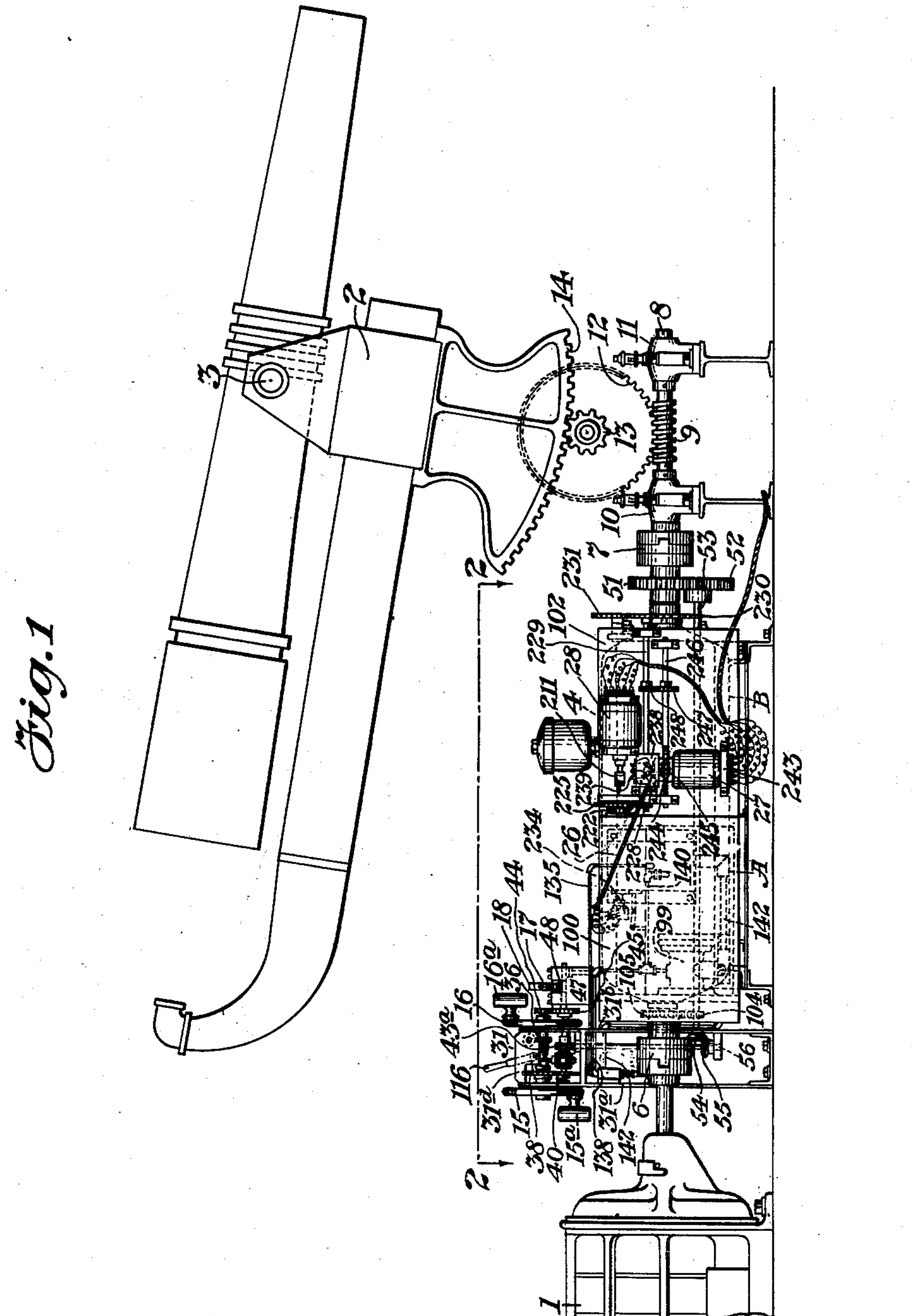
Oct. 21, 1952 E. J. POITRAS ET AL 2,614,390

REVERSIBLE HYDRAULIC DRIVE, INCLUDING FOLLOW-UP SYSTEM

Filed Jan. 11, 1935 8 Sheets-Sheet 1



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REVERSIBLE HYDRAULIC DRIVE, INCLUDING FOLLOW-UP SYSTEM

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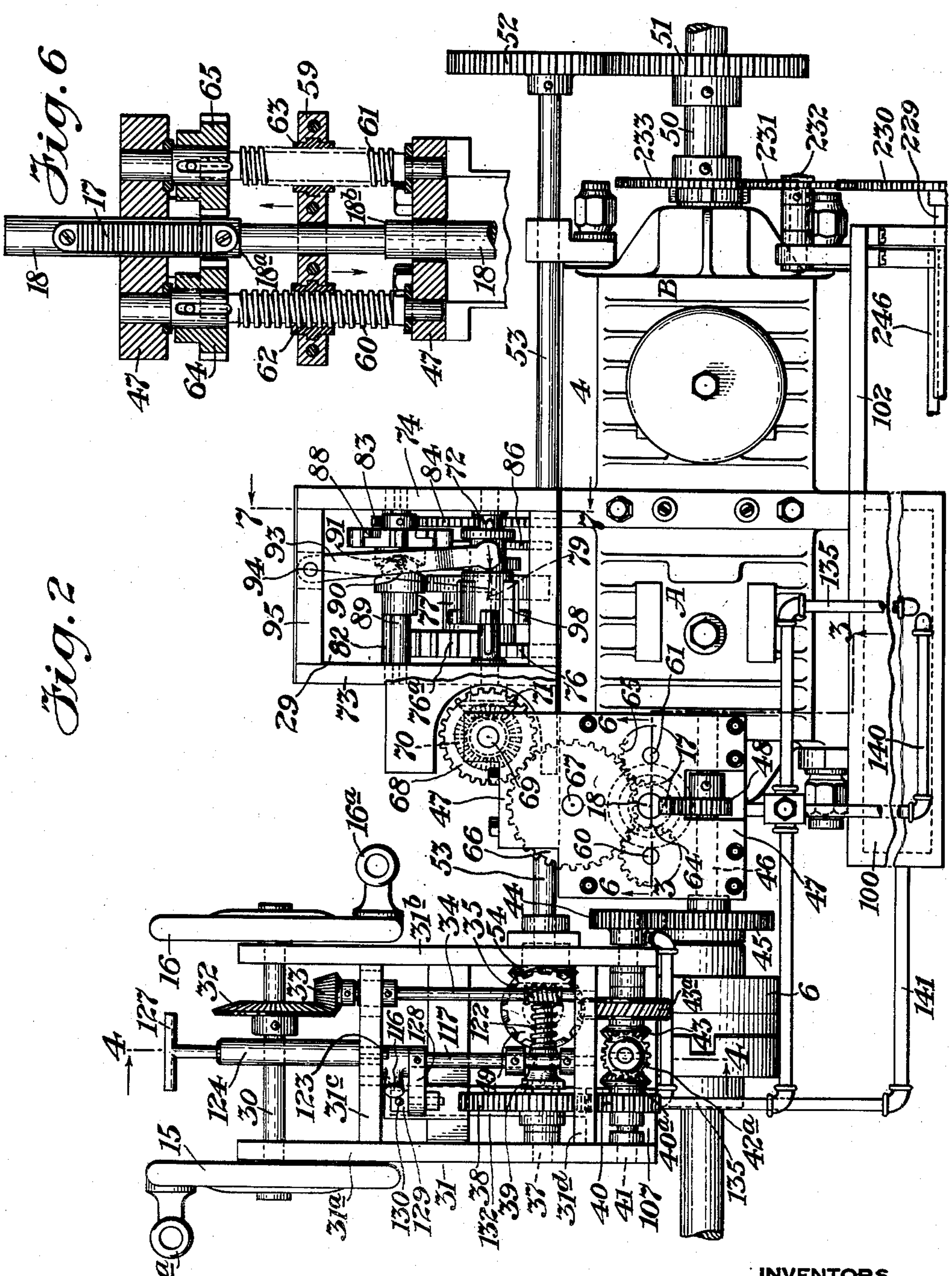

Fig. 6

Fig. 2

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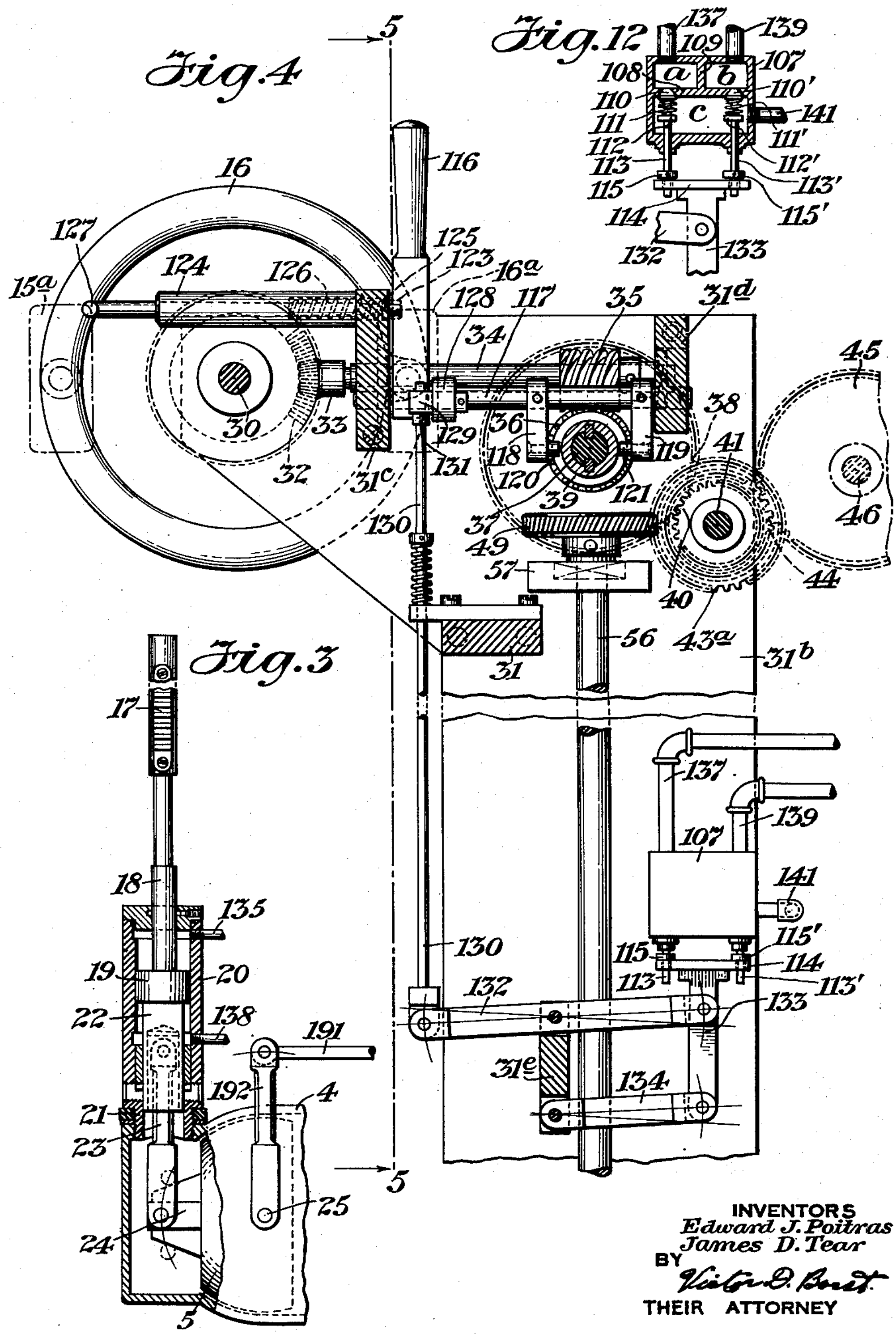

Fig. 4

Fig. 12

Fig. 3

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Fig. 5

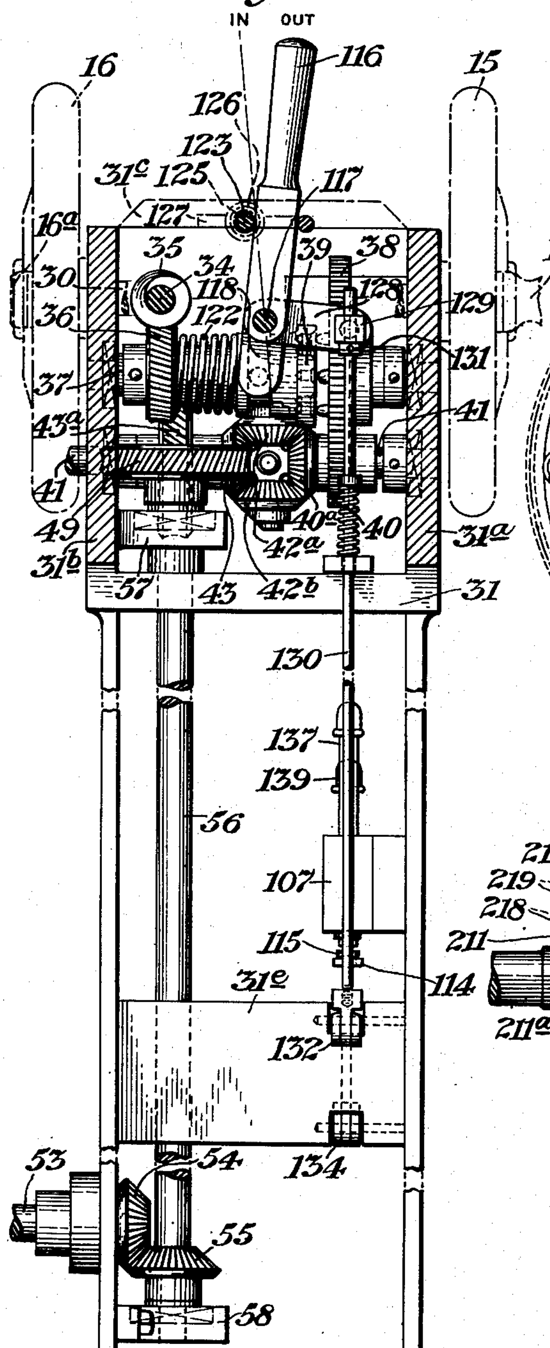


Fig. 14

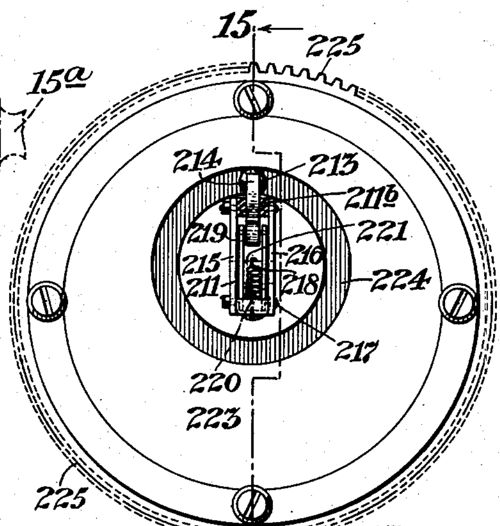
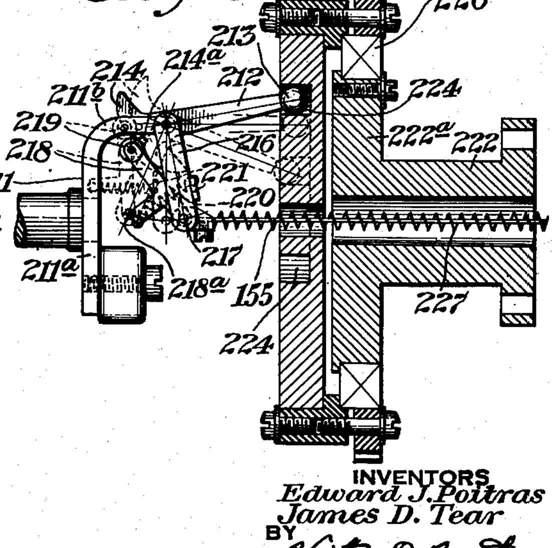


Fig. 15



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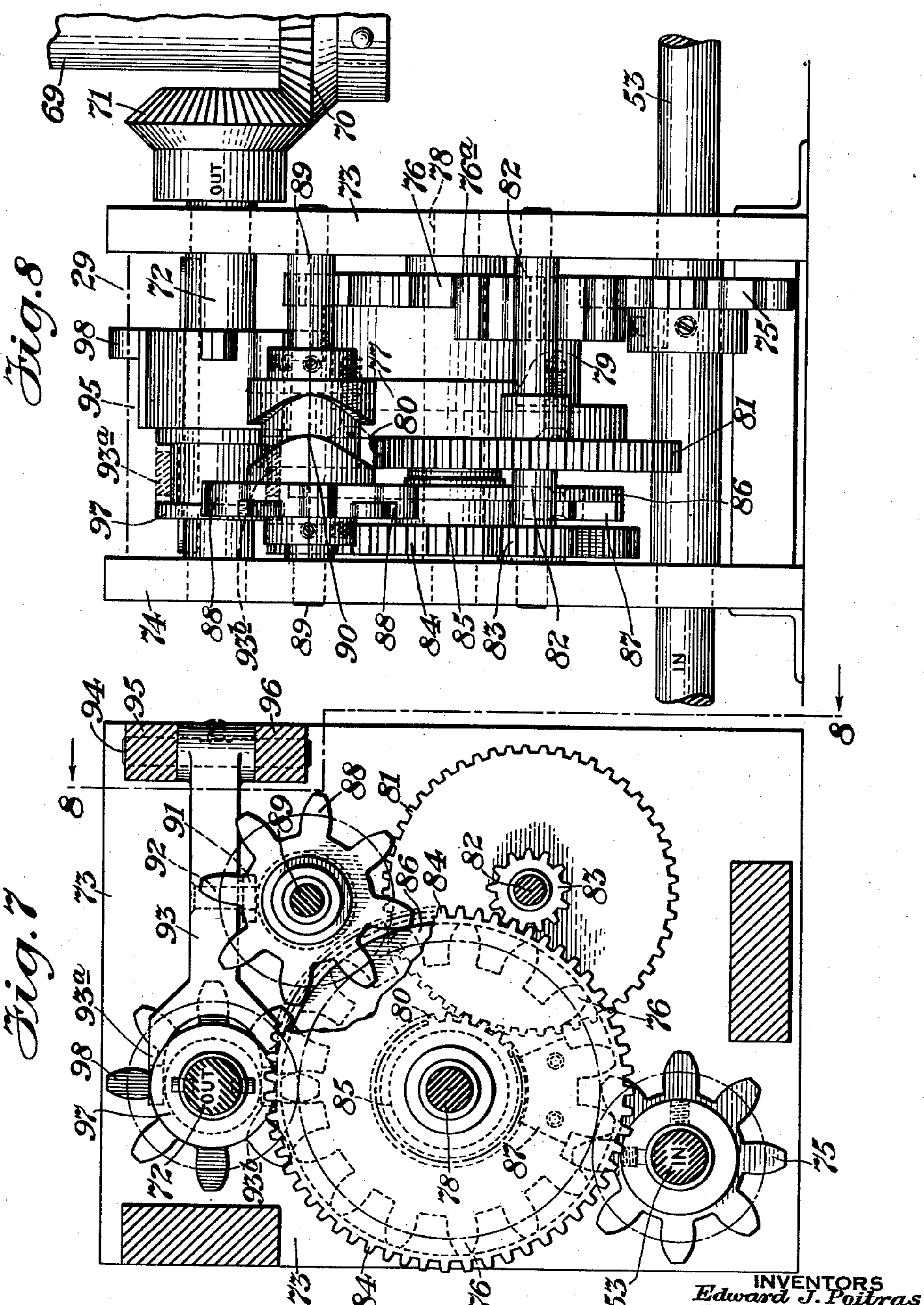

Fig. 8

Fig. 7

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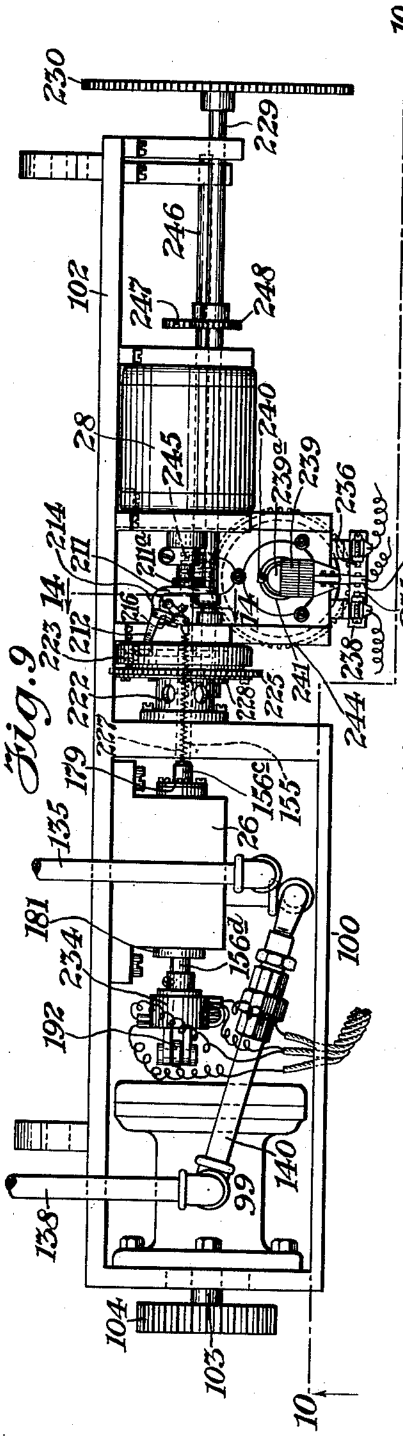
Fig. 9

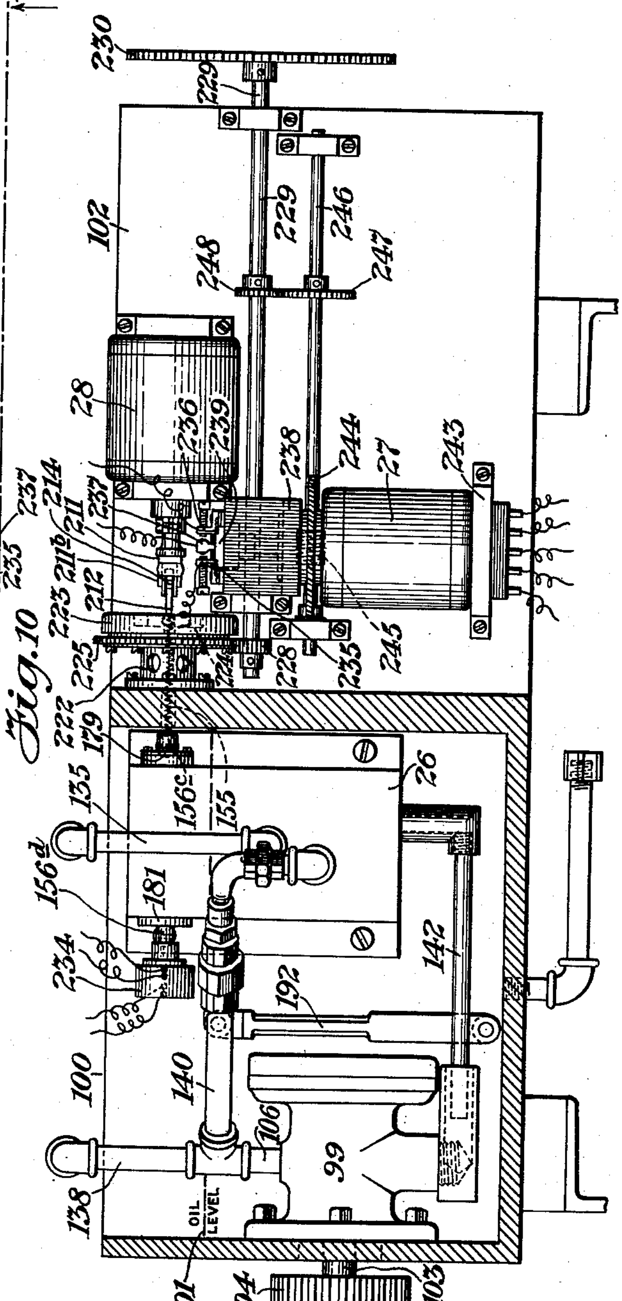
Fig. 10

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Fig. 16

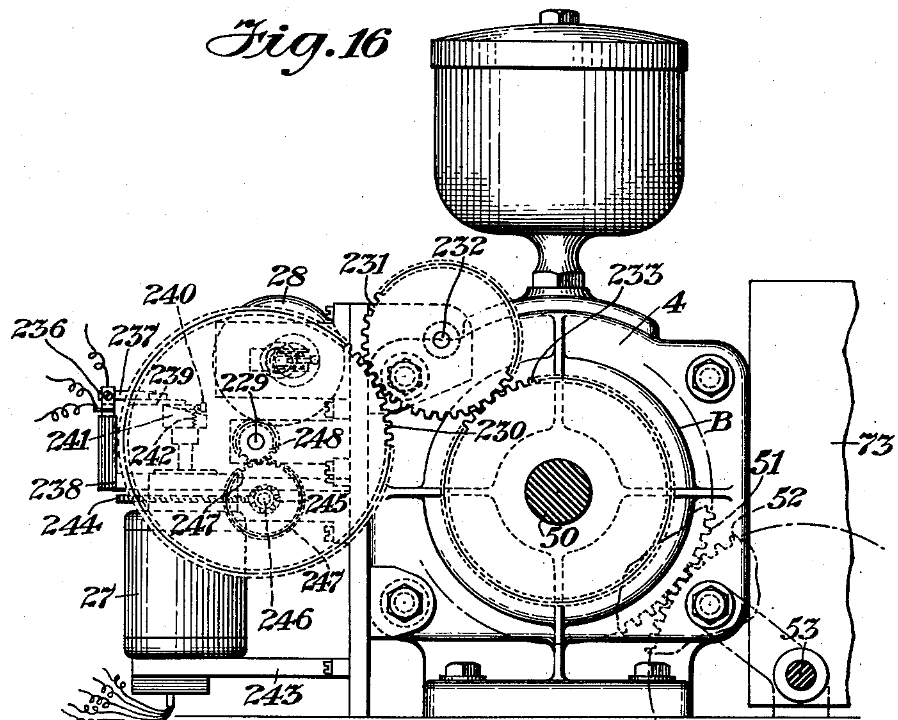

Fig. 17

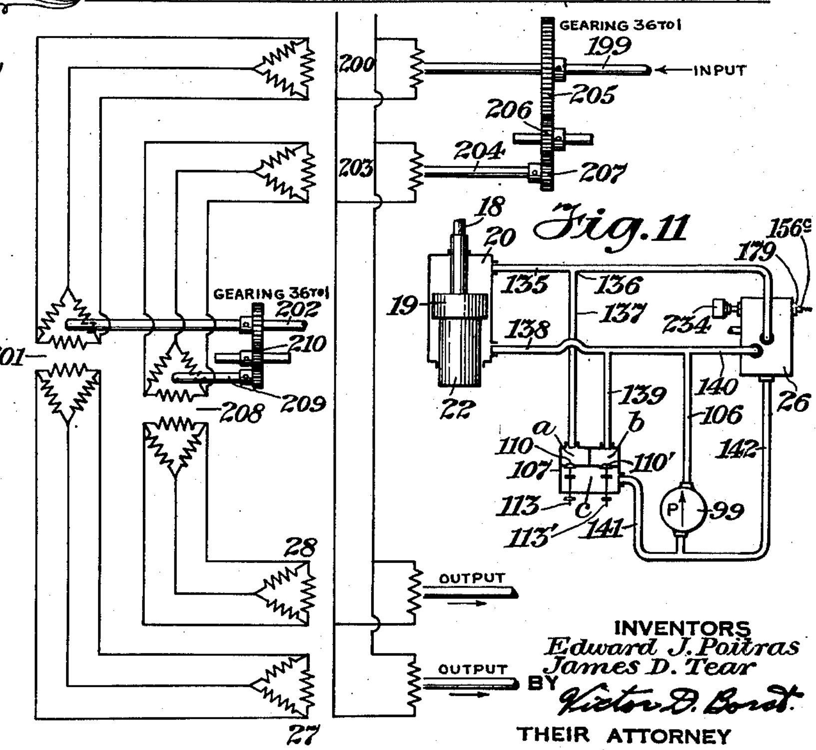

Fig. 11

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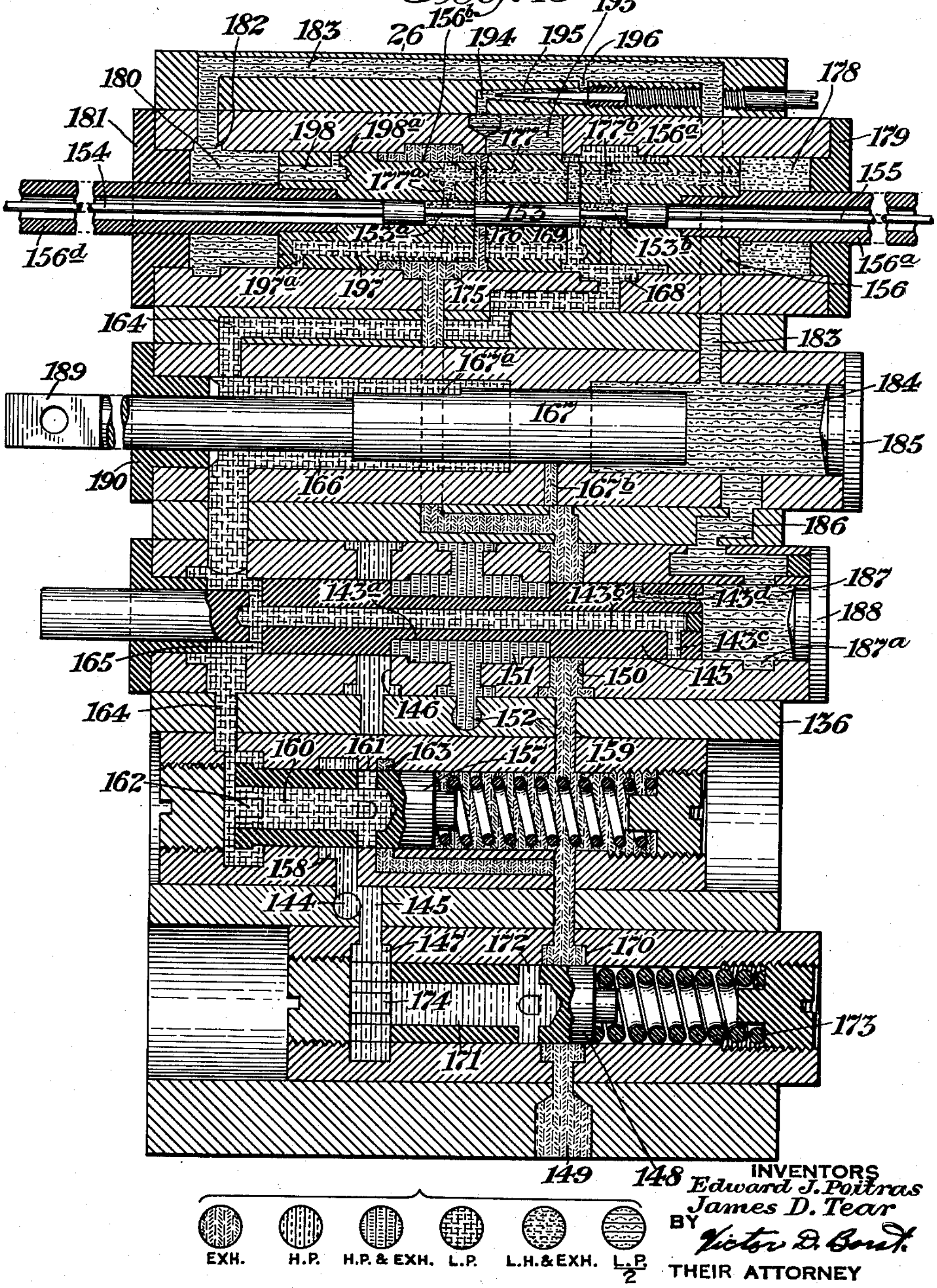
Fig. 13



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Patented Oct. 21, 1952 2,614,390

UNITED STATES PATENT OFFICE

2,614,390

REVERSIBLE HYDRAULIC DRIVE, INCLUDING FOLLOW-UP SYSTEM

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Application January 11, 1935, Serial No. 1,290

19 Claims. (Cl. 60—53)

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The invention herein described comprehends a system and mechanism for controlling the velocity and movement of a driven object. While the invention as thus considered has numerous applications, the embodiment described, in its entirety, is particularly useful in the control of ordnance.

Large ordnance pieces are commonly mounted for movement about two axes at right angles to each other, one, the train axis, perpendicular to the foundation on which the gun is mounted and the other, the elevation axis, at right angles thereto. The aiming of a gun for firing upon a target involves a problem the solution of which gives the necessary movement of the gun in elevation and train. The problem varies with the character of the foundation of the gun and the target, the most exacting conditions being encountered in directing, at a moving target, a gun mounted on an unstable platform such as, for example, the deck of a ship. For the solution of these problems, there is now employed director gun fire control mechanism which from certain observed and generated data related to the line of sight to the target constantly determines and transmits the correct setting of the gun in train and elevation. The inherent characteristics of these gun fire controls and the transmission systems used are such that the power available is little more than enough to operate a signal, such as a dial. Consequently, the gun elevation and train are usually expressed as angular quantities which are transmitted as signals to the gun turret where operators, through manually operative controls, effect movement of the gun to direct it in accordance with the signals.

Several attempts have been made to provide automatic mechanisms and systems in which the gun is made to follow the signals in elevation and train without the intervention of a human agency. There has not been, heretofore, however, an automatic system or mechanism available which is satisfactory, particularly for large guns, and it is one of the objects of this invention to produce a satisfactory system for this purpose. In a complete system for synchronizing the movements of a gun with a signal there should be provisions for moving the gun out of synchronism and returning the gun to synchronism. This is necessary, for example, in order that the gun may assume loading position, or in order that the gun may be prevented from moving beyond the limits of the turret or into positions where if fired it would strike some part of

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the vessel itself. The accomplishment of these functions in such a system is another object of this invention. A further object of the invention is to provide an automatic system of this type which may, if found desirable, be operated manually and independently of the automatic control.

These and other objects are realized in the embodiment of the invention illustrated in the accompanying drawings in which:

Fig. 1 is a side elevation of the system connected for moving a gun mount in elevation;

Fig. 2 is a plan taken on the line 2—2 of Fig. 1;

Fig. 3 is a fragmentary sectional elevation of the mechanism for operating the tilting box of a hydraulic gear, taken along the irregular line 3—3 of Fig. 2;

Fig. 4 is an end, sectional elevation of the hand control mechanism taken along the irregular line 4—4 of Fig. 2;

Fig. 5 is a side elevation of the same, partly in section taken along the line 5—5 of Fig. 4;

Fig. 6 is an enlarged, fragmentary section taken along the line 6—6 of Fig. 2;

Fig. 7 is a sectional elevation of the intermittent gear taken along the line 7—7 of Fig. 2;

Fig. 8 is an end elevation of the same taken along the line 8—8 of Fig. 7, the gear shift lever being omitted;

Fig. 9 is an enlarged plan of the automatic control mechanism for controlling movement of the tilting box of the hydraulic gear;

Fig. 10 is a side elevation of the same taken along the line 10—10 of Fig. 9;

Fig. 11 is a diagrammatic layout of the hydraulic connections;

Fig. 12 is a section of a relief and by-pass valve;

Fig. 13 is a sectional layout of the valve block;

Fig. 14 is a sectional elevation on enlarged scale of a part of the hydraulic control mechanism taken along the line 14—14 of Fig. 9;

Fig. 15 is a sectional side elevation of the same, taken along the line 15—15 of Fig. 14;

Fig. 16 is an end elevation of the hydraulic gear; and

Fig. 17 is a wiring diagram of the signal transmission system.

The general construction and operation of the system as it is used in moving a gun in elevation will first be described. Thereafter, the details of the construction of the several parts will be described and their particular functions and the manner of operation explained.

The system includes a constant speed electric motor 1 of sufficient capacity to readily move the gun mount 2 about its elevation axis 3. The motor is connected to the gun mount through a variable velocity hydraulic transmission or gear 4 of a type well known and illustrated in United States Letters Patent No. 925,148 granted June 15, 1909 to H. D. Williams for Variable Speed Gear. Essentially, the hydraulic gear 4 includes a hydraulic pump and a hydraulic motor known respectively as the A and B ends. The pump and motor are identical in construction with one exception. Each consists of a series of cylinders about and parallel to the longitudinal or shaft axis of the gear, the cylinders of the pump and the motor extending in opposite directions and separated by a stationary valve plate through which they communicate. The rotary motion applied to the pump is converted to reciprocatory motion of the pump pistons, and the hydraulic pressure created by the reciprocation of the pump pistons causes reciprocatory movement of the motor pistons, which movement is converted to rotary movement of the motor shaft, the conversions being effected by plates or boxes inclined to the axis of the gear shaft. The inclination of the motor plate is fixed but the plate 5, or tilting box as it is commonly called (Fig. 3), of the pump is mounted so that its angle of inclination may be varied to vary the length of the stroke of the pistons, and so that its direction of inclination may be reversed. It is by shifting this tilting box that the speed and direction of angular movement of the motor may be varied.

The shaft of the electric motor 1 is directly connected to the pump or A end of the hydraulic gear through a coupling 6 and the motor or B end of the hydraulic gear is connected through a coupling 7 to a shaft 8 which carries a worm 9 between bearings 10 and 11. A worm gear 12 meshes with the worm 9 and carries a pinion 13 that meshes with an arcuate rack 14 on the gun mount 2. Through this train of gears, the B end of the hydraulic gear moves the gun mount in elevation; it being understood that through suitable gearing the movement of the B end can also be used to train the gun.

Movement of the gun is thus controlled by moving the tilting box of the hydraulic gear, the direction of movement of the gun being dependent upon the direction in which the tilting box is moved from its neutral or vertical position and the speed of the gun movement being dependent upon the magnitude or extent the tilting box is moved from its neutral position. In this system, the tilting box may be moved manually or hydraulically, the latter being used for the automatic control of the movement of the gun in accordance with a signal. The manual movement of the tilting box is effected through hand-wheels 15 and 16 which operate through a train of gears upon a rack 17 mounted upon a rod 18 which extends from a piston 19 mechanically connected to the tilting box (see Fig. 3). The piston 19 is mounted in a cylinder 20 having a threaded extension 21 which is threaded into an opening in the case of the hydraulic gear. The piston is provided with a skirt 22 of less diameter than the piston which extends down through the extension of the cylinder and into the tilting box casing. The skirt of the piston is hollow and receives a connecting rod 23 which is pivotally secured to the piston at one end and to a bracket 24 at the other end. The bracket is an integral part of the tilting box 5 which is mounted for movement about a horizontal axis on a shaft 25 which extends through and is journaled in the casing of the hydraulic gear.

The hydraulic operation of the tilting box is effected by controlling the flow of oil under pressure to the cylinder 20 and thereby controlling the actuation of the piston 19. This control is exercised through a valve block 26 containing an hydraulic power amplifier which is operated by a pair of Selsyn receivers 27 and 28 (Figs. 9 and 10), the angular movement of which represents the signal. Whether the manual or the hydraulic operation is used, the effect is the same. A force is applied through the piston and the connecting rod to the bracket 24. The force is in a vertical direction and effects movement of the tilting box about its horizontal axis perpendicular to the longitudinal axis of the tilting box. In either case, there is also a mechanical reference back from the output shaft of the hydraulic gear which acts in opposition to the movement which originated through the control mechanism. This reference back is a restoring mechanism acting in opposition to the movement of the control mechanism initiating the movement of the tilting box, and its effect is to return the tilting box to the neutral position after a definite movement proportioned to the initiating movement of the control mechanism. In this way the total movement of the gun is proportional to the movement acting to displace the tilting box from its neutral position. In the hydraulic control mechanism, there is provided means for eliminating lag by accelerating the movement of the gun to bring it into positional agreement with the signal during movement of the signal.

Both the hydraulic and the manual controls are subservient to limit controls. The limit controls act through an intermittent gear 29 (Figs. 2, 7 and 8) and when the gun reaches a predetermined position the limit control mechanically returns the tilting box to the neutral position.

Hand control

The hand-wheels 15 and 16 provided for the manual control of the movement of the gun are mounted on the ends of a shaft 30 extending through and journaled in a frame 31. On each hand-wheel there is a grip, 15*a* and 16*a*, which is rotatably mounted on the hand-wheel and through which the wheels may be rotated. The hand-wheels are secured to the shaft 30 so that the shaft rotates therewith. On the shaft 30 and between the vertical frame members 31*a* and 31*b* there is mounted a beveled gear 32 which meshes with a beveled gear 33 secured on the end of a shaft 34 extending through and journaled in transverse frame elements 31*c* and 31*d*. Adjacent the frame element 31*d*, there is mounted on the shaft 34 a worm 35. The worm 35 is in mesh with a worm gear 36 (Fig. 4) mounted upon and secured to a shaft 37. Also mounted on the shaft 37 but rotatable thereon there is a spur gear 38 (Fig. 2). The spur gear 38 may be and, when the hand control is being used, is connected for rotation with the shaft 37 through a clutch 39. The gear 38 meshes with a gear 40 rotatably mounted upon a shaft 41 which extends between the frame members 31*a* and 31*b*. Formed integral with the gear 40 there is a bevel gear 40*a* which meshes with two bevel gears 42*a* and 42*b* (Fig. 5) which are mounted upon the spider of a differential which spider is secured to the shaft 41 for rotation therewith. Also meshing with the bevel gears 42*a* and 42*b* there is a bevel gear 43 rotatably mounted on the shaft

41 and formed integral with a helical gear 43a. The shaft 41 extends through the side 31b of the frame and on the end of the shaft there is mounted for rotation therewith a spur gear 44. This latter gear meshes with a gear 45 mounted upon the end of a shaft 46 which extends through a frame 47 mounted on the end of the hydraulic gear. Intermediate the ends of the frame, there is mounted on the shaft 46 for rotation therewith a pinion 48 which meshes with the rack 17 on the rod 18.

Assuming that the clutch 39 is engaged, rotation of the hand-wheels 15 and 16 will effect rotation of the shaft 34 through the bevel gears 32 and 33. Through the worm and worm gear connection 35 and 36, the clutch and the spur gear 38 are rotated with the shaft 37. The worm and worm gear form a non-reversible connection, that is, the hand-wheels could not be rotated through the gear 38. Rotation of the gear 38 effects, through its connection with the gear 40, rotation of the gear 40a forming one end of the differential. The helical gear 43a meshes with a gear 49 which forms a non-reversible gear train, that is, movement of the gear 43 can only be effected by movement of the gear 49. Thus, upon the initial movement of the hand-wheels and rotation of the gear 40a one side of the differential is stationary and rotation of the gear 40a effects a rotational movement of the spider of the differential and consequently of the shaft 41. Rotation of the shaft 41, through the gears 44, 45, 48 and the rack 17, causes vertical movement of the rod 18 and, consequently, a displacement of the tilting box 5 of the hydraulic gear. Upon the displacement of the tilting box from its neutral position, the shaft 50 of the B end of the hydraulic gear will be rotated in a direction and at a speed corresponding to the extent of the movement of the tilting box and through its connection with the shaft 8 will cause the gun to move in elevation.

On the shaft 50, there is mounted a gear 51 which meshes with a gear 52 secured on the end of a shaft 53 running alongside of, and journaled in bearings secured to the hydraulic gear. The shaft 53 extends parallel to the axis of the hydraulic gear and through the frame member 31b. On that end of the shaft 53 which extends through the frame member 31b, there is mounted a bevel gear 54 (Fig. 5) which meshes with a bevel gear 55 mounted on, and adjacent the lower end of a vertical shaft 56. The vertical shaft 56 is mounted in two thrust bearings 57 and 58 extending from the frame member 31b. The helical gear 49 is mounted on the upper end of the shaft 56. Through this train of shafts and gears, the gear 49 is rotated in accordance with the rotation of the shaft of the B end of the hydraulic gear. Rotation of the gear 49 effects rotational movement of the gear 43a, and, consequently, of the bevel gear 43 forming one end of the differential. If the hand-wheels are rotated a certain amount and stopped, the gear 40a will rotate the spider a corresponding amount. Upon rotation of the shaft 50 of the hydraulic gear, the gear 43 of the differential will be rotated so as to rotate the spider in a direction opposite to that in which it was rotated by movement of the gear 40a. This will continue until the tilting box of the hydraulic gear is returned to its neutral position in which position movement of the gun ceases. Thus, the total amount of movement of the gun is proportional to the amount of movement of the hand-wheels. If the hand-wheels and the gun are moving at the same speed, the tilting box will remain in a particular position and the gun will be elevated or depressed at a speed corresponding to the movement of the hand-wheels. In order for the B end of the hydraulic gear to attain a particular velocity in accordance with the movement of the hand-wheels, it is necessary that the gun lag behind the movement of the hand-wheels. If the hand-wheels are rotated beyond a certain amount in either direction, the operation of the tilting box will be effected by the hydraulic gear through the limit stops.

The limit stops

The limit stops act mechanically upon the shaft 18. As shown in Fig. 6, the shaft 18 is reduced in diameter just below the rack so that two shoulders 18a and 18b are formed thereon. The reduced portion of the shaft extends through a movable block 59 having an opening of a diameter sufficient to receive the reduced portion of the shaft and which is adapted to engage the shoulders 18a and 18b. The block 59 is moved vertically through a pair of screw shafts 60 and 61 journaled in the upper and lower elements of the frame 47 and nuts 62 and 63 secured to the block 59 and engaging the threads of the screw shafts 60 and 61 respectively. Adjacent the upper end of the screw shaft 60 there is a spur gear 64 secured to the shaft for rotation therewith. A similar gear 65 is secured to the shaft 61. These two gears both mesh with a gear 66 mounted on a vertical shaft 67 which is also journaled in the frame 47. The gear 66 also meshes with a gear 68 secured on the end of a vertical shaft 69 journaled at its upper end in an extension of the upper member of the frame 47. The lower end of the vertical shaft 69 carries a bevel gear 70 (see Fig. 8) which is secured thereto for rotation therewith and which meshes with a bevel gear 71 secured on the end of a shaft 72 which extends through and is journaled in the sides 73 and 74 of the casing of the intermittent gear 29.

The intermittent gear 29 is an instrumentality by which a constant input effects an intermittent output at a speed equal to the input. That is, if the input shaft is rotated continuously, the output shaft will commence rotating only after a definite predetermined number of revolutions of the input shaft from any given relationship of the parts, and at a speed equal to the speed of the input shaft. In the intermittent gear 29 the input is taken off from the shaft 53 which as heretofore explained is connected to the shaft 50 of the B end of the hydraulic gear. After the shaft 53 has rotated a definite predetermined number of revolutions from a given starting position, the shaft 72 rotates and through the beveled gears 70 and 71 rotates the shaft 69. Rotation of the shaft 69 causes, through the gears 68, 66, 65 and 64, rotation of the screw shafts 60 and 61. Upon rotation, these shafts cause the block 59 to move vertically along the shaft 18 and engage one or the other of the shoulders 18a or 18b. The direction of movement of the block is such as to move the rod 18 in such direction as to return the tilting box of the hydraulic gear to the neutral position. When this has been done, the B end of the hydraulic gear can not be rotated further in this direction. Suitable provision is made to allow of the backward rotation of the gear 48 and connected parts when the rack 17 is operated by the limit stop to restore the tilting box to neutral and the clutch 39 is engaged. For example the clutch may be constructed to slip when thus reversely driven.

The intermittent gear accomplishes its function through the mechanism illustrated in Figs. 2, 7 and 8. On the shaft 53 which extends through the sides 73 and 74 of the intermittent gear frame, there is mounted a spur gear 75. This spur gear meshes with a larger gear 76 formed on the end of a drum 77 mounted on a shaft 78 extending between and journaled in the sides 73 and 74 of the intermittent gear housing. The drum 77 is reduced in diameter adjacent the gear 76 and the shoulder formed by this reduction in diameter is notched as indicated at 79 for a purpose which will hereinafter appear. The shaft 78 also carries a gear 80 which meshes with a gear 81 mounted upon and secured to a shaft 82 extending between and journaled in the frame members 73 and 74. On the shaft 82, there is a gear 83 which meshes with a larger gear 84 mounted upon the shaft 78 for rotation relative thereto. The gear 84 is connected through a hub 85 with a disk 86 having a single tooth-receiving slot cut therein. A plate 87 comprising two gear teeth is secured against the inner face of the disk 86 with the space between the teeth aligned with the slot in the disk. The slot of the disk 86 and the teeth of the plate 87 are positioned to engage a gear 88 mounted upon a shaft 89 extending between the sides 73 and 74. The teeth on the gear 88 are alternately wide and narrow.

As the disk 86 rotates and the plate 87 approaches the gear 88, teeth of the greater width engage the periphery of the disk and prevent the gear 88 from rotating in either direction. Between the two teeth of greater width of the gear 88 which engage the periphery of the disk 86, a tooth of lesser width extends into the space formed between the gear 84 and the disk 86 and surrounding the hub 85. As the plate 87 approaches the gear, the forward tooth thereon engages the narrow tooth of the gear 88 which extends between the disk and the gear 84 and the tooth of greater width enters the slot in the disk and the gear is rotated until the next succeeding tooth of greater width engages the edge of the disk. In this way, the gear 88 is rotated two teeth for each revolution of the gear 84, the speed of rotation of which has been reduced considerably from the speed of rotation of the shaft 53 through the reduction gearing. A gear of this general type is shown in Ford Patent No. 1,811,624.

The shaft 89 also carries a cam 90. This cam is formed by cutting a circumferential slot in a cylindrical piece of metal, the slot or groove having one axial offset to form the cam rise, as shown in Fig. 8. Into this groove a cam follower 91 extends and cooperates with the sides of the groove. The cam follower 91 is mounted upon the end of a vertical pin 92 which extends transversely through and is secured in a shift lever 93 which is pivotally mounted at one end on a vertical pin 94 which extends through a hub on the end of the lever 93 and spaced arms or lugs 95 and 96 extending from the side of the intermittent gear housing. The other end of the shift lever 93 is bifurcated and the arms 93*a* and 93*b* thereof embrace and are received between the sides of a collar 97 which is an integral part of a gear 98 mounted upon the shaft 72. The gear 98 is splined upon the shaft 72 and is free to move longitudinally of the shaft but constrained to rotational movement with the shaft. The gear is so mounted as to be movable longitudinally of the shaft 72 into and out of engagement with the gear 76, its movement being dependent upon the cam 90. Like the gear 88, the gear 98 has successive teeth of different widths. Two teeth of greater width on the gear 98, when that gear is positioned so as to be out of engagement with the gear 76, as illustrated in Fig. 8, engage the periphery of the drum 77 so that the gear is held in a particular position. A tooth of lesser width between the two teeth of greater width which engage the drum 77 extends radially into the circumferential groove formed by reducing the diameter of the drum.

For a portion of the circumference of the gear 76, the gear teeth are widened as indicated at 76*a*. These teeth of greater width are so angularly disposed that the end tooth engages the tooth of the gear 98 extending into the groove formed by the reduction of the diameter of the drum 77, when the gear 98 is moved slightly along the shaft 72 in a direction towards the gear 76, that is, to the right as indicated in Fig. 8. The drum 77 is notched at 79 so as to provide a clearance for the tooth of greater width on the gear 98 engaging the drum when that gear is first engaged by the gear 76. It will be apparent that if the shaft 53 is rotated in one direction a sufficient number of revolutions for the cam shaft 89 to be rotated into a position where the offset portion of the cam engages the cam follower 91, the gear 98 will be moved into mesh with the gear 76. The offset of the cam 90 is sharp so that the gear 98 moves into mesh with the gear 76 rapidly. Upon meshing with the gear 76, the gear 98 and the shaft 72 are rotated at the same speed as the shaft 53. The limit stops are thus brought into action and continue to operate until the tilting box of the hydraulic gear is in the neutral position, in which position the hydraulic gear can not rotate further in the direction in which it was rotating when the limit stops came into action. The hydraulic gear thus measures its own limits of movement, the intermittent gear operating in both directions. The relation of the gears in the intermittent gear is such that the movement of the gun in elevation is restricted within the desired limits.

Hydraulic control

The piston 19 in the cylinder 20, through which the hydraulic control acts, is operated by a suitable liquid such as oil under pressure. The oil under pressure is supplied by a pump 99 (Figs. 9 and 10) mounted in a box 100 which contains a reserve supply of oil, the box being filled with oil to a level indicated by the dotted line 101. The box 100 is secured on to a plate 102 which is mounted upon the side of the hydraulic gear. The pump is of the rotary type and the shaft 103 thereof extends through the side of the box 100. On the end of the shaft 103 there is mounted a gear 104 which meshes with a gear 105 (Fig. 1) mounted upon and secured to the shaft of the A end of the hydraulic gear. Thus, whenever the motor 1 is operating, the pump is functioning. It is usually satisfactory for the pump to supply oil under a pressure of 600 lbs. per square inch. The oil is delivered through a pipe 106 communicating with the pump at its upper end.

The hydraulic control of the tilting box is dependent upon a duplex relief valve 107. The construction of this valve is shown in Fig. 12.

The valve consists of a casing which is divided into three chambers *a*, *b* and *c* by a horizontal partition 108 and a vertical partition 109. Each of the chambers *a* and *b* communicates with the chamber *c* through openings in the wall 108. The opening between the chamber *a* and the chamber *c* is normally closed by a poppet-valve 110. The valve is resiliently urged to the closed position by a spring 111 which abuts against the valve at one end and against the plate 112 at the other end. The plate 112 is mounted upon a rod 113 which extends through the bottom of the casing and is slidable therein, and through a plate 114. On the rod 113 there is a collar 115 which abuts against the upper surface of the plate 114. A similar valve with a similar arrangement is provided for normally closing communication between the chambers *b* and *c*. This valve and the corresponding parts thereof are numbered similarly to the valve 110 with the numbers primed. With the parts in the position shown in Fig. 12, the force exerted by the springs 111 and 111′ is very great, being sufficient to resist a pressure in the chambers *a* and *b* of 600 lbs. per square inch, for example, the total pressure of the pump. If, however, the pressure exceeds this, the valves will open. This is the position of the valves when the hydraulic control is in operation.

The force exerted by the springs 111 and 111′ may be lessened considerably and is lessened for the purpose of hand operation. This is accomplished by shifting a lever 116 mounted adjacent the hand-wheels 15 and 16. The lever 116 is secured to a shaft 117 journaled in transverse members 31*c* and 31*d* extending between the side frame members 31*a* and 31*b* (Fig. 4). This shaft also carries secured thereto a pair of spaced arms 118 and 119 forming a dog for operating the clutch 39. The clutch has therein a circumferential groove in which engage pins 120 and 121, extending inwardly from the arms 118 and 119 respectively. The clutch is splined upon the shaft 37 for longitudinal movement with respect thereto but restrained against relative rotational movement. A spring 122 (Fig. 2) urges the clutch into the engaged position.

When the hydraulic control is being used, the clutch is restrained from moving into the engaged position under the action of the spring 122 by a pin 123 which engages the lever 116. The pin 123 extends through a cylindrical housing 124 mounted upon the transverse frame member 31*c*. Adjacent the end of the pin 123 which engages the lever 116, there is a collar 125 which forms an abutment for one end of a spring 126 mounted in the housing 124 and abutting at its other end against a shoulder therein. The pin 123 is slidable within the housing 124 longitudinally thereof and at its opposite end is provided with a cross bar 127 by means of which it may be moved out of engagement with the lever 116 against the action of the spring 126. When this is done, the lever 116 moves to the right as seen in Fig. 2 under the influence of spring 122 and the clutch 39 becomes engaged.

A lever 128 secured on to the shaft 117 extends radially therefrom. This lever carries at its end a block 129 through which the end of a rod 130 extends. The rod 130 has a collar 131 thereon which abuts against the block 129. The opposite end of the rod 130 is pivotally secured to one end of a lever 132 which is pivotally mounted between its ends in a transverse frame member 31*e*. The other end of the lever

132 is pivotally secured to a link 133 to which is secured the plate 114 through which the valve rods 113 and 113′ extend. A link 134 is pivotally secured at one end to the transverse frame member in which the lever 132 is pivotally mounted and at the other end it is pivotally secured to the link 133. This additional link provides a straight line movement of the link 133. It will be observed that when the lever 116 is released and the clutch 39 is permitted to engage, the pressure on the collar 131 by the block 129 will be released and the plate 114 may move downwardly thus relieving the tension of the springs 111 and 111′. The holes in the plate 114 for the valve rods 113 and 113′ will provide the necessary looseness to allow of the slight relative lateral movement of the plate due to the arcuate movement of its pivots.

The hydraulic connections to the cylinder 20 are illustrated diagrammatically in Fig. 11. The upper chamber of the cylinder is connected through a pipe line 135 to the valve block 26 in which there is included a piston valve for controlling the supply of oil under pressure to the upper chamber of the cylinder. The valve block is mounted in the box 100 in which the pump is mounted. The upper end of the cylinder is also connected through a T-fitting 136 and by a pipe 137 to the chamber *a* of the valve 107. The lower chamber of the cylinder is connected through a pipe 138 to the high pressure side of the pump and through a T-fitting and a pipe 139 to the chamber *b* of the valve 107. The high pressure side of the pump is also connected through a pipe 140 to the valve box. The chamber *c* of the valve 107 is connected through a pipe 141 to the intake side of the pump to which there is also a connection through a pipe 142 to the valve block. It will be apparent from the connections illustrated diagrammatically in Fig. 11 that both the upper and lower chambers of the cylinder 20 are connected to the low pressure side of the pump through the valve box 107. The pressure relief valves in the valve box have a dual function. When the hydraulic gear is operating and controlled hydraulically, the relief valves are under a heavy spring tension. However, if the hydraulic gear runs against a limit stop and the tilting box and piston 19 are moved by the force of the hydraulic gear to a neutral position, a high pressure would be created in the cylinder 20. This pressure is relieved by the pressure relief valves in the valve box 107. On the other hand, when the hand control is effective, the light tension on the springs provides a ready escape of the oil to the low pressure side of the pump, thus, in effect, short-circuiting the pressure and rendering the hydraulic control ineffective.

The operation of the piston 19 in the cylinder 20 is accomplished by maintaining pressure on both sides of the piston. It will be noted that the high pressure side of the pump is directly connected to the lower chamber of the cylinder. The surface area of the piston upon which this pressure acts is, however, approximately one-half of the surface area of the opposite face of the piston. Consequently, the same pressure in the upper chamber would create a force twice as great as the force created by the pressure in the lower chamber. When, however, the pressure in the upper chamber is relieved, the pressure in the lower chamber causes the piston to rise in the cylinder. With this arrangement, the movement of the tilting box can be, and is accomplished by controlling the connection of the upper chamber to the high pressure and the exhaust.

This control is effective through the piston valve 143 in the valve block 26 (see Fig. 13). Oil under pressure is supplied to the valve box through the pipe 140 which communicates with a transverse passage 144 in the valve block. The passage 144 communicates with the passage 145 which extends between an annular port 146 surrounding the piston valve 143 and an annular port 147 surrounding a constant pressure valve 148 which is mounted in the valve block and utilized for the purpose of maintaining the high pressure at a constant pressure of, for example, approximately 600 lbs. per square inch. The pipe 142 communicates with a passage 149 in the valve block. This passage extends vertically through the valve block and communicates with a port 150 surrounding the piston valve. The piston valve 143 is of the usual type in that it has a reduced central portion 143a which is adapted upon movement of the valve to lap either the high pressure or the exhaust port.

The circumferential chamber formed by reducing the valve diameter communicates with a port 151 which, through a passage 152, communicates with the pipe 135 connected to the upper chamber of the cylinder 20. As viewed in Figs. 11 and 13, it will be apparent that when the piston valve 143 is moved to the left the upper chamber of the cylinder will be placed into communication with the high pressure, and when the valve is moved to the right the upper chamber of the cylinder will be placed into communication with the exhaust. The length of the reduced portion of the valve is such that intermediate these two positions and in the position illustrated in Fig. 13 both the high pressure and exhaust ports are cut off from communication with the upper chamber of the cylinder.

The valve 143 is shifted to control the communication with the upper chamber of the cylinder 20 and the high pressure or exhaust by what might well be termed an hydraulic amplifier. This hydraulic amplifier is operated through a pilot valve 153 which is provided with a rod, 154 and 155, extending from each end thereof through the valve block. The pilot valve 153 is mounted in a plunger 156 which acts upon oil in a substantially closed system connecting a chamber formed at one end of the plunger with a chamber formed at one end of the valve 143. The pilot valve controls the movement of the plunger which upon moving varies the pressure in the closed system.

A low pressure supply of oil, 100 lbs. per square inch, is provided for the purpose of operating the plunger 156. Oil at this pressure is supplied by a reducing valve 157, having a port 158 communicating with the passage 144. The reducing valve consists of a piston having a spring 159 acting on one end thereof, the force of which is regulated so that the valve supplies the requisite pressure. A longitudinal passage 160 in the piston communicates with a transverse passage 161. When the pressure of the oil in the passage 160 is reduced below 100 lbs. per square inch, the spring 159 causes the piston 157 to move to the left, as shown in Fig. 13. Upon movement in this direction, the transverse passage 161 is placed into communication with the port 158 and oil at 600 lbs. per square inch is permitted to enter the passage 160 which communicates with a port 162. An exhaust port 163 is provided so that in the event that the pressure in the passage 160 exceeds 100 lbs. per square inch the piston 157 is moved to the right against the action of the spring 159 and the transverse passage 161 is placed into communication with the exhaust port 163 until the pressure in the passage 160 is reduced so that the spring moves the piston 158 to the left and the passage 161 is cut off from communication with the port 163.

The port 162 communicates with a passage 164 which communicates with a chamber 165 formed at the left-hand end of the piston valve 143. The passage 164 also communicates with a chamber 166 formed at the left-hand end of a plunger 167 and supplies oil to the pilot valve 153 for which purpose it communicates with a port 168. Port 168 is normally in communication with a transverse passage 169 through the plunger 156, the plunger 156 having a reduced portion 156a with which the port 168 and the passage 169 communicate. In the position of the parts shown in Fig. 13 the passage 169 in the plunger 156 is cut off by the pilot valve 153. The pilot valve 153 has two reduced portions 153a and 153b. A common exhaust passage similar to the passage 164 is also provided in the valve block. This exhaust passage extends from the port 149 to a port 170 surrounding the piston 148 of the constant pressure valve.

The constant pressure valve is similar to the reducing valve in that it contains a piston having a longitudinal central passage 171 and a transverse passage 172 and a spring 173 acting upon the piston with a constant pressure. One end of the passage 171 in the piston communicates with the port 147 through a passage 174. The passage 172 is adjacent the exhaust port 170 and when the pressure in the passage 171 exceeds 600 lbs. per square inch the piston 148 is moved to the right against the action of the spring 173 and the passage 172 and the port 170 are placed in communication until the pressure is reduced to 600 lbs. per square inch.

The exhaust passage extends through the block and communicates with the port 150 and a port 175 surrounding a reduced portion 156b of the plunger 156. A transverse passage 176 extends through the plunger and normally communicates with the exhaust port 175. This passage, like the passage 169, is cut off by the pilot valve in the position shown in the drawings. The reduced section 153a of the pilot valve 153 forms a chamber which communicates with a passage 177a which is a branch of a passage 177 extending longitudinally of the plunger and opening at the end of the plunger to a chamber 178 formed between the end of the plunger and a cap 179. The passage 177 also has a branch passage 177b which communicates with the chamber formed by the reduced portion 153b of the pilot valve. Through these several ports and passages, the chamber 178 may be placed into communication with either the intermediate pressure or the exhaust. For example, if the pilot valve is moved to the left from the position shown in Fig. 13, the passage 169 will communicate with the passage 177b through the chamber formed by the reduced portion 153b of the pilot valve. Oil at the intermediate pressure will therefore pass through the passage 177 into the chamber 178. If the pilot valve, on the other hand, is moved to the right from the position shown in Fig. 13, the passage 176 which communicates with the exhaust will be placed into communication with the chamber formed by the reduced portion 153a of the pilot valve and through the chamber and the passages 177a and 177, the chamber 178 will be placed into communication with the exhaust.

Movement of the plunger 156 in response to movement of the pilot valve acts upon and affects the pressure of the oil contained in a closed system of chambers and passages. This system includes a chamber 180 formed between the left-hand end of the plunger and a cap 181, a port 182 communicating with the chamber 180, a passage 183 communicating with the port 182, a chamber 184 formed between the right-hand end of the plunger 167 and a cap or end-piece 185, and a passage 186 which communicates with a chamber 187 formed between the right-hand end of the piston valve 143 and an end cap 188. The pressure in this system is, in the neutral position of the plunger, that is, the position shown in Fig. 13, approximately one-half of the low pressure or 50 lbs. per square inch. In the position shown, the pressures in the chambers 178 and 180 are balanced. Upon movement of the pilot valve to the left, however, the chamber 178 is placed into communication with the low pressure and the pressure in the chamber 178 is increased. The plunger thereupon moves to the left and thus decreases the volume of the chamber 180. The necessary consequence of such a movement of the plunger is an increase in pressure in the chamber 180 and the passages and chambers to which it is connected, principally in the chamber 187. It will be noted that the surface area of the end of the piston valve 143 forming one end of the expansible chamber 165 is approximately one-half of the surface area of the end of the piston forming one end of the expansible chamber 187. Consequently, an increase in pressure in the chamber 187 will cause the piston valve to move to the left and thus place the upper chamber of the cylinder 20 into communication with the high pressure.

If it be assumed that the pilot valve is moved to the right from the position shown in Fig. 13, and the chamber 178 placed into communication with the exhaust in the manner heretofore explained, the pressure in the chamber 178 will be reduced and due to the pressure in the chamber 180 it will move to the right following the movement of the pilot valve. Thus, the volume of the chamber 180 will be decreased and the pressure in the closed system of which the chamber forms a part will be correspondingly reduced. The result of this reduction in pressure, in so far as the piston valve 143 is concerned, is that the total force exerted by the oil in the chamber 165 will exceed the total force exerted by the oil in the chamber 187 and the piston valve 143 will be moved to the right. Thus, the upper chamber of the cylinder 20 will be placed into communication with the exhaust passage 150. It is to be noted that the plunger 156 follows the movement of the pilot valve. In practice the difference in movement of these two elements is imperceptible. In view of this action, the valve and plunger tend to retain the same relation as that illustrated in Fig. 13.

The valve mechanism just described, if unmodified, would operate so as to cause a movement of the piston 19 in the cylinder 20 which would not be definitely related to the movement of the pilot valve 153 since the piston valve 143 would remain open when the pressures in the chambers 187 and 165 were equalized by movement of the valve 143. The action is, therefore, modified so that the piston 19 is returned to the off-position on the completion of a definite movement of the pilot valve, and, consequently, the movement of the tilting box is proportional to the movement of the pilot valve. This is accomplished through the action of the plunger 167 which forms with the plunger 156 and the valve 143, acting in the closed system, a hydraulic differential in which the movement of the valve 143 is the algebraic sum of the movements of the plungers 156 and 167. Like the piston valve 143, the plunger 167 has end faces of different areas. The end face forming one side of the chamber 166 is approximately half the area of the surface extending into the chamber 184. Thus, in the normal condition the resultant force on this plunger is equalized. The surface area of the end of the plunger and the chamber 166 is cut down by a rod 189 which is integral with the plunger and which extends through the cap 190 forming one end of the chamber 166. The end of the rod 189 is pivotally secured to one end of a link 191 (Fig. 3), the other end of which link is pivotally secured to one end of a lever 192 which is mounted upon and secured to the shaft 25 extending through the casing of the hydraulic gear and which rotates with the movement of the tilting box. Thus, the plunger 167 is moved to the right or to the left depending upon whether the piston 19 is moved upwardly or downwardly. Since the piston 19 moves downwardly when the piston valve 143 is moved to the left, the plunger 167 will also be moved to the left and conversely, that is, the plunger 167 is moved through the action of the tilting box in the same direction as the piston valve 143. Movement of the plunger 167 changes the volumetric space of the closed system. The result of its movement in the same direction as the piston valve is that it increases the volume of the closed system when the plunger 156 acts in a direction to decrease the volume, and when the plunger 156 moves in a direction to increase the volume, the plunger 167 is moved in a direction to decrease the volume. That is, its tendency is to offset the effect of the movement of the plunger 156. Therefore the plunger 167 responds to the movement of the tilting box and restores the balance on the piston valve 143. The plunger 167 is therefore denominated a response plunger. It will be apparent, however, that movement of the plunger 167 is dependent upon movement of the tilting box so that upon movement of the plunger 156 the pressure in the closed system is increased or decreased and this increase or decrease remains constant for a space of time. In other words, the movement of the plunger 167 lags behind the movement of the plunger 156, the lag being occasioned by the fact that the plunger 167 is moved by the tilting box. However, upon movement, the plunger tends to and does ultimately reestablish equilibrium with the valve 143 in the position shown in Fig. 13. The result is that if the pilot valve is moved in either direction a definite amount and then held in the position to which it is moved, the tilting box will be moved through an angle proportional to the movement of the pilot valve and held in that position. This would be equivalent to a response to a signal requiring movement of the gun at a constant velocity.

With the arrangement described, there is necessarily a phase difference or velocity lag of the movement of the gun behind the signal as represented by the movement of the pilot valve. This lag, in the system so far as it has been described, is necessary in order to secure a velocity in the movement of the gun. In practice, when the signal demands a high acceleration of the gun or movement of the gun at a constant velocity, this lag between the signal and the gun is undesirable as the ideal desired is a synchronous movement of the gun and signal without any phase difference. There is provided, therefore, in the system a means for removing the lag. This consists of a communication between the intermediate pressure or exhaust and the closed system acting between the plunger 156 and the piston 143. This communication is through a port 193, a passage 194, a needle valve 195 and a passage 196. The port 193 is placed in communication with the exhaust port 175 or the intermediate pressure port 168 depending upon the movement of the plunger 156. If the plunger 156 is moved to the left from the position illustrated in Fig. 13, the port 193 is placed in communication with the intermediate pressure port 168 and oil flows from the intermediate pressure port 168 through the passage 194, the needle valve 195 and the passage 196 so as to increase the pressure in the passage 183. On the other hand, if the plunger moves to the right, the passage 183 is placed into communication, through the needle valve, with the exhaust port 175. Thus, the communication with the passage 183 through the needle valve is such as to augment the effect of the movement of the plunger. The needle valve is adjustable and is set in such position as gives the best results. The result is that the pressure in the passage 183 and the closed system is augmented or decreased gradually. The piston valve 143 is thus moved beyond the point to which it would otherwise be moved upon movement of the pilot valve. This additional movement causes the hydraulic gear to assume a velocity in addition to that which is proportional to the movement of the pilot valve. This advances the gun into synchronism with the signal. Oil passing through the needle valve also maintains the oil in the closed system at the proper volume. The needle valve must be adjusted empirically in each particular system.

Under extreme conditions of acceleration the pilot valve and consequently the plunger 156 may be moved rapidly from one extreme position to the other. To guard against any of the parts assuming a position from which they could not be moved, there is provided certain limiting ports and passages. In the plunger 156 two such passages are provided 197 and 198. The passage 197 extends between the passage 169 and a transverse branch passage 197*a*, the branch passage extending between the passage 197 and the circumferential surface of the plunger 156. The passage 198 communicates with the chamber 180 through the end of the plunger and a branch passage 198*a* which extends between the passage 198 and the surface of the plunger. The passage 197*a* is so located that if the plunger 156 is moved to the left from the position illustrated in Fig. 13, until the end thereof is adjacent the cap 181, the passage 197*a* will be in communication with the port 182 and also the low pressure port 168. In this condition, oil under the low pressure will be admitted to the port 182 and move the piston to the right until the passage 197*a* is covered. On the other hand, if the pressure in the chamber 180 becomes such as to force the plunger to the right against the cap 179, the passage 198*a* will communicate with the exhaust port 175 and thus exhaust oil from the chamber 180. The oil from the chamber 180 will be exhausted until the pressure in the chamber 178 is sufficient to move the plunger to the left a distance sufficient to cut off the communication between the passage 198*a* and the exhaust port 175.

Similar ports and passages are provided in the plunger 167 and the valve 143. In the surface of the plunger 167 there is formed a cut indicated at 167*a*. This cut is such that when the plunger is moved a sufficient distance to the right from the position shown in Fig. 13, the chamber 184 will be placed in communication with the chamber 166 so that oil under pressure will pass from the chamber 166 through the cut 167*a* and into the chamber 184. When the plunger 167 is moved to the left a sufficient distance it uncovers a port 167*b* which places the chamber 184 into communication with the exhaust. The piston valve has a longitudinal central passage 143*b* which communicates at one end with the chamber 165 and at the other end with a radial passage 143*c*. Upon movement of the piston valve a sufficient distance to the right, the passage 143*c* communicates through a groove or port 187*a* with the chamber 187 and thus connects the chamber 187 to the chamber 165. Upon movement of the piston valve to the extreme left an L-shaped passage 143*d* connects the chamber 187 to the exhaust port 151. These limiting ports and passages also establish the volume of the oil in the closed system.

The various ports and passages are related to the oil pressures existing therein as indicated by the legend at the lower end of the drawing containing Fig. 13. In this legend the following abbreviations are used:

EXT.—Exhaust
H. P.—High pressure or the pressure of 600 lbs. per square inch
H. P. and EXH.—High pressure and exhaust or cylinder pressure
L. P.—Low pressure or the pressure of 100 lbs. per square inch
L. P. and EXH.—Low pressure and exhaust
L. P.—One-half of the low pressure or 50 lbs. per square inch

Automatic operation

The amplification of the hydraulic arrangement in the valve block is such that the hydraulic gear is responsive to the least force on the valve rod 154 or 155. Because of this sensitiveness the system may be operated directly from receivers of a Selsyn system. In the automatic operation of the system, the pilot valve is under the joint control of the two Selsyn receivers 27 and 28 of a Selsyn system. The receivers operate synchronously with the combined output of transmitters and transformers. The manner in which these receivers are connected to a Selsyn system including transmitters and transformers is illustrated in Fig. 17. In Fig. 17 a shaft 199 represents, when rotated, the necessary angular movement of the gun as a result of certain conditions. This shaft is directly connected to the rotor of a transmitter 200. The stator of the transmitter 200 is connected to the winding of the rotor of a transformer 201. The stator of the transformer 201 is connected to the stator of the Selsyn receiver 27. The rotor of the transformer 201 is rotated by a shaft 202 which represents the angular amount by which the rotation of the shaft 199 must be modified because of other conditions, and the transformer in effect adds or subtracts the rotational movement of the shaft 202 to or from the rotational movement of the shaft 199. The output of the receiver 27 is therefore the exact movement required of the gun in elevation.

The rotor of a transmitter 203 is connected to a shaft 204 which is geared to the shaft 199. The gearing arrangement including gears 205, 206 and 207 between the shafts 199 and 204 is such that the shaft 204 turns thirty-six revolutions for each revolution of the shaft 199. The stator of the transmitter 203 is connected to the rotor of a transformer 208, the shaft 209 of which is connected through gearing 210 to the shaft 202 which operates the rotor of the transformer 201. The ratio of the gears 210 is such that the shaft 209 rotates thirty-six revolutions for each revolution of the shaft 202. The stator of the transformer is connected to the stator of the receiver 28 so that the output of the stator 28 is related to the output of the receiver 27 as thirty-six is to one. The receiver 28 acts in the nature of a vernier to provide a fine adjustment of the pilot valve in accordance with the signal as represented by the receiver 27.

Rotational movement of the rotor of the receiver 28, relative to the movement of the gun, is translated into longitudinal movement of the pilot valve through a cam and lever arrangement. The receiver is secured to the plate 102 in the position shown in Figs. 9 and 10. On the shaft of the rotor of this receiver there is mounted an L-shaped bracket 211. One arm of the bracket 211*a*, is secured to the end of the shaft and extends perpendicular thereto, and the other arm 211*b* of the bracket extends parallel to the shaft, offset from the axis thereof, as best shown in Figures 14 and 15. The arm 211*b* is bifurcated and between the arms thereof there is pivotally mounted a lever 212 which carries on one end a ball cam follower 213 and at the other end a cam 214. Pivotally mounted on the same pin on which the lever 212 is mounted there are a pair of levers 215 and 216 which act conjointly. The free end of the levers 215 and 216 are secured together by a pin 217 and they act as a single lever. A lever 218 is pivotally mounted on the pin 217 at one end and at its opposite end carries a rotatably mounted cam follower 219 which engages the cam 214 on the lever 212. The cam 214 is in the nature of a constant rise cam. At its center it is provided with a notch 214*a* in which the cam follower 219 normally rests. The cam follower 219 is normally urged against the cam face 214 by a spring 220 which acts between a bar 221 extending across the levers 215 and 216 and a flange formed on the end of a branch arm 218*a* of the lever 218, the branch arm 218*a* extending at right angles to the arm 218. As this lever 212 is swung about its pivot, the engagement of the cam follower 219 and the notch of the cam 214 will cause the levers 215 and 216 to follow the movement of the lever 212. The valve rod 155 of the pilot valve is secured to the pin 217, and it is reciprocated in accordance with the movement of the levers 215 and 216.

The valve rod 155 extends through a hollow extension 156*c* (Fig. 13) extending from the end of the plunger 156 and through the cap 179. The valve rod also extends through the side of the box 100 in which the valve block is mounted and through a cylindrical mounting 222 (Fig. 15) secured to the side of the box 100 and upon which a cam 223 is rotatably mounted. The cam 223 is provided with a circular groove 224 which is eccentric to the axis of the cam 223 and the rotor shaft of the receiver 28 and which receives the ball cam follower 213 on the lever 212. The cam and a gear 225 to which it is secured are rotatably mounted on a circular flange 222*a* of the mounting 222, ball bearings 226 being interposed between the flange and the cam. If the cam 223 is held stationary and the shaft of the rotor of the receiver 28 is rotated, the arm 212 will be moved about its pivotal connection to the bracket 211 an amount proportional to the relative movement of the rotor shaft. This movement of the arm 212, through the links and levers previously described, will effect longitudinal movement of the pilot valve 153. A spring 227 surrounds the rod 155 and abuts against the pin 217 and against the end of the extension 156*c*. This spring is for the purpose of taking up any slack in the assembly.

The gear 225 is in mesh with a gear 228 (Fig. 10) mounted upon a shaft 229 journaled in bearings secured to the plate 102. On the end of the shaft 229, there is secured a gear 230 which meshes with a gear 231 mounted upon a stub shaft 232 (Fig. 2). The gear 231 is also in mesh with a gear 233 secured on the shaft of the B end of the hydraulic gear (see Fig. 16). The gear 225 is thus rotated in accordance with the rotational movement of the hydraulic gear. The gear train connecting the gear 225 to the shaft of the hydraulic gear is such that the gear 225, and, consequently, the cam 223, is rotated at a speed which is commensurate with a ratio of thirty-six to one to the gun. It is also rotated in the same direction as the rotor shaft of the receiver 28. Thus, it is the relative movement between the gun and the rotor shaft of the receiver 28 which effects movement of the lever 212 and consequently of the valve 153.

It will be noted that the cam groove 224 in the cam 223 is continuous and consequently the gun and signal may get out of positional agreement or synchronous operation. If, however, the cam and the rotor shaft of the receiver 28 are out of positional agreement by an angle of 90°, which represents a few minutes of angular difference between the gun and the signal, the receiver 27 takes over control of the pilot valve. This is accomplished through a solenoid 234 (Figs. 9 and 10) which is mounted upon an extension 156*d* extending from the end of the plunger 156 and through the cap 181. The solenoid acts upon the valve rod 154 to operate the valve. The solenoid has directional characteristics and is energized through stationary electrical contacts 235 and 236, and a movable contact 237 which is mounted between and cooperates with the stationary contacts 235 and 236 (Fig. 9). The stationary contacts 235 and 236 are mounted upon brackets extending from a plate 238 secured to the plate 102. The movable contact is brought into and out of engagement with the stationary contacts by the relative movement of the rotor of the receiver 27 with respect to the gun.

The movable contact is secured to a bracket 239 which is rotatably mounted upon the rotor shaft of the receiver 27 for relative movement with respect to the rotor shaft. The bracket carries a pivotally mounted section 239a on which there is rotatably mounted a cam follower 240. The cam follower 240 rides upon a cam 241 secured to the rotor shaft of the receiver. The cam follower is held against the cam surface by a spring 242. The details of the construction of this relief cam are illustrated and described in our copending application Serial No. 14,814, filed April 5, 1935, now Patent No. 2,134,488. As seen in Fig. 16, the cam surface has a low point at which the cam follower is normally located. From this low point the cam rises to a high point diametrically opposite to the low point. As the rotor of the receiver 27 moves, the cam 241 is rotated therewith. Since the spring 242 offers a resistance to the movement of the cam follower 240 relative to the cam 241, the bracket 239 will move with the rotor until the movable contact 237 is brought into engagement with either of the stationary contacts 235 or 236 depending upon the direction of rotation of the receiver rotor. If the cam rotates beyond this point, the cam follower 240 will remain stationary and ascend the rising cam surface. Upon the release of the pressure exerted by the continued movement of the cam 241, the cam follower will again assume its position at the low point of the cam.

The movement of the rotor of the receiver 27 is an expression of the relative movement between the signal and the gun. The receiver 27 is in effect an electrical differential. The stator is rotatably mounted in the bracket 238 and a bracket 243. A worm gear 244 is secured on to the stator of the receiver. This worm gear meshes with a worm 245 secured on a shaft 246 mounted and journaled in bearing brackets secured to the plate 102. The shaft 246 is connected to the shaft 229 through gears 247 and 248. Through the gears 247 and 248 the shaft 246 is rotated in accordance with the movement of the gun. The relation of the gear train from the gun to the stator of the receiver 27 is such that the stator of the receiver is rotated exactly the same angular amount as the gun and in the opposite direction to the signal as indicated by the rotation of the rotor. Thus it is that the rotor expresses the difference in angular movement between the gun and the signal.

When this difference is such, in the system shown, that the cam 223 is one quarter revolution out of phase with the shaft of the receiver 28, which represents a few minutes of angular movement between the signal and gun the movable contact engages one of the stationary contacts and energizes the solenoid. The solenoid acts upon the valve rod 154 and moves the valve 153 with great rapidity and provides a rapid movement of the tilting box to advance the gun. In order that the solenoid may ta'-e control of the pilot valve, the flexible connection of the cam 214 and the cam roller 219 permits the valve to move in response to the force exerted by the solenoid. This action is permitted by the cam roller 219 moving from the central groove 214a of the cam 214 and along the constantly rising surface of the cam. When the solenoid is deenergized by the breaking of the electrical contact, the cam follower rides back into the notch 214a and the valve rod is operated through the lever 212. The receiver 27 thus maintains the gun and the signal in approximate positional agreement and the receiver 28 provides a vernier or some sensitive adjustment.

If while the automatic control is operating, the limit stops come into operation, they will take control of the tilting box in the manner previously described. However, the signal will continue to operate and the cam follower 240 will ascend the constant rise of the cam 241 thus maintaining the solenoid energized. When the signal is reversed and the limit stops drop out of action, the pilot valve is in such relation, either under the control of the receiver 27 or the receiver 28, that the gun will be moved into positional agreement with the signal.

From the above description of the system illustrated in the drawings it will be apparent that there is provided a system which may be operated manually or automatically and when the automatic operation is functioning, the gun will be in positional agreement with the signal, the effect of the signal being advanced by the action of the needle valve 195 so that the gun has a velocity of movement commensurate with the change in signal but without a lag behind the signal.

It will be obvious that various changes may be made by those skilled in the art in the details of the embodiment illustrated in the drawings and described above within the principle and scope of the invention as expressed in the appended claims.

We claim:

1. In a mechanism for driving an object in accordance with the movement of another object, the combination comprising a movable object, a driven object, and means for controlling the movement of the driven object including a control unit comprising a differential connected in a one to one ratio with respect to the driven and movable objects to produce the difference in movement thereof, another differential connected in a higher ratio with respect to the movable and driven objects, a plunger operative upon a hydraulic medium, a control valve for effecting movement of the plunger, a cam and lever connection between the output of the second mentioned differential and the control valve for moving the control valve, additional means acting on the control valve and operative to effect movement thereof and control means therefor actuated by the first mentioned differential.

2. In a mechanism for driving an object in accordance with the movements of another object, the combination comprising a movable object, a driven object, variable-speed, power-driven operating means for the driven object, a control for the driving means comprising a movable element adapted upon movement to effect changes in speed of the operating means, control means for said movable element, actuating means operable in accordance with the difference in movement between the movable and driven objects, means actuated by said movable element, and means under the differential control of the actuating means and of the means actuated by the movable element for effecting the operation of said movable-element control means.

3. In a mechanism for driving an object in accordance with the movements of another object, the combination comprising a movable object, a driven object, variable-speed, power-driven, operating means for the driven object, a control for the operating means comprising a hydraulically operated movable element adapted upon movement to effect changes in the speed of the operating means, and means for controlling the movement of the movable element comprising a

hydraulically operated valve for controlling the application of a hydraulic medium to the hydraulically operated element, and actuating means for the valve comprising a closed volume hydraulic system formed in part by one end of the valve, a plunger operable to displace the hydraulic medium in the system in accordance with differences in movement between the movable and driven objects, and another plunger operable to displace the hydraulic medium in the system and actuated by movement of said movable element, the displacement of the hydraulic medium in the system being effective to actuate the valve.

4. In a control system for variable-speed, power-driven driving means having a movable control element adapted upon movement to effect changes in the speed of the driving means, the combination comprising means for moving said control element, control means for said control element moving means, actuating means for effecting the actuation of the control means, and a hydraulic differential interconnecting the control element of the driving means, the actuating means and said control means.

5. In a control system for variable-speed, power-driven driving means having a movable control element adapted upon movement to effect changes in the speed of the driving means, the combination comprising means for moving said control element, control means for said control element moving means, actuating means for effecting the actuation of the control means, and a hydraulic differential interconnecting the control element of the driving means, the actuating means and said control means including a closed, variable-volume hydraulic system containing a hydraulic medium under substantially constant pressure, means for varying the volume of the closed system in accordance with the movements of the actuating means, and additional means for varying the volume of the closed system in accordance with movements of the control element of the driving means.

6. In a control system for variable-speed, power-driven driving means having a movable control element adapted upon movement to effect changes in the speed of the driving means, the combination comprising means for moving said control element, control means for said control element moving means, actuating means for effecting the actuation of the control means, a hydraulic differential interconnecting the control element of the driving means, the actuating means and said control means including a closed, variable-volume hydraulic system containing a hydraulic medium under substantially constant pressure, means for varying the volume of the closed system in accordance with the movements of the actuating means, and additional means for varying the volume of the closed system in accordance with movements of the control element of the driving means, and means for augmenting the effect of movement of the actuating means.

7. A control for a variable-speed, power-driven driving means, comprising a movable element adapted upon movement to effect changes in the speed of the driving means, manually operative means directly connected for effecting movement of said movable element, a motion receiving element, power amplifying means responsive to said motion receiving element and adapted to operate the movable element, means for selectively rendering said manually operative means and said motion receiving element effective for effecting movement of the movable element, and means controlled by the movement of the driving means and operative upon a predetermined movement of the driving means in either direction to incapacitate the driving means for further movement in the same direction.

8. A control for a variable-speed, power-driven driving means comprising a movable element adapted upon movement to effect changes in the speed of the driving means, manually operative means for effecting movement of said movable element, two motion receiving elements, power amplifying means responsive to either of said motion receiving elements and adapted to operate the movable element, means for selectively connecting said manually operative means and said motion receiving elements to effect movement of the movable element, and means controlled by the movement of the driving means and operative upon a predetermined movement of the driving means in either direction to incapacitate the driving means for further movement in the same direction.

9. A control for a variable-speed, power-driven driving means comprising a movable element adapted upon movement to effect changes in the speed of the driving means, manually operative means for effecting movement of said movable element, two motion receiving elements, power amplifying means responsive to either of said motion receiving elements and adapted to operate the movable element, means for selectively connecting said manually operative means and said motion receiving elements to effect movement of the movable element, means for rendering one of said motion receiving elements ineffective upon the operation of the other, and means controlled by the movement of the driving means and operative upon a predetermined movement of the driving means in either direction to incapacitate the driving means for further movement in the same direction.

10. In a mechanism for driving an object in accordance with the movements of another object, the combination comprising a movable object, a driven object, power-driven operating means for the driven object, a variable velocity transmission connecting the power-driven operating means and the driven object, means for regulating the variable velocity transmission including a member differentially connected to the movable and driven objects, and means controlled by the movement of the driving means and operative upon a predetermined movement of the driving means in either direction to incapacitate the driving means for further movement in the same direction.

11. In a mechanism for driving an object in accordance with the movements of another object, the combination comprising a movable object, a driven object, power-driven operating means for the driven object, a variable velocity transmission connecting the power-driven operating means and the driven object, means for regulating the variable velocity transmission including a member differentially connected to the movable and driven objects, manually operative means for regulating the variable velocity transmission, means for selectively connecting said differentially operated member and said manually operative means for regulating the variable velocity transmission, and means controlled by the movement of the driving means and operative upon a predetermined movement of the driving means in either direction to inca-

pacitate the driving means for further movement in the same direction.

12. In a control system for variable-speed, power-driven driving means, the combination comprising a piston valve for controlling said power-driven driving means, an actuating plunger, a hydraulic linkage interconnecting the plunger and the piston valve including a closed hydraulic system containing a hydraulic medium under a predetermined pressure, means for maintaining said predetermined pressure, and means for applying a balancing force on said piston valve in opposition to said predetermined pressure, whereby movement of the plunger displaces the hydraulic medium in the closed system and actuates the piston valve.

13. In a control system for variable-speed, power-driven driving means having a movable control element adapted upon movement to effect changes in speed of the driving means, the combination of a control plunger, a control piston valve and a response plunger, a closed hydraulic system the chamber of which includes one end of each of the said plungers and piston valve, means for maintaining a normal pressure in the closed system, means for exerting a balancing force on the opposite end of the control piston valve equal to the force on the piston valve of said normal pressure, means operatively connecting the control piston valve and the movable control element, means operatively connecting the movable control element with the response plunger, and means for operating the control plunger to cause a displacement of the medium in the closed system, whereby a movement of the control plunger disturbs the balance on the piston valve and the resultant movement of the response plunger restores said balance.

14. In a mechanism for driving an object in accordance with the movement of another object, the combination of a movable object, a driven object, variable-speed driving means for the driven object having a movable control element, a control plunger, a piston valve and a response plunger, a closed hydraulic system the chamber of which includes one end of each of said plungers and piston valve, means for maintaining a normal pressure in the closed system, means for producing balancing hydraulic pressures on the opposite ends of both the control plunger and piston valve, hydraulic means under the control of the piston valve for actuating the control element, means operatively connecting the control element with the response plunger, a pilot valve responsive to the relative movement of the movable and driven objects operative to alter the pressure on the said opposite end of the control plunger and thereby cause the plunger to move and displace the medium in the closed system.

15. In a mechanism for driving an object in accordance with the movements of another object, the combination comprising a variable speed operating means for the driven object, a control for the operating means comprising a movable element adapted to effect changes in the speed of the operating means, and means for controlling the movable element comprising a valve controlling the application of a medium to the element and actuating means for the valve comprising a closed volume hydraulic system formed in part by one end of the valve, a plunger operable to displace the hydraulic medium in the system, and another plunger actuated in the system by movement of said movable element.

16. In hydraulic follow-up control means for a controlled device, a first chamber of variable volumetric capacity, means for altering the volumetric capacity of said first chamber, a second chamber of variable volumetric capacity, means active to alter the volumetric capacity of said second chamber conformably with and responsive to operation of the controlled device, a third chamber of variable volumetric capacity, the volumetric capacity of said third chamber being altered upon change in volumetric capacity of said first and second chambers, a control element arranged to move responsive to change in volumetric capacity of said third chamber and active to regulate the operation of said controlled device, means connecting said chambers with one another to form a closed hydraulic circuit and means arranged to maintain a substantially constant predetermined pressure in said circuit during the time that the volumetric capacities of all of said chambers remain constant, departure from said predetermined pressure in said circuit being accompanied by change in the volumetric capacity of said third chamber and said control element being responsively moved in a direction to cause operation of the controlled device in a direction and to an extent to effect corrective change in the volumetric capacity of said second chamber to thereby restore the predetermined pressure in said circuit.

17. In control means for a power-operated device, a closed hydraulic control circuit comprising at least two chambers of variable volumetric capacity, means establishing a predetermined pressure value to exist in all parts of said circuit including both of said chambers when the capacities of all chambers thereof remain constant, an element movable to control the operation of said power-operated device, said element having a neutral position in which said power-operated device is rendered inoperative and being movable from said neutral position responsive to pressure in said circuit above said predetermined value to cause operation of said power-operated device in one direction and movable responsive to pressure in said circuit below said predetermined value to cause operation of said power-operated device in the opposite direction, means for altering the pressure in said circuit relative to said predetermined pressure value and means responsive to operation of said power-operated device and active to correctively alter the capacity of one of said chambers to restore said predetermined pressure value in said circuit.

18. In control means for an hydraulic-power-operated device, a closed hydraulic control circuit comprising at least two chambers of variable volumetric capacity, means establishing a predetermined pressure value to exist in all parts of said circuit including both of said chambers during the time that the capacities of all chambers thereof remain constant, a valve element movable to control the flow of operating pressure fluid to and the exhaust of fluid from said device to thereby control its operation, said valve element having a neutral position in which said device is rendered inoperative and being movable from said neutral position responsive to pressure in said circuit above said predetermined value to cause operation of said device in one direction and movable responsive to pressure in said circuit below said predetermined value to cause operation of said device in the opposite direction, means for altering the pressure in said circuit relative to said predetermined pressure value and

means responsive to operation of said device and active to correctively alter the capacity of one of said chambers to restore said predetermined pressure value in said circuit.

19. In power operated means for regulating the output of a variable capacity pump, said pump having an element movable in one direction to increase its capacity and movable in the opposite direction to decrease its capacity and a fluid pressure operated motor for moving said element, in combination, a first chamber of variable volumetric capacity, means for altering the volumetric capacity of said first chamber, a second chamber of variable volumetric capacity, means active to alter the volumetric capacity of said second chamber conformably with and responsive to operation of said fluid pressure operated motor, a third chamber of variable volumetric capacity, the volumetric capacity of said third chamber being altered upon change in volumetric capacity of said first and second chambers, control valve means arranged to move responsive to change in volumetric capacity of said third chamber and active to regulate the operation of said fluid pressure operated motor, means connecting said chambers with one another to form a closed hydraulic circuit and spring means arranged to maintain a substantially constant predetermined pressure in said circuit during the time that the volumetric capacities of all of said chambers remains constant, departure from said predetermined pressure in said circuit being accompanied by change in the volumetric capacity of said third chamber and said control valve means being responsively moved in a direction to cause operation of the fluid pressure operated motor in a direction and to an extent to effect corrective change in the volumetric capacity of said second chamber to thereby restore the predetermined pressure in said circuit.

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