Jan. 9, 1934. J. K. DOUGLAS 1,943,061

HYDRAULIC DRIVE FOR MACHINE TOOLS

Filed Oct. 21, 1929 2 Sheets-Sheet 1

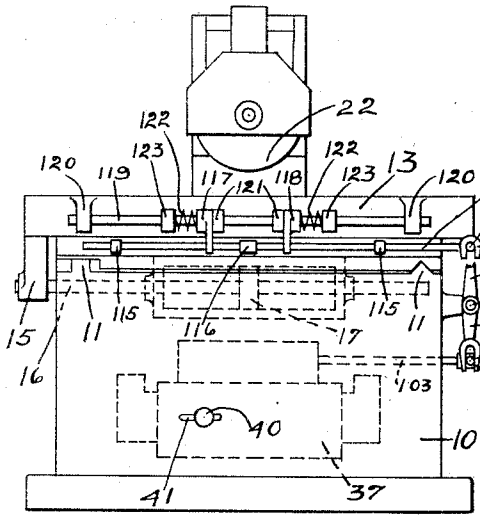


Fig. 1.

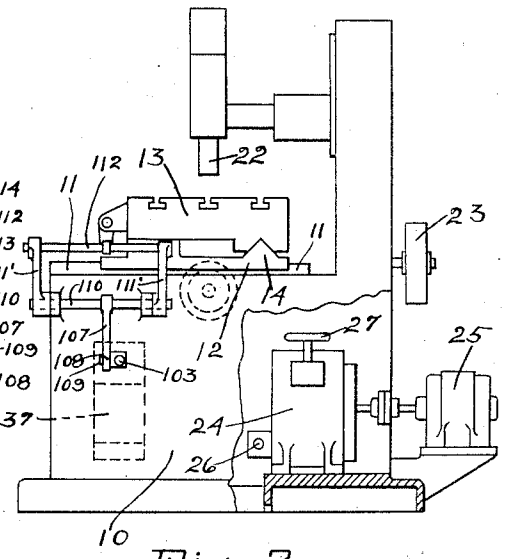


Fig. 2.

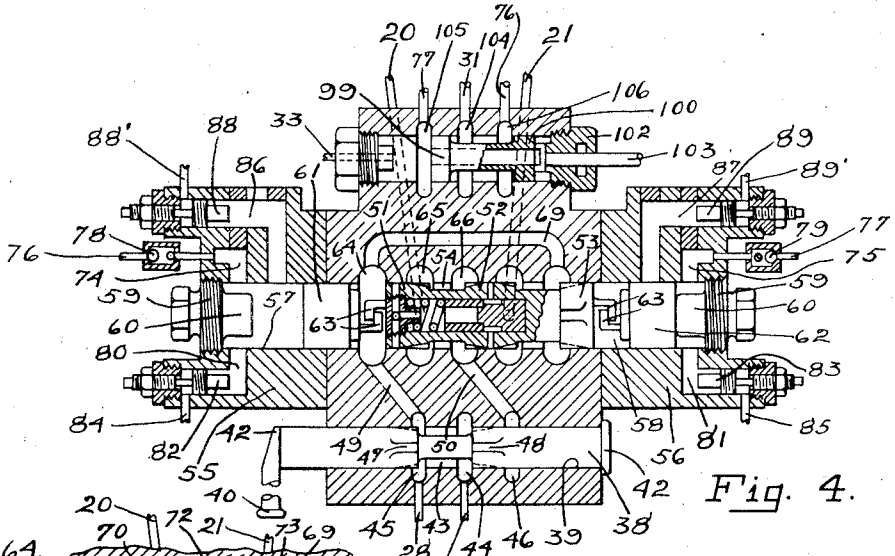


Fig. 4.

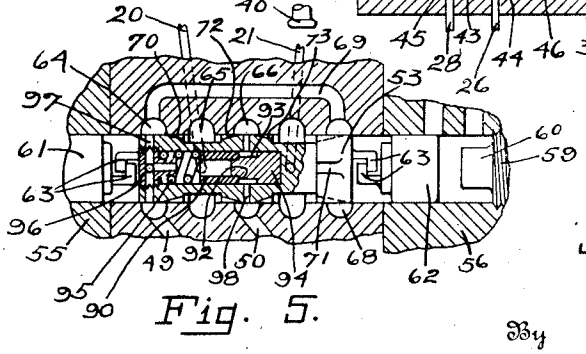


Fig. 5.

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HYDRAULIC DRIVE FOR MACHINE TOOLS

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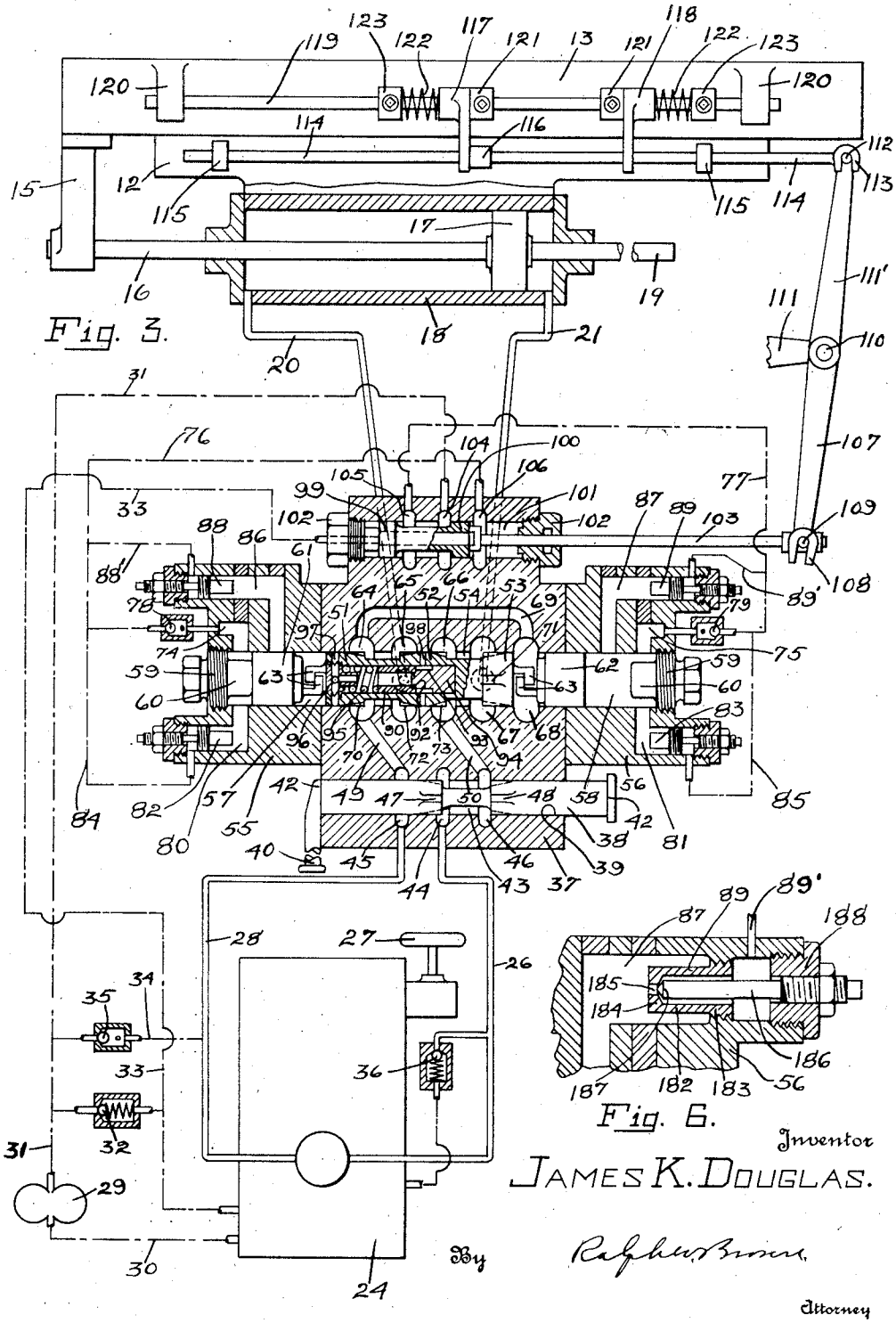


Fig. 5.

Fig. 6.

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UNITED STATES PATENT OFFICE

1,943,061

HYDRAULIC DRIVE FOR MACHINE TOOLS

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5 Claims. (Cl. 60—52)

This invention relates to hydraulic drives of the continuously reciprocating type such as are ordinarily employed for operating grinder tables although the invention is applicable as well to various other machines such as planers, shapers, and the like.

Present commercial forms of hydraulically operated grinders are driven by constant displacement gear pumps of a well known type under the control of a reversing valve which ordinarily functions automatically to effect reversal at the end of each stroke. Due to the low pressures involved and the hydraulic slip inherent in pumps of that type, each reversal, though smooth, is sluggish and the driven part fails to reassume its normal operating speed with the promptness desired, entailing a considerable loss of time. Furthermore it is impossible with pumps of that type to maintain an operating speed with any degree of accuracy and impracticable to regulate the speed to suit the various working requirements.

One object of the present invention is the provision of an improved hydraulic drive for the purposes mentioned capable of effecting a smooth and prompt reversal of the driven reciprocating part at the end of each stroke. This I accomplish by the use of a positive delivery pump and a reversing valve which functions automatically to regulate the rate of deceleration and acceleration during each reversal. The pump employed is preferably of the variable displacement type in order to provide an accurate regulation of the rate of movement of the driven part.

Another object is the provision in a valve controlled hydraulic circuit of means for regulating the rate of movement of the valve to thereby regulate the rate of reversal of the hydraulic circuit controlled thereby.

Another object is the provision in a reversible hydraulic circuit of means automatically effective during periods of reversal only to reduce and predetermine the hydraulic pressures available for acceleration.

Other objects and advantages will appear from the following description of an illustrative embodiment of the present invention.

In the accompanying drawings:—

Figure 1 is a front elevation of a grinder equipped with a hydraulic drive embodying the present invention.

Fig. 2 is an end elevation of the grinder.

Fig. 3 is a diagram of the hydraulic circuit.

Fig. 4 is a sectional view of the valve assembly shown in Fig. 3 with the parts thereof in different positions.

Fig. 5 is a sectional view of the reversing valve shown in Figs. 3 and 4 illustrating a third position thereof.

Fig. 6 is a sectional view on a larger scale of the choke device shown in Figs. 3 and 4.

The grinder shown is of a well known type. It has its mechanism carried by a frame 10 provided with horizontal ways 11 upon which a carriage 12 is mounted for adjustment forwardly and rearwardly relative to the frame. A table 13 is mounted for lengthwise reciprocation upon the customary longitudinal ways 14 formed on the carriage 12. In this instance a bracket 15 depending from one end of the table 13 is connected through a rod 16 with a piston 17 by which the table is reciprocated. The piston 17 reciprocates in a cylinder 18, fixed to the bottom of the carriage 12, and is provided with a tail rod 19 of a diameter equal to that of the rod 16 so that the piston is non-differential. The cylinder 18 is supplied with driving liquid, preferably oil, through pipes 20 and 21 connected to the opposite ends thereof and controlled by a reversing valve to be later described. The work is clamped to the table 13 in the usual manner and passes back and forth therewith beneath the rotating grinder element 22 during reciprocation of the table. The grinder element 22 is mechanically driven from a pulley 23 in a well known manner by mechanism which forms no part of the present invention.

The pipes 20 and 21 constitute two sides of a hydraulic circuit which is supplied with liquid from a positive delivery, variable displacement pump 24 of a well known type. The pump shown is fully described in the patent to Walter Ferris No. 1,558,002. It is driven at constant speed by appropriate means such as an electric motor 25 and delivers liquid through pipe 26 at a rate corresponding to the setting of an appropriate control element such as a hand wheel 27. Liquid returns to the pump through a pipe 28. A gear pump 29 of a well known type, ordinarily contained in the pump housing, receives liquid through a pipe 30 from a reservoir provided in the base of the pump housing, and discharges into a pipe 31. A low pressure relief valve 32 connected to the pipe 31 determines gear pump pressure. This valve discharges into a drain pipe 33 which leads back to the reservoir in the pump housing. Pipe 31 also connects with the return pipe 28 through a pipe 34 and check valve 35 by which the return side of the main circuit

is maintained flooded with liquid at low pressure. A high pressure relief valve 36, connected to the delivery pipe 26, protects the pump and main circuit against excessive pressures.

Pipes 26 and 28 lead to and from a manually controlled starting valve which in this instance is contained in a valve block 37 mounted within the frame 10 of the machine. The starting valve shown comprises a plunger 38 closely fitted for lengthwise movement within a bore 39 formed in the lower portion of the block 37 and equipped with an operating handle 40 which projects through a slot 41 in the front wall of the frame 10. Stop flanges 42 on the opposite ends of the valve plunger limit the movement thereof. The plunger 38 is reduced intermediate its ends to provide an annular chamber 43 which is open at all times to the delivery pipe 26 through an annular groove 44 formed in the bore 39. Two additional annular grooves 45 and 46, formed in the bore 39 at opposite sides of the groove 44, are arranged to communicate alternately with the chamber 43 in the two extreme positions of the valve plunger. Two series of tapered grooves 47 and 48, formed on the valve plunger 38 and communicating with the opposite ends of the chamber 43, cooperate with the grooves 45 and 46, respectively, to effect a graduated flow of liquid through this valve as the plunger 38 approaches one extreme position or the other. Groove 45 is permanently connected to the return pipe 28, and passages 49 and 50, communicating with grooves 45 and 46, respectively, lead to a reversing valve to be later described.

The arrangement is such that when the valve plunger 38 is in the right extreme position of Fig. 3 groove 45 is blocked from groove 44 so that the entire discharge from the pump 24 through pipe 26 is directed through passage 50 to the reversing valve, and liquid returning through the reversing valve passes through passage 49, groove 45 and pipe 28 to the pump. When the valve plunger 38 is shifted into its left extreme position of Fig. 4 groove 46 is blocked and groove 45 is open to chamber 43, so that liquid discharged from the pump through pipe 26 passes back to the pump through groove 45 and pipe 28, so that the pump circuit is by-passed and the machine is at rest. During movement of the valve plunger 38 from its right extreme position the tapered grooves 48 cooperate with the groove 46 to gradually reduce the flow from chamber 43 into groove 46, and similarly during movement of the valve plunger from its left extreme position the tapered grooves 47 cooperate with the groove 45 to gradually reduce the flow from chamber 43 into groove 45.

The reversing valve shown comprises a plunger having three heads 51, 52, and 53 closely fitted for lengthwise movement within a bore 54 formed in the central portion of the valve block 37. Auxiliary blocks 55 and 56 fixed to the ends of the block 37 contain bores 57 and 58 which are aligned with bore 54 and form in effect extensions thereof. The outer ends of bores 57 and 58 are closed by screw plugs 59, each having an inwardly directed stop lug 60 thereon. Pistons 61 and 62 fitted in the bores 57 and 58 are flexibly connected through interlocking lugs 63 with the opposite ends of the valve plunger so as to move therewith. Five annular grooves 64, 65, 66, 67 and 68 are formed in the bore. Groove 64 communicates with the passage 49 and also with the other groove 68 through a passage 69; the intermediate groove 66 communicates with the passage 50; and the grooves 65 and 67 communicate with the pipes 20 and 21, respectively, leading to the opposite ends of the cylinder 18.

The arrangement is such that when the valve plunger is in the left extreme position of Fig. 3 liquid supplied from the pump through passage 50 is directed through pipe 21 into the right end of the cylinder 18 to drive the piston 17 and table 13 toward the left, liquid from the left end of the cylinder returning to the pump through pipe 20, grooves 65 and 64, passage 49, and pipe 28; and when the valve plunger is shifted into the right extreme position of Fig. 4 liquid delivered through passage 50 is directed through pipe 20 to the left end of the cylinder to drive the piston and table 13 toward the right, liquid from the right end of the cylinder returning to the pump through pipe 21, grooves 67 and 68, passages 69 and 49, and pipe 28.

A series of tapered grooves 70 formed in the periphery of the head 51 at the inner end thereof cooperate with the groove 64 to gradually reduce the rate of admission of liquid into groove 64 from groove 65 during movement of the valve plunger from the left extreme position of Fig. 3, to thereby gradually reduce the rate of flow from the left end of the cylinder 18 and a similar series of grooves 71 in the head 53 cooperate in a similar manner with the groove 68 to gradually reduce the rate of escape of liquid from the right end of the cylinder during movement of the valve plunger from its right extreme position of Fig. 4. Two similarly formed series of tapered grooves 72 and 73 in the opposite ends of the head 52 cooperate with the central groove 66 in a similar manner; grooves 73 gradually reduce the rate of escape of liquid from groove 66 into groove 67 during movement of the valve plunger from the left extreme position of Fig. 3, to thereby gradually reduce the rate of admission of liquid to the right end of the cylinder, and grooves 72 gradually reduce the rate of flow from groove 66 into groove 65 and thus to the left end of the cylinder during movement of the valve plunger from the right extreme position of Fig. 4. After the valve plunger passes the middle position of Fig. 5 the function of the grooves 72 and 73 is reversed, one or the other series serving to gradually increase the rate of flow from groove 66 into the right or left end of the cylinder dependent upon whether the valve plunger is then moving toward the right or left from the middle position. It will thus be noted that upon each reversal of the valve the rate of liquid supplied to and discharged from the cylinder 18 is gradually reduced to thereby decelerate the motion of the table 13 after which the direction of flow is reversed and the rate of supply gradually increased to thereby accelerate the table until it reaches its normal speed in the opposite direction.

Provision is made for operating the plunger of the reversing valve at a rate properly regulated so that each movement thereof is substantially synchronized with the deceleration and acceleration periods of the table at each reversal to thereby effect a smooth reversal. In this instance the valve plunger is hydraulically actuated by liquid supplied to the outer ends of the bores 57 and 58 through passages 74 and 75 in the blocks 55 and 56, respectively. Liquid supply pipes 76 and 77 are connected to these passages through check valves 78 and 79 which prevent the escape of liquid from the passages therethrough. Outlet passages 80 and 81 permanently connected to the outer ends of the bores 57 and 58, communi-

cate through choke valves 82 and 83, respectively, with pipes 84 and 85 which lead to the pipes 76 and 77. Additional outlet passages 86 and 87 in the blocks 55 and 56, controlled by the pistons 61 and 62, respectively, communicate through choke valves 88 and 89, and pipes 88' and 89', with the pipes 76 and 77, respectively.

The several choke valves 82, 83, 88 and 89 are designed to respond automatically to variations in temperature so as to automatically compensate for variations in the viscosity of the oil (constituting the hydraulic medium) and thus maintain a substantially constant resistance to the flow of oil therethrough at all temperatures. Valve 89 is shown in detail in Fig. 6. It comprises an elongated thimble 182 of metal having a low coefficient of expansion, such as invar metal. The open end 183 of the thimble is threaded into the passage 87 and the other end 184 thereof contains a small inlet passage 185. A valve stem 186 of metal having a relatively high coefficient of expansion, such as bronze, projects into the thimble 182, and the end 187 thereof cooperates with the end 184 of the thimble to restrict the flow of oil therethrough. The stem 186 is screwed into a plug 188 which serves to close the passage. The arrangement is such that as the temperature increases, the length of the stem 186 increases faster than the length of the thimble 182, so that the end 187 of the stem approaches the end 184 of the thimble to thereby reduce the clearance therebetween. This compensates for the reduced viscosity of the oil as the temperature increases, so that the valve offers substantially the same resistance to flow at all temperatures.

The arrangement is such that when pipe 76 is exposed to fluid pressure the piston 61 and the valve plunger are driven toward the right from the position of Fig. 3, at a rate dependent upon the rate of escape of liquid from the bore 58 through the choke valves 83 and 89 and pipes 85, 89', and 77; thence to drain pipe 33. But as the valve plunger passes through its mid-position the piston 62 in its advance chokes off and finally completely blocks the outlet passage 87, and thereafter the plunger continues to move toward the right but at a slower rate as determined by the restricted flow through the choke valve 83 alone. Experience shows that under ordinary conditions the period of deceleration of the table may be shorter than the acceleration period so that in this instance the valve plunger is permitted a more rapid movement during the first half of its stroke (the deceleration period) than during the remainder of its stroke (the acceleration period). Similarly when pipe 77 is exposed to pressure the piston 62 and valve plunger are driven toward the left, first at a rate depending upon the rate of escape through both choke valves 82 and 88, thence to pipes 84, 88', 76 and 33, and later, after passage 86 is blocked by piston 61, at a reduced rate depending upon the rate of escape through the choke valve 82 alone.

It will be noted that during this reduction in the rate of flow in the motor circuit (the pipes 20 and 21 and cylinder 18) which thus occurs during each reversal the pump continues to deliver liquid at a constant rate dependent upon the set position of the hand wheel 27. This would ordinarily result in a marked increase in pressure up to the maximum limit determined by the high pressure relief valve 36, which is ordinarily greater than that required or desired during reversal. Provision has therefore been made for automatically mantaining a predetermined pressure during reversal, which is ordinarily considerably lower than the maximum pressure determined by the valve 36, and which may be adjusted so as to predetermine the accelerating force available to return the table to normal speed after each reversal. This is accomplished in this instance by the use of a pressure relief valve within the plunger of the reversing valve and so arranged as to be connected to the pressure passage 50 throughout the major portion of the stroke of the valve plunger. The relief valve shown is in the form of a sleeve 90 closely fitted for reciprocation within a bore formed in one end of the valve plunger. One end of this sleeve 90 is normally seated on the shouldered end 92 of the reduced portion 93 of a plug 94 seated within the inner end of the bore. A spring 95 is interposed between the other end of sleeve 90 and a plug 96 screwed into the outer end of the bore. Ports 97 in the periphery of the plunger head 51 communicate with the interior of the sleeve 90 through passages in the plug 96, and ports 98 in the central peripheral portion of the plunger head 52 communicate with the annular chamber surrounding the reduced portion 93 of plug 94.

The arrangement is such that when the reversing valve plunger is in either of the extreme positions shown in Figs. 3 and 4 the ports 98 of the relief valve are blocked by one or the other of the lands at the opposite sides of the groove 66, so that this relief valve is then ineffective. But upon shifting the reversing valve plunger slightly from either of those positions ports 98 are opened to the groove 66, and ports 97 are opened to the groove 64, this condition being maintained throughout the major portion of the movement of this plunger. So long as this condition exists the annular chamber surrounding the reduced portion 93 of plug 94 is exposed to the pressure in the groove 66 and passage 50, and as this pressure rises sufficiently to force the sleeve 90 to the left against the resistance of spring 95, liquid may escape through the sleeve 90, plug 96, and ports 97 into the groove 64 and back to the pump. The resistance of the spring is determined by adjustment of the plug 96 and is ordinarily such as to permit movement of the sleeve 90 at pressures considerably lower than the pressure required to open the relief valve 36.

The pipes 76 and 77, leading to the bores 57 and 58 of the reversing valve, are supplied with fluid pressure from any appropriate source, such as the gear pump 29, and are controlled by a pilot valve preferably such as will now be described. This valve comprises a hollow plunger having two spaced heads 99 and 100 closely fitted for lengthwise movement in a bore 101 formed in the upper portion of the valve block 37. The opposite ends of this bore are closed by screw plugs 102 one of which is drilled to receive a valve stem 103 connected with the plunger to operate the same. The other contains drain pipe 33. A central annular channel 104 in the bore communicates with the pressure pipe 31 leading from the gear pump, and two annular channels 105 and 106, at opposite sides of the channel 104, are connected with the pipes 77 and 76, respectively. When the plunger of the pilot valve is shifted into the left extreme position of Fig. 3 pipe 77 is connected with pressure pipe 31 and pipe 76 is connected to drain pipe 33, and when it is shifted into the right extreme position of Fig. 4 pipe 76 is connected with pressure pipe 31 and pipe 77 is connected to drain pipe 33.

The pilot valve just described is operated and

controlled by the table 13 through appropriate mechanism such for instance as will now be described. This mechanism includes an arm 107 having a forked end 108 engaged with a pin 109 on the stem 103. The arm 107 is fixed to a rock shaft 110, journaled in brackets 111 on the end of the frame 10 of the machine. A pair of upright arms 111', fixed to the opposite ends of the rock shaft 110, carrying a bar 112 which is engaged within a forked member 113 formed on the end of a rod 114 mounted for lengthwise movement within brackets 115 on the carriage 12. The rod 114 carries a stop collar 116 fixed thereto for coaction with a pair of dogs 117 and 118 adjustably fixed on a bar 119 fixed in brackets 120 carried by the table. In this instance each dog 117–118 is yieldably retained against a collar 121, adjustably fixed to the bar 119, by a spring 122 which bears against another collar 123, adjustably fixed to the bar 119.

The arrangement is such that during each movement of the table 13 toward the right, dog 117 engages stop collar 116 and forces the rod 114 to the right to thereby shift the pilot valve plunger into the left extreme position of Fig. 3 and thus effect movement of the reversing valve into the left extreme position of Fig. 3 to thereby reverse the table. Then as the table travels toward the left dog 118 ultimately strikes stop collar 116 so as to shift the rod 114 to the left to thereby shift the pilot valve plunger into the right extreme position of Fig. 4 and thus shift the reversing valve and again reverse the table. The springs 122 behind the respective dogs permits the table to continue movement to the point of actual reversal after the rod 114 and pilot valve plunger have reached the limit of their strokes. It will be noted that the engagement of the member 113 with the bar 112 permits the rod 114 to travel forwardly and rearwardly with the carriage 12 without destroying the operating connections between the rod 114 and the stem 103 of the pilot valve.

Various changes may be made in the embodiment of the invention hereinabove specifically described without departing from or sacrificing the advantages of the invention as defined in the appended claims.

I claim:—

1. In a machine of the character described the combination of a translatable member, hydraulically actuated means for driving said member, a positive delivery pump, hydraulic connections including a valve for delivering driving liquid from said pump to said hydraulically actuated means, means automatically operable at the end of each stroke of said member for operating said valve to thereby reverse the direction of motion of said member, said valve functioning to reduce the rate of liquid delivered to said hydraulically actuated means during reversal, means for regulating the rate of operation of said valve, and means rendered effective by said valve for controlling the escape of excess liquid supplied from said pump during reversal.

2. In a machine of the character described the combination of a translatable member, hydraulically actuated means for driving said member, a positive delivery pump, hydraulic connections including a valve for delivering driving liquid to said hydraulically actuated means, means for operating said valve to thereby reverse the direction of motion of said member, said valve functioning to reduce the rate of liquid delivery to said hydraulically actuated means during reversal, and means rendered effective by said valve for controlling the escape of excess liquid supplied from said pump during reversal.

3. In a machine of the character described the combination of a driven member, hydraulically actuated means for driving said member, a pump, hydraulic connections including a reversing valve between said pump and said hydraulically actuated means, means for operating said valve to reverse the direction of motion of said member, and means controlled by said valve for limiting the working pressure in said connections during reversal.

4. In a machine of the character described the combination of a driven member, hydraulically actuated means for driving said member, a pump, hydraulic connections including a reversing valve between said pump and means, means for operating said valve to reverse the direction of motion of said member, and a pressure relief valve controlled by said reversing valve for limiting the pressures in said connections during reversal.

5. In a machine of the character described the combination of a reciprocating member, hydraulically actuated means for driving the same, a reversing valve operable to reverse the direction of motion of said member, fluid actuated means for operating said valve, and means including a temperature responsive valve connected with said fluid actuated means for controlling the rate of operation of said reversing valve.

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