Aug. 26, 1941. C. HERFURTH ET AL 2,254,010

GRINDING MACHINE

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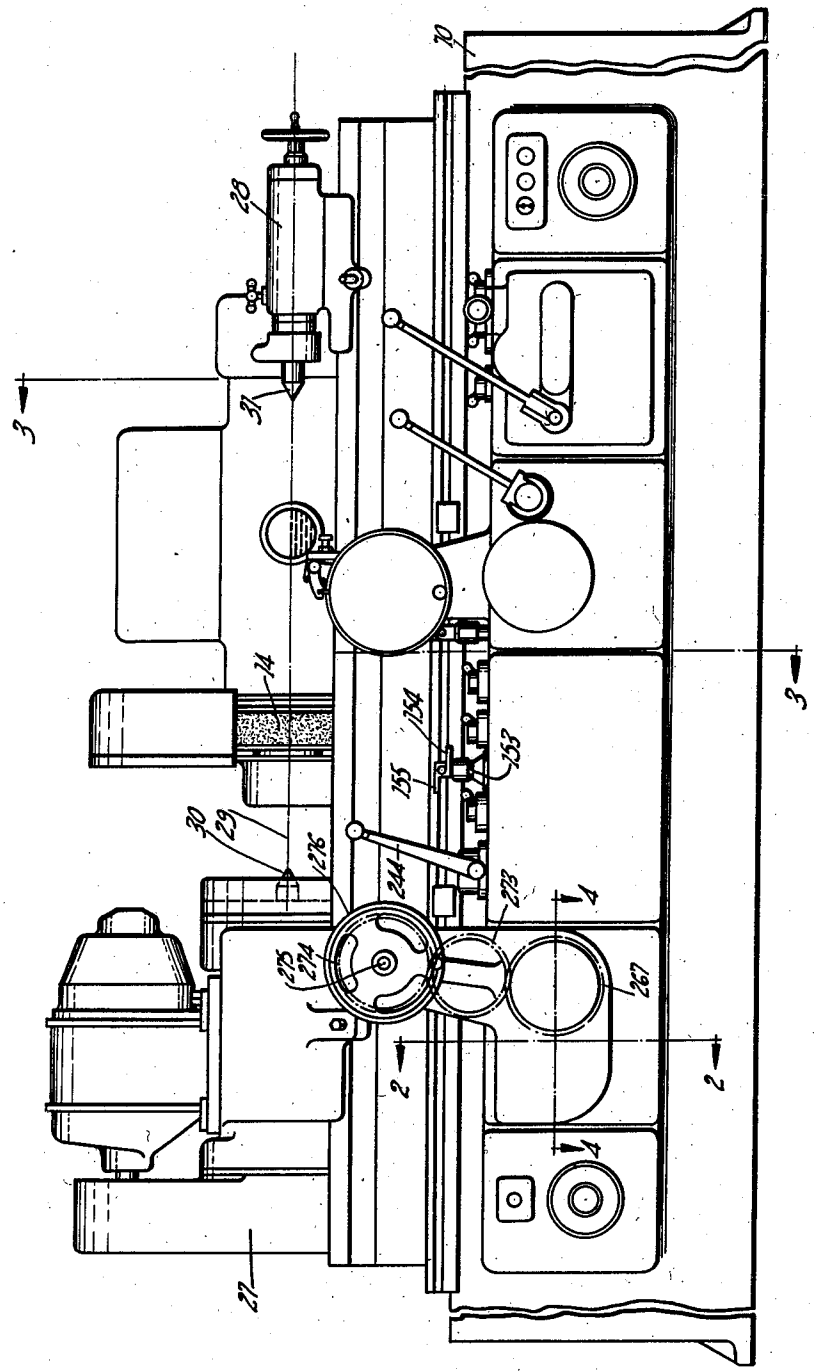


Fig. 1

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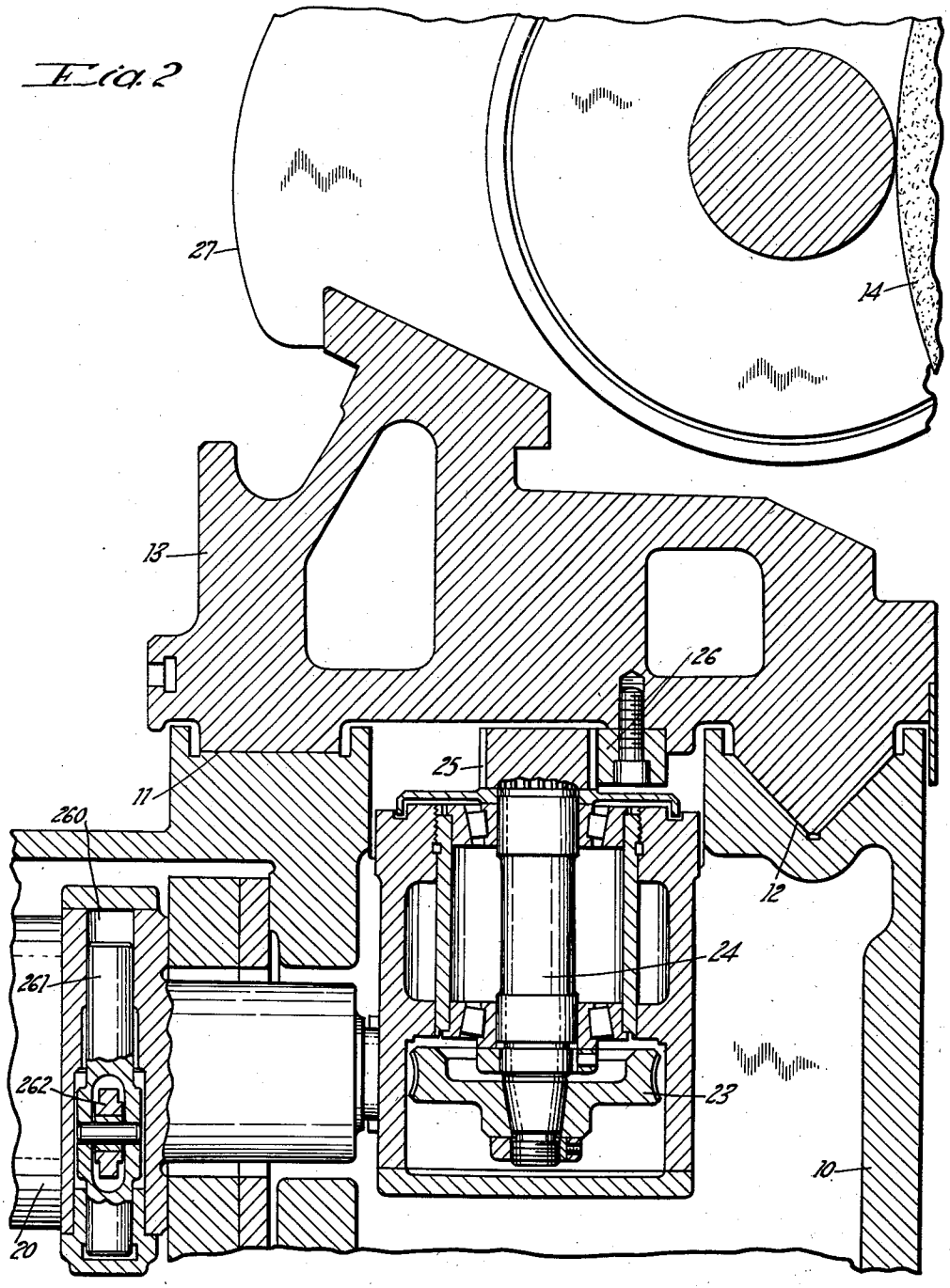
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GRINDING MACHINE

Filed April 25, 1938 9 Sheets-Sheet 2

FIG. 2



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Filed April 25, 1938 9 Sheets-Sheet 3

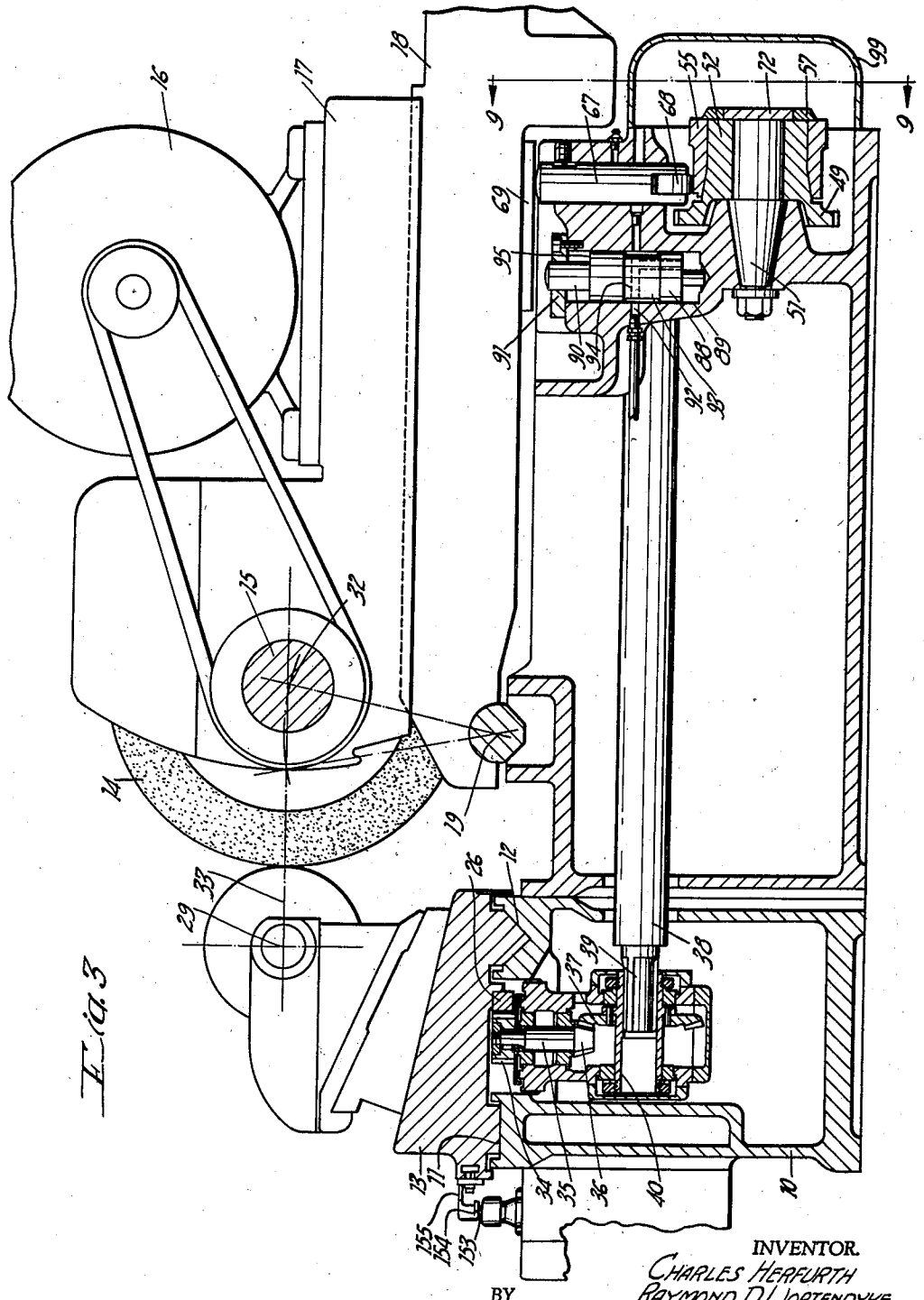


Fig. 3

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GRINDING MACHINE

Filed April 25, 1938 9 Sheets-Sheet 4

Fig. 6

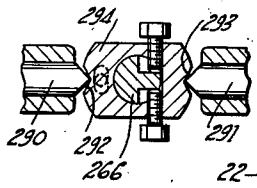


Fig. 8

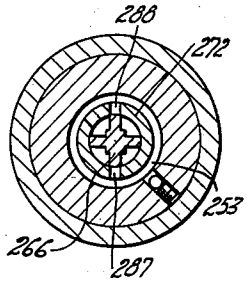


Fig. 4

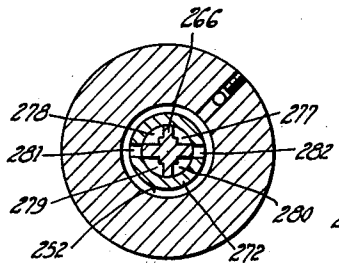


Fig. 5

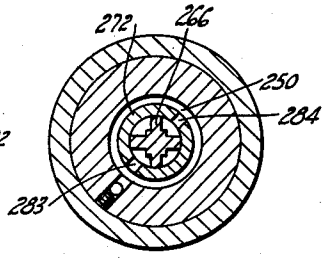


Fig. 6

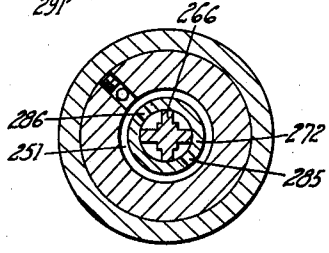


Fig. 7

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GRINDING MACHINE

Filed April 25, 1938 9 Sheets-Sheet 5

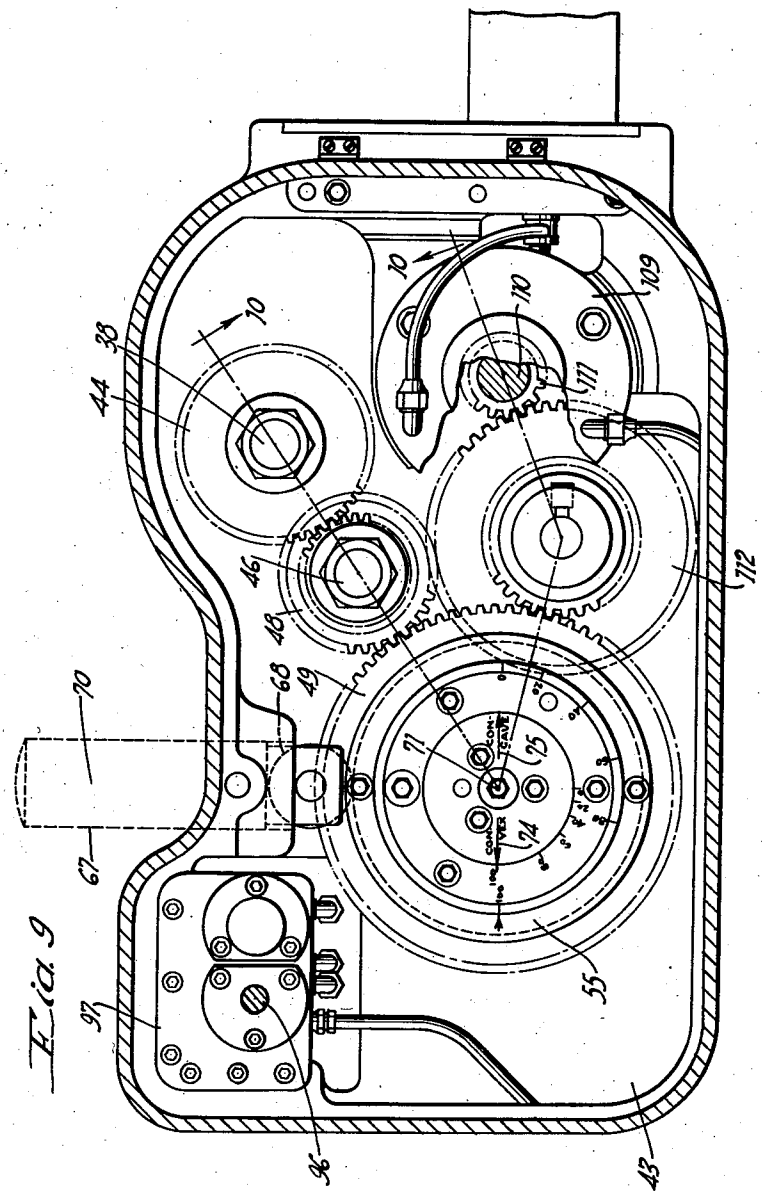


Fig. 9

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Aug. 26, 1941. C. HERFURTH ET AL 2,254,010

GRINDING MACHINE

Filed April 25, 1938 9 Sheets-Sheet 6

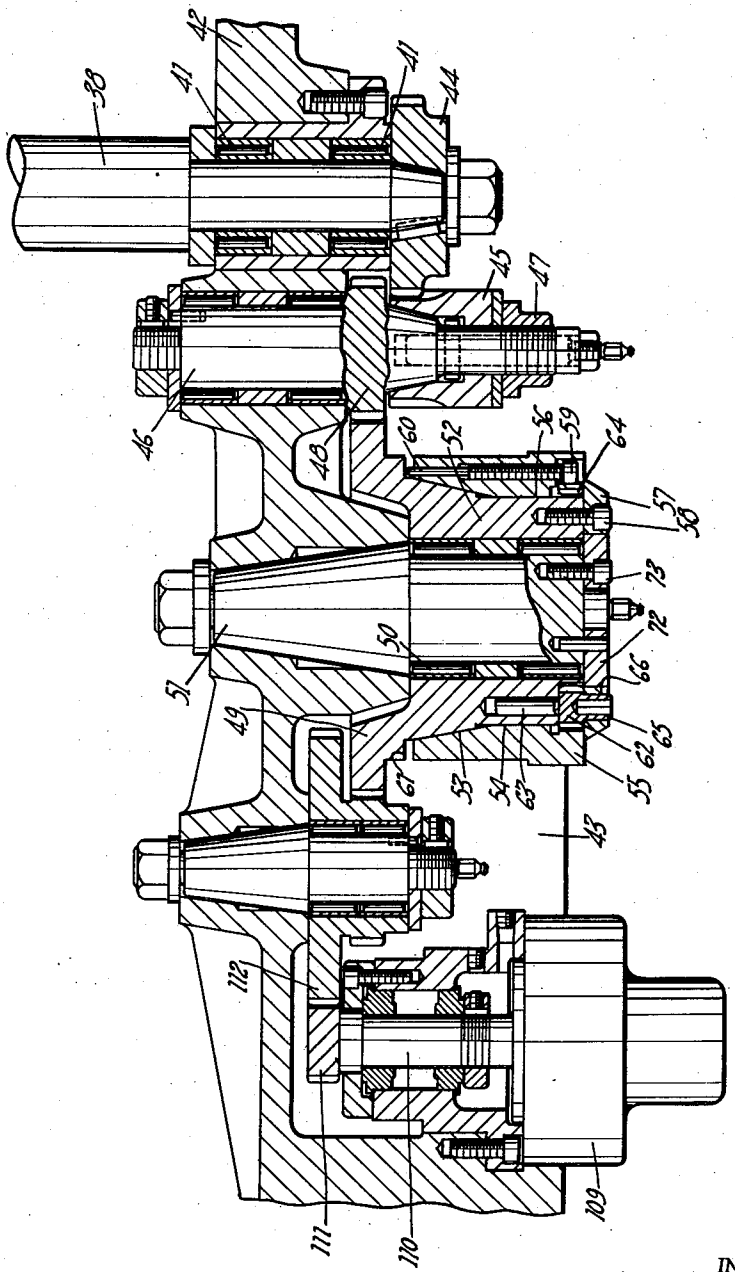


Fig. 10

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Aug. 26, 1941. C. HERFURTH ET AL 2,254,010

GRINDING MACHINE

Filed April 25, 1938 9 Sheets-Sheet 7

Fig. 11

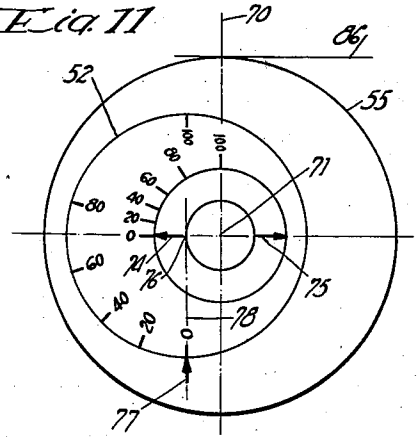


Fig. 11a

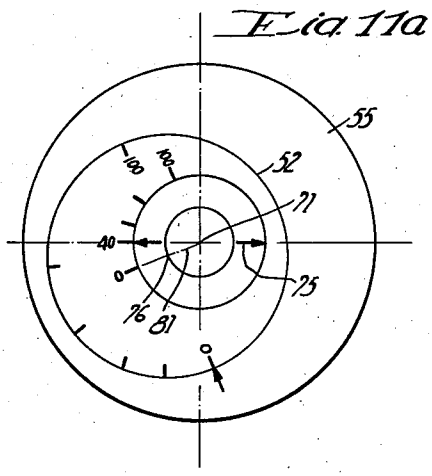


Fig. 12

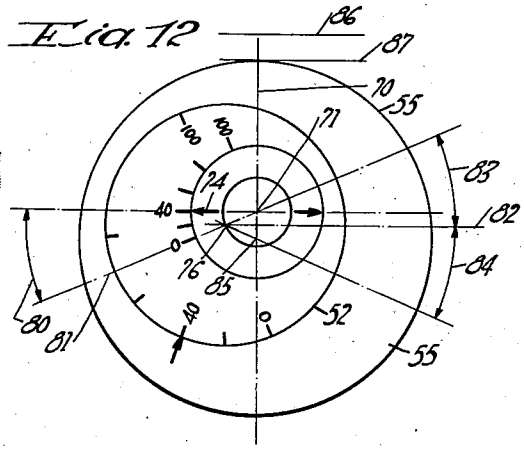


Fig. 13

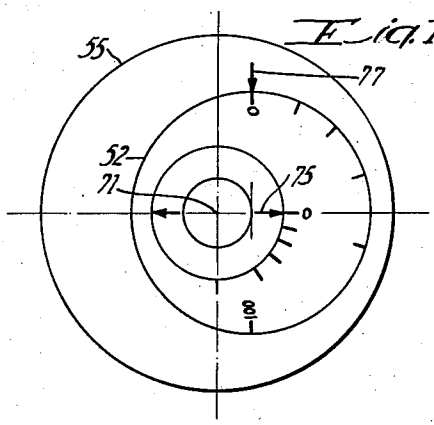
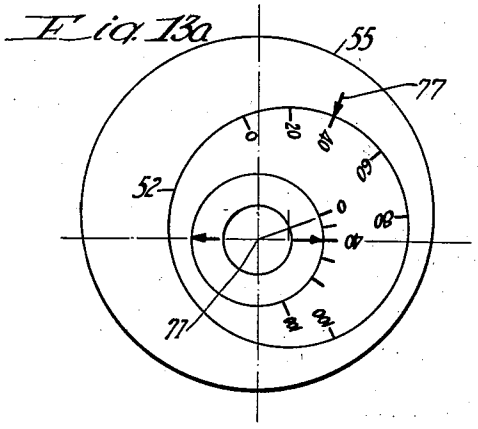


Fig. 13a



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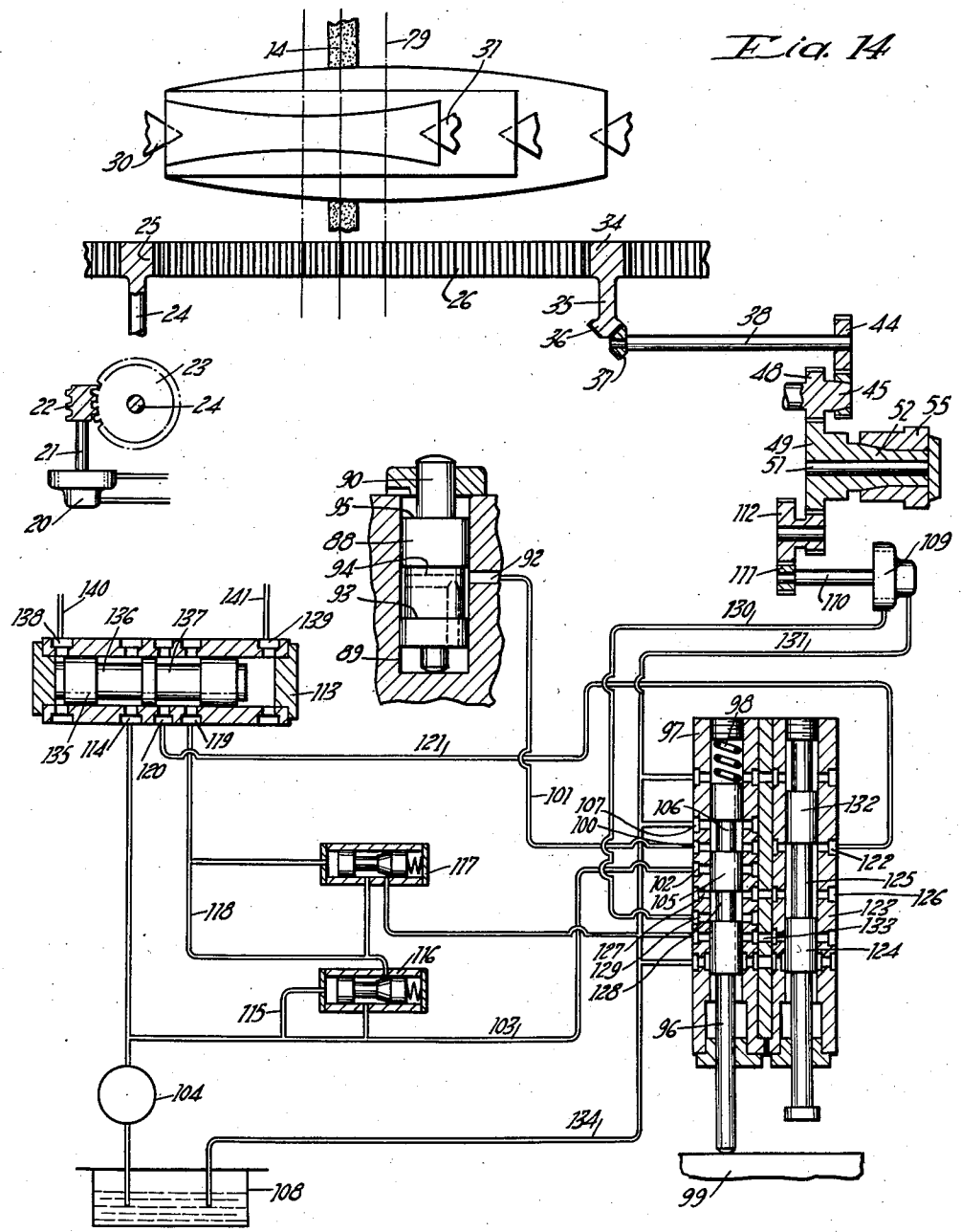
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GRINDING MACHINE

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Fig. 14



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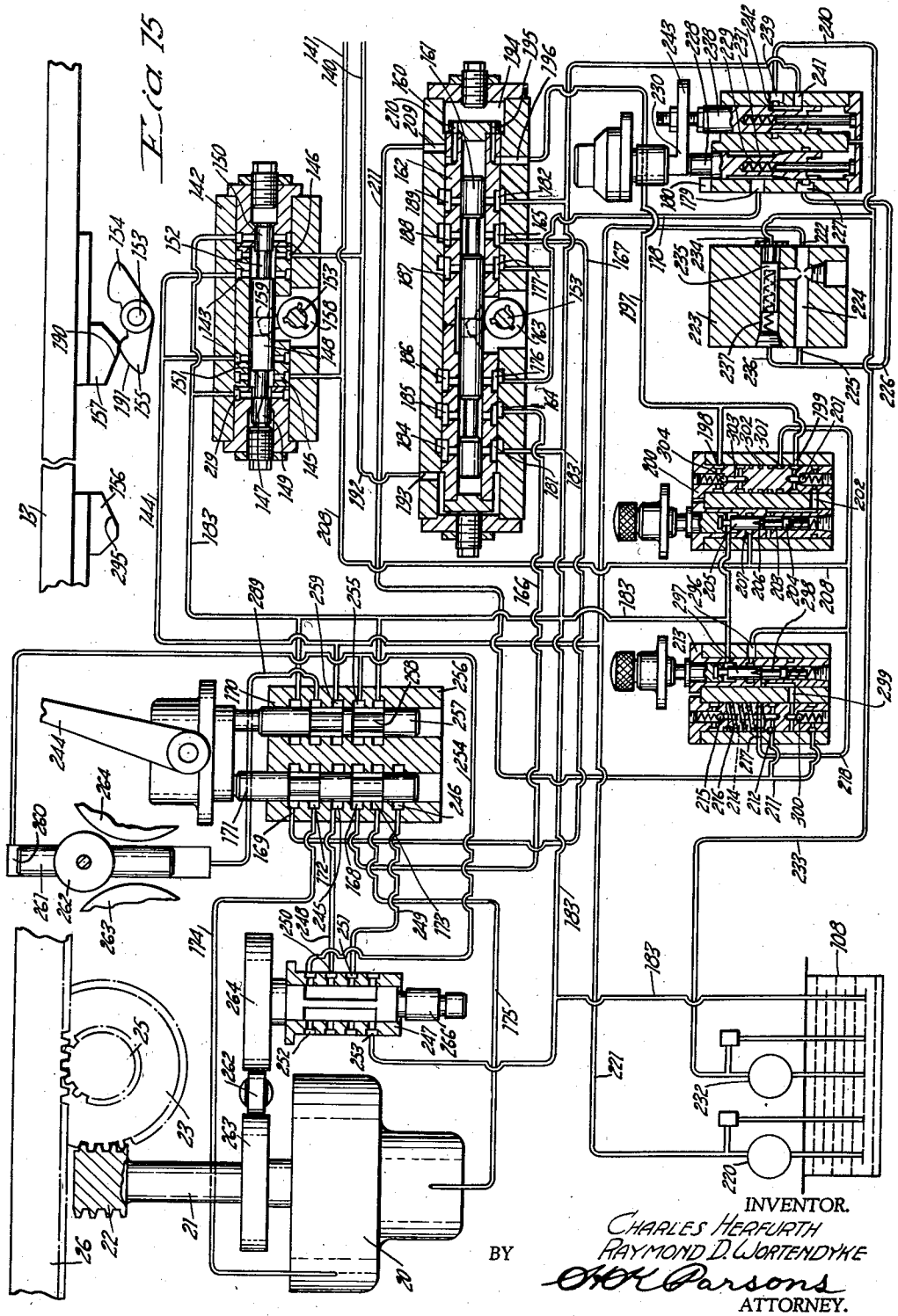
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GRINDING MACHINE

Filed April 25, 1938 9 Sheets-Sheet 9

FIG. 15



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Patented Aug. 26, 1941 2,254,010

UNITED STATES PATENT OFFICE

2,254,010

GRINDING MACHINE

Charles Herfurth and Raymond D. Wortendyke, Cincinnati, Ohio, assignors to Cincinnati Grinders Incorporated, Cincinnati, Ohio, a corporation of Ohio

Application April 25, 1938, Serial No. 204,083

14 Claims. (Cl. 51—33)

This invention relates to grinding machines and more particularly to improvements in the transmission and control mechanism therefor.

One of the objects of this invention is to provide a transmission mechanism for a grinding machine which will execute an improved traverse grinding cycle whereby sudden changes in the rate of traversing movement will be eliminated so that a more uniform grinding operation may be obtained.

Another object of this invention is to provide a control mechanism for the traversing movement between a work piece and a grinding wheel, which has versatility as respects both adjustment and control, whereby variations in rate of any portion of the automatic cycle may be readily made, and selection between different types of cycles may be easily obtained.

A further object of this invention is to provide improved means for synchronizing the movement of a work support and a camber control mechanism.

An additional object of this invention is to provide in a hydraulic transmission, improved and simplified control valve mechanism which may be adjusted and set to yield variable deceleration, variable tarry and variable acceleration at each reversal of the traversing movement.

Other objects and advantages of the present invention should be readily apparent by reference to the following specification considered in conjunction with the accompanying drawings illustrative of one embodiment thereof, but it will be understood that any modifications may be made in the specific structural details within the scope of the appended claims without departing from or exceeding the spirit of the invention.

Referring to the drawings in which like reference numerals indicate like or similar parts:

Figure 1 is a front elevation of a machine tool embodying the principles of this invention.

Figure 2 is a section on the line 2—2 of Figure 1 showing the gear train between the table motor and the table.

Figure 3 is a vertical section through the machine taken on the line 3—3 of Figure 1 showing part of the drive from the table to the cambering mechanism.

Figure 4 is a horizontal section taken on the line 4—4 of Figure 1 showing the servo-motor control mechanism for the table motor.

Figure 5 is a section on the line 5—5 of Figure 4.

Figure 6 is a section on the line 6—6 of Figure 4.

Figure 7 is a section on the line 7—7 of Figure 4.

Figure 8 is a section on the line 8—8 of Figure 4.

Figure 9 is a rear elevational view of the cambering attachment gear box as viewed on the line 9—9 of Figure 3.

Figure 10 is an expanded view of the gearing shown in Figure 9 and as viewed on the line 10—10 of that figure.

Figures 11, 11A, 12, 13 and 13A are views showing different settings of the camber control cam for obtaining different degrees of camber.

Figure 14 is a diagrammatic view of the hydraulic control for obtaining synchronism between the movement of the table and the movement of the camber control cam.

Figure 15 is a diagrammatic view of the hydraulic traverse transmission and control mechanism for the table.

Figure 16 is a detail section on the line 16—16 of Figure 4.

A machine embodying the principles of this invention is shown in Figure 1 of the drawings, and in general comprises a bed 10 upon the top of which are formed guideways 11 and 12 as more particularly shown in Figure 2. A work supporting table 13 is mounted on these guideways for traversing movement relative to a grinding wheel 14.

As shown in Figure 3, the grinding wheel 14 may be supported on a spindle 15 for rotation by a motor 16. The grinding wheel and motor are mounted on a subslide 17 which is movable relative to a support 18, which in turn is pivoted for movement about an axis 19 located under the grinding wheel.

Suitable mechanism may be employed for effecting movement of the slide 17, but since this mechanism forms no part of the present invention, it has not been illustrated.

Referring to Figures 2 and 4, the mechanism for traversing the table comprises a rotary hydraulic motor 20 which is operatively connected to a drive shaft 21 for rotating a worm 22 antifrictionally supported in the bed. This worm meshes with a worm wheel 23 secured to the lower end of shaft 24 which is also antifrictionally supported in the bed. The upper end of this shaft has an integral pinion 25 meshing with a rack bar 26 secured to the under side of the table between the guideways 11 and 12. It will now be apparent that upon rotation of the hydraulic motor in one direction or the other that the table may be traversed relative to the grinding wheel.

The table 13 has a headstock 27, and a tailstock 28, mounted thereon for adjustment to accommodate different lengths of work pieces. The

axis 29 which connects the points of the work supporting centers 30 and 31 may be defined as the rotational axis of the work. This machine is especially suitable for grinding rolls and is provided with an adjustable camber control mechanism which operates to oscillate the support 18 about the axis 19. Since this axis is under the grinding wheel, the lever arm which connects this axis to the center 32 of the grinding wheel moves practically at right angles to a line 33 which connects the axis of rotation 29 to the center of the grinding wheel so that for small movements of the grinding wheel sufficient to produce the necessary camber on the work, it may be said that the grinding wheel moves toward and from the work along substantially a radial line thereof.

The mechanism for oscillating the support 18 to produce the cambering action consists of a camber control mechanism which is driven from the table rack 26 through a synchronized transmission which maintains the camber control mechanism in exact step with the movement of the table. The adjustable camber control mechanism to be described does not constitute part of the present invention, but is described and claimed in a separate copending application, Serial Number 204,082, filed April 25, 1938. The rack 26 engages a pinion 34 which is keyed to the upper end of the shaft 35. This shaft is connected through a bevel pinion 36 and a bevel gear 37 to a longitudinal shaft 38. This shaft has a splined end 39 fitting in a splined bore 40 integral with the bevel gear 37 so as to facilitate assembly of the parts. Referring to Figure 10, the shaft 38 is antifrictionally supported by bearings 41 in the wall 42 of the cambering mechanism gear box indicated generally by the reference numeral 43. The shaft 38 has a spur gear 44 keyed to the end thereof and intermeshing with a gear 45 keyed to a stub shaft 46. The gear 45 is held in position on the end of shaft 46 by a nut 47 threaded on the end of shaft 46. The gears 44 and 45 are removable and gears of different sizes may be substituted therefor whereby they constitute change gears for varying the rate of actuation of the remainder of the gear train.

The shaft 46 also has a gear 48 which intermeshes with a large gear 49. The gear 49 is supported by antifriction bearings 50 on a fixed shaft 51. The gear 49 has an elongated eccentric hub 52 having a tapered portion 53 and a straight portion 54. A circular cam member 55, having an eccentric bore 56, is mounted on the hub of the gear 49. This eccentric bore has a straight portion and a tapered portion whereby the cam may be forced on the tapered portion and frictionally held for rotation with the gear.

The cam is clamped to the member 49 by means of a clamping ring 57 which overlaps the end of the cam and is secured to the gear 49 by bolts 58. When these bolts are tightened, the cam is forced upon the tapered portion 53. When adjustment is to be made, the clamping bolts 58 are loosened, and the cam is forced off of the tapered portion by means of a pair of screws 59 which are threaded in the cam and have reduced ends 60 which engage the face 61 of the gear 49 whereby the reaction on the screw threads will force the cam off of the taper 53. When the cam has been loosened in this manner, a small pinion 62 which is journaled in the gear member 49 by means of an integral stud 63, is rotated, and through interengagement with internal gear teeth 64 carried by the cam member, the same may be rotated relative to the gear 49.

It will be noted that the pinion 62 has a reduced portion 65 which passes through a hole 66 formed in the clamping ring 57, and since the pinion 62 is fixed with the gear 49, the clamping ring 57 will also be held with the gear, whereby the cam 55 may be rotated relative to both of these parts.

As shown in Figures 3 and 9, a cam follower 67 is supported for vertical reciprocation in the bed 10, and one end of this follower is provided with a roller 68 which engages the periphery of the cam 55, and the other end engages a wear plate 69 mounted on the under side of the pivoted support 18. It will be noted from Figure 9 that the vertical axis 70 defines a vertical plane in which the follower reciprocates, and that the axis passes through the axis of rotation of the gear 49 and cam 55. Since the peripheral surface of the cam is a true circle, it should be evident that if the eccentricities of the hub 52, and the bore 56 are equal, and positioned opposite to one another, that the eccentricities will cancel, whereby the true center of the circular surface of the cam may be made to lie on the center of rotation 71, whereby no movement of the follower will result upon rotation of the cam.

In order that the operator may know the relative positions of the eccentrics for set-up purposes, a plate 72 is attached to the end of the fixed shaft 51 by means of screws 73, and this plate is provided with two diametrically opposite index marks 74 and 75 that lie in a plane at right angles to the vertical axis 70. Referring to Figure 11 in which the eccentricities are exaggerated for explanatory purposes, the point 76 indicates the true center of the hub 52, and therefore the distance between the axis of rotation 71 and the point 76 represents the effective radius or eccentricity. When the cam 55 is rotated relative to the hub 52, the point 76 becomes the axis of rotation about which the cam 55 moves. When the cam is in the position shown in Figure 11, the true center of the cam lies on the axis of rotation 71 because its eccentricity, which may be measured to the right of the point 76, is equal to and cancels the eccentricity of the hub 52, which is measured to the left of the axis of rotation 71. The result is that the true center of the cam lies on the axis of rotation, and the resultant eccentricity is equal to zero. Therefore with the parts in a zero eccentric position, the end face of the hub 52 is graduated, beginning with a "0" mark opposite the arrow 74, and other marks extending throughout the 90° arc clockwise to the axis 70.

Since the cam 55 moves about the center 76 when it is adjusted relative to the hub 52, it is provided with an arrow 77 which is located in the plane of the radius 78 passing through the center 76 and parallel to the vertical axis 70. The end face of the hub 52 is graduated from this point through an arc of 180° in a clockwise direction, the graduations being equal in number to the first named graduations but having twice the angular extent.

The graduations shown indicate percentages of the maximum camber which can be obtained. This maximum camber will be equal to twice the radius 71—76.

To set the machine up for obtaining a camber on the work, the table is traversed until the longitudinal center of the work as represented by the axis 79, Figure 14, is opposite the center of the grinding wheel 14, and then the gear 45 is withdrawn from mesh with the gear 44, so

that the gear 49 may be rotated in a counterclockwise direction, as viewed in Figure 11, until the desired camber percentage is opposite the arrow 74.

The gear 45 is then replaced in mesh with the gear 44, and the cam 55 loosened and rotated relative to the gear 49 until the arrow 77 is opposite a graduation mark having a value equal to the value of the graduation mark opposite the arrow 74. The effect of this is to shift the true center of the cam circle 55 along the axis 70, and, in this case, downward or below the center 71 which will thereby cause the grinding wheel support 18 to move clockwise as viewed in Figure 3, thus determining the maximum diameter or crown of the work at its longitudinal center. It will now be evident that after the cam 55 is reclamped to the gear 49, rotation of the cam in either direction will cause the effective radius to increase. This will cause counterclockwise movement of the support 18, whereby the grinding wheel will move toward the work and thus reduce its diameter.

In Figure 12, the parts are shown in a position to yield a camber which is 40% of the maximum camber obtainable. To obtain this set-up, the gear and cam are rotated about the axis 71 in a counterclockwise direction through an angle indicated by the reference numeral 80 until the graduation mark 40 is opposite the arrow 74. The parts will then be in the position indicated in Figure 11A. It will be noted that the true center of the cam 55 still remains in juxtaposition to the center 71. The center 76 of the eccentric hub 52 has moved and lies on the radial line 81. The cam 55 is now rotated about the center 76 and through an angle equal to twice the angle 80 so that the true center of cam 55 will lie on the vertical line 70.

It is desirable that the maximum offset or eccentricity be measurable along the line 70, because this position of the cam corresponds to the central position of the work with respect to the grinding wheel and whatever maximum displacement the cam can effect should be effected while the work is central of the grinding wheel. If a radial line 82 is drawn from the center 76 parallel to the diameter through the arrows 74—75, the angle 83 between this line and the prolongation of radial line 81, will be equal to the angle 80. If the angle 84 is made equal to the angle 83, the true center 85 of the circle 55 will lie on the axis 70 and the eccentricity is then the distance between the line 86, which was tangent to the circle 55 in Figure 11 and the new tangent 87 shown in Figure 12, this distance being measured along the axis 70.

The above set-up was for obtaining convex cambers on the work. To set up for concave members, the parts shown in Figure 11 are rotated through 180° so that the inner "0" mark on the hub 52 is opposite the arrow 75, as shown in Figure 13. The gear and cam will then rotate together in a counterclockwise direction until the proper graduation mark is opposite the arrow 75, and then the cam is rotated relative to the gear until the arrow 77 is opposite the same graduation mark on the outer periphery of the hub 52. An example of such a set-up is shown in Figure 13A.

Since the weight of the grinding wheel support 18 rests on the follower pin 67, it is of course desirable that this weight be removed in order to facilitate adjustment of the cam during set-up. Power operated means have been provided for lifting the support 18 a sufficient distance that no weight will be on the pin 67 during adjustment. This mechanism comprises a piston 88, Figures 3 and 14, which is reciprocably mounted in a cylinder 89 formed in the bed 10. The piston has a reduced end 90 which passes through the cylinder head 91 for engagement with the plate 69. Hydraulic pressure is admitted to port 92 which is in constant communication with an annular groove 93 formed on the periphery of the piston. An interdrilled passage 94 connects this groove to the bottom of the cylinder whereby the fluid pressure will act on the under side of the piston to raise the same. The shoulder 95 on the upper end of the piston limits its outward movement and determines the amount of lift that will be imparted to the support 18.

The admission of fluid pressure to the cylinder 89 is controlled automatically by a valve plunger 96 which is reciprocably mounted in a valve housing 97. A spring 98 is interposed in the end of the valve housing for continuously urging the valve plunger in one direction. The valve housing is contained within the gear box 43 and this gear box is provided with a removable cover 99. When the cover is closed, it engages the end of the valve plunger 96 and moves the same inwardly into the position shown in Figure 14. The valve housing has a port 100 which is connected by a channel 101 to the port 92 of the cylinder. It is also provided with a port 102 to which fluid is supplied through a channel 103 from a suitable source of pressure, such as a supply pump 104. When the valve plunger is in the position shown, a spool 105 closes the port 102 thereby disconnecting it from the port 100. When the door 99 of the gear box is opened, the spring 98 moves the plunger outward, and an annular groove 106 formed in the plunger interconnects the port 102 with port 100 whereby the fluid pressure flows to the cylinder 89 and causes outward movement of the piston 88. When the door is closed, the annular groove 106 interconnects port 100 with an exhaust port 107 whereby the fluid in the cylinder is exhausted to the reservoir 108.

As previously mentioned, it is desirable that the camber control cam rotate in exact synchronism with the table movement and this means that there should be no lost motion in the gearing between the table and the cam because otherwise the maximum point of camber would shift for each direction of movement of the table. Means have therefore been provided for taking the backlash or lost motion out of the drive connection between the table and the cam, and this means or mechanism acts in such a manner that the backlash is always taken out in one direction only and regardless of the direction of movement of the table. This is necessary in order to prevent shifting of the maximum point of camber relative to the work.

The mechanism for accomplishing this comprises an hydraulic motor 109, which is mounted in the gear box, as shown in Figure 10. The drive shaft 110 of this motor is provided with a spur gear 111, which is connected by the double gear 112 to the gear 49. It will now be apparent that when the shaft 38 is rotated to cause actuation of the gear 49, that the gear 49 in turn will drive or tend to drive the gear 111 on the end of the motor shaft 110. In order that the backlash may always be taken up in one direction regardless of the direction of movement of the table, or, in

other words, the direction of rotation of the shaft 38, means have been provided whereby the pressure in the motor 109 may be varied so that during rotation of the shaft 38 in one direction, the motor will act as a retarding or braking means, and during rotation of the shaft 38 in the other direction, the pressure in the motor will be increased so that the motor will serve as a driving means. Although the motor 109 would otherwise be able to do the driving and feed the table, it must be remembered that the feed motor 20 is connected to the table through a worm and worm gear drive and therefore the actual rate of movement of the table will still be determined by the fed motor so that the motor 109 although actually moving the table will do so at a rate determined by the rate of rotation of the feed motor 20.

This automatic change in the unidirectional pressure acting on the motor 109 is automatically effected when the work supporting table is reversed and is effected by the automatic reversing mechanism.

The two different pressures needed for the operation of motor 109 are provided by the pump 104, and a selector valve 113 is provided for alternately connecting these pressures to the motor. The higher pressure is provided by connecting the output of pump 104 directly to port 114 of the selector valve. The lower pressure is obtained by connecting a branch line 115 in series with a pair of relief valves 116 and 117. The valve 116 is set to yield the higher pressure in channel 103, and cause a pressure drop in the fluid flowing to channel 118. This channel is connected to the relief valve 117 which is set at a lower pressure, and channel 118 is connected to port 119 of valve 113 so that this predetermined lower pressure may be connected to the motor 109.

The two relief valves act as hydraulic resistances for determining an intermediate pressure in the channel 118, the valve 116 serving as a resistance "in," and the valve 117 serving as a resistance "out."

The selector valve 113 has a port 120 which is connected by a channel 121 to port 122 of a cut out valve 123 for the motor 109. This valve has a plunger 124 in which is formed an annular groove 125 which, in the position shown, serves to connect the port 122 to port 126. The port 126 is permanently connected to port 127 of valve 97. The valve plunger 96 has an annular groove 128, which is in such a position, when the door of the gear box is closed, to interconnect port 127 to port 129 and thereby to channel 130 which leads to the hydraulic motor 109. The other port of the motor is permanently connected through channel 131 to the reservoir 108. Whenever the valve plunger 124 is pulled out, a spool 132 closes the pressure port 122 and the annular groove 125 interconnects port 126 to an exhaust port 133. Thus when the gear box is closed, and the valve plunger 124 is pulled out, both ports of the motor 109 are connected to reservoir thereby offering no resistance or braking action on the table drive mechanism. When the valve plunger 124 is in the position shown in Figure 14, and the gear box cover is opened, the shifting of valve plunger 96 will disconnect port 126 from port 129, and the annular groove 128 will interconnect the port 129 to the exhaust channel 134 whereby the pressure will be off of the motor 109, so that the gear 49 may be rotated readily for set-up purposes.

When the valve plungers 96 and 124 are in the position shown in Figure 14, the selection of pressure delivered to the motor 109 is determined by the selector valve 113 and to this end the valve has a plunger 135 in which are formed two annular grooves 136 and 137. When the plunger 135 is shifted to the left, the annular groove 137 interconnects the lower pressure port 119 to port 120, and when the plunger is shifted to the right, the annular groove 136 interconnects the higher pressure port 114 to port 120.

The plunger 135 is shifted automatically in accordance with the direction of movement of the table, and the ends of the valve housing 113 are provided with ports 138 and 139 which are connected respectively through channels 140 and 141 to a pilot valve 142, shown in Figure 15. The pilot valve 142 has pressure ports 143 which are connected by a channel 144 to a suitable source of operating pressure; and ports 145 and 146 on opposite sides thereof to which are connected the channels 141 and 140 respectively. The valve plunger 147 is self actuating in that it has a central spool 148 which is larger in diameter than the two end spools 140 and 150. When the plunger is to the left, the annular groove 151 interconnects pressure port 143 to port 146 to cause shifting of the plunger 135 to the right. A trip operable mechanism is provided for shifting the plunger 147, but it will be noted that after the plunger passes the pressure port 143 in either direction, the pressure differential created by the difference in area between the spool 148 and the end spools will complete the shifting movement thereby simulating the action of a conventional detent mechanism.

The trip mechanism comprises a trip plunger 153 from which projects a pair of wings 154 and 155, the wings lying in different planes, as shown in Figure 1. These wings are alternately engaged by dogs 156 and 157 carried by the table 13 for rotating the plunger. The plunger has a ball ended lever 158 which engages a groove 159 formed in the valve plunger 147 for shifting the same. It will thus be evident that the direction of movement of the table determines the position of the selector valve plunger 135 and thereby the value of the unidirectional pressure to be applied to the motor 109.

The trip plunger not only determines the pressure to be applied to the backlash motor, but also controls the reversing of the feed motor 20, and in such manner as to cause deceleration of the motor at a uniform rate prior to the shifting of the reverse control valve. The entire cycle of reversing, however, includes not only the deceleration of the table, but also means for effecting a predetermined dwell of the table, and a controlled uniform acceleration in the opposite direction. This manner of reversing the table eliminates any sudden changes in rate which might cause undesirable vibrations and their resultant effect upon the finish of the work. These various results are obtained by a reversing control valve 160, which has an acceleration and deceleration control valve plunger 161, and a reversing control sleeve 162. The plunger 161 is connected by a ball ended crank 163 to the trip plunger 153 for actuation thereby.

The reversing valve 160 has a pair of motor ports 164 and 165, which are connected by channels 166 and 167 respectively to ports 168 and 169 of a start-and-stop control valve 170. This valve has a plunger 171 which, when in the running position shown, interconnects ports 169 and

168 to ports 172 and 173 and thereby through channels 174 and 175 to the motor 20.

The valve 160 also has a pair of pressure ports 176 and 177 which are supplied through a channel 178 from port 179 of a rate control valve indicated generally by the reference numeral 180. The valve also has a pair of exhaust ports 181 and 182 which are connected to the return channel 183. The reversing valve sleeve 162 has a series of annular grooves 184, 185, 186, 187, 188, and 189 which are in continuous communication with ports 181, 164, 176, 177, 165, and 182 respectively. Each of these annular grooves has a series of radial holes drilled in the bottom thereof and communicating with the interior of the sleeve, thus forming ports for control by the inner plunger.

In the position of the parts shown in Figure 15, the table 13 has been moved toward the left and the beveled face 190 on the dog 157 has engaged the curved face 191 on the wing 155 and rotated the plunger 153 in a counterclockwise direction, and this rotation has taken place through a predetermined length of movement of the table. This means that the plunger 161, which was in a right hand position interconnecting pressure port 176 to motor port 164, and motor port 165 to exhaust port 182, was moved toward the left at a predetermined rate to gradually close port 176 and thereby throttle the flow to the motor 20 to effect deceleration thereof. After the trip plunger rotated through half of its angle of movement, the automatic self actuation of the pilot valve quickly completed the movement. This resulted in deceleration and final stoppage of the motor 20 and thereby of the table 13.

The final shifting of the pilot valve caused admission of pressure to port 146 and thereby to branch channel 192 leading to port 193 located in the left hand end of valve housing 160. This resulted in pressure being admitted to cause shifting of the reversing sleeve 162. The space 194 in the other end of the valve housing is in communication through an interdrilled passage 195 in the sleeve to port 196 and this port is connected by channel 197 to ports 198 and 199 of a combination tarry and acceleration control valve 200. The fluid entering port 199 of this valve passes through a spring closed check valve 201 to an interdrilled passage 202 which terminates in an annular groove 203 formed in the tarry control valve plunger 204. A spool 205 on one side of this groove has a series of V-shaped grooves 206 formed in its periphery which control the rate of flow to a port 207. The port 207 is connected by channel 208 to port 145 of the pilot valve 142, and at this time this port is connected to the return channel 183. Thus the rate of movement of the sleeve 162 toward the right depends upon the setting of the tarry control valve plunger 204. This will determine the length of tarry of the table, which will continue until the sleeve 162 is shifted toward the right, a sufficient distance to interconnect the pressure port 177 with the motor port 165.

At this point in the cycle of movement of the sleeve toward the right, the port 196 will close and a second port 209 will just start to open. In other words, the end of the sleeve is provided with a second interdrilled passage 210 which will now be placed in communication with the port 209. This means that the fluid that is being exhausted from the space 194 will stop flowing through port 196 and will start to flow through port 209 and channel 211 to port 212 of a combination tarry and acceleration control valve indicated generally by the reference numeral 213. This port communicates with a T-shaped interdrilled passage 214 which terminates in a spring pressed ball check valve 215. The fluid flows past this check valve into a spiral groove 216 which has sufficiently small pitch to offer a definite resistance to the flow of fluid through it.

This spiral groove terminates in a port 217 which is connected by channel 218 to channel 208 leading to port 145 of the pilot valve 142. The port 145 is connected at this time to an exhaust port 219 through which the fluid flows to the return channel 183. Thus the rate of movement of the reversing valve sleeve 162 will depend upon first the resistance of the tarry control grooves 206, which may be different from the resistance of the acceleration control spiral groove 216. Therefore a first means has been provided for determining the rate of the first half of the stroke of the reversing valve sleeve to determine the amount of tarry of the table, and a second means for determining the rate of the last part of the stroke of the reversing valve sleeve to determine the rate of acceleration of the table in the new direction.

After the pressure port 177 has become fully connected to port 165, the rate of fluid supply to the motor 20 is determined by the rate control valve 180. The supply of operating fluid for the motor 20 comes from a separate pump 220 through channel 221 to port 222 of a combining valve 223. This valve has a straight passage 224 through it, which terminates in port 225 that is connected by channel 226 to port 227 of the rate control valve 180.

The rate valve has an adjustable plunger 228 which is urged in one direction by a spring 229 to close port 227, and in the other direction by a separate plunger 230 which is threaded in the bed of the machine. The plunger has a series of longitudinally extending V-shaped grooves 231 which decrease in depth at one end so that as the plunger 228 is moved against the resistance of spring 229, the resistance of port 227 may be gradually reduced. In this manner the flow of fluid to the hydraulic motor 20 is throttled to yield various feed rates.

When the rate valve is adjusted to a position to yield the higher feed rates, the pump 220 will not deliver sufficient volume to produce these rates, and therefore an auxiliary pump has been provided together with suitable control means for automatically combining or adding the delivery of the auxiliary pump to that of the main pump. In order, however, to prevent the auxiliary pump from laboring when not being utilized, a by-pass connection has been provided and this by-pass control is so arranged that the discharge from the auxiliary pump is gradually throttled as the setting of the rate valve goes up, so that desirable division of flow of the output of the auxiliary pump is made in accordance with the demands for extra fluid and in accordance with the degree of increase in the rate.

As shown in Figure 15, the auxiliary pump 232 is connected by a channel 233 to port 234 of the combining valve 223. The port 234 is normally maintained closed by a spring pressed plunger 235 and the spring is assisted by the pressure existing in channel 226, there being a branch connection 236 from this channel to the end of the bore 237 which contains the plunger. The

pressure that will exist in the channel 233 tending to shift the valve plunger 235, will depend upon the setting of an auxiliary valve plunger 238 which is mounted in the same block with the rate valve and adjusted by the same control. This valve has a port 239 to which a branch 240 of channel 233 is connected, and when the rate valve is set for low feed rates, the port 239 is connected to a reservoir port 241 by means of an annular groove 242 formed in the plunger 238.

When the plunger 230 of the rate valve is adjusted in a direction to increase the feed rate, a laterally extending arm 243 which is connected to the plunger 238 moves this plunger simultaneously with plunger 228 and in a direction to gradually close the port 242 and thereby increase the pressure in channel 240 to such a point that when the pressure in the channel 224 drops below a predetermined minimum, the valve 235 will open and thereby combine the delivery of the auxiliary pump with that of the main pump.

Automatic reciprocation of the table is stopped by shifting the plunger 171 of the start and stop control valve downward, as viewed in Figure 15, by rotating the control handle 244 to a central position. This will close ports 169 and 168 thereby stopping the flow to and from the motor. The table movement control mechanism just described is specifically claimed in our copending application, Serial No. 372,240.

When it is desired to control the movement of the table manually, the lever 244 is rotated to the left of its central position thereby depressing the valve plunger 171 a sufficient amount to interconnect the motor port 172 with port 245, and the motor port 173 with port 246. This serves to interconnect the hydraulic motor 20 with a servo control valve 247 through channels 248 and 249 which terminate in ports 250 and 251 of the servo-valve. The servo-valve also has a pressure port 252 and an exhaust port 253. The pressure port 252 is connected by channel 254 to port 255 of an auxiliary control valve 256 which has a plunger 257 which is simultaneously reciprocated with the plunger 171 under control of lever 244, but in an opposite direction. Therefore, the plunger 257 moves up when the plunger 171 moves down, and this results in port 255 being interconnected by the annular groove 258 to port 259. The port 259 is connected by a branch channel to the main pump supply channel 144. Therefore, the flow from pump 220 passes through port 259, annular groove 258, and port 255 to channel 254 and thereby to the pressure supply port 252 of the servo-valve.

At the same time the pressure in channel 254 flows to a cylinder 260 in which is slidably mounted a piston 261 carrying a roller 262. This presses the roller into frictional engagement with the periphery of two discs 263 and 264. As shown in Figure 4, the disc 263 is fixed on the drive shaft 21 for rotation by the hydraulic motor 20. The disc 264 is supported on the end of a shaft 265 which is antifrictionally supported in the bed for rotation and is operatively connected to the rotary plunger 266 of the servo-valve.

There is a gear 267 which is supported for rotation on a reduced diameter 268 of the servo-valve housing and this gear is connected by a series of pins 269 to an equalizer plunger housing 270 supported on the end of the rotary plunger 266 for movement relative thereto but connected as by pins 271 to the rotary valve sleeve 272.

In other words, when the sleeve 272 is rotated by the gear 267, the motor 20 is caused to rotate and through the feed-back connection comprising the disc 263, the roller 262 and the disc 264, the plunger 266 is caused to rotate and thereby follow up the movement of the sleeve in a direction to stop operation of the motor 20.

The gear 267, as shown in Figure 1, is connected by a gear train indicated generally by the reference numeral 273 to gear 274 secured to shaft 275 which carries a hand wheel 276. Thus, by rotation of the hand wheel 276, the servo-valve sleeve may be rotated to cause operation of the hydraulic motor 20 in opposite directions.

The manner in which the hydraulic connections to the motor are effected by the servo-valve will now be explained. The servo-valve plunger 266 has a series of longitudinally cut away portions whereby a cross section through the plunger resembles a quadrate cross. This forms four quadrants 277, 278, 279, and 280. As shown in Figure 5, the pressure groove 252 has a pair of radial ports 281 and 282 which are normally closed when the valve is in a stop position. From this it will be seen that when the sleeve 272 is rotated in a counterclockwise direction relative to the plunger 266, that the quadrants 277 and 279 will interconnect the radial pressure ports 281 and 282 with radial ports 283 and 284 in the groove 250 which leads to one port of the motor. At the same time the radial ports 285 and 286 in groove 251, as shown in Figure 7, will be connected to the radial ports 287 and 288 in exhaust groove 253, as shown in Figure 8. Thus the other port of the motor will be connected to reservoir. It will be noted that the pair of ports shown in each of the Figures 5, 6, 7 and 8 are arranged diametrically opposite to one another thereby equalizing the opposing pressures on the rotary plunger 266.

If the sleeve 272 is rotated clockwise, then the pressure and exhaust connections are reversed with respect to the motor ports, and the motor will rotate in an opposite direction.

When the lever 244 is moved back to a central position and the plunger 257 moves upward to a central position, the pressure port 259 will be closed, but upon further downward movement into the position shown in Figure 15, the pressure port 259 will be connected to a channel 289 leading to the other end of cylinder 260 whereby the plunger 261 will be moved upward and thereby break the feed-back connection from the motor 20 to the servo-valve. In order to prevent the plunger 266 and the sleeve 272 from getting out of phase during automatic operation of the machine, the equalizer plunger housing 270 is provided with two diametrically opposed equalizing plungers 290 and 291, Figure 16, which have V-shaped ends fitting in V-notches 292 and 293 formed in opposite sides of a collar 294 fixed with the rotary plunger 266.

When the table 13 moves toward the right, as viewed in Figure 15, the inclined surface 295 on the dog 156 engages the wing 154 of trip plunger 153 rotating the same in a clockwise direction, thereby shifting the pilot valve control plunger to the right and simultaneously moving the deceleration control valve plunger 161 in the same direction. The result is the gradual closing of pressure port 177, whereby the rate of movement of the table will decrease.

The fluid pressure that is admitted to port 145 by the shifting of the pilot valve will flow through channel 288 to port 296 of control valve 213.

This valve has a throttle valve plunger **297** similar to valve plunger **205** which throttles the flow from port **296** into the annular groove **298**. The fluid continues through an interdrilled passage **299** and past spring closed check valve **300** to channel **211** through which it continues to the right hand end of valve **160**. This will cause movement of the sleeve **162** toward the left until the port **209** closes.

Since the fluid can no longer flow through valve **213** from channel **208** on account of the closing of port **209**, it will be forced through port **301** and spiral groove **302** of valve **200**, to the interdrilled passage **303**, which terminates in a spring closed check valve **304**. This valve will open, and the fluid will continue through port **198** and channel **197** to port **196** of valve **160**, thereby continuing the movement of the sleeve **162** to the left and causing the motor to accelerate in the new direction. It is thus possible to cause the table to decelerate to a stop, tarry, and accelerate in a new direction at each end of the table stroke.

There has thus been provided an improved transmission and control mechanism for a grinding machine which is capable of yielding an improved traverse cycle; which is adjustable to vary the feed rate and the length of tarry; which is provided with a camber control mechanism that is accurately synchronized with the table movement and adjustable to yield variable cambers; and which is provided with a manual control mechanism having intermediate servo-motor actuating mechanism, whereby manual operation and/or automatic cyclic operation, with or without a cambering mechanism, may be obtained.

What is claimed is:

1. In a machine tool having a work support and a tool support, the combination of transmission means for effecting relative movement between the supports, including a first power driven train for moving one of said supports, a second train in driving engagement with the movable support and including an actuator for the second support, and means to eliminate backlash from said second train to maintain synchronism between said actuator and the movable support, including a hydraulic motor connected to the second train, and means for maintaining unidirectional pressure on said motor whereby the motor will act as a retarding means during one direction of movement and as a driving means during the other direction of movement of said second train.

2. In a grinding machine having a work support and a grinding wheel support, the combination of transmission means including a rack fixed to said work support, a reversible hydraulic motor connected by a motion transmitting train to said rack to control the reciprocation of the work support, a control cam for shifting the grinding wheel toward and from the work to camber the same, a motion transmitting train coupled with said rack driving connections between said cam and train for rotating said cam, said last named train terminating in a hydraulic motor, and means to apply a variable unidirectional hydraulic pressure on said motor whereby during one direction of movement of the work support the motor will act as a retarding means, and during the other direction of movement the motor will serve as a driving means.

3. In a grinding machine having a work support and a grinding wheel support, the combination of means for reciprocating one of said supports and oscillating the other support in synchronism therewith, including a pair of hydraulic motors, a motion transmitting train extending from one of said motors to the other, serially arranged means in said train for actuating each of said supports, means for applying a uniform hydraulic pressure to one of said motors, and a relatively lower pressure to the other of said motors to effect actuation of the train in one direction, and means to reverse said one motor and increase the pressure on the other motor to effect actuation of the train in an opposite direction.

4. In a grinding machine having a work support and a grinding wheel support, the combination of means for reciprocating one of said supports and oscillating the other support in synchronism therewith, including a pair of hydraulic motors, a motion transmitting train extending from one of said motors to the other, serially arranged means in said train for actuating each of said supports, means for applying a uniform hydraulic pressure to one of said motors, and a relatively lower pressure to the other of said motors to effect actuation of the train in one direction, and means to reverse said one motor and increase the pressure on the other motor to effect actuation of the train in an opposite direction.

5. In a grinding machine having a reciprocating work support and a grinding wheel movable toward and from said support the combination of a reversible hydraulic motor, a motion transmitting train connecting said motor to the work support, a unidirectionally driven motor, a train extending from the work support to the last named motor and having means therein for oscillating the grinding wheel, a valve for reversing the fluid pressure on the first named motor, a pressure selector valve for the second named motor, and a common pilot control valve for shifting said reverser and selector valves.

6. In a machine tool having a relatively movable tool support and work support, the combination with a reversible power operable motor for determining the rate of said relative movement in opposite direction, of fluid operable means connected to the moving support for opposing one direction of movement and assisting the other direction of movement thereof, a pressure determining valve having two positions for connecting fluid supplies of different pressure to said means, valve means for controlling said motor, including a valve member having movement through a neutral zone for creating a tarry in said motor, and a pilot valve having fluid connections for simultaneously shifting said pressure determining valve and said valve member.

7. In a machine tool having a relatively movable tool and work support, the combination of a hydraulic control circuit for effecting said movement in opposite directions, including a pair of fluid operable driving motors, a two-pressure system for selective actuation of one of said motors, including a shiftable pressure selector valve, a decelerator valve member and a tarry valve member for controlling said driving motors, a pilot valve, a trip plunger having operative connection for simultaneously shifting said pilot valve and decelerator valve, fluid operable means for completing the movement of said trip plunger and thereby of the connected members, and fluid pressure means connectible by said pilot valve for shifting of said pressure selector valve and of said tarry control valve member.

8. In a machine tool having a work support and a tool support mounted for relative movement in two angularly related directions, means for controlling the relative movements of said supports including a first motor driven train connected to one of said supports, an actuating motor therefore, means for reversing the directional actuation of the motor, a second train coupled to the motor for transmitting motion to the second support in synchronized relation to the movement of the first support, and hydraulic means for eliminating backlash in the driving trains for the supports including a unidirectionally actuated hydraulic device reacting on said trains in opposition to the motor.

9. In a machine of the character described including a shiftable part, a pair of simultaneously power actuated hydraulic motors coupleable with the part in opposed driving relation for effecting a shifting movement thereof, means for determining the rate of actuation of one of said motors, and additional means for determining the relative power reaction of the motors whereby they selectively act as drivers during joint power actuation for movement of the part at the determined rate.

10. In a machine of the character described including a shiftable part, a pair of simultaneously power actuated hydraulic motors coupleable with the part in opposed driving relation for effecting a shifting movement thereof, means for determining the rate of actuation of one of said motors, and additional means for determining the relative power reaction of the motors whereby they selectively act as drivers during joint power actuation for movement of the part at the determined rate, said means including an element activated by movement of the part for effecting reversal of said movement.

11. In a machine of the character described including a shiftable part, a power source a pair of hydraulic motors simultaneously coupleable with the part and with said power source for jointly effecting a shifting movement thereof, means for determining the rate of actuation of one of said motors, additional means for determining the relative driving reaction of the motors for movement of the part at the determined rate, said means including an element activated by movement of the part for effecting reversal of said movement, and a direction determinator effective as respects one of said motors actuable by movement of the element.

12. In a machine of the character described including a shiftable part, a pair of hydraulic motors simultaneously coupleable with the part for effecting a shifting movement thereof, means for determining the rate of actuation of one of said motors, additional means for determining the relative driving reaction of the motors for movement of the part at the determined rate, said means including an element activated by movement of the part for effecting reversal of said movement, a direction determinator for one of said motors, and an activating pressure determinator for the other of said motors actuable by said element.

13. In a machine of the character described including a shiftable part, a pair of hydraulic motors simultaneously coupleable with the part for effecting a shifting movement thereof, means for determining the rate of actuation of one of said motors, additional means for determining the relative driving reaction of the motors for movement of the part at the determined rate, said means including an element activated by movement of the part for effecting reversal of said movement, a direction determinator for one of said motors, and an activating pressure determinator for the other of said motors actuable by said element, one of said determinators including hydraulically operable means for completing the actuation of the element and associate parts as initiated by movement of the shiftable part.

14. In a machine of the character described including a shiftable part, a pair of hydraulic motors simultaneously coupleable with the part for effecting a shifting movement thereof, means for determining the rate of actuation of one of said motors, additional means for determining the relative driving reaction of the motors for movement of the part at the determined rate, said means including an element activated by movement of the part for effecting reversal of said movement, a direction determinator for one of said motors and an activating pressure determinator for the other of said motors actuable by said element, both of said determinators including hydraulically operable means for completing the actuation of the element and associate parts as initiated by movement of the shiftable part.

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