Dec. 1, 1942. C. HERFURTH ET AL 2,303,639

GRINDING MACHINE

Original Filed April 25, 1938 3 Sheets-Sheet 1

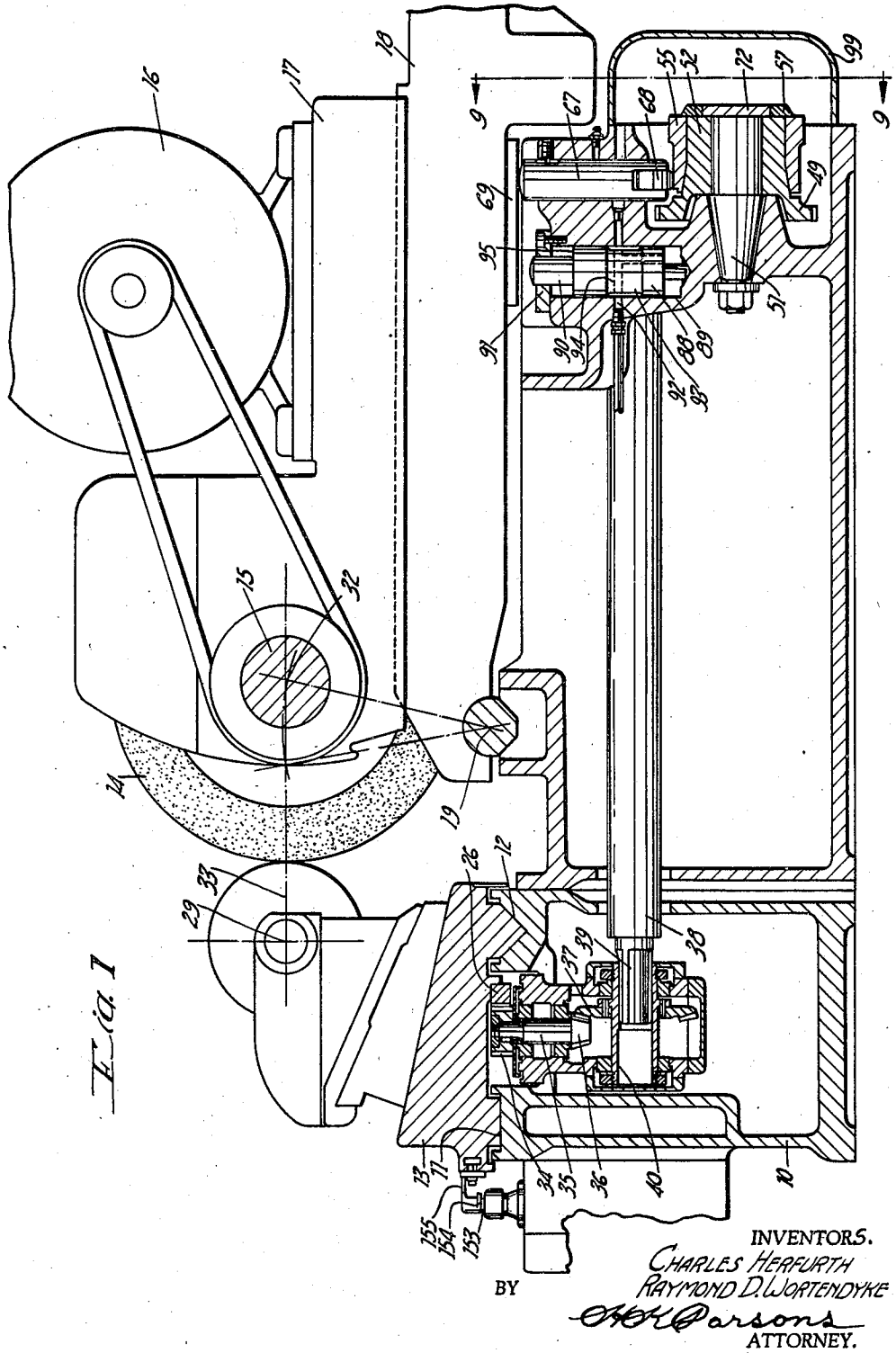


Fig. 1

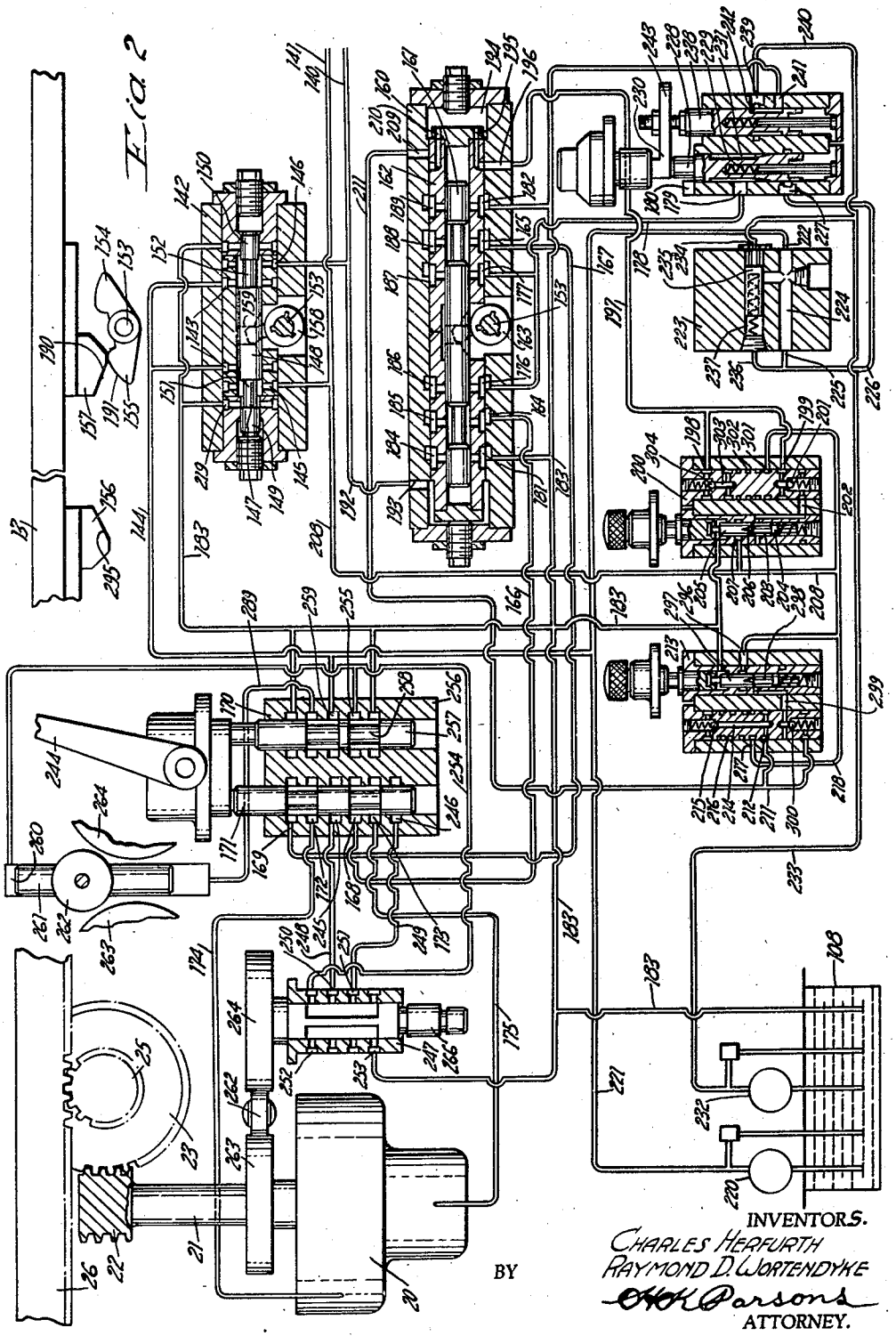
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FIG. 2



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Fig. 3

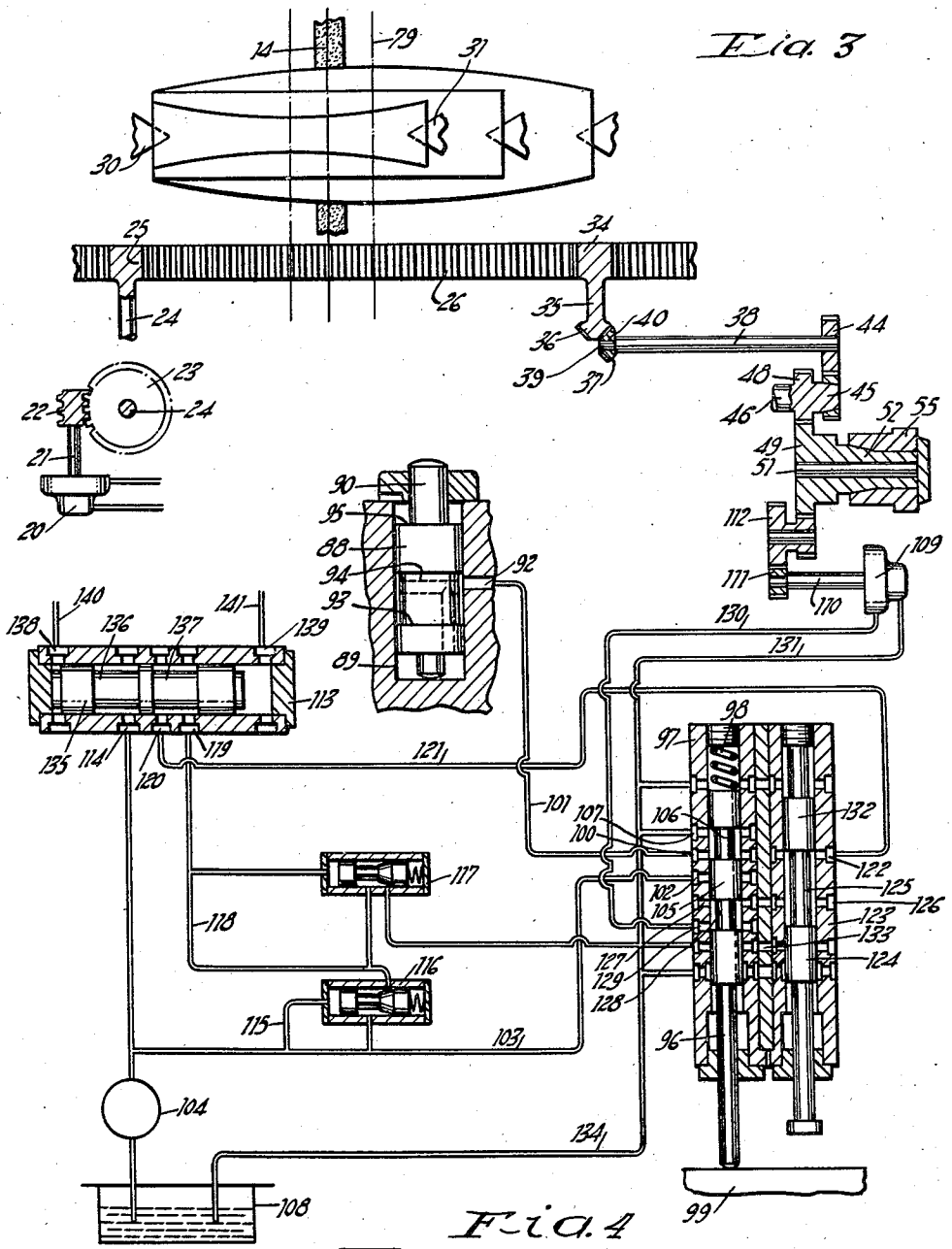
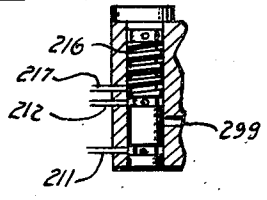


Fig. 4



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UNITED STATES PATENT OFFICE

2,303,639

GRINDING MACHINE

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Original application April 25, 1938, Serial No. 204,083, now Patent No. 2,254,010, dated August 26, 1941. Divided and this application December 30, 1940, Serial No. 372,240

11 Claims. (Cl. 121—45)

This invention relates to improvements in grinding machines and more particularly to improvements in the transmission and control mechanism therefor, and is a division of copending application Serial No. 204,083, filed April 25, 1938, issued as Patent No. 2,254,010.

One of the principal objects of the present invention is the provision of an improved actuating mechanism for a machine of this character which will effect a smooth and uniform relative shifting of the parts of the machine and shockless reversal of the direction of their relative movement.

A further object of the present invention is the provision in a transmission of this character of controllable means for independent determination of the rates of deceleration and acceleration in connection with reversal or change of direction of relative movement of the parts.

Another object of the invention is the provision of a novel control mechanism for effecting the changes of rate and direction of movement of the parts.

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 forming a part thereof and it is to be understood that any modifications may be made in the exact structural details there shown and described, 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 represents a transverse sectional view of the machine constructed in accordance with and embodying the principles of the invention.

Figure 2 presents a diagrammatic view of the essential parts of the machine and the hydraulic control circuits therefor, the controlling valve elements being indicated in section, and

Figure 3 represents a diagrammatic view of the hydro-mechanical drives for reciprocation of the shiftable element of the machine and certain of the controls for the hydraulic circuit.

Figure 4 presents a view of a detail of one of the valve devices of the machine.

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 1. A work supporting table 13 is mounted on these guideways for traversing movement relative to a grinding wheel 14.

As shown in Figure 1, 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 3, 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 204,032, filed April 25, 1938, issued as Patent No. 2,167,948. 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. 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 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 48 is supported by a shaft 51.

Means have been provided for taking the backlash or lost motion out of the drive connection for the table, 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. 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 feed 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 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 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 3, 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 3, 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 is 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 2. 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 a 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 2, 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 completes the movement. This results in deceleration and final stoppage of the motor 20 and thereby of the table 13 by trip effected positioning of member 161 which acts as a stop valve counteractible by subsequent shifting of the enclosing reversing valve sleeve 162.

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 2, 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 2, 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.

When the table 13 moves toward the right, as viewed in Figure 2, 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 208 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.

What is claimed is:

1. In a machine tool having a reciprocating support, the combination of a hydraulic control circuit for said support, including a fluid operable motor, a source of pressure, a valve interposed between said source of pressure and said motor including telescoping members, an actuator operatively connected for moving one of said members in a direction to reduce the flow of fluid to said motor, fluid operable means for shifting the second of said members, and a control valve operatively connected to said actuator for subsequently effecting the admission of shifting pressure to said fluid operable means to effect reversal of said motor.

2. In a grinding machine having a reciprocable support, the combination of a hydraulic motor for effecting said reciprocation, a control circuit for said motor, including a decelerating valve and a reversing valve, a rotatable trip plunger, means connecting said plunger to the decelerating valve, a pilot valve for controlling the hydraulic shifting of the reversing valve, said pilot valve being connected to said trip plunger, and dog means carried by the table for oscillating said plunger whereby said table will be decelerated and then reversed.

3. In a machine tool having relatively movable tool and work supports and a fluid operable motor for controlling reciprocation of one of said supports relative to the other, the combination of a hydraulic control circuit for said motor, including a valve housing having telescoping decelerator and reversing valve members, mechanical means operable by the table for shifting said decelerator valve, cylinders formed on the ends of said valve housing, piston portions on the end of said reversing valve member reciprocable within said cylinder portions, means operable by the moving support for admitting fluid pressure to one of said cylinders, means to control the exhaust from the other cylinder, including an adjustable throttle valve, a hydraulic resistance, and means responsive to a partial movement of

the decelerator valve member for disconnecting the exhaust from said throttle valve and connecting it to said hydraulic resistance.

4. In a machine tool having relatively movable tool and work supports, the combination of a hydraulic motor for controlling reciprocation of one of said supports relative to the other, a hydraulic control circuit for said motor, including a decelerator valve member and a reversing valve member, piston and cylinder means for shifting said reversing valve member in opposite directions, a pilot valve operable by the table for alternately connecting a source of fluid pressure to one of said piston and cylinder combinations, one of said connections including a throttle valve and a hydraulic resistance connected in parallel to one of said cylinders, means controlled by the shiftable valve for alternately establishing said connections whereby the throttle valve will control the rate of the first part of said shifting movement, and the hydraulic resistance will control the rate of the remainder of said shifting movement.

5. In a machine tool having relatively movable tool and work supports and a fluid operable motor for controlling reciprocation of one of said supports relative to the other, the combination of a hydraulic control circuit for said motor including a hydraulically actuable reversing valve for the motor circuit, a pilot valve for determining the coupling of hydraulic medium with the reversing valve for effecting alternate directional movements thereof, a pair of resistance elements intervening the pilot valve and one end of the reversing valve, means in the reversing valve for sequentially coupling the resistance elements in controlling relation to the said valve to determine successive rates of movement thereof during unidirectional shifting under influence of the pilot valve circuit, an additional plurality of resistance elements, conduit means potentially coupling said additional resistance elements in serial relation between the pilot valve and said end of the reversing valve, and means for establishing alternate couplings of the sets of resistance elements with one end of the reversing valve for determination of its respective successive rates of movement in opposite directions.

6. In a machine tool having relatively movable tool and work supports and a fluid operable motor for controlling reciprocation of one of said supports relative to the other, the combination of a hydraulic control circuit for said motor including a hydraulically actuable reversing valve for the motor circuit, a pilot valve for determining the coupling of hydraulic medium with the reversing valve for effecting alternate directional movements thereof, a pair of resistance elements intervening the pilot valve and one end of the reversing valve, means in the reversing valve for sequentially coupling the resistance elements in controlling relation to the said valve to determine successive rates of movement thereof during unidirectional shifting under influence of the pilot valve circuit, an additional plurality of resistance elements, conduit means potentially coupling said additional resistance elements in serial relation between the pilot valve and said end of the reversing valve, means for establishing alternate couplings of the sets of resistance elements with one end of the reversing valve for determination of its respective successive rates of movement in opposite directions, and means individual to each set of resistances for varying one of said resistances whereby the rate of movement of the reversing valve during the effectiveness of said individual resistance with respect thereto may be adjustably determined.

7. In a machine tool having a movable support, a fluid operable motor for controlling reciprocation of the support and a hydraulic operating circuit including a source of hydraulic pressure medium for activation of the motor, means for controlling the action of the medium with respect to the motor including a valve housing in the hydraulic circuit having a plurality of pressure inlet and outlet ports formed therein, a reversing sleeve valve slidably mounted within the casing having conduit portions respectively communicating with the several ports of the casing, a cylindrical valve spindle slidably mounted within the sleeve and having cannelures selectively positionable for variable coupling of the conduits in the sleeve, means operable by the motor for determining the position of the valve spindle and thus of the cannelures with respect to the conduits in the sleeve, and additional means for effecting movement of the sleeve with respect to the spindle to vary the coupling of the conduits in the sleeve with the cannelures for a prescribed positioning of the valve spindle.

8. In a machine tool having a movable support, a fluid operable motor for controlling reciprocation of the support and a hydraulic operating circuit including a source of hydraulic pressure medium for activation of the motor, means for controlling the action of the medium with respect to the motor including a valve housing in the hydraulic circuit having a plurality of pressure inlet and outlet ports formed therein, a reversing sleeve valve slidably mounted within the casing having conduit portions respectively communicating with the several ports of the casing, a cylindrical valve spindle slidably mounted within the sleeve and having cannelures selectively positionable for variable coupling of the conduits in the sleeve, means operable by the motor for determining the position of the valve spindle and thus of the cannelures with respect to the conduits in the sleeve, additional means for effecting movement of the sleeve with respect to the spindle to vary the coupling of the conduits in the sleeve with the cannelures for a prescribed positioning of the valve spindle, and a plurality of rate determining devices adapted to control the rate of adjusting movement of the sleeve, the sleeve having portions effective during movement thereof to determine the action of the rate control devices with respect thereto.

9. In a machine tool of the character described, the combination with a movable support and a hydraulic motor for controlling the rate and direction of support movement, of means actuable by movement of the support for controlling the motor including a pilot valve and a stop valve each mechanically actuable by the support, said stop valve having variably positionable flow controlling portions, a source of hydraulic pressure medium, a reversing valve having ports variably positionable for serial cooperation with the flow controlling portions of the stop valve, and conduits serially coupling the source, the reversing and stop valves with the motor whereby the relative positions of the stop and reversing valves will be determinative of the directional operation of the motor, piston means for actuating the reversing valve, and auxiliary pilot conduits interconnecting the pilot valve, the source of pressure and said piston means whereby support determined positioning of the pilot valve will cause

the hydraulic pressure to act on the reversing valve for effecting a reverse hydraulic positioning thereof.

10. In a machine tool of the character described, the combination with a movable support and a hydraulic motor for controlling the rate and direction of support movement, of means actuable by movement of the support for controlling the motor including a pilot valve and a stop valve each mechanically actuable by the support, said stop valve having variably positionable flow controlling portions, a source of hydraulic pressure medium, a reversing valve having ports variably positionable for serial cooperation with the flow controlling portions of the stop valve, and conduits serially coupling the source, the reversing and stop valves with the motor whereby the relative positions of the stop and reversing valves will be determinative of the directional operation of the motor, piston means for actuating the reversing valve, auxiliary pilot conduits interconnecting the pilot valve, the source of pressure and said piston means whereby support determined positioning of the pilot valve will cause the hydraulic pressure to act on the reversing valve for effecting a reverse hydraulic positioning thereof, a pair of hydraulic resistances interposed in the pilot circuit between the pilot valve and the actuating pistons for the reversing valve, and means for sequentially coupling said resistances with the reversing valve during unidirectional movement thereof whereby the time of movement of the reversing valve through tarry and reverse accelerating positions may be controlled.

11. In a machine tool of the character described, the combination with a movable support and a hydraulic motor for controlling the rate and direction of support movement, of means actuable by movement of the support for controlling the motor including a pilot valve and a stop valve each mechanically actuable by the support, said stop valve having variably positionable flow controlling portions, a source of hydraulic pressure medium, a reversing valve having ports variably positionable for serial cooperation with the flow controlling portions of the stop valve, and conduits serially coupling the source, the reversing and stop valves with the motor whereby the relative positions of the stop and reversing valves will be determinative of the directional operation of the motor, piston means for actuating the reversing valve, auxiliary pilot conduits interconnecting the pilot valve, the source of pressure and said piston means whereby support determined positioning of the pilot valve will cause the hydraulic pressure to act on the reversing valve for effecting a reverse hydraulic positioning thereof, a pair of hydraulic resistances interposed in the pilot circuit between the pilot valve and the actuating pistons for the reversing valve, means for sequentially coupling said resistances with the reversing valve during unidirectional movement thereof whereby the time of movement of the reversing valve through tarry and reverse accelerating positions may be controlled, additional hydraulic resistances, and means for effecting sequential operative coupling of said additional hydraulic resistances sequentially in controlling relation to the reversing valve upon opposite actuation of the pilot valve whereby to control the period of tarry and acceleration of the motor and support in an opposite direction.

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