Aug. 15, 1950 W. A. COX 2,518,600

INFEED ATTACHMENT FOR GRINDING MACHINES

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Fig. 1

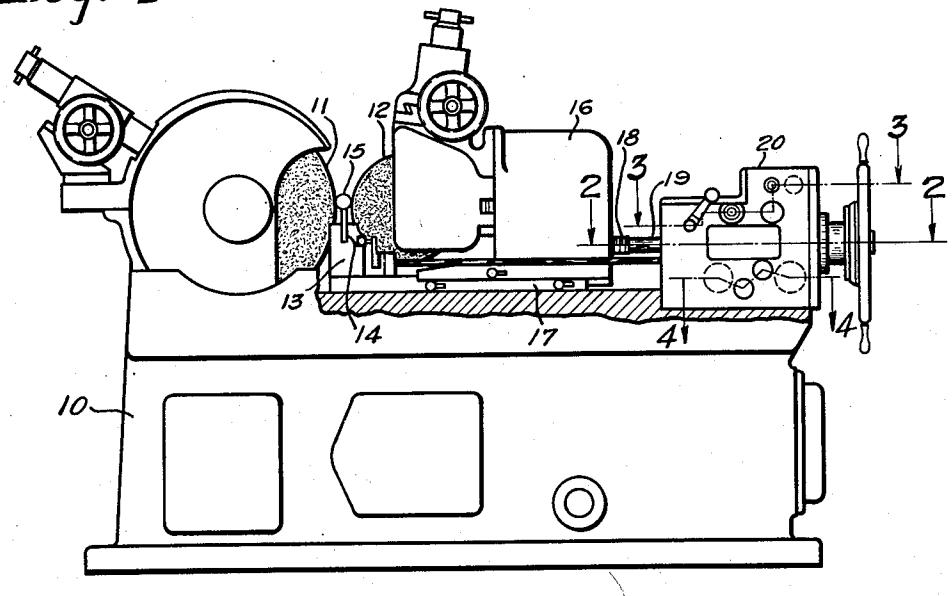
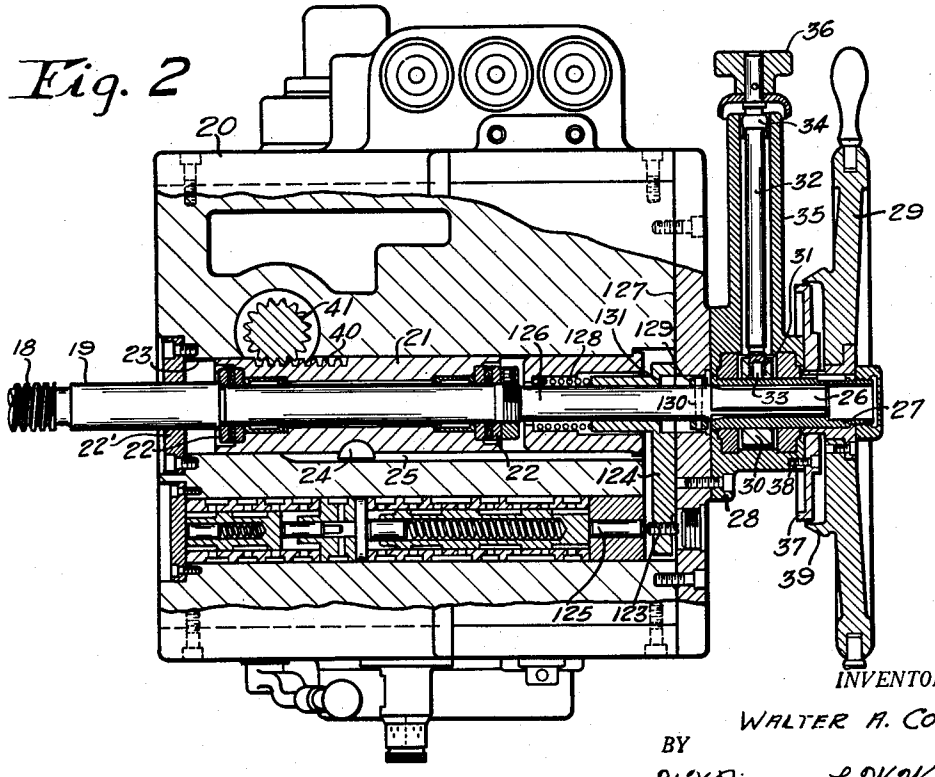


Fig. 2



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Fig. 3

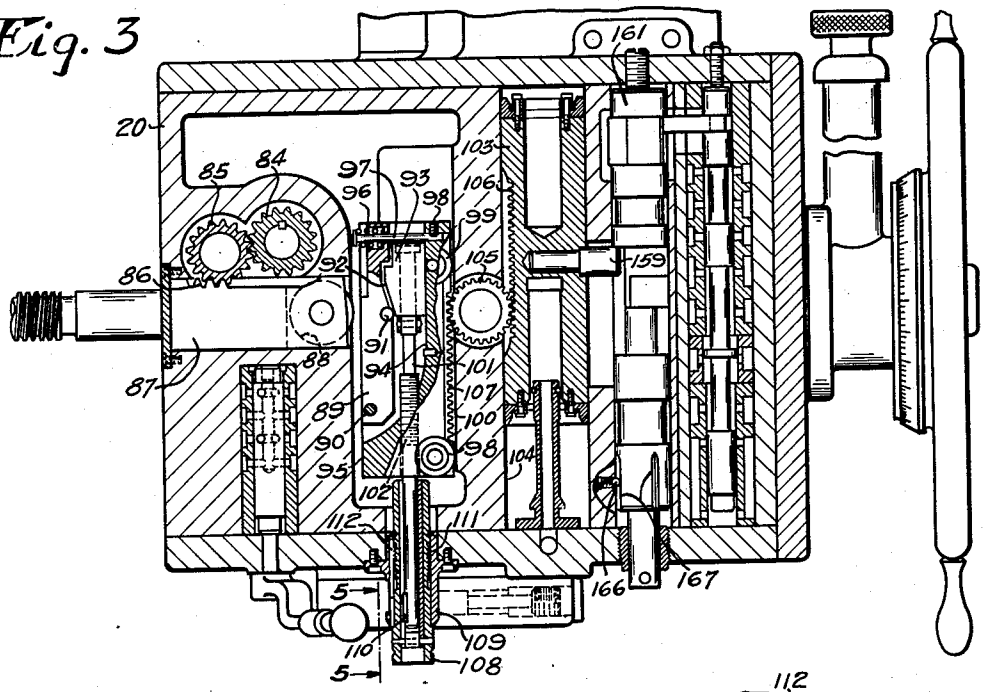


Fig. 4

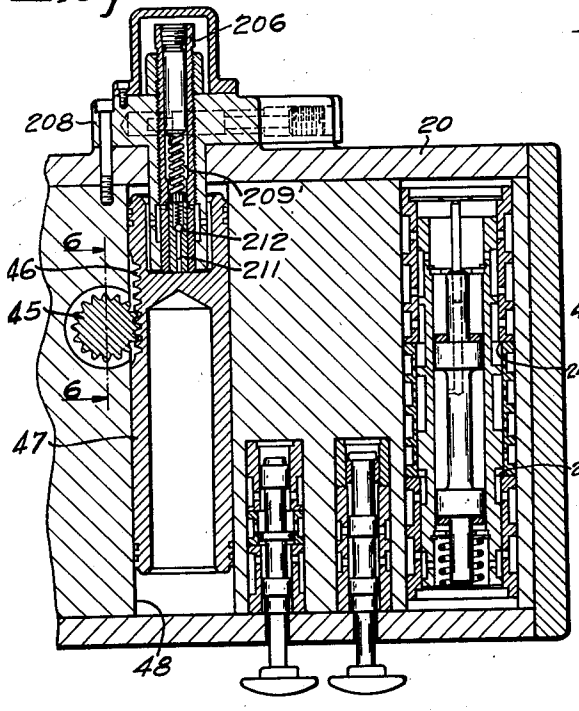


Fig. 5

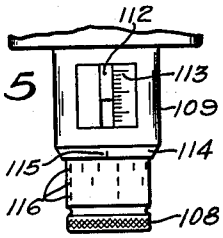
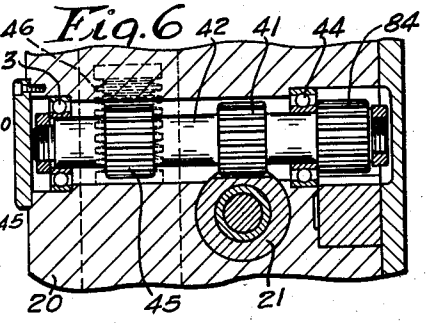


Fig. 6



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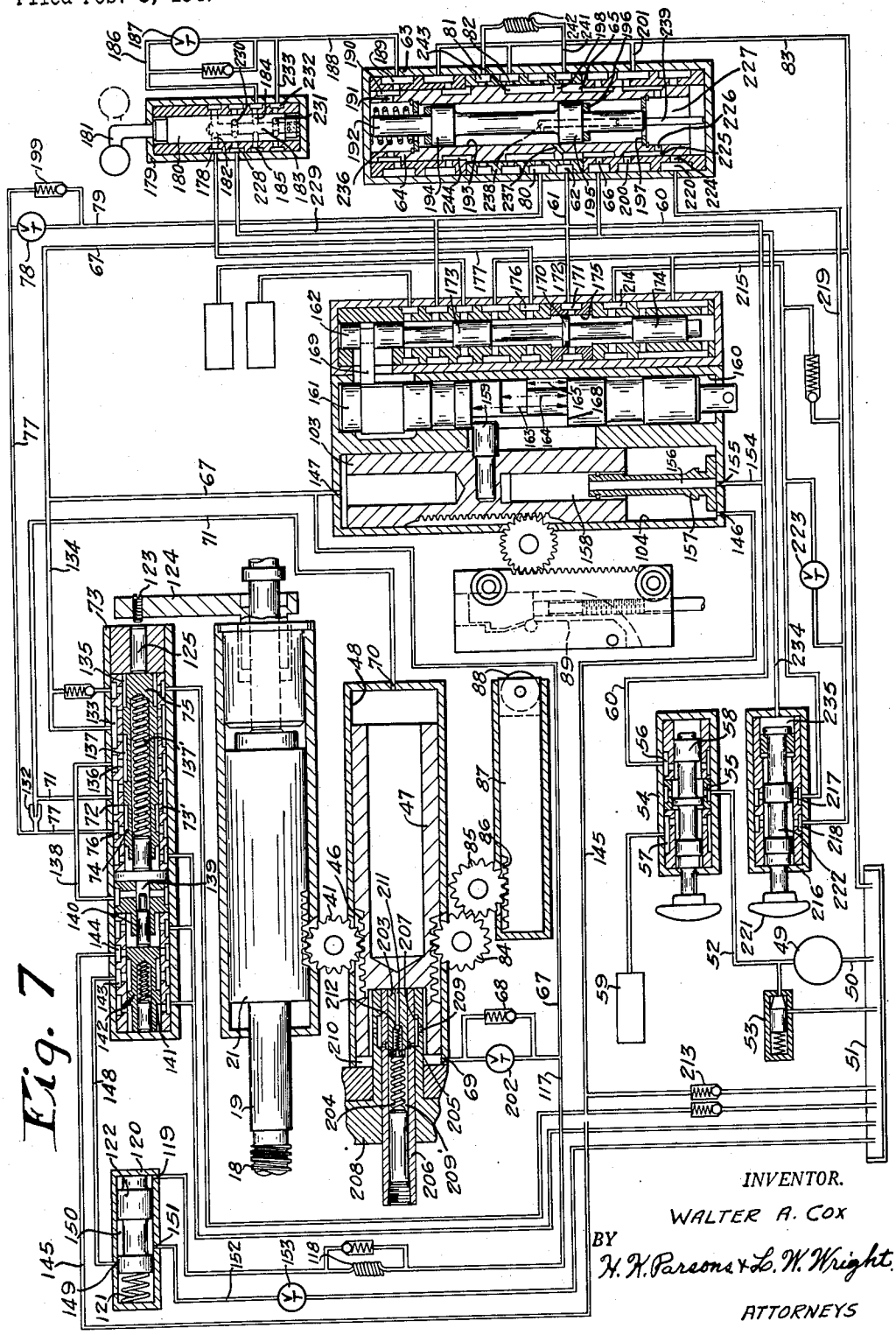


Fig. 7

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2,518,600

INFEED ATTACHMENT FOR GRINDING MACHINES

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Application February 8, 1947, Serial No. 727,436

9 Claims. (Cl. 51—2)

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This invention relates to machine tools and more particularly to improvements in grinding machines.

One of the objects of this invention is to provide an improved automatic infeed cycle control mechanism for a grinding machine.

Another object of this invention is to provide an improved mechanism of the character described which has greater accuracy in sizing the finished work piece over existing mechanisms.

Another object of this invention is to provide a mechanism of the character described which can be built as a compact unit for attachment to existing machines.

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 is an elevation of a machine tool embodying the principles of this invention.

Figure 2 is a section through the mechanism as viewed on the line 2—2 of Figure 1.

Figure 3 is a section on the line 3—3 of Figure 1.

Figure 4 is a section on the line 4—4 of Figure 1.

Figure 5 is a detail view of the feed rate control micrometer as viewed on the line 5—5 of Figure 3.

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

Figure 7 is a diagram of the hydraulic control circuit.

Referring to the drawings and more particularly to Figure 1 the reference numeral 10 indicates the bed of a machine tool in the form of a grinding machine which for explanatory purposes has been illustrated as a centerless grinder. Such a machine has a pair of opposed grinding wheels, the wheel 11 serving the purpose of a grinding wheel, and the wheel 12 acting as a regulating wheel. Interposed between these wheels is a work support 13 which carries a work rest blade 14 upon which a work piece, indicated by reference numeral 15, is supported during a grinding operation thereupon. The regulating wheel 12 is supported for rotation on a slide 16 which is reciprocably mounted on a second slide 17 which, in turn, is reciprocably mounted upon the bed 10. The slide 17 serves to adjust the

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work support 13 relative to the grinding wheel and, after this has been locked in position, the slide 16 is moved relative to the slide 17 to adjust the regulating wheel 12 toward and from the work support 13 as well as toward and from the grinding wheel whereby the size of the work depends upon how far the regulating wheel is moved toward the grinding wheel.

The slide 16 may be advanced and retracted by means of a feed screw 18 which is threaded in a suitable nut member not shown in the slide 16, and this screw has a reduced or shaft-like portion 19 which extends through a feed rate control box 20 that is mounted on the end of the bed 10. This box contains the mechanism forming the subject matter of this invention. As shown in Figure 2, the portion 19 is supported in a sleeve 21 by thrust bearings 22, the sleeve being supported in a bore 23 formed in the box but keyed at 24 to a spline groove 25 formed in the bore. The result is that the sleeve 21 is subject to axial movement in the bore but held against rotation, while the portion 19 is subject to free rotation relative to the sleeve 21 and also responsive to axial movement of the sleeve 21. In addition, the shaft portion 19 has a reduced splined end 26 by which it is operatively connected to a sleeve 27 which is supported for rotation in a bracket 28 attached to the end of the box. The sleeve 27 has a wheel 29 secured thereto whereby rotation of the hand wheel will affect rotation of the screw 18.

The sleeve 27 may be clamped against rotation by means of a ring 30 which surrounds the sleeve and on one side has a threaded bore 31 in which is threaded a clamping member 32 that carries a shoe 33 for engagement with the periphery of the sleeve whereby, upon rotation of the member 32, the shoe 33 is pulled into clamping engagement with the sleeve. The member 32 is supported at its outer end by a spherical bearing 34 mounted in the end of a tubular portion 35 of the bracket 28. A suitable operating knob 36 is secured to the end of the member 32 for imparting rotation thereto.

The hand wheel 29 is utilized for effecting manual rotation of the screw 18 for set-up and adjusting purposes and also for manual controlled grinding operations, and to facilitate this control the bracket 28 has a disc 37 secured thereto, as by screws 38, and this disc has a suitable index mark or reference point for cooperation with graduation marks formed on the beveled face 39 of the hand wheel 29.

Advance and retraction of the wheel 12 is ef-

fected by power through the sleeve 21 which is provided with rack teeth 40 engaging with a pinion 41. It will now be obvious that rotation of the pinion 41 will effect axial movement of the screw 18 without rotation to thereby cause sliding movement of the support 16. As shown in Figure 6, the pinion 41 is integral with a shaft 42 which is journaled at 43 and 44 in the housing 20, and this shaft also carries a second integral pinion 45 which meshes with rack teeth 46, Figure 4, formed on a piston member 47 that is reciprocable in a cylindrical bore 48 formed in the housing 20. It should now be obvious that by introducing pressure to one end of the cylinder 48 and connecting the other end to exhaust that power reciprocation of the piston 47 and thereby power movement of the shaft 19 shown in Figure 2 may be effected.

The automatic operation of the machine is controlled by the hydraulic control circuit shown in Figure 7 to which reference is now had and in that Figure the reference numeral 49 indicates a pump for supplying fluid pressure to the circuit. This pump has an intake 50 through which oil is withdrawn from a reservoir 51 and delivered under pressure into line 52 to which is branch connected a relief valve 53 for determining the high limit of pressure in the system. A selector valve 54 has a pressure port 55 to which the line 52 is connected, and a pair of ports 56 and 57 to which the port 55 is selectively connected by the valve plunger 58. The port 57 leads to a hydraulic truing circuit indicated generally by the reference numeral 59, but the plunger 58 is normally in the position shown whereby the port 55 is connected to port 56 and channel 60. The channel 60 has a branch connection 61 to pressure port 62 of a reversing valve indicated generally by the reference numeral 63.

This valve has an axially movable sleeve 64 in which is formed an annular groove 65 which serves to connect the port 62 with a port 66 when the sleeve is in the position shown. The port 66 is connected by channel 67 through a check valve 68 to port 69 in one end of the cylinder 48. It will now be obvious that with the parts in the position shown that the piston 47 will move toward the right at a rapid traverse rate.

Fluid will exhaust from the other end of cylinder 48 through port 70 and channel 71 to port 72 of a pilot control valve 73. The port 72 which is in the form of an annular groove has a radially extending port 73' which is in communication with an annular groove 74 formed in an axially reciprocable sleeve 75. In the position shown the annular groove 74 connects the port 73 with a port 76 to which is connected a channel 77 which leads to a throttle valve 78. This throttle valve may be adjusted to regulate the rapid traverse rate of advance of the regulating wheel toward the work, the fluid exhausting from the throttle valve through channel 79 to port 80 of the reversing valve. The port 80 is connected by an annular groove 81 formed in the valve sleeve 64 to port 82 and exhaust channel 83 which leads back to the reservoir 51.

As the piston 47 advances it also causes rotation of a gear 84 which as shown in Figure 6 is integrally formed with the shaft 42. The pinion 84 is connected through an idler pinion 85, Figure 3, to rack teeth 86 formed on the periphery of a plunger 87 which carries a follower roller 88. Thus, as the piston 47 advances toward the right as viewed in Figure 7 it causes the

plunger 87 to advance and move the roller 88 into engagement with a feed rate control cam 89 which is more clearly shown in Figure 3. This cam is pivoted on a pin 90 and carries a shoe 91 in the form of a pin which is adapted to ride on a cam surface 92 of a feed rate determinator 93.

The feed rate determinator is slidable on a surface 94 which is formed in the cam carrier 95 at right angles to the direction of movement of the plunger 87. This carrier also supports the pin 90. It will be obvious that by longitudinally shifting the member 93 on its supporting surface 94 that the inclined face 92 will rotate the cam member 89 in a counterclockwise direction as viewed in Figure 3 about its pivot pin 90 and thereby present an inclined surface to the roller 88. The cam member 89 is held in contact with the inclined surface 92 by means of a spring 96 which surrounds a headed pin 97, this pin being secured in the carrier 95 by a set screw 98. Thus, as the cam 93 is shifted upward as viewed in Figure 3 the cam 89 will be caused to move clockwise about its pin 90 and follow up any drop in the surface 92. The cam carrier 95 is provided with a pair of rollers 98 and 99 which ride on the surface 100 formed on a fixed part of the bracket 20. It should now be evident that the surface 94 of the cam carrier 95 constitutes a fixed abutment normal to the direction of movement of the plunger 87.

The member 93 is adjusted downward to incline the cam 89 by threading an actuating rod 101 at 102 in the carrier 95 whereby rotation of the rod 101 will cause the member 93 to be adjusted relative to its supporting member 95. During the actuation of the machine the member 95 is fed upward as viewed in Figure 3 by a feed piston 103 which is slidable in a cylinder 104, the piston being connected to the carrier 95 by means of a pinion 105 which is interposed between the parts and which interengages with rack teeth 106 formed on the piston 103 and rack teeth 107 formed on the carrier 95.

The rod 101 is adjustable from the outside of the box 20 by means of a micrometer mechanism including a sleeve 108 which is rotatable in a bracket 109 that is secured to the box 20. The sleeve 108 has a splined connection at 110 with the rod 101 whereby rotation of the sleeve will effect adjustment of the member 93 relative to its support 95 but upon movement of the support 95 upwardly as viewed in Figure 3 the rod 101 will slide relative to the bore of the sleeve 108. This sleeve also has a threaded exterior portion 111 which meshes with a slider or indicator 112 whereby upon rotation of the sleeve the indicator 112 will be moved longitudinally as viewed in Figure 5 relative to some graduations 113 which are formed on the fixed bracket 109. In addition, the fixed bracket has a beveled end surface 114 upon which is a reference mark 115 and the periphery of the sleeve 108 is provided with a series of graduation marks 116 similar to the barrel on a micrometer to facilitate adjustment and setting of the cam 89.

The cam carrier 95 is shown in its starting position in Figure 3, and it should now be obvious that with the cam member 89 inclined at some angle that the cam will act as a positive stop to limit the rapid traverse movement of the wheel head as effected by the piston 47. When this occurs the pressure in the supply line 67 to the port 69 of the rapid traverse cylinder 48 will naturally rise, and this increase in pres-

sure will be transmitted through the branch line 117, Figure 7, and resistance 118 to port 119 of a time delay valve 120. The rise in pressure will overcome the resistance of a spring 121 and move the valve plunger 122 to the left as viewed in Figure 7.

Simultaneously with the engagement of the cam roller 88 with the cam 89 a set screw 123, Figure 2, is so adjusted relative to its supporting arm 124 that it will engage a pin 125 and shift the pilot valve 75. The trip actuation of the arm 124 is more clearly shown in Figure 2 from which it will be seen that the arm 124 is slidably mounted on a reduced portion 126 of the member 19 and held against the end wall 127 of the box 20 by a spring 128. The portion 126 carries a ring 129 which is secured thereto by a pin 130 and as the member 126 moves toward the left as viewed in Figure 2 the ring 129 will engage the annular surface 131 on the arm 124 and shift the same toward the left against the resistance of spring 128. It will be noted that there is a certain amount of lost motion between the ring 129 and the surface 131 so that the arm 124 only has a limited movement and the actual timing of the shifting of the valve plunger 75 may be determined by adjusting the set screw 123.

When the valve 75 is shifted to the left, as viewed in Figure 7, it will shift the annular groove 74 sufficiently to the left to disconnect port 72 from port 76 and thereby close a by-pass which existed around a throttle valve 132.

This now causes the return oil from cylinder 48 to be forced through the throttle valve 132, thereby decelerating the rapid traverse movement of the wheel head to reduce the impact effect of the follower roller 88 when it engages the cam 89. The shifting of valve plunger 75 will also connect port 133, to which is connected a branch line 134 of the rapid traverse cylinder supply line 67, to an annular groove 135 whereby fluid pressure may flow behind the piston 75 and shift it the full length of its stroke independent of the pin 125. This will also disconnect the port 133 from a port 136 to which it was connected by the annular groove 137. When the ports 133 and 136 are connected, fluid pressure flows through the line 138 to a cylindrical space 139 and moves plunger 140 against the resistance of a strong spring 141 to hold a valve element 142 in the position in which it is shown in Figure 7, but upon disconnection of these ports the port 136 becomes connected to port 72 and thereby line 71 which is at lower pressure than the pump pressure in channel 134 whereby the spring 141 is capable of shifting the valve plunger 142 against the pressure in the cylindrical space 139 resulting in the interconnection of a pair of ports 143 and 144.

The port 144 is connected by a channel 145 to a port 146 located in one end of the feed cylinder 104. As previously mentioned, this cylinder contains a piston 103 and the other end of this cylinder has a port 147 which is connected to a branch from line 67 which is under pressure. Therefore, there is an urge on the piston 103 to move downward as viewed in Figure 7, but the line 145 from port 146 located at the other end of the cylinder has been blocked at port 144. By the shifting of the valve member 142 the port 144 becomes connected to port 143 whereby the fluid in the lower end of cylinder 104 may now pass through line 148 to port 149 of the interlock valve 122. As previously explained, the rise in pressure in line 67, caused by

the follower roller 88 engaging its cam 89, will cause shifting of the valve plunger 122, resulting in port 149 being connected by annular groove 150 in the valve member to port 151 and channel 152. This channel has a feed rate throttle valve 153 serially connected therein so that by variably throttling the oil the feeding rate of the feeding piston 103 can be determined. Thus, the shifting of the valve piston 122 institutes the feeding movement of the wheel 12.

Attention is invited to the fact that the pump pressure supply line 60 has a branch connection 154 to a second port 155 located in the end of cylinder 104, and this port communicates with an axial bore 156 formed in a member 157 which is mounted within the cylinder 104 and has a telescoping connection with a drilled hole 158 formed in the end of the piston 103. This means that pump pressure continuously exists in the space 158 exerting a continuous urge on the piston 103 in an upward direction as viewed in Figure 7, but the pump pressure entering the port 147 has a larger area of the piston 103 to work on and therefore will force part of the oil in space 158 back into the supply channel 60 during the feeding movement.

It will thus be seen in the cycle of operation so far that a rapid traverse movement was first instituted, followed by a deceleration of the rapid traverse movement at the time of engagement with the feed cam, and that the pressure still existing in the cylinder 48 also effected the feeding movement but the rate of the feeding movement is controlled by the rate of movement of the cam. The length of the feeding movement is determined by the collar 22 on the sleeve 21 engaging the positive stop 22' which is the closure member for the end of the bore 23. Thus the infeed movement always stops by engagement of these parts and if the size of the work is not correct, adjustment is made by the hand wheel 29 rotating the screw 18.

This method of stopping is also independent of the feed cam, in that the feed cam can still keep on going, and therefore has nothing to do with the final size of the work. The continuation of its movement, however, serves to operate the reversing mechanism to effect retraction of the wheel by means of the following mechanism.

The piston 103 carries a pin 159 which extends into an adjacent bore 160 in which is slidably mounted a trip member 161 for the reversing valve pilot plunger 162. The member 161 has three flats formed in the periphery thereof, one of which has a length as indicated by the line 163, the second of which has a length as indicated by the line 164, and the third of which has a length as indicated by the line 165. The member 161 may be rotated to present any one of these three flats to the pin 159 and is adapted to be held in any one of these positions by a spring pressed detent 166 as shown in Figure 3 and engageable with longitudinally extending detent grooves indicated generally by the reference numeral 167.

These different flats provide different lengths of lost motion between the member 161 and the pin 159. It will be noted that the shoulder 168 is the common end wall for all of these flats and is the one which determines the limit of the infeeding movement. The amount that the wheel returns is determined by the length of the flat which is effectively presented to the pin 159. It will thus be seen that three different lengths of feeding strokes may be selected.

When the pin 159 engages the wall 168 during a feeding movement it shifts the plunger 161 and thereby, through its connecting arm 169, the valve plunger 162 and more particularly the spool 170 on the valve plunger relative to the pressure port 171 which is supplied through branch line 172 from the pump pressure supply line 60. It will be noted that the spool 170 is larger in diameter than the adjacent spools 173 and 174 which are formed on the valve and that the spool 170 reciprocates in a bore 175 which is larger in diameter than the bores in which the spools 173 and 174 slide, and therefore as the spool 170 passes to the other side of the pressure port 171 a hydraulic detenting action will take place to complete the shifting of the valve plunger to the end of the enlarged bore 175.

This will result in the connection of pressure from port 171 to port 176 and channel 177 to port 178 of a run and return valve 179. This valve has a rotatable valve spool 180 which may be manually rotated by a crank lever 181 secured to the end of the plunger. When the valve plunger 180 is in the position shown, the port 178 is connected to a radial port 182 which communicates with an axial bore 183 formed in the plunger which has a second radial passage 184 which at this time is in communication with a port 185. The port 185 has a channel 186 connected to it which leads to a throttle valve 187 for throttling the flow through line 188 to the end port 189 of the reversing valve cylinder 63. The purpose of the throttle valve 187 is to establish a predetermined delay in the shifting of the reversing valve sleeve 64.

The port 189 is connected through an annular groove 190 and radial port 191 to the interior of the sleeve 64 for the purpose of shifting a piston plunger 192 which is slidably mounted in a bore 193 formed in the valve sleeve 64. This plunger has a pair of spools 194 and 195 formed thereon and the plunger is capable of lost motion movement with respect to the sleeve. Thus, upon admittance of pressure through port 191 the plunger 192 will be shifted relative to the sleeve.

The reversing operation was set into motion at this time, but the actual shifting of the reversing valve sleeve 64 cannot be effected until the piston 192 has taken up its lost motion with respect to the sleeve 64, the lost motion at this time being the distance between the shoulder 196 on the piston 192 and the shoulder 197 formed by a disc which is integrally connected with the sleeve. Upon engagement with this shoulder the sleeve 64 is shifted, causing the spool 198 formed on the sleeve to be moved past the pressure port 62, thereby connecting this port to port 80 and at the same time disconnecting port 80 from the exhaust port 82. Fluid pressure will now flow through line 79, bypassing the throttle valve 78 through check valve 199 and channel 77 to port 76 of valve 73.

At the same time, port 66 of the reversing valve is connected to port 200 which has the return channel 201 connected thereto. This relieves the pressure in channel 67 and branch line 134 of valve 73 whereby the fluid in the end of valve 73 may be forced out to reservoir by the spring 137', shifting the valve plunger 75 back to its normal position in which it is shown in Figure 7. This results in the line 77, now under pressure, being connected through the annular groove 74 directly to port 72 and line 71 whereby the fluid pressure will enter the cylinder 48 and return the rapid traverse piston 47 to its starting position.

This will simultaneously cause retraction of the follower roller 88 from its control cam 89 through the rack and gear connection. To prevent the return movement of piston 47 from being at too fast a rate, a throttle valve 202 is connected in parallel with the check valve 68 so that as the check valve is closed by the returning fluid, the returning fluid is forced to pass through the valve 202. In addition, the end of the cylinder 48 is provided with a dashpot mechanism comprising a central plunger 203 which is caused at all times to follow up the movement of piston 47 by means of a spring 204. It will be obvious then that as the member 203 moved forward it uncovered a port 205 in the sleeve member 206.

The port 205 is in continuous communication with a wide internal annular groove 207 formed in the bracket 208 secured to the outside wall of the box 20, and this groove is connected by radial ports 209 to the end space 210 of cylinder 48. The member 206 is threaded in the bracket 208 as shown in Figure 4 for adjustment relative thereto whereby the relative position of the port 205 may be adjusted longitudinally to determine the amount of return of the piston 46, or in other words, to determine its starting position for the advance stroke. As the plunger 203 is moved to the left by the returning piston 46 it throttles the escape of fluid through the port 205 and thereby builds up a hydraulic cushion to decelerate the movement of the piston 47.

Should the port 205 become closed to such an extent that upon advancing movement of the piston 47 the oil could not return fast enough through the throttled port 205 to the spring space 209' the end of the piston 47 would momentarily leave the end of the plunger 203, in which event the pressure fluid entering the space 210 would be able to pass through an axial bore 211 formed in the end of the plunger 203 and force itself by the check valve 212, thereby forming an auxiliary supply to the cylindrical space 209'.

As previously recited the channel 67 which has a branch connection 147 to the feed piston 103 was connected to reservoir on the shifting of the reversing valve, and this relieves the pressure on the upper end of piston 103 as viewed in Figure 7 whereby the constant pressure which exists in the space 158 becomes effective to return the feed piston 103 in an upward direction as viewed in Figure 7. Since the line 145 leading to the lower end of the cylinder might be closed a suction check valve 213 is branch connected between the reservoir 51 and the line 145 whereby if the returning movement of the piston 103 is at such a rate that it might tend to cause a vacuum in the lower end of the cylinder 104 oil from the reservoir may be drawn through the check valve 213 to prevent this condition.

Return movement of the piston 103 will cause shifting of the trip member 161 and connected pilot valve 162 whereby the pressure port 171 of the pilot valve will be connected to port 214 and line 215 to the cycle start valve 216 having a port 217 which forms the terminus of the line 215. This port is normally disconnected from a port 218 which is connected by line 219 to port 220 located in the end of the reversing valve 63. It will thus be seen that the reversing valve will not be shifted to a forward position to start a new cycle until the cycle start valve plunger 221 is manually moved to the right as viewed in Figure 7 so that the annular groove 222 can interconnect the ports 217 and 218. Thus, the cycle will automatically stop so that the operator may manually load a new work piece in the machine.

However, if the machine has an automatic loading attachment and the time can be determined that it takes to load a work piece in the machine a delay throttle valve 223 is interconnected to lines 215 and 219 in parallel relation to the cycle start valve whereby the valve 223 may be adjusted to produce a predetermined timed delay before the reversing valve is shifted to start a new cycle.

In any case, the sleeve 64 in the reversing valve will be in its down position as viewed in Figure 7, and when fluid pressure does enter port 220 it will pass through the radial ports 224 into the annular groove 225 formed in the periphery of sleeve 64 and which at this time is in registry with the ports 224. By means of the radial passages 226 the fluid pressure will flow from the groove 225 into the space 227 and cause upward movement of piston 63 which, in turn, will cause upward shifting of the reversing valve sleeve 64 to its forward running position.

At any time during the operation of the machine and especially in case of an emergency the valve 181 may be actuated to cause immediate separation of the grinding wheels. As previously noted, the valve 181 is shown in its run position but upon rotation of the valve plunger 180 to its retracted position a port 228 which is continuously connected by a line 229 directly to the pump pressure will be connected through radial grooves 230 and the axial bore 183 to radial ports 231 which will now be in registry with ports 232 which communicate with a line 233 which is connected in parallel across the throttle valve 187, thus forming a direct connection for the admittance of pump pressure to line 188 and shifting of the reverse valve to its retract position.

Attention is invited to the fact that when the cycle control valve 216 is utilized to start the cycle that there is a branch line 234 from the main line 67 which supplies fluid pressure to the rapid traverse and feed cylinders so that immediately upon this line becoming connected to pressure that some of the fluid pressure is admitted to chamber 235 of the cycle start valve 216 to shift valve plunger 221 automatically to its stop position.

It should also be noted that the reversing valve has a spring 236 mounted in one end which acts on one end of the valve sleeve to hold it in its down position as viewed in Figure 7. When the piston member 192 moves the sleeve up sufficiently to connect pressure port 62 to annular groove 65, the fluid pressure flows through a drilled hole 237 in member 192 to an interdrilled passage 238, in the end of which is slidably mounted a piston pin 239. The hydraulic pressure acting on the pin serves to hold the piston member in its upward shifted position especially when reversal takes place and the lower end of the valve is connected to reservoir.

Also, when port 62 is connected to groove 65, a differential hydraulic pressure is created to hold the sleeve up. This is accomplished by making the spool 198 larger in diameter than the adjacent spool, and having it slide in a larger bore, thus creating a shoulder area 240 which is of sufficient area to hold the sleeve up against the resistance of the spring and independent of the piston member 192.

When the piston member moves the sleeve down, hydraulic pressure is connected from the port 62 through channel 241 and resistance 242 to port 243 which becomes connected to groove 244, and thus by means of shoulder 245 shown in Figure 4 a hydraulic detenting pressure is created to shift the sleeve to the end of its stroke.

There has thus been provided an improved infeeding mechanism which is very efficient in operation, economical to manufacture, and extremely accurate in sizing work.

What is claimed is:

1. In a grinding machine having a shiftable wheel support, the combination of an infeeding mechanism for said support including a first fluid operable device for advancing said support at a rapid traverse rate, means for throttling said device to cause movement at a slower rate, a second fluid operable device for controlling subsequent movement at a feed rate, a pilot valve operable by said second device, and a reversing valve controlled by the pilot valve for reversing the direction of movement of both of said devices to effect retraction of the wheel support.

2. In an infeeding mechanism for a grinding machine having a wheel support, the combination of a feed screw threaded in the support, a sleeve mounted on the feed screw and interconnected therewith for joint axial movement, independent means for slidably supporting the sleeve and holding the same against rotation, a follower plunger, a fluid reciprocable member, rack and gear connections between said member and said sleeve and said plunger for imparting simultaneous movement thereto, a feed rate control cam arranged transversely of the path of movement of said plunger, power operable means for feeding said cam for controlling the feeding rate of said sleeve and screw effected by said fluid operable member, a positive stop engageable by said sleeve for limiting the advancing movement of said screw, and trip operable means for reversing actuation of the moving parts to a starting position.

3. In a grinding machine having a wheel support, the combination of power operable means for advancing and retracting said support, a source of pressure for said power operable means, a reversing valve for connecting said source of pressure with said power operable means including a shiftable valve sleeve, a piston member slidable within said sleeve, a pilot valve operable by said power operable means for connecting pressure to said piston member, a lost motion connection between said piston member and sleeve, and a throttle valve interposed between said pilot valve and said piston member for regulating the movement of said piston member in taking up said lost motion to thereby control the delay between actuation of the pilot valve and the shifting of said valve sleeve.

4. In a grinding machine having a wheel support, the combination of an infeed cycle control mechanism for said support, including fluid operable means for reciprocating said support, a source of fluid pressure, a reversing valve for selectively connecting said pressure for reverse operation of said fluid operable means, an automatically operable pilot valve for connecting said pressure to effect shifting of said reversing valve, a cycle control valve normally positioned for blocking the flow from the pilot valve to one end of said reversing valve to prevent shifting of the reversing valve to its support advancing position, said cycle control valve being manually shiftable to effect the interconnection to cause shifting of the reversing valve, and means responsive to subsequent shifting of the reversing valve to automatically effect shifting of the cycle control valve to its blocking position.

5. In a grinding machine having a wheel slide, the combination of an infeed cycle control mechanism for said slide including fluid operable means for advancing and retracting said slide, a source of fluid pressure, a reversing valve for connecting said source of pressure for reverse operation of said fluid operable means, a trip operable pilot valve responsive to movement of the parts for connecting fluid pressure to one end of the reversing valve to effect shifting of the valve to a slide retracting position, and means serially arranged in said last-named connection to connect the source of pressure directly to said reversing valve independent of the pilot valve to effect shifting of the reversing valve at will to its slide retracting position.

6. In a grinding machine having a wheel support, the combination of an infeeding mechanism for said support comprising fluid operable means for shifting said support, a source of pressure, a feed control cam for retarding the movement of said support by said fluid operable means to a predetermined feed rate, a fluid operable piston for actuating said cam, means connecting said source of pressure for exerting a constant urge on said piston toward its starting position, valve means for connecting said source of pressure to cause advancing movement of said piston simultaneously with the connection of pressure to said fluid operable means, a channel for conducting exhaust fluid away from the other end of said piston, serially arranged blocking valves in said channel, means responsive to a predetermined advance of said support for releasing one of said blocking valves, a positive stop for limiting the movement to be effected by said fluid operable means, and means responsive to engagement of said positive stop for releasing the second blocking valve.

7. In a grinding machine having a wheel support, the combination of an infeeding mechanism for said support comprising fluid operable means for shifting said support, a source of pressure, a feed control cam for retarding the movement of said support by said fluid operable means to a predetermined feed rate, a fluid operable piston for actuating said cam, means connecting said source of pressure for exerting a constant urge on said piston toward its starting position, valve means for connecting said source of pressure to cause advancing movement of said piston simultaneously with the connection of pressure to said fluid operable means, a channel for conducting exhaust fluid away from the other end of said piston, serially arranged blocking valves in said channel, means responsive to a predetermined advance of said support for releasing one of said blocking valves, a positive stop for limiting the movement to be effected by said fluid operable means, means responsive to engagement of said positive stop for releasing the second blocking valve, and a throttle valve in said channel for controlling the rate of movement of said piston.

8. In a grinding machine having a wheel support, the combination of an infeed cycle control mechanism for said support including a fluid shiftable piston, motion transmitting connections between said piston and said support, a source of fluid pressure, means to connect said source of fluid pressure to one end of said piston to effect advancing movement of said support, serially arranged throttle valves in the return connection from the other end of said piston and sequentially actuable to produce successively different rates of advancement of the grinding wheel support, means to reverse the connections to said piston and simultaneously by-pass said serially arranged throttles, an additional throttle automatically rendered effective upon reversal of flow to said cylinder to control the rate of support retraction.

9. In a grinding machine having a wheel support, the combination of an infeed cycle control mechanism therefor, including a fluid operable piston for advancing and retracting said support, a source of pressure, a reversing valve for connecting said source of pressure to said piston, said reversing valve having a shiftable sleeve valve member and an internal piston member having a lost motion connection with said sleeve, a spring normally maintaining said sleeve in a support retracting position, said piston being independently actuable to effect shifting of said sleeve, independent means for maintaining said piston in its shifted position, and hydraulic means for maintaining the sleeve in its shifted position.

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