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Coyne

[11] 3,759,335

[45] Sept. 18, 1973

[54] **MOLE HAMMER-CYCLE CONTROL**

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[52] **U.S. Cl.** **173/91,** 91/282, 173/134, 175/19

[51] **Int. Cl.** **E21b 3/12**

[58] **Field of Search** 173/134–138, 91; 91/282, 283, 291, 292; 175/19

[56] **References Cited**

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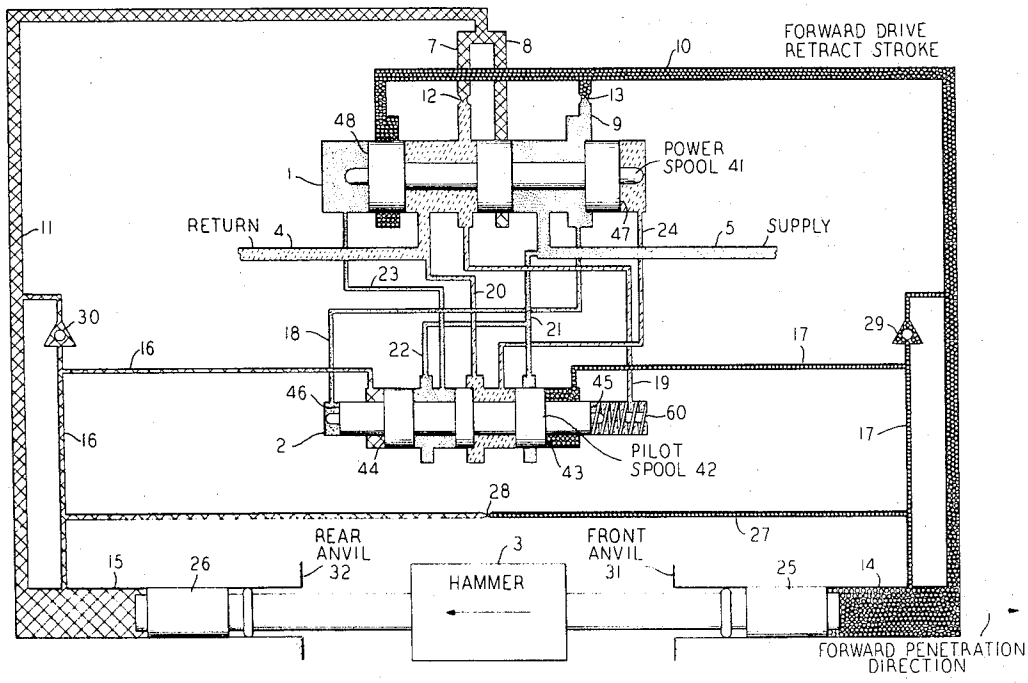
[57] **ABSTRACT**

This disclosure describes a scheme for switching the propelling fluid forces alternately to opposite sides of a Mole impacting hammer. The scheme is grounded on valving structures for sensing fluid pressure differences (1) between supply and return pressure and (2) on opposite sides of the hammer; and on orifice means for generating a controlled difference between (1) and (2). By selecting operating parameters in this system, the fluid force switching is always actuated at a point in the retract portion of the cycle so as to assure maximum hammer stroke.

5 Claims, 6 Drawing Figures



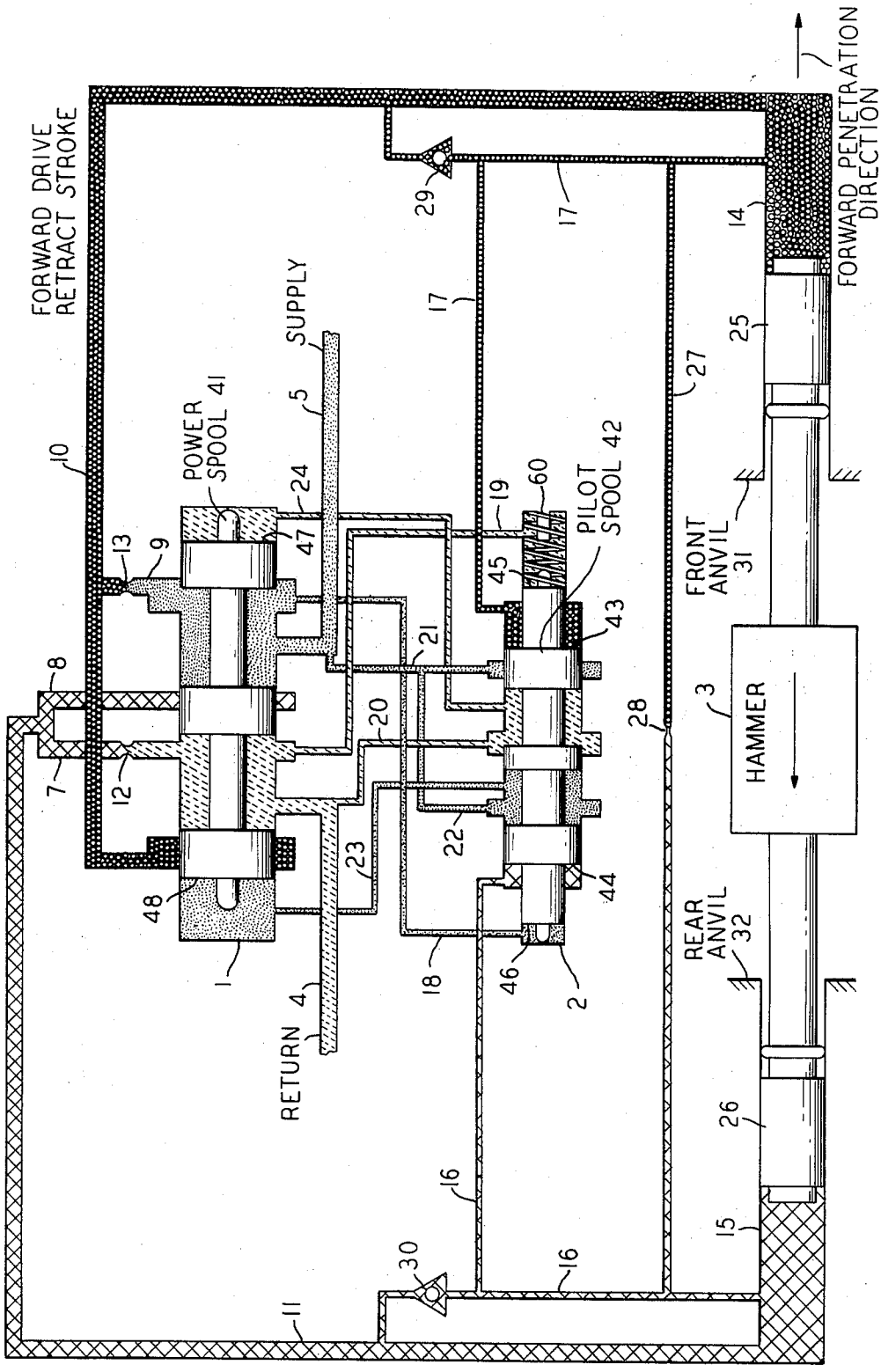

FORWARD DRIVE
RETRACT STROKE
FORWARD PENETRATION
DIRECTION
SUPPLY
RETURN
POWER
SPOOL 41
PILOT
SPOOL 42
FRONT
ANVIL
REAR
ANVIL
HAMMER
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FIG. 1

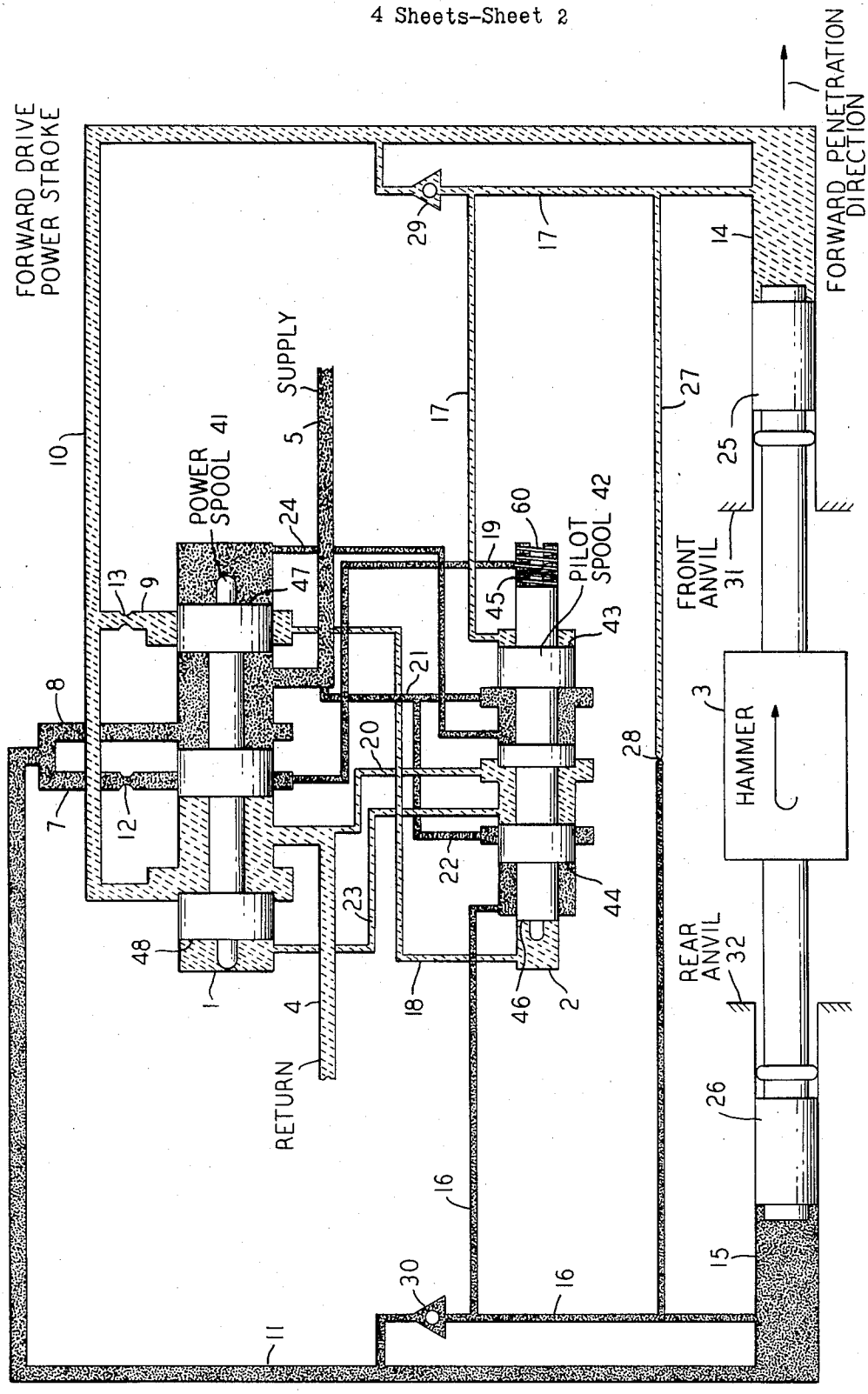
FORWARD DRIVE POWER STROKE
FORWARD PENETRATION DIRECTION
SUPPLY
POWER SPOOL 41
PILOT SPOOL 42
FRONT ANVIL
REAR ANVIL
HAMMER
RETURN
1
2
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FIG. 2

FIG. 3

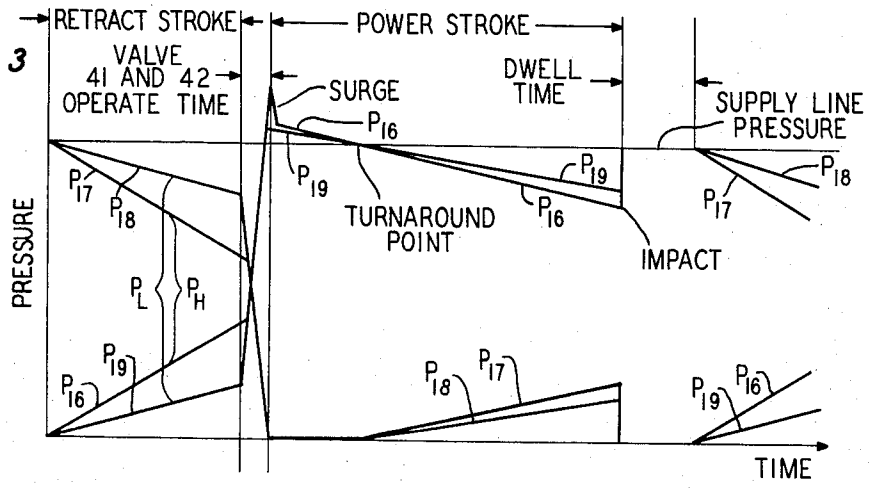


FIG. 4

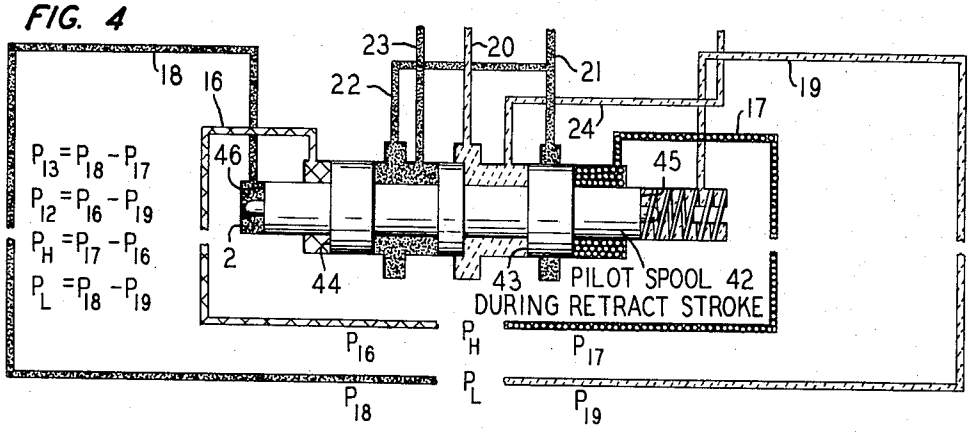


FIG. 5

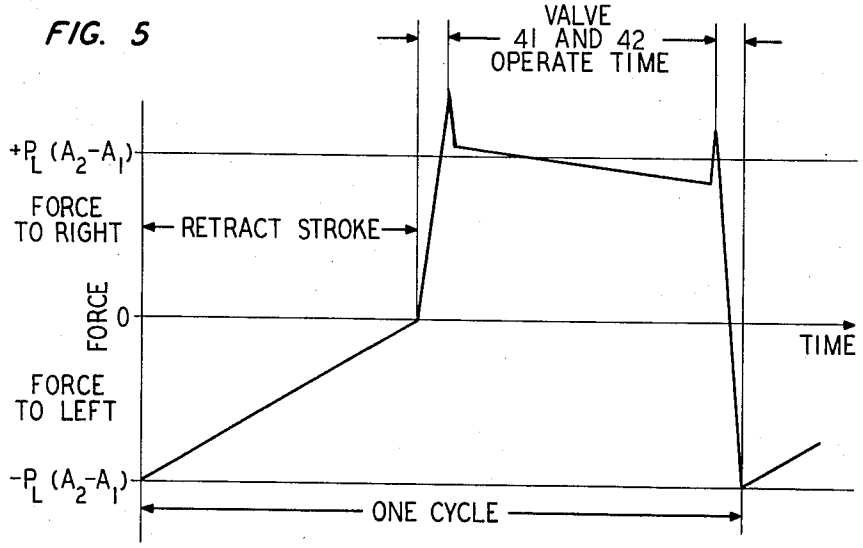
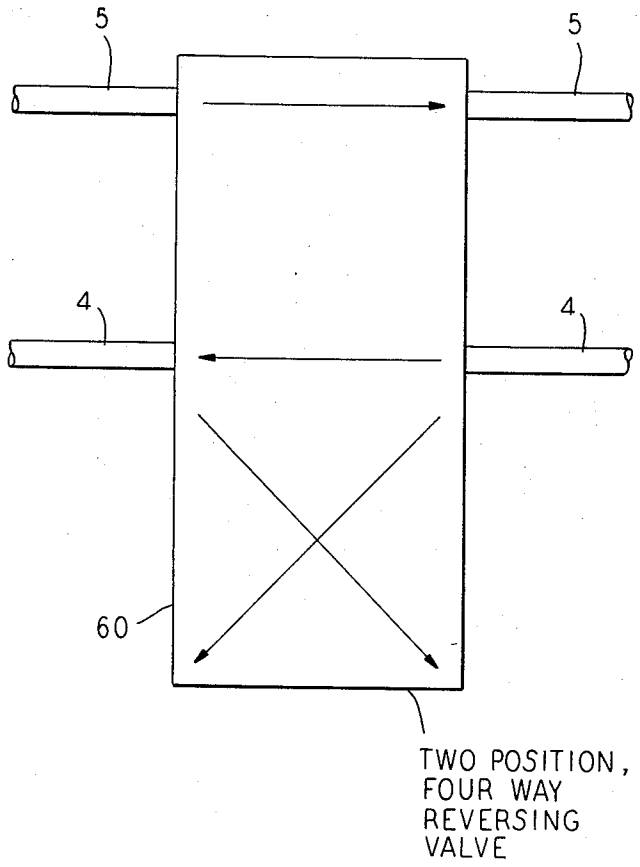


FIG. 6



MOLE HAMMER-CYCLE CONTROL

FIELD OF THE INVENTION

This invention relates in general to self-propelled earth penetrators known as Moles; and more specifically to a Mole hammer cycle control.

BACKGROUND OF THE INVENTION

Self-propelled earth penetrators or Moles typically are driven by an internal linearly impacting hammer that is free to slide for and aft. Between impacts of the hammer on a front anvil, the hammer must return to the rearmost possible position without impacting on the rear anvil, and without aid of bumpers or snubbers to stop the hammer motion. Accordingly, the hammer cycle must be completely controlled by the application and removal of hydraulic pressure at the proper times.

More specifically, after impact on the front anvil, at some proper "midstroke" point in the retract stroke while the hammer is being accelerated toward the rear anvil, the net force on the hammer must be switched to cause a further impact on the front anvil. Thus, propulsion pressures are switched, the hammer decelerates to a stop, reverses its direction of motion, and then accelerates toward the front anvil.

For forward penetration into the soil, the rear anvil must of course not be impacted; although it is desired that the maximum available stroke distance be utilized for hammer acceleration in the forward direction.

Thus the problem is one of switching the forces acting on the hammer at just the right time to make the hammer coast to a stop just before striking the rear anvil. At "end-of-stroke," after the hammer has impacted the front anvil, the forces must again be switched to restore the initial pressure conditions and commence the next cycle.

In practice, soil conditions, oil temperature, pump pressure, internal leakage rate and many other conditions will vary. Thus, if an extremely close control of hammer stroke over a wide range of operating conditions is desired, a closed-loop feedback system using some sort of proximity centers to generate an error signal to cause a correction to the point of switch on the next cycle, would be necessary. However, need for a closed-loop feedback system could be obviated if a dependable open-loop system existed.

Accordingly, one object of the invention is to retain the maximum hammer stroke length over all conditions of soil, oil temperature, viscosity, hydraulic frictions, etc., but without striking the rear anvil.

A further inventive object is to maintain tunnel traction as the Mole hammer cycles, so that the Mole does not slip back in the tunnel at any point.

A further inventive object is to achieve the foregoing objects with an open-loop system.

A further inventive object is to obviate the need for position sensors and mechanical linkages to the hammer.

A still further inventive object is to provide for hammer cycle reversal remotely, by interchanging at the ground surface supply and return hydraulic hoses.

SUMMARY OF THE INVENTION

It has been realized that a switching parameter

$$G = v^2/p \quad (1)$$

exists that can be used as the basis for initiating the midstroke switch of net actuating (or hydraulic forces on the hammer, where v is the instantaneous hammer velocity and p is the instantaneous value of line pressure.

Although the inventive switching function would appear difficult to implement at first glance, it is simple because of the proportionality that exists between turbulent flow pressure losses, and flow rate squared.

The invention and its further objects, features, and advantages will be fully apprehended from a reading of the description to follow of an illustrative embodiment.

BRIEF DESCRIPTION OF THE DRAWING

FIGS. 1 and 2 are schematic hydraulic circuit diagrams;

FIG. 3 is a graph tracing certain critical pressure variations;

FIG. 4 is a schematic diagram of the pilot spool which further identifies certain critical pressures;

FIG. 5 is a graph depicting force on pilot spool versus time; and

FIG. 6 is a reversing valve.

DETAILED DESCRIPTION OF ILLUSTRATIVE EMBODIMENT

Theory

Equation 2 below states the proportionality between turbulent flow (through a turbulent flow resistor such as an orifice), pressure losses, and flow rate squared.

$$p_R = C_1 q^2 \quad (2)$$

where p_R = the pressure loss across the turbulent flow resistor, q = flow rate, and C_1 is a constant depending on the geometry of the orifice. The proportionality of flow rate q to hammer velocity v is:

$$q = av \quad (3)$$

where a is the pressurized area of the hammer. Substituting Equation (2) and (3) into Equation (1) gives:

$$G = q^2/a^2 p_L = p_R/C_1 a^2 p_L \quad (4)$$

where p_L is line pressure. Equation (4) can be put into slightly different form by making the substitution

$$p_R = p_L - p_h \quad (5)$$

where p_h is the pressure difference across the hammer. This gives:

$$G = p_L - p_h/C_1 a^2 p_L \quad (6)$$

Accordingly, pursuant to the invention the hammer forces are switched when:

$$p_L - p_h/C_1 a^2 p_L \geq k \quad (7)$$

where k is a specified constant.

Pursuant to one further step of algebraic simplification the hammer forces are switched when:

$$p_h \leq C p_L \quad (8)$$

where C is a constant defined by:

$$C = 1 - C_1 a^2 k \quad (9)$$

Implementation

FIG. 1 is a complete schematic of the valving system pursuant to the present invention for cycle control of the Mole. The power valve 1 and pilot valve 2 can each be in one of two positions by virtue of the respective positions of power spool 41 and pilot spool 42. Thus,

taken together, the two valves 1 and 2 can be in four possible states. Each of these four states of the valves 1 and 2 corresponds to one of the four possible states of a linearly impacting hammer 3 as follows:

Condition	Pilot Spool	Power Spool	Hammer State	
			Drive	Stroke
1	left	right	forward	retract
2	right	left	forward	power
3	right	right	reverse	retract
4	left	left	reverse	power

FIG. 1 shows the valves 1, 2 in the first condition listed above: forward-drive retract-stroke. Pressurized oil enters the power valve 1 from a supply line 5 and leaves through line 9 in which is located an orifice 13. The pressure drop across orifice 13 is proportional to q^2 (the flow-rate squared). The flow passes through a line 10 to chamber 14 where the pressurized oil acts on an area a of the hammer 3 determined by the diameter of chamber 14, thus propelling the hammer toward rear anvil 32. Concurrently oil in chamber 15 flows through line 11 to line 7 and through orifice 12 to the power valve 1. The pressure drop across orifice 12 is also proportional to q^2 (the flow-rate squared).

The flow leaves the power valve 1 through return line 4 which carries the oil flow back to a sump (not shown). The lines 4, 5, 7, 9, 10, 11 carry the main propulsion flow and have large flow area (at least one-fourth inch I.D.) to minimize pressure losses. Lines 4 and 5 are long lengths of hose (250 feet or more), and are of one-half inch I.D.

In addition to these large power lines there are several small switching lines going to the pilot valve 2. The pressure in chamber 14 is communicated via line 17 to the right-hand annular area denoted 43 of the pilot spool 2. Likewise the pressure in chamber 15 is communicated via line 16 to the left-hand annular area 44 of the pilot spool. The pressure in line 7 is communicated via line 19 to the right-hand stub-area 45 of the pilot spool 2. Likewise the pressure in line 9 is communicated via line 18 to the left-hand stub-area 46 of the pilot spool 2. The annular areas 43, 44 on either side of the pilot spool are square; also the stub areas 45, 46 are equal. Thus, Equations (10) and (11) may be written:

$$\text{Area } 43 = \text{Area } 44 = A_2 \tag{10}$$

$$\text{Area } 45 = \text{Area } 46 = A_1 \tag{11}$$

The ratio of Areas A_1/A_2 is an important parameter in the system. This ratio pursuant to one aspect of the invention, must be less than unity. Thus:

$$A_1/A_2 = C < 1 \tag{12}$$

The pilot valve 2 has the function of switching high and low pressures to either side of the power spool 41. Lines 21 and 22 connect supply pressure from line 5 to the inlet of pilot valve 2. Likewise, line 20 connects return line pressure in or from line 4 to the inlet of the pilot valve 2. Lines 24 and 23 connect the outlet of the pilot valve 2 to the right-hand end area denoted 47 and the left-hand end area denoted 48 of the power spool 41.

In the forward-drive retract-stroke state reflected in FIG. 1, the left-hand side of the power spool 41 is pressurized. Thus the power spool 41 will be in the right-hand position as shown in the Figure.

The changes in pressures on the pilot spool areas 43, 44, 45, 46 during a retract stroke will now be considered. At the beginning of the retract stroke the hammer 3 velocity is zero. Hence there is no flow through orifices 12 and 13 and the pressure in line 7 equals that in chamber 15. Likewise, the pressure in line 9 equals that in chamber 14. In other words referring to the FIG. 3 diagram, the pressure difference p_h across the hammer 3 equals the net line pressure p_L. Hence, the force on the pilot spool 42 is equal to $p_L(A_2-A_1)$ and is directed to the left. The pilot spool 42 is in the left-hand position as shown in FIG. 1.

The FIG. 3 diagram shows the changes of the four pressures acting on the four areas of the pilot spool 2. As the hammer 3 gains speed, the pressure losses across orifices 12 and 13 increase. As shown on FIG. 3 the pressure difference $p_{18} - p_{17}$ (subscript numerals refer to oil lines) as well as the pressure difference $(p_{16} - p_{19})$ both increase, due mostly to the throttling effect of the orifices 12 and 13.

As is evident from FIG. 3, the pressure difference across the hammer (p_H) decreases at a faster rate than the net line pressure p_L. Since p_H acts on the larger areas of the pilot spool 42, a point will be reached where the force on the pilot spool 42 reverses direction. At this point the pilot spool will shift, thereby initiating the midstroke operation of the valves.

The forces acting on the pilot spool 42 will now be examined to determine the critical point at which the pilot spool 42 operates. FIG. 4 shows the pilot spool 42 and the four pressures acting on its areas.

The force tending to push pilot spool 42 to the right is:

$$F = p_L A_1 - p_H A_2 \tag{13}$$

or

$$F = (p_{18} - p_{19})A_1 - (p_{17} - p_{16})A_2 \tag{14}$$

This force is shown graphically versus time in FIG. 5. The critical point at which the force becomes positive is found by setting F (Equation 14) equal to zero. Thus, the pilot valve oprerates when:

$$p_H/p_L \leqq A_1/A_2 = C \tag{15}$$

This is the desired switching criterion previously discussed. It is seen that the ratio A_1/A_2 must be a positive fraction. Advantageously, it has been found that this ratio is optimal if falling in the range of 0.6 to 0.9.

As the pilot spool 42 shifts to the right, the pressure in line 24 rises while the pressure in line 23 drops, and the power spool 41 starts to shift to the left. High pressure oil entering the power valve from line 5 now leaves the valve through line 8, by-passing the orifice 12. The oil flows through line 11 to chamber 15, where it collides with the oncoming hammer 3, causing the pressure surge denoted p_{16} in FIG. 3. At the same time, line 10 is being connected through the power valve 1 to return line 4; and the pressure in line 10 and chamber 14 drops to return line pressure. The hammer, however, continues its motion to the left causing oil to be drawn into chamber 14 from the return line 4. This causes the oil pressure in chamber 14 to drop further, to a value less than return line pressure.

The power spool 41, having shifted to the left, blocks flow through lines 7 and 9 in or out of the power valve 1. The flow by-passes the orifices 12, 13 and consequently the pressures in lines 7 and 9 become equal to

the pressures in lines 11 and 10 respectively. The pressures p_H and p_L differ now by only the amount of pressure loss in the valve 1 and lines. As FIG. 3 shows, p_{16} differs only slightly from p_{19} after the midstroke valve operation. Likewise p_{17} differs only slightly from p_{18}. The small differences are due to pressure losses in the valve and lines.

A second result of the power spool shifting to the left is that of interchanging the pressures in lines 18 and 19. P_{18} which had been supply-line pressure now becomes return-line pressure and p_{19} which had been return-line pressure now becomes supply-line pressure. Thus, the net line pressure is still applied on the stub areas of the pilot spool but now with opposite polarity. Since the polarity of both p_L and p_H across the pilot spool 42 has been reversed, the force tending to push the pilot spool to the right becomes the negative of Equation (13).

Thus: $$F = -p_L A_1 + p_H A_2 \quad (16)$$

The pilot spool becomes latched to the right as in FIG. 2 because $A_2 > A_1$ and $p_H > p_L$. The second inequality is true because the hammer 13 is being decelerated by pressure of the supply line 5. At this point in the cycle the hammer 3 is pumping into the supply line 5 and drawing oil out of the return line 4. Naturally, p_H will exceed the net line pressure. Also, the pressure surge mentioned earlier, occasioned by the rapid opening of the supply line 5 into the oncoming hammer 3, helps to latch the pilot valve to the right.

For reasons discussed above, the force on the pilot spool 42 during the midstroke valve operation rises very rapidly from zero to a value in excess of p_L (A_2-A_1), as shown in FIG. 5, thereby latching the pilot spool 42 to the right. As the hammer 3 decelerates, the latching force diminishes. The hammer 3 coasts to a stop at a point just short of the rear anvil 32, reverses direction, and begins to accelerate toward the front anvil. At the hammer turn-around point, the latching force on the pilot spool equals $p_L(A_2 - A_1)$.

As the hammer 3 accelerates forward, the latching force decreases due to the pressure losses in the valve and lines. Since the orifices 12, 13 are by-passed, the latching force does not decrease as rapidly as it did during the retract stroke.

The operation of the end-of-cycle valve operation will now be described. Just before the hammer 3 impacts the front anvil 31, seal 25 closes off the opening from chamber 14 into line 17. Leakage of high pressure oil through line 27 and orifice 28 into line 17 elevates the pressure in line 17, thereby increasing the force on the right-hand annular area 43 of the pilot spool. The pressure in line 17 builds up at a rate governed by the orifice 28 size and the oil compressibility. At some point in time (for example after impact has occurred) the net force on the pilot spool reverses direction, shifting the pilot spool 42 to the left. The pressure level in line 17 at which the pilot spool shifts to the left is approximately equal to $(1 - A_1/A_2$ times line pressure.

The shifting of the pilot spool 42 to the left interchanges high and low pressure on the ends of the power spool 41, thereby causing the power spool to shift to the right. This power valve operation pressurizes line 10 and de-pressurizes line 11. The leakage flow through line 27 and orifice 28 reverses direction. Since the opening from chamber 14 into line 17 is still closed, the pressure in line 17 would leak away and allow the pilot spool to shift back to the right if it were not for check valve 29. The function of check valve 29 is that of providing a pressure path from line 10 to line 17 thereby maintaining the net leftward directed force on the pilot spool immediately after the end-of-cycle valve operation.

The re-pressurization of chamber 14 causes the hammer 3 to accelerate again toward the rear anvil 32. The hammer seal 25 moves away, reopening line 17 into chamber 14 and the system is back in its original state having gone through one complete cycle.

The spring of pilot valve 42 has two functions. The first is to compensate for the small variations in stroke versus supply line pressure caused by actual (non-ideal) conditions, primarily the finite operate time of the spools 41, 42. Experiments have shown that a hammer stroke variation of less than 10 percent can be achieved over a 2:1 range in supply line pressure with use of the proper choice of spring and precompression. The second function of spring 60 is to cause a short stroke, and hence a weaker impact, when the Mole is in the reverse mode of operation. For backing out of a tunnel, a high energy impact is not necessary; and in fact is undesirable because of wear and fatigue of parts. Advantageously, the spring 60 causes the hammer to couple with a short stroke at a rapid cycle rate in the reverse mode.

The detailed description given above is for the forward drive mode of operation. To operate the Mole in reverse it is only necessary to interchange the supply and return lines at the pump using, for example, a standard two-position four-way valve 60 shown in FIG. 6. Then the hammer will impact at the rear anvil 32 instead of the front anvil 31. In this reverse drive the operation of the valving system is substantially identical to that for the forward drive mode. The basic system is symmetrical. Thus, for example, check valve 30 performs the same function in the reverse drive as check valve 29 does in the forward drive.

It is to be understood that the embodiments described herein are merely illustrative of the principles of the invention. Various modifications may be made thereto by persons skilled in the art without departing from the spirit and scope of the invention.

What is claimed is:

1. In a Mole comprising an internal front anvil, a hammer having front and rear ends, pressurized fluid drive means having a power stroke mode and a retract stroke mode and including a fluid switch alternately applying said fluid to said hammer rear and front ends to respectively affect said two modes, and wherein said hammer travels between said front anvil and a retracted position defining therebetween an optimum hammer stroke length, the improvement in controlling the operation of said fluid switch comprising:

means for continuously deriving a switching parameter $G = v^2/p$ where v is the instantaneous velocity of said hammer and p is the instantaneous pressure of said fluid, and

means responsive to sensing when said parameter G reaches a predetermined value, said fluid switch, said predetermined value being selected to assure said optimum hammer stroke length.

2. The apparatus of claim 1, further comprising a rear anvil rearwardly adjacent to said retracted position, and wherein said fluid drive means further comprises means for effecting hammer impact on said rear anvil

only, by the same said controlled operation of said fluid switch.

3. A Mole system comprising:

an elongated body,

a front anvil linked to said body,

a hammer having front and rear hydraulic drive ends,

means for mounting said hammer within said body for linear movement between said front anvil and a retracted position defining therebetween a maximum hammer stroke length physically permissible,

pressurized hydraulic drive means having a power stroke state and a retract stroke state and comprising supply and return hydraulic lines,

means, operative only during said retract stroke and including an orifice connected in series with said hydraulic drive means, for generating a pressure difference $P_L - P_H$, where P_L is the difference between said supply and said return line pressures, and P_H is the difference in pressure between said hammer front and rear ends,

means including power spool means for alternately supplying hydraulic fluid from said supply line to said hammer rear and front ends to respectively affect said two modes, and

pilot valve means having first and second states, and connected to said power spool means and having a first pair of opposed pressure regions each of area A_2 connected to said hammer front and rear ends,

a second pair of opposed pressure regions each of area A_1 connected to said supply and to said return lines, where the ratio A_1/A_2 is a positive fraction,

said pilot valve means further comprising means operating when $(A_1/A_2)\ P_L - P_H \geq 0$, to effect a switch in said states, thereby to switch said drive means, from said power stroke mode to said retract stroke mode, and

third means responsive to a predetermined value of difference between said first sensed pressure and said second sensed pressure the latter scaled by the factor A_1/A_2 sensed by said first and second sensing means for operating said pilot spool means.

4. Apparatus pursuant to claim 3, further comprising means including movement of said power spool means for placing said orifice means in series with said supply line during said retract stroke, said orifice means being by-passed by said supply line flow during said power stroke, means connecting said orifice means during said retract stroke to said hammer front end, said orifice developing a pressure difference thereacross that is proportional to the velocity squared of said hammer, the proportionally constant being selected to assure said maximum hammer stroke length physically permissible and wherein the ratio A_1/A_2 is in a range between 0.4 and 0.7.

5. Apparatus pursuant to claim 3, further comprising a rear anvil rearwardly adjacent to said retracted position, and wherein said fluid drive means further comprises means for effecting hammer impact on said rear anvil only, by the same said controlled operation of said fluid switch.

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