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HYDRAULIC CONTROL MECHANISM FOR A MACHINE TOOL

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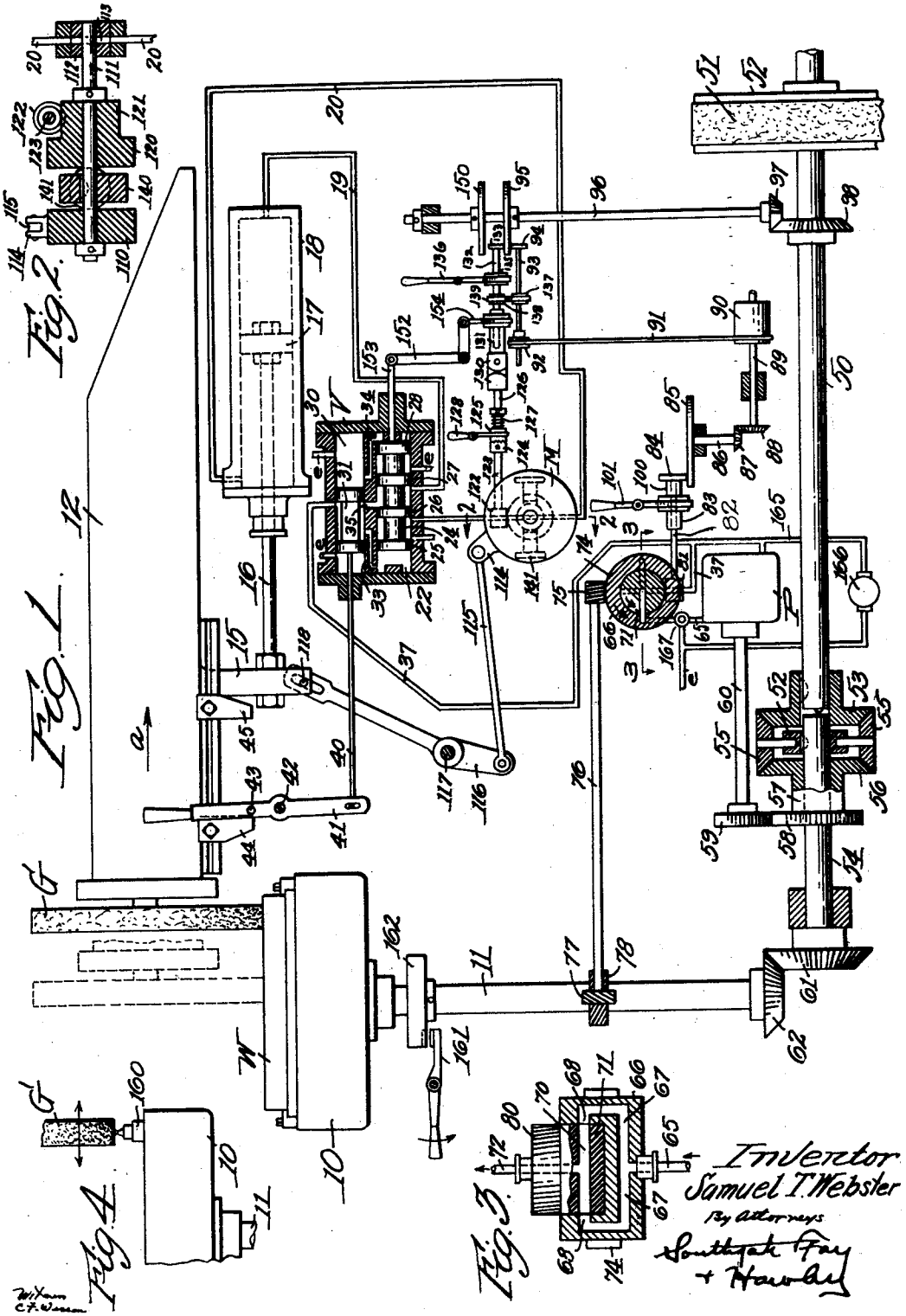


Fig. 1.

Fig. 2.

Fig. 3.

Fig. 4.

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## HYDRAULIC CONTROL MECHANISM FOR A MACHINE TOOL

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This invention relates to machine tools, such as grinders, in which one part of the machine is reciprocated by fluid-operated or hydraulic mechanism and in which another part of the machine is rotated by continuous acting means, said means and said mechanism preferably being actuated from a single external source of power, such as a driving shaft.

It is the general object of my invention to provide means by which the speed ratio of these differently actuated parts may be maintained substantially constant.

Another object is to provide a construction in which the desired ratio will be retained when the actual rate of movement of one of said parts is increased or decreased. I further provide means for retaining the predetermined ratio when the direction of travel of one of said parts only is reversed.

An important feature of the invention also relates to the provision of a differential connection between the two parts of the apparatus for a purpose to be described and the provision of a pump operated in association therewith.

My invention further relates to arrangements and combinations of parts which will be hereinafter described and more particularly pointed out in the appended claims.

A preferred form of the invention is shown in the drawing, in which

Fig. 1 is a side elevation, partly in section, of a grinder embodying my improvements;

Fig. 2 is a detail sectional elevation, taken along the line 2—2 in Fig. 1;

Fig. 3 is a detail sectional plan view, taken along the line 3—3 in Fig. 1, and

Fig. 4 is a detail side elevation to be described.

### *General machine construction*

Referring to the drawing, I have shown parts of a rotary surface grinding machine comprising a work supporting head 10, mounted on a vertically disposed shaft 11, supported in suitable bearings, not shown.

The grinding wheel G is rotatably mounted on a wheel slide or carriage 12 and may be rotated in any convenient manner, as by a motor (not shown) mounted within the slide 12.

The slide 12 is supported in guideways on the machine frame for reciprocating movement at right angles to the axis of the shaft 11. An arm 15 projects downward from the slide 12 and is connected by a rod 16 to a piston 17 slidable in a fixed cylinder 18. Oil or other fluid under pressure is supplied to the opposite ends of the cylinder 18 by pipes 19 and 20.

The pipe 19 is connected to a port near the center of a main valve cylinder 22, formed within the casing of a reversing valve mechanism V. A main valve 24 is slidable in the cylinder 22 and comprises four spaced piston portions 25, 26, 27 and 28.

The pipe 20 is connected through a throttle valve mechanism M, to be described, to another port in the lower side of the cylinder 22.

An upper or control cylinder 30 is provided in the valve mechanism V and a piston 31 is slidable in the cylinder 30 and is provided with two spaced piston portions.

Passages 33 and 34 connect ports in the cylinder 30 to the extreme end portions of the main valve cylinder 22. An additional port 35 directly connects the middle portions of the cylinders 30 and 22.

Suitable exhaust pipes *e* are provided for the cylinders 22 and 30, these exhaust pipes returning the oil through connections, not shown, to the operating pump P or to a suitable storage. Oil under pressure is delivered from the pump P to the valve mechanism V through a feed pipe 37.

The control piston valve 31 is connected by a rod 40 to a reversing lever 41, pivoted at 42 and having a stud 43 positioned for engagement by dogs 44 and 45 adjustably mounted on the wheel slide 12.

In the position of the parts shown in Fig. 1, it is assumed that the wheel slide 12 is just completing its travel to the right, as indicated by the arrow *a*, and that the dog 44 has engaged and shifted the reversing lever 41, moving the piston valve 31 to the left.

In this position, the piston valve 31 con-

nects the supply pipe 37 through the passage 33 to the left-hand end of the main valve cylinder 22, while at the same time the right hand end of the cylinder is connected through the passage 34 and cylinder 30 to the exhaust.

Consequently the main valve 24 is shifted to the right to the position shown in Fig. 1, in which position the supply pipe 37 is connected through the port 35 to the space between the main valve portions 26 and 27 and thence through the pipe 19 to the right-hand end of the cylinder 18.

At the same time the left-hand end of the cylinder 18 is connected through the pipe 20 and the throttle valve mechanism M to the cylinder space between the piston portions 25 and 26, which space is connected to the exhaust. Accordingly, the wheel slide 12 and the grinding wheel G are about to commence their travel to the left from the full line to the dotted line position shown in Fig. 1.

When this travel to the left is completed, the dog 45 shifts the reversing handle 41 to move the control valve 31 to the right, causing movement of the main valve 24 to the left and initiating return movement of the wheel slide 12 to the right, thus completing the cycle of operations.

The hydraulic actuating mechanism just described is of a usual construction and in itself forms no part of my present invention, except as to the throttle valve mechanism M.

*Work driving mechanism*

My improved machine is provided with a drive shaft 50, rotated at constant speed from any convenient source of power, as by a belt 51 and pulley 52.

The drive shaft 50 is supported in suitable bearings and is provided with a bevel gear 53 forming part of a differential gear set. A shaft 54 is supported in bearings aligned with the shaft 50 and an intermediate carrier frame or spider 52 is keyed thereon, on which frame a plurality of bevel pinions 55 are rotatably mounted.

The pinions 55 engage a bevel gear 56 mounted to rotate freely on the shaft 54 and having a hub 57 provided with a gear 58 meshing with a pinion 59 on a shaft 60, which constitutes a drive shaft for the pump B.

The shaft 54 is connected by bevel gears 61 and 62 to the shaft 11 which supports the head 10 by which the work W is rotated. The pump P may be of any suitable type but is preferably a comparatively simple and inexpensive pump of the rotary constant displacement type.

If the pump P was entirely without load and rotating freely, the bevel gear 56 would be free to rotate and the differential mechanism would transmit no power to the shaft 54. Consequently the shaft 54 would remain stationary and the work W would not rotate.

If, on the other hand, the delivery from the pump P was blocked, so that the pump could not rotate, the bevel gear 56 would be held in fixed position and the differential gearing would rotate the shafts 54 and 11 and the work W at maximum speed.

Obviously, under intermediate conditions, the pump P will be rotated at a certain speed, delivering oil at a certain pressure, and the work W will also be rotated at a definitely related speed. The faster the pump P rotates, the slower the work W is rotated.

*Speed control for work*

In order to control the speed of the work W I provide certain automatic adjusting mechanism for regulating the delivery of the pump P.

For this purpose, I provide a return pipe 65 for the pump P, which return pipe is connected to an axial opening in a rotating valve member 66 (Fig. 3). This axial opening is connected through branch passages 67 to inwardly directed ports 68, which are more or less aligned with the ends of a cross-passage 70 in an inner rotatable valve member 71 having an axial port or opening connected by a pipe 72 (Fig. 1) to the supply pipe 37 previously described. This pipe 37 is also connected to the pressure side 165 of the pump P and therefore these connections constitute a by-pass by which the load on the pump may be varied.

A worm gear 74 is formed on the outer rotatable valve member 66 and is engaged by a worm 75 mounted on a shaft 76 provided with a spiral pinion 77 engaging a spiral gear 78 on the work-supporting shaft 11. Consequently the outer valve member 66 rotates in timed relation to the rotation of the shaft 11 and work W.

A worm wheel 80 is formed on the inner rotatable valve member 71 and is engaged by a worm 81 on a shaft 82 having a telescoping connection to the elongated hub 83 of a friction roll 84. The roll 84 engages the upper surface of a friction disc 85, mounted on a short vertical shaft 86 connected by bevel gears 87 and 88 to a horizontal shaft 89 supporting a drum 90.

The drum 90 is connected by a belt 91 to a pulley 92 on a shaft 93 slidable in fixed bearings (not shown), and provided with a friction roll 94 engaging the under side of a friction disc 95. The disc 95 is mounted on an upright shaft 96 connected by a bevel pinion 97 and bevel gear 98 to the main drive shaft 50 of the machine.

Through these connections the inner rotatable valve member 71 is rotated in pre-

determined speed relation to the speed of rotation of the drive shaft 50, while the outer rotatable valve member 66 is rotated in predetermined speed relation to the rotation of the shaft 11 and work W.

If the work W tends to lag behind its predetermined speed, the outer valve member 66 will rotate more slowly than the inner valve member 71. The ports 68 and cross passage 70 are so disposed that this relatively slower movement of the outer valve member 66 will cause a decrease in the available port opening.

As the pump P is a constant discharge pump, the decrease of the discharge opening will increase the load on the pump, thus tending to slow down the pump and the connected bevel gear 56. As previously explained any slowing down of the gear 56 causes a corresponding increase in the speed of rotation of the shaft 54 and of the work W connected therewith.

If on the other hand the work W tends to rotate too fast relatively to the driving shaft 50, the outer valve member 66 will move ahead of the inner valve member 71, opening the port 68 to a wider extent and thus decreasing the load on the pump. This permits the bevel gear 56 to rotate more rapidly, which in turn tends to slow down the work W.

If it is desired to change the speed ratio between the shaft 50 and the shaft 11 and the work W, this may be accomplished by moving the friction roll 84 radially of the friction disc 85, for which purpose a grooved collar 100 and manual speed-control lever 101 is provided. The telescoping connections between the shaft 82 and the hub 83 of the friction roll permit such speed adjustment.

*Hydraulic speed control*

I will now describe the mechanism by which the rate of movement of the piston 17 in the cylinder 18 is controlled. For this purpose I provide the throttle mechanism M previously mentioned. This comprises a friction disc 110 (Fig. 2) mounted to rotate freely on a shaft 111, to which an inner valve member 112 is secured. The valve member 112 is rotatable in a fixed valve casing and is provided with a cross passage 113 which connects the upper and lower portions of the pipe 20 which extends from the main valve cylinder 22 to one end of the cylinder 18.

The friction disc 110 (Fig. 2) is provided with an ear 114 (Fig. 1) connected by a link 115 to a lever 116 which is pivoted at 117. The upper end of the lever 116 has a pin-and-slot connection at 118 with the depending arm 15 of the slide 12. As the slide is reciprocated, the friction disc 110 is rocked back and forth through the described connections and in timed relation to the movement of the slide.

A second friction disc 120 (Fig. 2) is freely rotated on the shaft 111 and is provided with a worm gear portion 121 engaged by a worm 122 mounted on a worm shaft 123 and having a clutch collar 124 secured at one end. A second clutch collar 125 is keyed to a shaft 126 and is held in engagement with the collar 124 by a spring 127. A handle 128 is provided for disengaging the clutch whenever desired. Such disengagement causes the friction disc 120 to stand still, with excessive resultant angular movement of the valve member 112 and thus stops the movement of the wheel slide by shutting off the oil supply.

The shaft 126 is connected by a swivel joint 130 to a hub member 131 having telescoping connection with a shaft 132 keyed to said hub member. The shaft 132 is provided with a friction roll 133 engaging the upper face of the friction disc 95 previously described.

A grooved collar 135 and handle 136 are provided for adjusting the shaft 132 and roll 133 radially with respect to the disc 95.

Grooved collars 137 and 138 on the shafts 93 and 132 respectively are connected by a yoke member 139, so that the shafts 93 and 132 will be simultaneously adjusted radially of the disc 95, and so that the rolls 94 and 133 will always engage disc portions of the same diameter on the opposite faces of the disc 95.

The parts are so proportioned that when the slide 12 is moving at the desired speed with respect to the speed of rotation of the shaft 50, the friction discs 110 and 120 will be moved angularly at the same speed, but in opposite directions.

A frame or carrier 140 (Fig. 2) is keyed to the shaft 111 previously described, and is provided with friction rolls 141 engaging the inner faces of the discs 110 and 120 at a fixed distance from the axis. If the discs 110 and 120 move angularly at the same speed but in opposite directions, the rolls 141 will rotate idly and the frame or carrier 140 and shaft 111 will remain fixed in position. If, however, one of the discs moves more or less rapidly than the other disc, the carrier 140 and shaft 111 will be moved angularly, increasing or decreasing the port opening between the passage 113 and the pipe 20, and correspondingly varying the delivery of oil through the cylinder 18.

The ports are so arranged that if the slide 12 tends to move too rapidly, the port openings will be closed, reducing the rate at which oil may be discharged from the left-hand end of the cylinder 18, and if the slide 12 tends to move too slowly, the port opening will be increased, thereby decreasing the resistance to the flow of oil from the left-

hand end of the cylinder 18 and consequently causing the rate of movement of the slide 12 to be increased.

Obviously, the angular movement of the friction disc 110 will be reversed whenever the direction of travel of the slide 12 is reversed, and consequently it is necessary to correspondingly reverse the direction of rotation of the friction disc 120.

This is accomplished by providing a second friction disc 150 on the shaft 96 above the friction roll 133, and by providing means for shifting the contact of the roll 133 from one disc to the other upon reversal of movement of the slide 12.

For this purpose, a bell crank 152 has its upper arm pivotally connected to a rod 153 extending outward from the main piston valve 24 and has its horizontal arm connected by a link 154 to the hub member 131. As the main valve 24 moves from one end to the other of the cylinder 22, the shaft 132 and the roll 133 are correspondingly raised and lowered, thus reversing the direction of movement of the friction disc 120 substantially simultaneously with the reversal of the friction disc 110.

*Wheel truing*

I have made special provision for truing the grinding wheel G. For this purpose, a truing device 160 is mounted on the work table 10 and the work table is held from rotation in some convenient manner, as by a latch 161 engaging a notched collar 162 on the shaft 11. The grinding wheel is then moved back and forth over the device 160, thus truing the wheel.

In order to reciprocate the slide 12 and the wheel G, it is necessary that the pump P remain operative. But the outer valve member 66 connected to the shaft 11 will now be held stationary, consequently causing the pump control mechanism to operate irregularly, as the inner valve member 71 will continue to rotate while the other valve member 66 remains stationary. This intermittent opening and closing of the by-pass connection through the pipes 72 and 65 would cause a jerky or irregular movement of the wheel slide 12.

Accordingly I provide a three-way valve 167 which is shifted to shut off the valve opening 65 from the exhaust side of the system. Since the shaft 11 is now locked in position and unable to rotate, the pump P will be rotating at its maximum speed and delivering the maximum volume of oil, which volume will be much in excess of that required to operat the piston 17 which reciprocates the wheel head 12. To take care of this excess oil, I provide a by-pass relief valve 166 which is connected across the exhaust *e* and pressure sides of the pump P. This relief valve 166 is set to open at a pressure in excess of the pressure required to move the wheel head 12 by piston 17.

Before grinding is resumed, the valve 167 is reset so that the valve opening 65 is freely connected to the exhaust side of the system and the latch 161 is moved outward to free the shaft 11.

*Operation*

By the mechanism described I maintain the rate of travel of the slide 12 in predetermined relation to the speed of rotation of the drive shaft 50, and I also maintain the rate of rotation of the work W in predetermined relation to the speed of the shaft 50. Consequently the rotation of the work W and the reciprocating movement of the grinding wheel G are coordinated and are maintained in a desired speed relation.

If it is desired to speed up the operation of the entire machine, the rolls 94 and 133 are adjusted radially of the friction disc 95 by use of the handle 136. If on the other hand it is desired to change the ratio of movement between the work and the grinding wheel, the roll 84 may be adjusted radially of the disc 85 by use of the handle 101.

I have thus provided mechanism by which a rotated member and a reciprocated member may be driven from a common source of power and by which the movements of the two members may be coordinated in any desired ratio, together with means for varying the ratio of movement between the rotated and reciprocated parts, as well as for varying the actual rate of movement relative to the speed of the common driving means.

Having thus described my invention and the advantages thereof, I do not wish to be limited to the details herein disclosed otherwise than as set forth in the claims, but what I claim is:—

1. In a machine tool, a reciprocating member, hydraulic actuating mechanism therefor including a pump and a cylinder and piston associated therewith, one of which is connected to said reciprocating member, a rotated member, mechanical means to rotate said member, one of said members being connected to move a piece of work and the other member being connected to move a tool relative to the work while said tool is operating on the work, a common initial driving element for said pump and said rotating means, and means to coordinate the rates of movement of said reciprocating and rotating members and to thereby maintain a fixed speed ratio between the work and tool, said latter means including differential gearing connected to drive said pump from said initial driving element and said differential gearing having an internal gear-carrying frame connected to drive said rotated member, a by-pass connection for said pump and a two-

part rotating speed control valve in said by-pass connection with one part thereof rotated in fixed speed relation to the rotation of said rotated member and the other part of said valve rotated in fixed speed relation to the rotation of said initial driving element.

2. The combination in a machine tool as set forth in claim 1 in which the pump is connected to operate in inverse speed relation to the rotated member.

3. The combination in a machine tool as set forth in claim 1 in which the pump is connected to operate in inverse speed relation to the rotated member, and in which the pump is of the constant displacement type and is effective to act as a brake on said rotated member.

4. In a machine tool, a reciprocating member, hydraulic actuating mechanism therefor including a pump and a cylinder and piston associated therewith, one of which is connected to said reciprocating member, a rotated member, mechanical means to rotate said member, one of said members being connected to move a piece of work and the other member being connected to move a tool relative to the work while said tool is operating on the work, a common initial driving element for said pump and said rotating means, and means to coordinate the rates of movement of said reciprocating and rotating members and to thereby maintain a fixed speed ratio between the work and tool, said coordinating means including a two-part speed control valve having one valve part operated in timed relation to said rotated member and having the other valve part operated in timed relation to the initial driving element and said valve varying the load on said pump and inversely varying the speed of said pump in accordance with variations in the relative velocities of said valve parts.

5. The combination in a machine tool as set forth in claim 4, in which means is provided to reverse the direction in which one of said parts is driven by said initial driving element when the direction of travel of said reciprocated member is reversed.

6. In a machine tool, a reciprocating member, hydraulic actuating mechanism therefor including a pump and a cylinder and piston associated therewith, one of which is connected to said reciprocating member, a rotated member, mechanical means to rotate said member, one of said members being connected to move a piece of work and the other member being connected to move a tool relative to the work while said tool is operating on the work, a common initial driving element for said pump and said rotating means, a two-part speed control valve for said hydraulic mechanism having one part connected for oscillation by said reciprocated member and having a second part connected for rotation from said initial driving element, and said latter connections including a pair of friction discs continuously rotated, a friction roll mounted between said discs, and means to shift said roll from one disc to the other as the reciprocated member is reversed.

7. The combination in a machine tool as set forth in claim 6 in which the shifting means includes a reversing valve for said hydraulic mechanism and a connection through which said valve is effective to move said friction roll.

8. In a machine tool, a reciprocating member, hydraulic actuating mechanism therefor including a pump and a cylinder and piston associated therewith, one of which is connected to said reciprocating member, a rotated member, mechanical means to rotate said member, one of said members being connected to move a piece of work and the other member being connected to move a tool relative to the work while said tool is operating on the work, a common initial driving element for said pump and said rotating means, means to coordinate the rates of movement of said reciprocating and rotating members and to thereby maintain a fixed speed ratio between the work and tool, and manual means to simultaneously increase the speed of both of said members relative to said driving element while maintaining the same speed ratio between said members.

9. In a machine tool, a reciprocating member, hydraulic actuating mechanism therefor including a pump and a cylinder and piston associated therewith, one of which is connected to said reciprocating member, a rotated member, mechanical means to rotate said member, one of said members being connected to move a piece of work and the other member being connected to move a tool relative to the work while said tool is operating on the work, a common initial driving element for said pump and said rotating means, means to coordinate the rates of movement of said reciprocating and rotating members and to thereby maintain a fixed speed ratio between the work and tool, manual means to simultaneously increase the speed of both of said members relative to said driving element while maintaining the same speed ratio between said members, and additional manual means to change said speed ratio between said members.

In testimony whereof I have hereunto affixed my signature.

SAMUEL T. WEBSTER.