under the laws of the State of Delaware, United States of America, of 4751 Harrison Avenue, Rockford, Illinois 61101, United States of America, do hereby declare the invention for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The invention relates to dual path transmission units and, more particularly, to operating structures for control thereof.

The invention provides an actuator system for the control of a pair of transmission units provided with first means controllable to establish a uniform output speed and output direction for both units and with second and third means associated one with each transmission unit and each being controllable to modify the output speed of one unit independently of the first means, the actuator system comprising a control lever assembly movable about a first axis to control the first means and about a second axis to control the second and third means, and first, second and third motion-transmitting linkages operatively associated with the first, second and third means respectively and with the control lever assembly and arranged to transmit movement of the control lever assembly about the first axis to the first means only and to transmit movement of the control lever assembly about the second axis to one only of the second and third means dependent upon the direction of movement of the control lever assembly about the second axis.

In a preferred embodiment of the invention, the first motion-transmitting linkage includes a rotatable shaft having an input member operatively connected to the control lever assembly and an output arm connected by a link to an external operating member for a valve member of a speed control valve comprised in the first means, each of the second and third motion-transmitting linkages includes a first rod connected to the control lever assembly, a second rod coupled to the first rod, a rotatable shaft, an input member on the shaft which input member is operatively connected to the second rod, an adjustable stop for the input member, means for urging the input member against the stop, an output arm on the rotatable shaft and a link connecting the output arm to an external operating member for a valve member of a steering control valve, the steering control valves associated with the second and third motion-transmitting linkages being comprised in the second and third means respectively, the three rotatable shafts being coaxial and extending to a control console mounting the speed and steering control valves, the arrangement being such that movement of the control lever assembly about the first axis causes rotation of the rotatable shaft of the first motion-transmitting linkage, movement of the control lever assembly in one direction about the second axis moves both of the first rods and one but not the other of the second rods to move the input member operatively connected with the said one of the second rods away from its stop and thereby rotate the associated shaft, and movement of the control lever assembly in the other direction about the second axis moves both of the first rods and the said other but not the said one of the second rods to move the input member operatively connected with the said other of the second rods away from its stop and thereby rotate the associated shaft.

The invention is illustrated by the drawings, of which:

Figure 1 is a rear elevation of the actuator system and associated structure shown assembled with components of a vehicle and with certain parts broken away;

Figure 2 is a vertical sectional view taken generally along the line 2–2 in Figure 1;

Figure 3 is a schematic view of a pair of hydrostatic transmission units and the control system therefor as known in the prior art;

Figure 4 is a sectional view, taken about the line 4–4 in Figure 2 and on an enlarged scale;

Figure 5 is a sectional view taken about the line 5–5 in Figure 1 and on an enlarged scale;

Figure 6 is a sectional view, taken about the line 6–6 in Figure 1 and on an enlarged scale and being a partial system view showing parts of a single motion-transmitting linkage; and

Figure 7 is a sectional view, on an enlarged scale taken about the line 7–7 in Figure 1 and showing, in isolation, parts of another of the motion-transmitting linkages.

Referring first to Figure 3, this is a schematic block diagram representing the control apparatus described with reference to Figures 2A and 2B of our British Patent Specification No. 1,494,243. A pair of transmission units is indicated generally at 10 and 11, and each has a pair of power units 14 and 15, each pair being in the form of a pump-motor unit. Each of the power units 15 may be connected in a known manner to respective drive sides of a vehicle, such as a crawler tractor, and each of the power units 14 may be connected to a prime mover whereby in normal forward travel direction power input to the power units 14 causes rotation thereof hydraulically to drive the power units 15 and the outputs connected thereto. One or the other of each of the power units 14 and 15 has a variable displacement capability and in the preferred embodiment all power units 14 and 15 are of variable displacement. Additionally, each power unit 14 has over-centre displacement control for reverse direction of operation of the associated power unit 15. The control of the power units 14 and 15 is provided by displacement control units 20, fully described in the aforesaid British Patent Specification No. 1,494,243.

In the control system for the displacement control units 20, a control pump 25 directs fluid under pressure to a line 26, with a branch line 27 having a pressure relief valve 28 therein directing a pressure P_1 to a horsepower control valve 30. A valve 31 having a variable orifice adjustable unit operable by an external operating member 32 is connected in line 26 and has an output line 33 with a pressure regulator valve 34 therein whereby a regulated pressure P_R is delivered to a line 35 which extends to a ratio valve 36 in the form of a pressure-reducing valve to set the ratio of the hydrostatic transmission units and, therefore, the output speed thereof. The pressure P_R is also directed through a line 37 to constitute a pressure P_2 applied to the horsepower control valve 30 whereby the ratio of pressures P_1 and P_2 is monitored in the horsepower control valve 30 for sensing a potential stall condition of the prime mover which drives the power units 14 and also the control pump 25. This P_1–P_2 relation is basically established by the setting of the variable orifice valve 31.

The ratio valve 36, by the setting thereof, delivers a control pressure P_3 to a line 40 extended to the horsepower control valve 30. When the latter valve is inactive, the control pressure delivered thereto as P_3 is delivered at the same value as a control pressure P_4 to a line 41 and to a directional control valve 42.

The ratio valve 36 and the directional control valve 42 are controlled by an operating structure including an external operating member 43 which, by means of a shaft 44, is connected to an operator 45 connected to a stem 46 of the directional control valve 42 and to a multi-step cam 47 associated with a stem 48 of the ratio valve. Rotatable positioning of the external operating member 43 sets the position of the directional control valve 42 to either a neutral position or a forward or reverse position and the setting of the cam 47 sets a position of the ratio valve 36 to establish the value of the control pressure P_3 which is delivered to the horsepower control valve 30.

A pair of lines 50 and 51 connect to the outlet side of the directional control valve 42, with the line 50 having a branch line 50*a* extending to a steering control valve 52 and a branch line 50*b* extending to a steering control valve 53. The line 51 has a branch line 51*a* extending to the steering control valve 52 and a branch line 51*b* extending to the steering control valve 53. Depending upon the position of the directional control valve 42, other than in neutral position, the control pressure P_4 delivered from the horsepower control valve 30 to the line 41 will be applied to one or the other of the lines

50 and 51 for delivery to both of the steering control valves 52 and 53. Each of the steering control valves 52 and 53 has a pair of control lines 54 and 55 and 56 and 57 extended to the associated displacement control units 20 for applying the pressure P_4 to the displacement control units 20. Control pressure in either of the lines 56 or 57 determines whether the hydrostatic transmission will operate to provide forward or reverse travel for the vehicle. Each of the displacement control units 20 has a charge pump input 20*a* for delivery to the swashplate positioning means of the variable displacement mechanism dependent upon the positions of the displacement control units 20.

Each of the steering control valves 52 and 53 has an external operating member 60 and 61, respectively, whereby in one position thereof the steering control valve is in a neutral position and merely lets the control pressure signal P_4 pass therethrough to one of the control lines 54 to 57. As the external operating member 60 or 61 is moved from the neutral position, there is a gradual connection of the control pressure line to drain to reduce the control pressure signal delivered to the displacement control unit 20 associated therewith. At a certain position of the valve member of a steering control valve, there is a complete connection of the control pressure to drain, further movement of the valve member reversing the connection of control lines 54 to 57 to the line extending from the directional control valve 42 that has the control pressure whereby the displacement of the hydrostatic transmission power units 14 is reversed to have the rotational output of the power units 15 opposite from the previous output direction.

In applying this control to a crawler tractor or the like, the prime mover speed is established with a corresponding setting of the variable orifice valve 31 and drive of the control pump 25 as well as the transmission power units 14. The direction of travel either forward or reverse of the vehicle as well as the speed thereof is established by operation of the external operating member 43 for positioning of the directional control valve 42 and the ratio valve 36. When it is desired to steer the vehicle to make either a right turn or a left turn, one of the external operating members 60 or 61 is operated to vary the basic control pressure delivered to one of the displacement units without affecting the control pressure delivered to the other, with a consequent reduction in the speed of the one transmission unit whereby there is a change in direction of the vehicle. A spin turn capability is obtained by a total movement of one steer valve member from its neutral position to its opposite limit position. In the event there is a stall condition on the prime mover, this is sensed by the horsepower control valve and causes control pressure P_4 to drop to a lower value that the pressure P_3 received by the last-mentioned valve to lower the control pressure delivered to both displacement controls for a reduction in output speed of the transmission units.

The foregoing is a general summary of the control apparatus disclosed in the aforesaid British Patent Specification No. 1,494,243, the disclosure of which is incorporated herein by reference.

The control system for establishing a control pressure and the value thereof that will be delivered to the displacement control units 20 for the pair of hydrostatic transmissions is built into a control console, indicated generally at C in Figures 1, 2 and 4. The control console has a basic housing supported upon a frame structure, indicated generally at F, of the vehicle, such as a crawler tractor. The location of this console may be visualized by the showing of an operator's seat S in Figures 1 and 2, the rear of the seat S being shown in Figure 1, whereby the view is one looking in the direction of forward travel of the vehicle. Forward vehicle travel would be toward the left as shown in Figure 2.

The four external operating members 32, 43, 60 and 61 of the control apparatus described with reference to Figure 3 are identified in Figure 4 in a plan view of the control console, two of the members being at one side thereof and the other two at the other side of the control console.

A throttle lever 70, shown in Figures 1 and 2, is located forwardly and to the right of an operator in the seat S and is movable between the full line and broken line positions shown in Figure 2 to control the throttle of the prime mover. In rotating about a pivot 71 supported by a part 72 of the vehicle frame, the throttle lever 70 moves a link 73 connected to the external operating member 32 for the variable orifice valve 31.

The control of the external operating members for setting a speed and direction of travel of the vehicle by operation of the ratio valve 36 and the directional control valve 42 as well as operation of the steering control valves 52 and 53 is effected by use of a control lever 80. The control lever 80 is movable between a neutral position, shown in full lines in Figure 2, and two broken line positions to either side thereof to establish either forward or reverse travel of the vehicle, the movements being along a path coincident with straight forward travel of the vehicle. For example, if the control lever 80 is moved counterclockwise, as shown in Figure 2, this is movement in the direction of forward travel of the vehicle and correspondingly causes the hydrostatic transmissions to operate in a direction to cause forward vehicle travel. The control lever 80 can also move along a path in directions as indicated from the neutral position, shown in full lines in Figure 1, to broken line positions to either side thereof. Broken line positions 80*a* and 80*b* represent the extent of movement of the control lever to either side of central position for either a full left turn or full right turn with the positions corresponding to actual direction of vehicle turning. When the control lever 80 is in the position 80*a*, the left transmission unit, as for example the one driving the left crawler track will be at zero speed whereby the right crawler track is still operating at set speed and causing the vehicle to travel toward the left. The broken line positions 80*c* and 80*d* represent additional extents of movement of the control lever 80 in either direction, with the movement beyond the full turn position just described causing a counter rotate action and with the movement of the control lever to either of the positions 80*c* and 80*d* causing full reverse output of the associated transmission unit for a spin turn of the vehicle and less movement causing a pivot turn. With the mounting of the control lever as subsequently described, it should be noted that the different movements of the control lever 80 as described for clarity in connection with Figures 1 and 2 for steer and speed control individually may actually be a compound or universal movement whereby there is movement of the control lever in either direction from the full line position of Figure 2 as well as movement in either direction from the full line position of Figure 1 for conjoint control of both the ratio valve for setting a common control pressure applied to both hydrostatic transmissions as well as a modification thereof with respect to the control pressure delivered to one hydrostatic transmission whereby both vehicle speed and turning radius may be variably controlled. With the control apparatus of Figure 3 and the operating structure therefor including control lever 80, it is possible to have the same radius of vehicle turn for steering control at any ratio setting of the ratio valve 36.

The control lever 80 is mounted at a control station having a control pedestal 81 extending upwardly adjacent the left side and slightly forward of the operator's seat S. A pair of upstanding spaced-apart pedestal panels 82 and 83 mount a pair of plates 84 and 85 carrying journal members 86 and 87 (Figure 5). A control housing 88 has parts 89 and 90 extending from opposite ends thereof rotatably journalled in the journal members 86 and 87. The control housing 88 is thus rotatable about its journal axis with corresponding movement of a control pin 91 which is connected to a first motion-transmitting linkage, described hereinbelow, extending to the external operating member 43 for the ratio valve 36 and the directional control valve 42. The control lever 80 is mounted within an upper end 92 of a tubular member 93 rotatably mounted on the pin 91 by sealed bearings and the member 93 has a pair of lateral extensions 94 and 95 each of which is connected to a motion-transmitting linkage, described hereinbelow, which extends to the external operating members 60 and 61 for the steering control valves. The movement of the control lever 80 for forward and reverse and speed control of the vehicle is about the first axis while steer control with movement of the lever 80 to various positions generally illustrated in Figure 1 is about a perpendicular intersecting axis, the structure shown in Figures 2 and 5 permitting movement of the control lever about either axis without affecting movement of linkages operated by movement about the other axis.

With reference to Figures 1 and 4, a housing 100 connected to the control pedestal 81 by structure 100*a* mounts coaxial rotatable shafts 101, 103 and 105. The shaft 101 is mounted in a central passage of the housing 100 by bearings 102 and is of tubular form. The shaft 103 is mounted within the shaft 101 by bearings 104 and is also of tubular form. The shaft 105, which is solid, is mounted within the shaft 103 by bearings 106. Both ends of the shaft 105 extend beyond the ends of the shaft 103; both ends of the shaft 103 extend beyond the ends of the shaft 101 and both ends of the shaft 101 extend beyond the ends of the housing 100. Each of the shafts 101, 103 and 105 is provided at one of its ends with an input arm and at the other of its ends with an output arm, as described hereinbelow.

The outermost tubular shaft 101 is for control of the external operating member 61 for the steering control valve 53 and has an output arm 110 adjacent an end of the housing 100, which output arm 110 extends generally upwardly from the shaft 101 and has a swivel connection 111 to a link 112 which, through a swivel connection and a spacer 115, connects to the external operating member 61. The tubular coaxial shaft 103 has an upwardly-extending output arm 120 which, by means of a swivel connection 121, connects to a link 122 extending forwardly to the control console C for connection to the external operating member 43 for the ratio valve 36 and the directional control valve 42. The shaft 105 mounts an output arm 125 extending generally upwardly therefrom which, by means of spacer structure 126 is connected by a swivel joint to a forwardly-extending link 127 which, by a swivel joint 128, connects to the external operating member 60 for the steering control valve 52. From the foregong it will be noted that operation of the ratio valve and the steering control valves is solely dependent upon rotation of a particular one of the coaxial shafts, the rotation of each one being independent of the others. The output arms 110, 120 and 125 are spaced apart from each other whereby the connecting linkages 112, 122 and 127 respectively to the external operating members 61, 43 and 60 respectively extend in non-interfering relationship.

Fastened to the shafts 101, 103 and 105 are input arms 130, 131 and 132 respectively (Figures 1 and 5). The input arm 130 is connected by a swivel joint 183 (Figure 7) to the lower end of a lower rod part 182 of a two part rod 161. The upper end of the lower rod part 182 is slidably received in a coupling member 180, the bore of which is provided with a shoulder 181 against which the said upper end may abut. The coupling member 180 is secured to the lower end of an upper rod part 175 of the two part rod 161. The upper end of the upper rod part 175 is connected to the lateral extension 95 of the tubular member 93 by a swivel joint 176, a spacer 177 and a threaded member 178 (Figure 5). The input arm 132 is connected to the lateral extension 94 of the tubular member 93 in a similar manner (see Figures 5 and 6), that is by a swivel joint 173, a lower rod part 172 of a two part rod 160, a coupling member 170 with a shoulder 171 in its bore, an upper rod part 165 of the two part rod 160, a swivel joint 166, a spacer 167 and a threaded member 168. The input arm 131 is connected by a swivel joint to a rod 151 (Figures 1 and 2) which in turn is connected by a swivel joint 150 to an end of the pin 91 which end extends from the control housing 88.

Movement of the control lever 80 in the forward travel direction, that is to the left as shown in Figure 2, causes elevation of the rod 151 and corresponding elevation of the input arm 131 to move the connecting link 122 forwardly, that is upwardly as shown in Figure 4, for corresponding movement of the external operating member 43. Movement of the control lever 80 in the reverse travel direction causes movement in the opposite direction of the associated parts of the motion-transmitting linkages.

It is preferable to operate one of the steering control valves 52 or 53 without operation of the other and it will be noted that rotation of the member 93 about an axis defined by the pin 91 will cause movement of both of the upper rod parts 165 and 175. It is the purpose of the slidable couplings 170 and 180 to cause operation of one steering control valve without the other, as described hereinbelow. It will be noted that rotation of the control housing 88 about the axis defined by the pin 91 does not have any effect on the rod 151 because of the intersecting relationship of the two axes of movement of the control lever 80

The independent operation, referred to above, of the steer control valves is obtained by the couplings 170 and 180 and also by the particular structure associated with the input arms 130 and 132. Each of these arms is urged to an inactive position against an adjustable stop by a spring 200 and 201, respectively, extending between fastening members carried by a pedestal plate 202 (Figure 1) and attaching members 203 and 204 secured to the respective input arms 130 and 132. The adjustable stop for the input arm 130 includes an L-shaped member 210 having one leg thereof fastened to the pedestal panel 83 and having the other leg thereof carrying an adjustable threaded member 211 having an end engageable with the upper surface of the input arm 130. The adjustable stop for the input arm 132 includes an L-shaped bracket 220 having one leg thereof fastened to the pedestal panel 82 and the other leg carrying an adjustable threaded member 221 having an end positionable for engagement with the upper surface of the input arm 132. Assuming the control lever 80 is moved in a direction to lower the control rod 160, the lower rod part 172 will move with the upper rod part 165 and move the input arm 132 away from the adjustable stop structure. At the same time, the upper rod part 175 of the control rod 161 is moving upwardly. However, the lower rod part 182 will remain stationary because of the slidable coupling 180, the input arm 130 being held against the adjustable stop element 211. Movement of the control lever 80 in an opposite direction about the axis defined by the pin 91 results in movement of the input arm 130 without movement of the input arm 132, the latter remaining in engagement against the stop member 221. In addition to holding the input arms against the adjustable stops, the springs 200 and 201 provide resistance against movement of the control lever 80 in a steer function. When the control lever 80 is moved into a counter range of movement, namely, between positions 80*a* and 80*c* and positions 80*b* and 80*d*, additional resistance is imparted to movement of the control lever by means of pre-loaded spring structures, shown in Figures 6 and 7. The pre-loaded spring structure shown in Figure 6 comprises a plate 250 attached to the pedestal 81 and mounting a sleeve member 251 housing a spring 252 which bottoms against a surface 253 of the housing. The spring 252 engages under a flange 254 of a movable stem 255 disposed within the spring and having a lower end extending outwardly of the housing 251 and having nuts 256 threaded thereon. The position of the nuts 256 may be varied to control the pre-loading of the spring 252 and these nuts also engage against the underside of the housing 251 to limit the upward movement of the stem 255. The flange 254 is positioned at a distance beneath the swivel joint 173 for the rod 160 such that the swivel joint 173 will come into engagement with the flange 254 when the control lever 80 is moved to position 80*a* and further movement of the control lever 80 towards the position 80*c* will be resisted by downward movement of the stem 255. The pre-loaded spring structure shown in Figure 7 is similar, housing 260 carried by the plate 250 having a spring 261 therein engaging a flange 262 on a stem 263 to urge the stem upwardly, the upward movement being limited by adjustable nuts 264 which also control the pre-loading of the spring. When the control lever 80 has moved to position 80*b*, the swivel joint 183 will be moved into engagement with the flange 262, further movement of the control lever 80 towards position 80*d* causing movement of the stem against the action of the spring 261.

The control lever 80 has a detented neutral position with respect to speed and direction control provided by a spring-urged detent 280 (Figure 1) on the pedestal engageable with a notch in the control housing 88 when the housing is in a neutral position. Preferably, the control lever 80 may be maintained in any position between the broken line extreme speed and direction control positions, shown in Figure 2, by means of friction discs 290 (Figure 5) acting between and carried by the journal member 86 and the lateral extension 89 of the control housing, respectively. The engagement force therebetween is established by a disc spring 291 and the compression may be varied by adjustable threaded members 292 carried on a threaded end 293 of the lateral extension 89.

It should be noted that, with the linkage connections as disclosed, the steering control valve 52 is operated when the vehicle is to be steered to the left, even though this particular valve is located at the right-hand side of the control console, as shown in Figure 4. The actual physical location of the steering control valves 52 and 53 in the control console C need not be related to the direction of steer, since the steering control valves may have their control lines 54 to 57 connected in any desired manner to the two transmission units.

In operation, an operator seated at the seat S may operate the throttle control rod 70 to set a desired engine speed and, at the same time, set the variable orifice valve 31. The control lever 80 may then be moved to a full forward position to set the ratio valve 36 for full forward speed of the vehicle or moved a lesser distance for a slower speed.

If the operator wishes to steer either to the right or left while the control lever is full forward, a slight movement is imparted to the control lever towards either of the positions 80*a* or 80*b*, shown in Figure 1, without this movement effecting the motion-transmitting linkage to the ratio valve 36. Because of the particular structure of the control lever and motion-transmitting linkage systems, there will be a slight movement of the end of the control lever 80 back from the full forward position, as the control lever is moved in a steering control direction. Similarly, the same type of action occurs when the control lever 80 is moved to a full reverse direction and speed of travel and if a steer control signal is inputted by movement of the control lever. With the structure and control system disclosed herein, it is possible to have the same radius of turn for steering of the vehicle at any setting of the ratio valve 36. If the control lever 80 is full forward and a steering control motion is imparted thereto, there will be a certain radius of turn established. However, even if the control lever 80 is now moved backwardly, to reduce the ground speed of the vehicle, the turning radius of the vehicle will remain the same. The inputting of a steer command may be at any speed in either direction of vehicle travel.

In the actuator system disclosed herein, the movements of the control lever 80 are adapted to normal arm and wrist movements of an operator. There is full capability for forward and reverse travel of the vehicle at varying speeds and with steer capability while travelling either forward or reverse and with spin or pivot turn capability. The basic speed and directional positioning of the control lever 80 may be maintained, without manual control, but the steer control function must be maintained by the operator for if the operator releases the control lever 80 during a steer the control lever 80 will return to a steer neutral position because of the urging of the springs 200 and 201 which bring the steer valves 52 and 53 back to a neutral or inactive position.

WHAT WE CLAIM IS:—

1. An actuator system for the control of a pair of transmission units provided with first means controllable to establish a uniform output speed and output direction for both units and with second and third means associated one with each transmission unit and each being controllable to modify the output speed of one unit independently of the first means, the actuator system comprising a control lever assembly movable about a first axis to control the first means and about a second axis to control the second and third means, and first, second and third motion-transmitting linkages operatively associated with the first, second and third means respectively and with the control lever assembly and arranged to transmit movement of the control lever assembly about the first axis to the first means only and to transmit movement of the control lever assembly about the second axis to one only of the second and third means dependent upon the direction of movement of the control lever assembly about the second axis.

2. An actuator system according to claim 1 in which the first motion-transmitting linkage is connected to a part of the control lever assembly on the second axis, the second and third motion-transmitting linkages are connected to parts of the control lever assembly on the first axis one on each side of the second axis, and movement of the control lever assembly about the second axis imparts motion to both the second and third motion-transmitting linkages, means being provided to render the motion imparted to one of the second and third motion-transmitting linkages ineffective to operate the associated one of the second and third means.

3. An actuator system according to claim 1 or claim 2 in which the first motion-transmitting linkage is connected to the valve member of a speed control valve comprised in the first means, and the second and third motion-transmitting linkages are connected to the valve members of first and second steering control valves respectively, the first steering control valve being comprised in the second means and the second steering control valve being comprised in the third means.

4. An actuator system according to any preceding claim in which each of the second and third motion-transmitting linkages includes a first rod connected to the control lever assembly, a second rod coupled to the first rod, a rotatable shaft, an input member on the shaft which input member is operatively connected to the second rod, an adjustable stop for the input member, and means for urging the input member against the stop, the arrangement being such that movement of the control lever assembly in one direction about the second axis moves both of the first rods and one but not the other of the second rods to move the input member operatively connected with the said one of the second rods away from its stop and thereby rotate the associated shaft and movement of the control lever assembly in the other direction about the second axis moves both of the first rods and the said other but not the said one of the second rods to move the input member operatively connected with the said other of the second rods away from its stop and thereby rotate the associated shaft.

5. An actuator system according to any preceding claim in which the first motion-transmitting linkage includes a rotatable shaft having an input member connected to the control lever assembly such that movement of the control lever assembly about the first axis causes rotation of the rotatable shaft of the first motion-transmitting linkage.

6. An actuator system according to claims 3, 4 and 5 in which the rotatable shaft of the first motion-transmitting linkage has an output arm connected by a link to an external operating member for the valve member of the speed control valve, and each rotatable shaft of the second and third motion-transmitting linkages has an output arm connected by a link to an external operating member for the valve member of the associated one of the steering control valves.

7. An actuator system according to claim 6 in which the three rotatable shafts are coaxial and extend to a control console mounting the speed and steering control valves.

8. An actuator system according to claim 3 or any claim appendant directly or indirectly to claim 3 in which the valve member of each steering control valve is movable beyond a specified position to reverse the output direction of the associated transmission unit, and means are provided for increasing the resistance to movement of the control lever assembly when either of the steering control valve

members is moved beyond the specified position.

9. An actuator system according to claim 8 in which the resistance-increasing means comprises a spring-loaded member for each of the second and third motion-transmitting linkages, each spring-loaded member being engageable by a part of its associated motion-transmitting linkage when the latter has moved a distance sufficient to move the valve member of the associated steering control valve beyond the specified position.

10. An actuator system according to any preceding claim in which the first and second axes are perpendicular to one another and the control lever assembly is mounted for universal movement about the first and second axes.

11. An actuator system according to any preceding claim further comprising a second control lever assembly and a fourth motion-transmitting linkage operatively connected with the second control lever assembly and with a valve for controlling the throttle setting of a prime mover for driving the transmission units.

12. An actuator system substantially as described herein with reference to the drawings.

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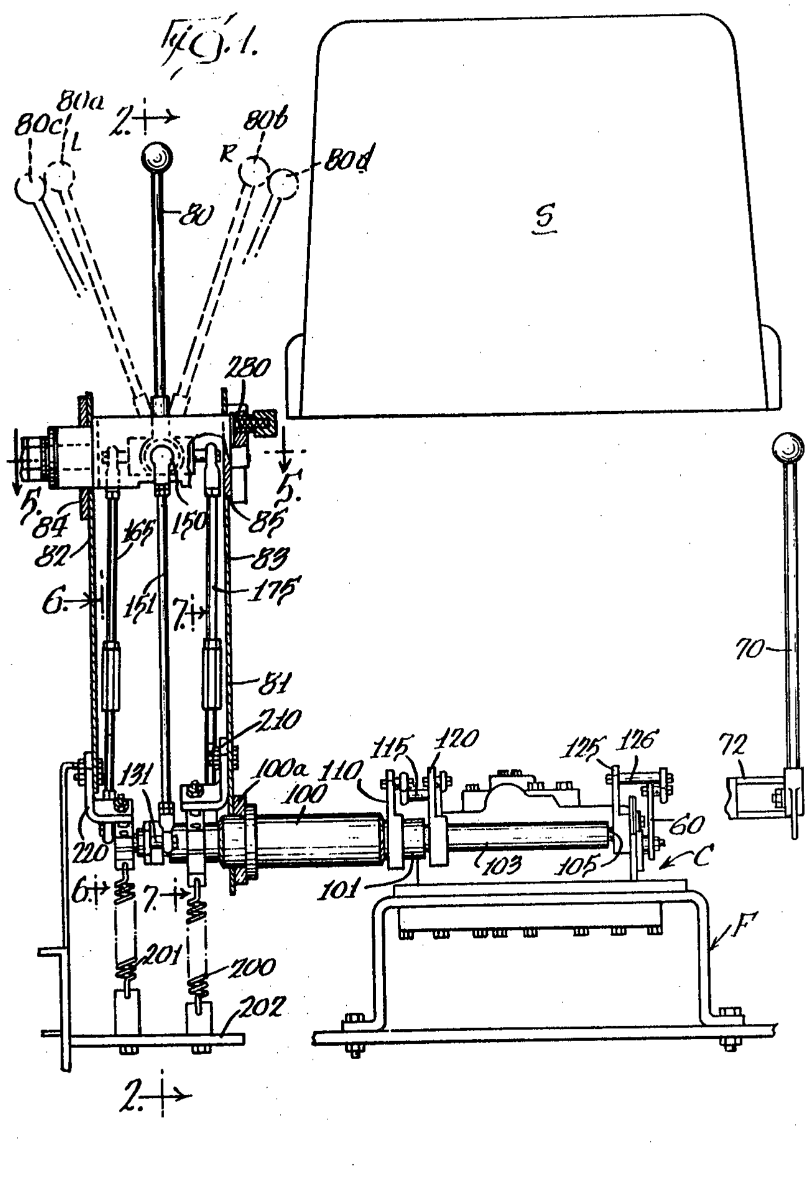

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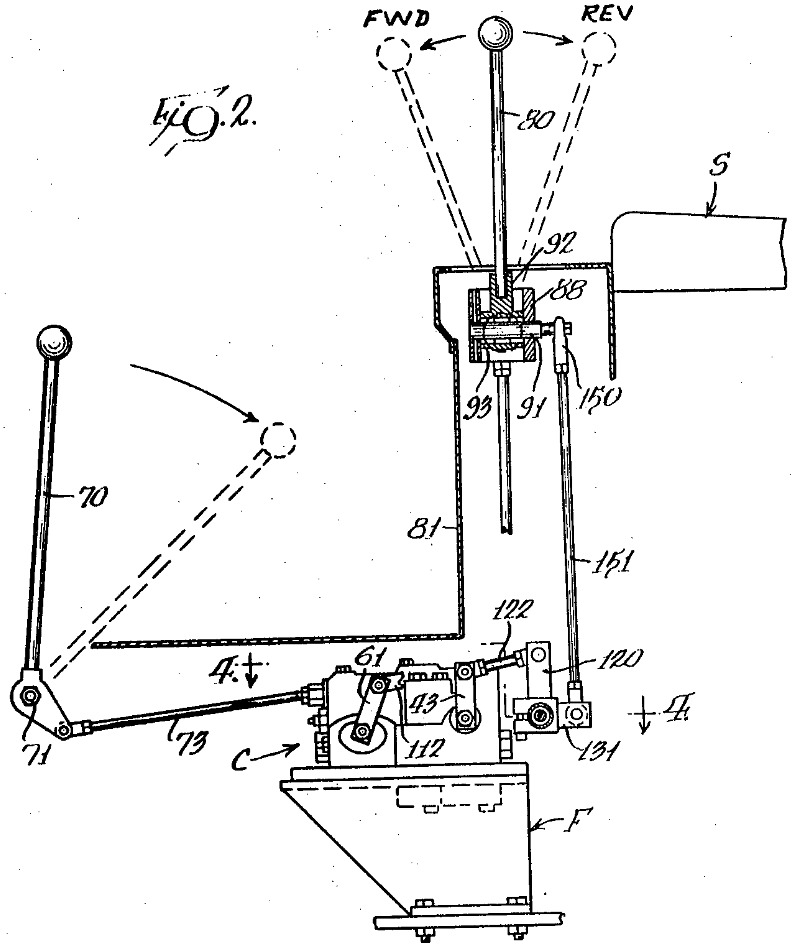
Fig. 2.



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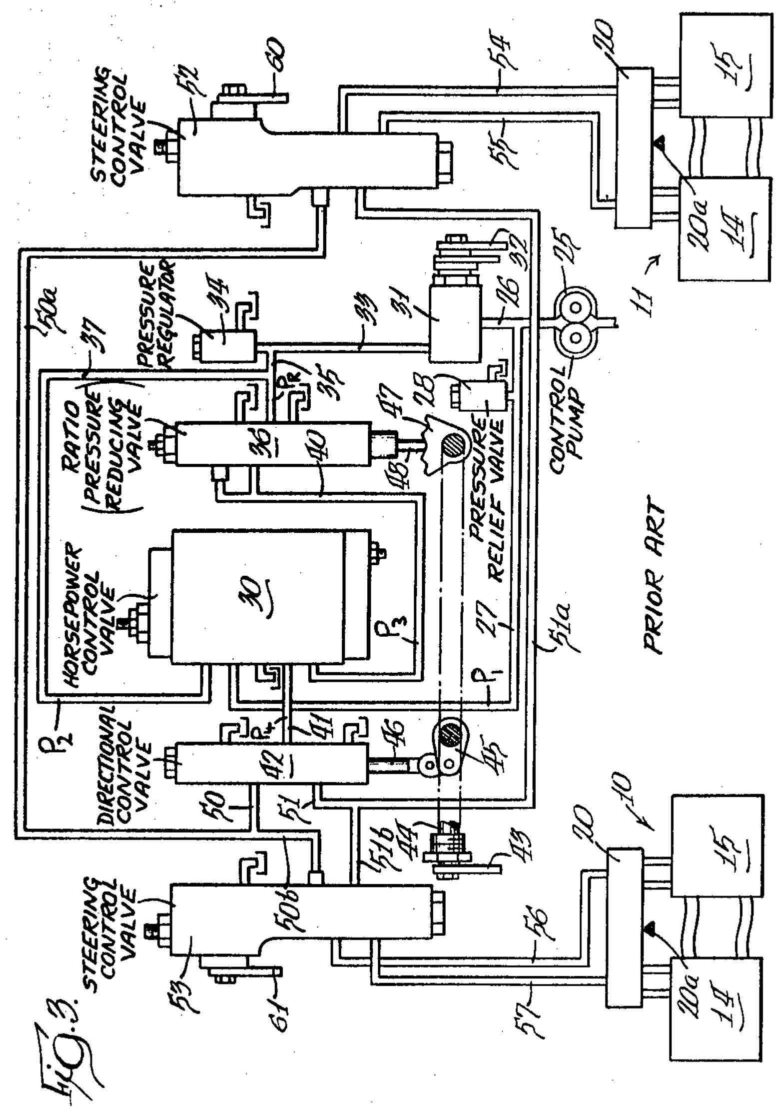

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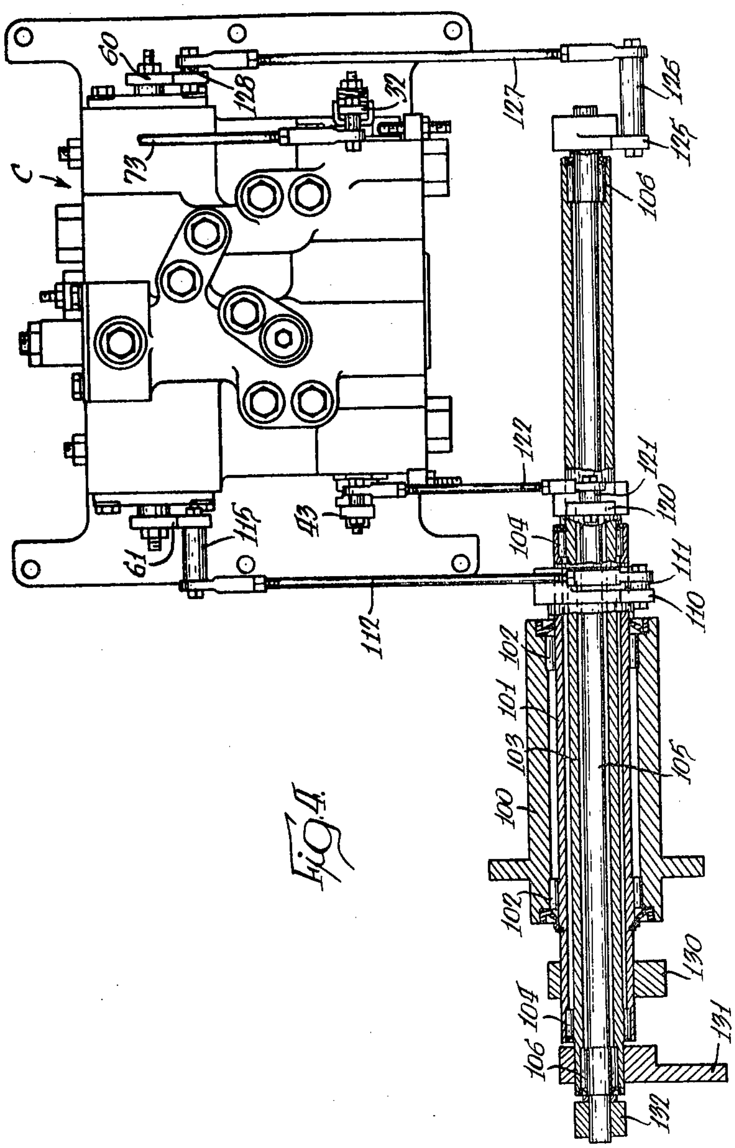

Fig. 4.

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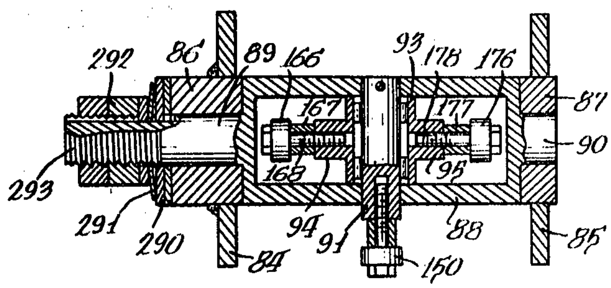

Fig 6.

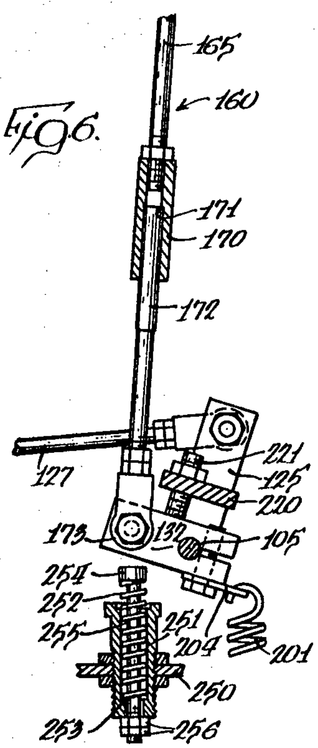

Fig 7