HYDRAULIC FLUID ACTUATED PERCUSSION TOOL

INVENTOR: Gunnar Vigg Riss Romell, Djursholm, Sweden

ASSIGNEE: Atlas Copco Aktiebolag, Nacka, Sweden

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ABSTRACT

A percussion tool has a hammer piston forming with the cylinder a rear and a forward variable volume work chamber which are alternately pressurized. A fluid accumulator is connected to the rear work chamber and it is loaded by a spring device which is adapted to be automatically pre-loaded in proportion to the pressure in the pressurized hydraulic fluid supplied to the percussion tool. Another identical fluid accumulator is shown connected to the forward pressure chamber.

7 Claims, 5 Drawing Figures
HYDRAULIC FLUID ACTUATED PERCUSSION TOOL

This invention relates to a hydraulic fluid actuated percussion tool of the kind which has a hammer piston reciprocating in a cylinder and adapted to deliver impact energy to a work tool in the form of a chisel, a drill steel, a ram or the like. Usually, such a tool has a pressure chamber for urging the hammer piston forwards and a pressure chamber for urging it backwards defined between the hammer piston and the cylinder, and at least one of the pressure chambers are alternately connected to a source of pressurized hydraulic fluid.

Different impact energy per blow will be required for different sizes of the work tool. If the work stroke of the hammer piston is shortened, the impact energy per blow will be reduced but the number of impacts per minute increases. When, however, so called extension drill steels are used, the number of impacts per minute may not be too high because the screw joints between the drill steel lengths and between the drill bit and the first length will become hot. Hand-held breakers or drills may not work with a too high number of impacts because of the operator will then risk Raynaud's disease. If the pressure of the drive fluid is varied, the number of blows per minute and the impact energy per blow will vary simultaneously. By varying both the average work stroke and the pressure of the drive fluid, one may therefore choose any desired combination of the number of blows per minute and impact energy per blow. If, however, a fluid accumulator is connected to a pressure chamber which is alternately pressurized and drained, its pre-load must be adapted to the pressure of the drive fluid. A manual adjustment of the pre-load takes time and will still not be very precise. It is therefore an object of the invention to provide for an automatic and precise adjustment of the accumulator when the pressure of the drive fluid varies.

The invention will be described in detail with reference to the accompanying drawings in which two embodiments of a percussion tool according to the invention are shown as examples.

In the Figures, FIG. 1 is a longitudinal section along line 1—1 in FIG. 2 through one embodiment of the percussion tool, FIG. 2 is a transverse section along line 2—2 in FIG. 1, FIG. 3 is a fragmentary longitudinal section along line 3—3 in FIG. 2, FIG. 4 shows diagrammatically the percussion tool of the FIGS. 1—3, and FIG. 5 shows diagrammatically the other embodiment. Corresponding details have been given the same reference numeral in the various figures.

The percussion tool shown in the figures is made as a rock drill which consists of a housing 10 which constitutes a cylinder 11 for a reciprocating hammer piston 12. It is to be understood that the expression "cylinder" is used in the claims and in the specification to define the chamber in which the hammer piston reciprocates, and that the cylinder is not limited to have a pure cylindrical form, but may for instance be stepwise cylindrical. The hammer piston consists of a cylindrical rod with two piston portions 13, 14 having piston surfaces 15, 16. These piston portions have annular pressure equalizing grooves (shown in FIG. 4 only) which improve the centering in the cylinder and counteract oblique load of the piston if such load should occur. The portion of the hammer piston that extends rearwardly from the piston portion 14 is denoted by 12a, and the portion that extends forwardly from the piston portion 13 is denoted by 12b. The rod portion between the piston portions 13, 14 is denoted by 12c. A sleeve 17 is arranged in the forward end of the housing 10 so as to receive a shank 18 of a drill string 19 (shown as a drill steel). The sleeve 17 has internal splines and the shank 18 has mating external splines so that the drill string will rotate conjointly with the sleeve. A pressurized hydraulic fluid driven rotary motor 20, e.g. a sliding vane motor is attached to the housing 10 and it has a drive gear 21 which rotates the sleeve 17 over a gear 22 which is partly concealed in FIG. 1. Flushing fluid is supplied to the axial hole of the drill rod 19 through a swivel 23 which is slipped onto the shank 18. The shank 18 can be moved into the sleeve 17 to abut against an axially movable anvil block 24 which then moves to a rear stop position as shown in FIG. 1. A spring 78 retains the shank 18. The anvil block is an anvil to the hammer piston 12 but it could be omitted and the end surface of the shank 18 be an anvil to the hammer piston. A rear annular pressure chamber 25 is defined by the cylinder 11, the rod portion 12a, the piston surface 16 on the piston portion 14, and the front surface of a sealing ridge 26. The oil that leaks through the circular clearance space between the sealing ridge 26 and the rod portion 12a is collected in a circular groove 27, which, as shown in FIG. 4, may be drained directly to tank through a conduit not shown but, advantageously, may be drained by means of a pump so that there will be suction in the groove. By this arrangement the fluid is prevented from leaking to a closed end chamber 28 which encloses the rear end of the rod portion 12a. The cylinder 11 has another circular groove 29 to which an accumulator 30 is connected by means of a passage 31. By means of a passage 32, another accumulator 33 is connected to the cylinder 11 at the rear of the pressure groove 29.

A forward annular pressure chamber 34 is defined in the same way by the cylinder 11, the rod portion 12b, the piston surface 15 on the piston portion 13, and the rear surface of a circular sealing ridge 35. Thus, the volumes of the pressure chambers 34, 25 varies with the position of the piston 12. Advantageously, a collecting groove 36 is drained in the same way as the collecting groove 27. Outside the collecting grooves 27, 36 sealing rings are disposed so as to seal against the hammer piston. The accumulator 30 has a piston 30a which is biased by a spring unit 41 in the form of a pile of Belleville springs. The pile 41 of springs is braced between this piston 30a and a piston 70 the latter being loaded by the pressure in a passage 71 with a narrow restriction 72.

In FIG. 4, an accumulator 38 is also shown. This accumulator has a piston 38a which is connected to an annular pressure groove 37 by means of a passage 39 and which is biased by a spring unit 73 (shown also in FIG. 2) in the form of a pile of Belleville springs. As shown diagrammatically in FIG. 4, the spring unit 73 is braced between the piston 38a and a piston 74 the latter being loaded by the pressure in a passage 75 with a narrow restriction 76. The passages 71 and 75 are branches of a passage 77 which is connected to a main supply conduit 43.

A distributing valve in the form of a slide 42 is supplied with pressurized hydraulic fluid through the main supply conduit 43. An accumulator 44 is continuously connected to the main supply conduit 43 so as to receive the hydraulic fluid that is supplied when the slide...
42, as shown in FIG. 2, blocks the main supply conduit 43 for a short moment when shifting position. The accumulators 32, 44 are so loaded that they are substantially inactive at the pressures at which the accumulators 30, 38 are active. The main supply conduit 43 leads to an annular inlet chamber 45 in the cylinder of the valve. The cylinder of the valve has also two annular outlet chambers 46, 47 to which return conduits 48, 49 are connected. These return conduits lead to a non-illustrated pump which pumps hydraulic fluid from the annular outlet chamber 45 to the annular inlet chamber 46, 47. The slide 42 is positively retained in its position because the pressure in the return conduit 48 is transmitted to the holding surface 58 of the slide. When the hammer piston 12 moves on forwards (to the right in FIG. 4), the control passage 56 is again opened so as to drain now into the drain passage 65. Then, when the piston portion 14 passes the opening of the control passage 57, it uncovers this opening into the rear pressure chamber 25 from which the pressure is conveyed through the control passage 57 to the end face 55 of the slide. Now, the slide shifts to its non-illustrated position (to the left in FIG. 4) so that the forward pressure chamber 34 is pressurized while the rear pressure chamber is drained 25. This takes place just before the hammer piston strikes the anvil block 24 and thus the hammer piston 12 is appreciably retarded before delivering the blow. The hammer piston 12 rebounds after impingement, but the flow of pressurized hydraulic fluid supplied through the supply passage 51 to the forward pressure chamber 34 is at the beginning more than what can be received by the pressure chamber. Therefore, the accumulator 38 receives fluid at the beginning, but when the hammer piston 12 has reached the speed which corresponds to the flow supplied by the pump, the accumulator 38 starts discharging to the pressure chamber 34 so as to further increase the speed of the hammer piston 12. The slide 42 remains in its left-hand position because the pressure in the supply passage 51 is conveyed to the holding surface 60 of the slide. The control passage 57 is already in communication with the drain passage 65 when the piston surface 15 of the piston portion 13 passes the branch passage 56a of the control passage 56 so that the pressure in the forward pressure chamber 34 is transmitted through the control passage 56 to the end face 54 of the slide. The slide 42 shifts therefore to its right-hand position (shown in FIG. 4) where it remains because of the fluid pressure upon the holding surface 58. Pressurized hydraulic fluid is now supplied from the inlet 43 to the rear pressure chamber 25 and the hammer piston 12 retards due to the hydraulic fluid pressure upon the piston surface 16. Now, the accumulator 30 receives the hydraulic fluid flow supplied through the supply conduit 50 as well as the hydraulic fluid that is forced out from the pressure chamber 25 because of the movement to the rear of the hammer piston 12 which decreases the volume in the pressure chamber 25. In normal return strokes, the hammer piston 12 returns because of the pressure in the rear pressure chamber 25 without moving so far to the rear that the piston surface 16 reaches the rear edge of the groove 29, and the accumulator 30 is therefore continuously in communication with the pressure chamber 25. The accumulator 32 is biased so as to be inactive during this operation. The accumulator 30 is supplied with pressurized hydraulic fluid also during the first part of a work stroke since the pressurized hydraulic fluid flow through the supply passage 50 is larger at the beginning than what can be received by the pressure chamber 25. However, when the hammer piston 12 reaches the speed that corresponds to this supplied flow, the accumulator 30 starts supplying pressurized hydraulic fluid to the pressure chamber 25 and thus further increases the speed of the hammer piston further.

Thus, kinetic energy of the hammer piston from the return stroke is stored in the accumulator 30 and this energy is delivered back to the hammer piston during the subsequent power stroke.
If the drill rod shank 18 is not pressed against the anvil block 24 or if no shank 18 is inserted through the sleeve 17, the hammer piston will not rebound even if the anvil block 24 should happen to be in the rear position, but it will proceed to its normal forward return point and its piston surface 15 will pass the groove 37 whereby the accumulator 38 and the supply passage 51 are shut off from the forward pressure chamber 34 which now acts as a damping chamber. However, the supply passage 51 is still in communication with the accumulator 38 through the pressure groove 37. The pressure in the pressure chamber or damping chamber 34 increases instantaneously to a degree which might be two or three times the pump pressure. Thereby, a hydraulic fluid flow is forced to the rear through the clearance space between the piston portion 13 and the cylinder 11 and out into the pressure groove 37. Simultaneously the slight amount of hydraulic fluid increases which passes the sealing ridge 35 out into the collecting groove 36. The hammer piston 12 will therefore retard rapidly but softly to a stop. Owing to leakage around the piston portion 13 from the pressure groove 37 to the pressure chamber 34, the hammer piston will then move slowly rearwards until the piston surface 15 reaches the pressure groove 37, at which moment the acceleration of the hammer piston 12 starts as a normal return stroke.

If the pressure of the drive fluid is varied, the pistons 70, 74 will vary their load on the springs 41, 73 in proportion to the variation in the pressure of the drive fluid. The pistons 30a, 38a of the accumulators 30 and 38 will therefore always be pre-loaded to an optimal degree. The restrictions 72, 76 shall be so narrow as to prevent the pistons 70, 74 to oscillate in time with the pistons 30a, 38a.

The accumulator 32 is so biased that it is normally inactive and it starts accumulating only in case the hammer piston will have such a hard rebound that the piston surface 16 moves behind the rear edge of the pressure groove 29 so that the pressure chamber 25 is isolated from the accumulator 30 and from the passage 50 and the pressure in the pressure chamber increases to for instance twice the normal pressure. However, the accumulator 30 will still be in communication with the passage 50.

The accumulator 32 can alternatively be omitted and part of the pressure chamber 25 which is behind the pressure groove 25 can be used as a dampering chamber only.

In FIG. 5 another percussion tool is shown and details corresponding to the details in FIG. 1–4 have been given these figures. Like FIG. 4, FIG. 5 is only diagrammatic and the various details are not shown in the same scale. The accumulator 30, for example, is shown far too small in relation to the pressure chamber 25.

The piston 12 is in this form a differential piston, the rear piston surface 16 of which has a greater area than the forward piston surface 15. The forward pressure chamber 34 is in continuous communication with the main inlet 43 while the rear pressure chamber 25 is alternately in communication with the main inlet 43 and the main outlet 65. As distinguished from the valve 42, the valve 79 is a differential valve and one of the ends of its slide 80 is continuously acted upon by the inlet pressure through a control passage 82 while the other end face 83 is alternately loaded by and relieved of pressure through a control passage 84 which is controlled by the piston portion 13.

The accumulator 30 is identical with and has the same function as the corresponding accumulator 30 in FIG. 4, but the accumulators 30a and 32 in FIG. 4 are replaced by a single accumulator 85. There is no accumulator shown that corresponds to the accumulator 32 in FIG. 4.

The invention is not limited to the illustrated embodiments but it may be varied in many ways within the scope of the claims.

What I claim is:

1. A hydraulic fluid actuated percussion tool comprising a housing forming a walled cylinder, a hammer piston reciprocating in the cylinder and adapted to deliver impact energy to a work tool, a first variable volume pressure chamber for urging the piston rearwards defined between the piston and the walls of the cylinder, a second variable volume pressure chamber for urging the piston forwards defined between the piston and the walls of the cylinder, means for connecting at least one of the pressure chambers to a source of hydraulic fluid for alternately pressurizing and depressurizing said chambers, an accumulator comprising a chamber connected to at least one of said pressure chambers and loaded by spring means including a spring in response to the pressure in the pressurized hydraulic fluid supplied to the percussion tool, said spring means comprising a device loaded by transmitting the pressurized fluid through a restricted passage, whereby oscillations of said device in timed relationship to the periodical operation of the accumulator will be prevented, said device comprising an axially movable piston coaxial with another axially movable piston closing the ends of the accumulator chamber, and said spring being braced between said two coaxial pistons.

2. A percussion tool as defined in claim 1, in which said accumulator is connected to said second pressure chamber.

3. A percussion tool as defined in claim 1, in which the spring means comprises a pile of Belleville-springs.

4. A percussion tool as defined in claim 1, in which a distributing valve is connected to the source of pressurized hydraulic fluid, to the first pressure chamber by means of a first connection conduit, to the second pressure chamber by means of a second connection conduit, said distributing valve having a first position in which it pressurizes the first connection passage and depressurizes the second connection conduit and a second position in which it depressurizes the first connection passage and pressurizes the second connection conduit, the valve being adapted to shift from its first position to its second position while said hammer piston is moving backwards, and said accumulator being connected to the second pressure chamber.

5. A percussion tool as defined in claim 4, in which said second connection conduit is connected to an annular groove in the cylinder and said accumulator is connected to the second pressure chamber by means of a conduit which is connected to this annular groove.

6. A percussion tool as defined in claim 4, in which a second accumulator is connected to the first pressure chamber and this second accumulator is loaded by a second spring means which is adapted to be automatically pre-loaded in response to the pressure in the pressurized hydraulic fluid supplied to the percussion tool.

7. A percussion tool as defined in claim 6, in which said second accumulator has an axially movable piston and said second spring means is braced between this piston and another piston which, through a restricted passage, is loaded by the pressure in the hydraulic fluid supplied to the percussion tool.