United States Patent

[11] 3,612,191

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[54] **PERCUSSION DRILLING TOOL**
12 Claims, 16 Drawing Figs.

[52] **U.S. Cl.** **173/73,** 173/136, 175/320

[51] **Int. Cl.** **E216 1/00**

[50] **Field of Search** 173/73, 90, 134–139, 162; 175/293, 320

[56] **References Cited**

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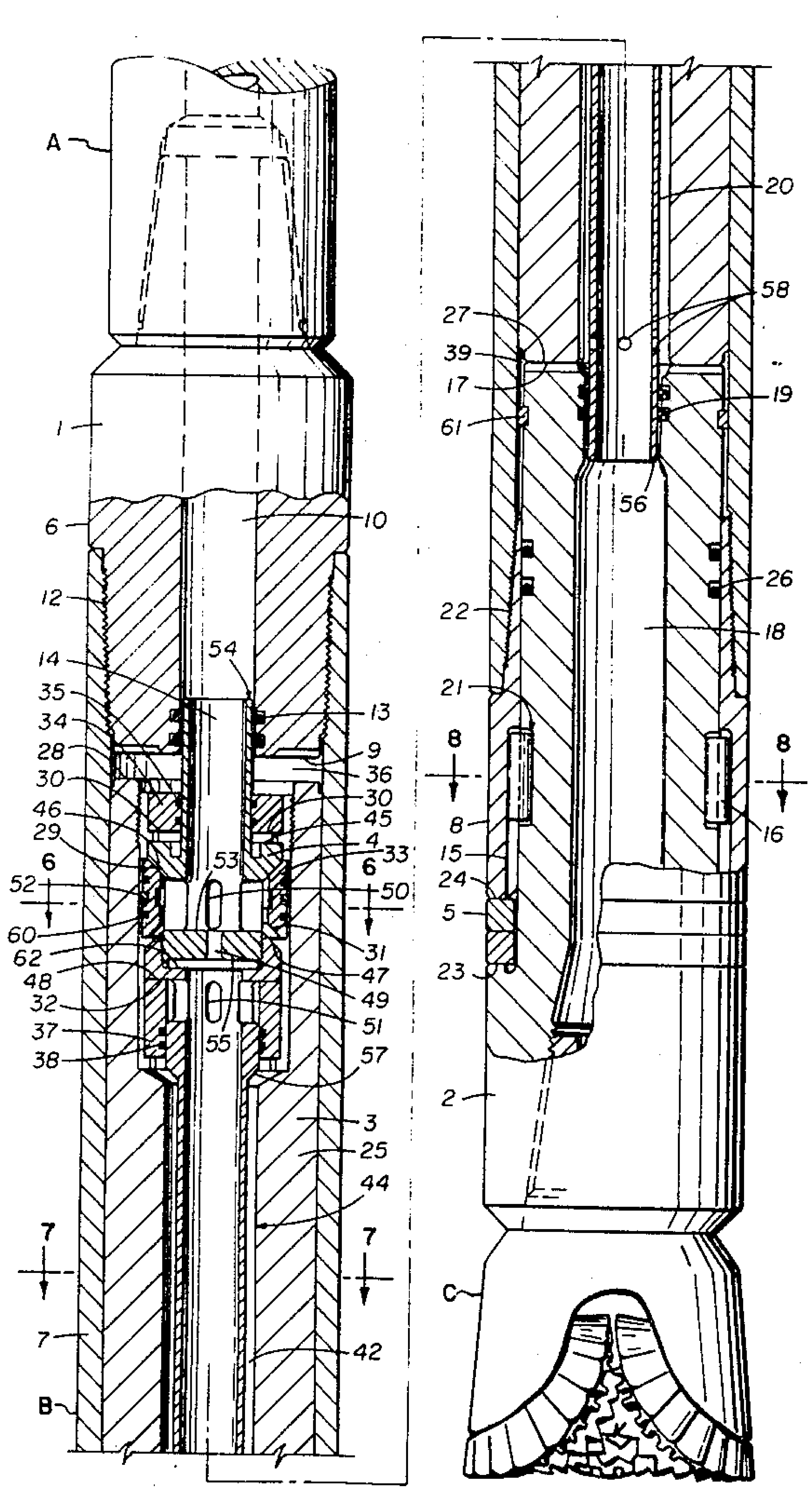
Primary Examiner—James A. Leppink
Attorney—Richards, Harris & Hubbard

ABSTRACT: A unique percussion motor is disclosed that is operable on liquid or gaseous fluid under pressure for rotary drilling of oil, gas, and water wells, geophysical holes, open strip mining blastholes, construction holes and the like for greatly increasing the rate at which said boreholes are drilled. This comparatively simple prime mover produces sustained high-frequency, high-longitudinal-force spikes on a drill bit by synchronizing application of percussive force and drill collar weight energy and superimposes one force upon the other to obtain instantaneous anvil accelerations of much greater magnitude than either force could, acting separately, to produce rock-crushing forces of greater effectivity. Rebound of tough elastic masses is also used to decided advantage for conserving system energy and applying it usefully, allowing this device to adjust cycle frequency and percussive blow force to the hardness of the formation being drilled. This tool has essentially a positive displacement allowing use of properly sized drill bit jet nozzles for hole bore cleaning.

This tool also incorporates a unique tubular single valving element and seat arrangement located totally in hammer allowing use of maximum hammer surface area for fluid-biased hammer accelerations in both directions, simultaneously and alternately permitting fluid flow to one pressure chamber while exhausting the other, and has an unusually fast, short-stroke valve-shifting action at the end of each hammer stroke, thereby eliminating precise part dimensions, hammer stroke, and anvil locations as well as reducing valve-timing and erosion problems. Limited-life tool components consisting of valve and valve seats are separate, easily replaceable and disposable.



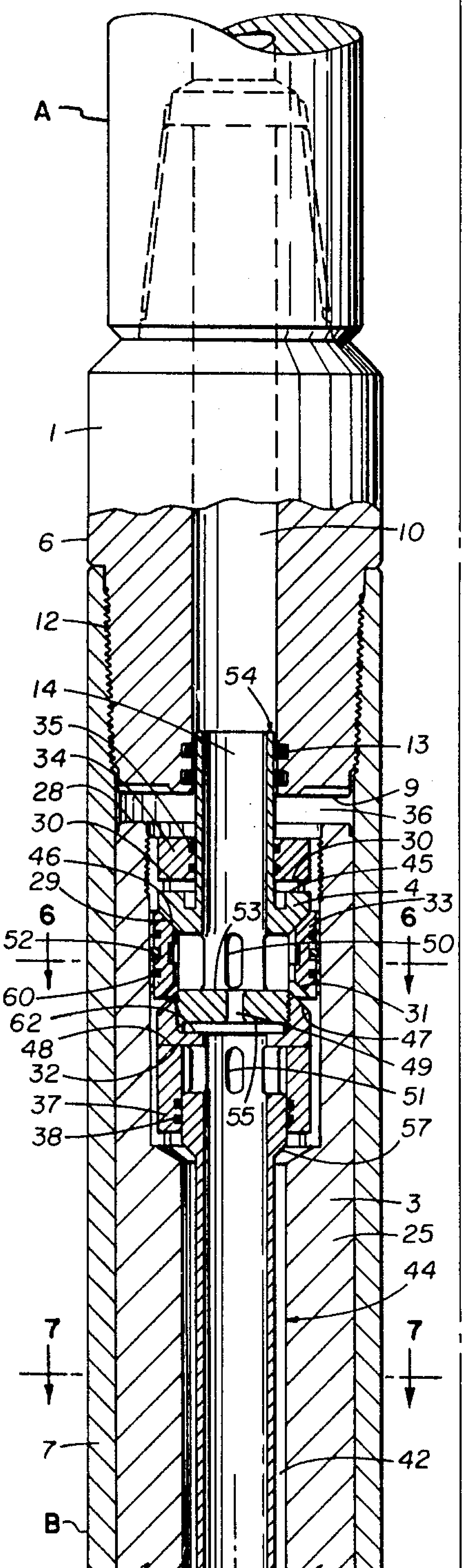


FIG. 1

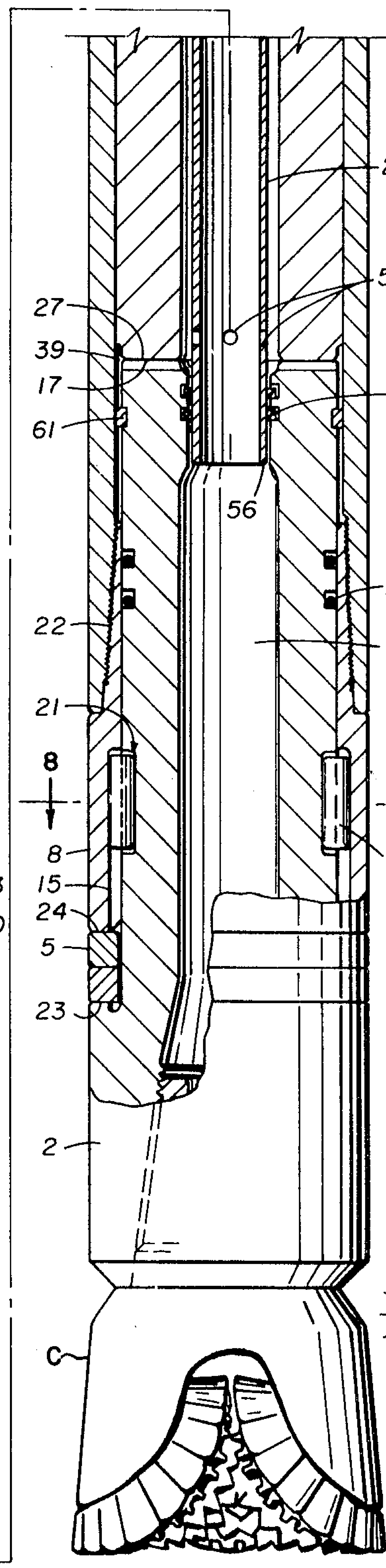


FIG. 2

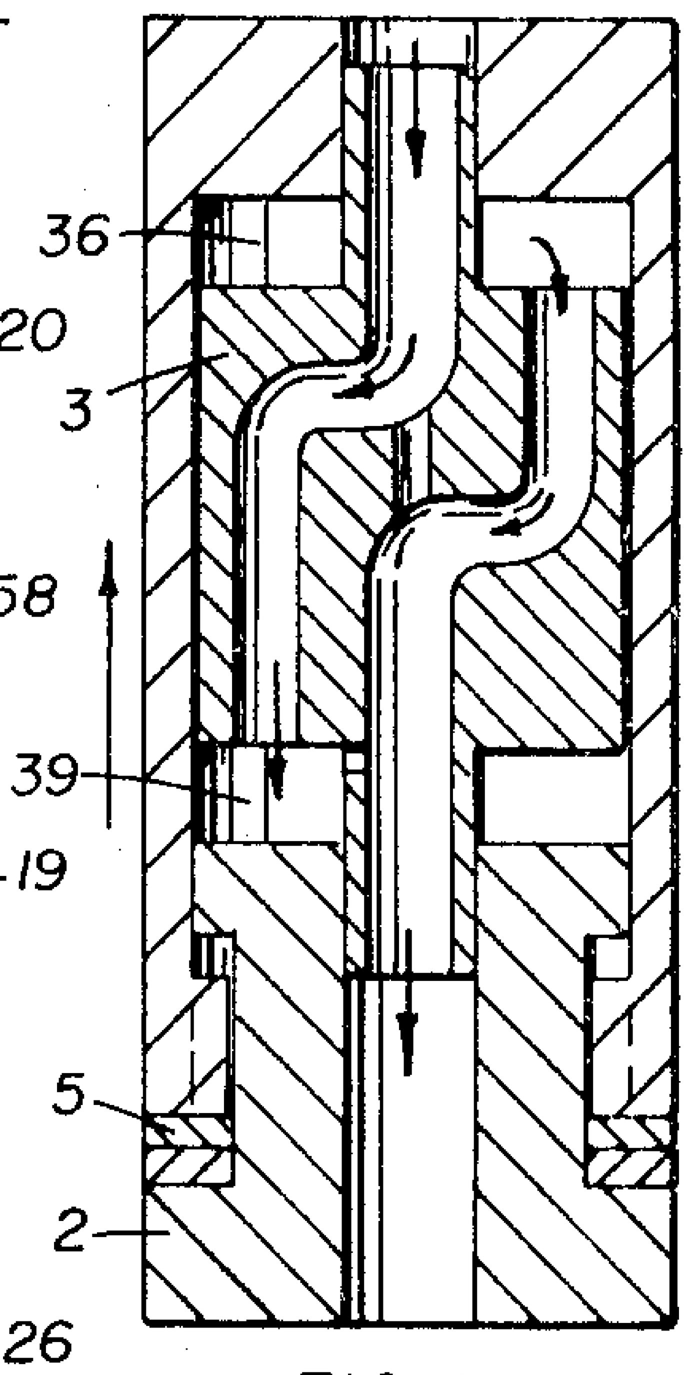


FIG. 3

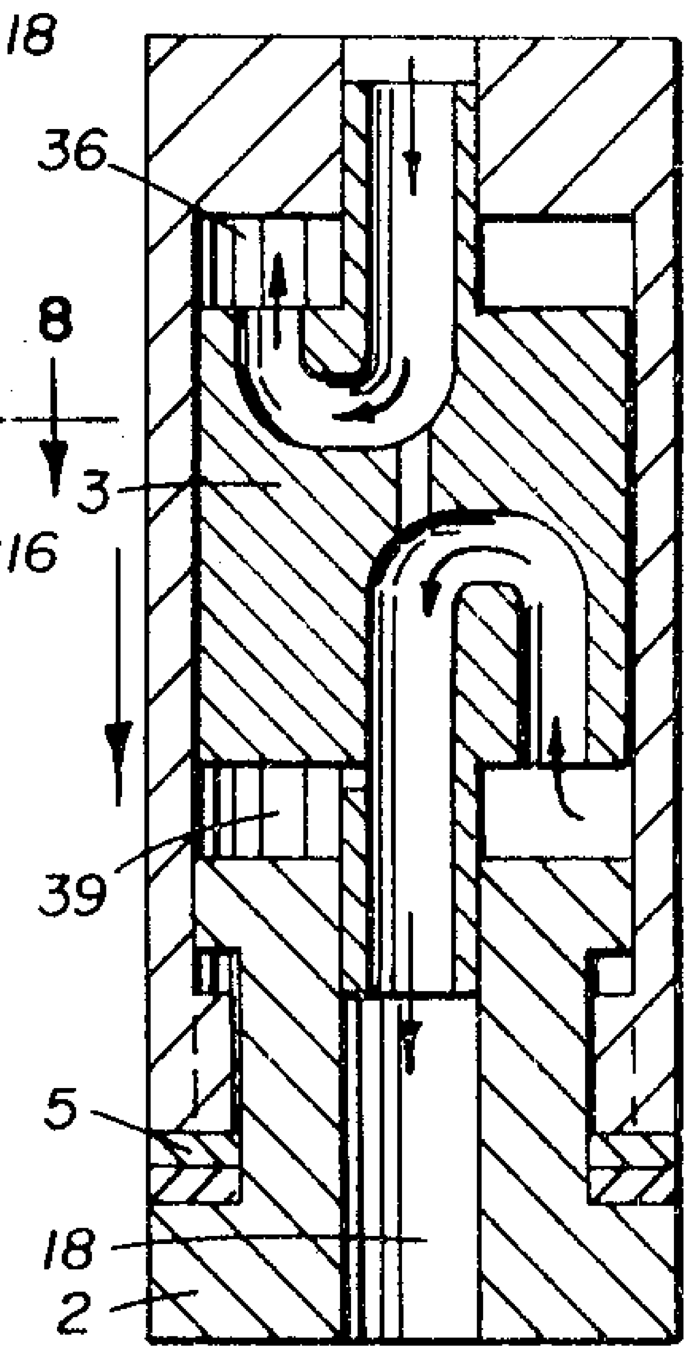


FIG. 4

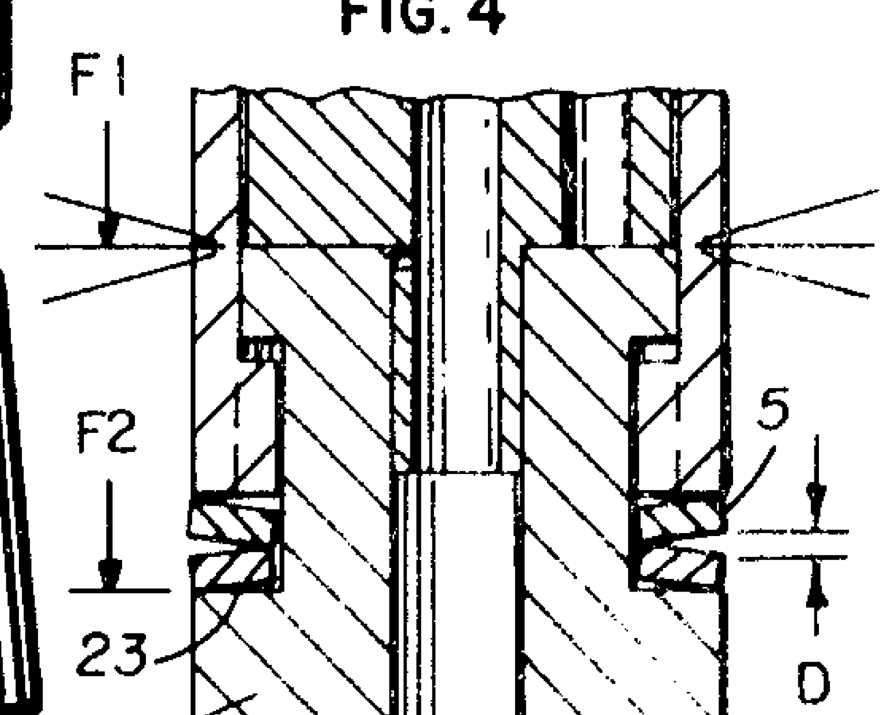


FIG. 5

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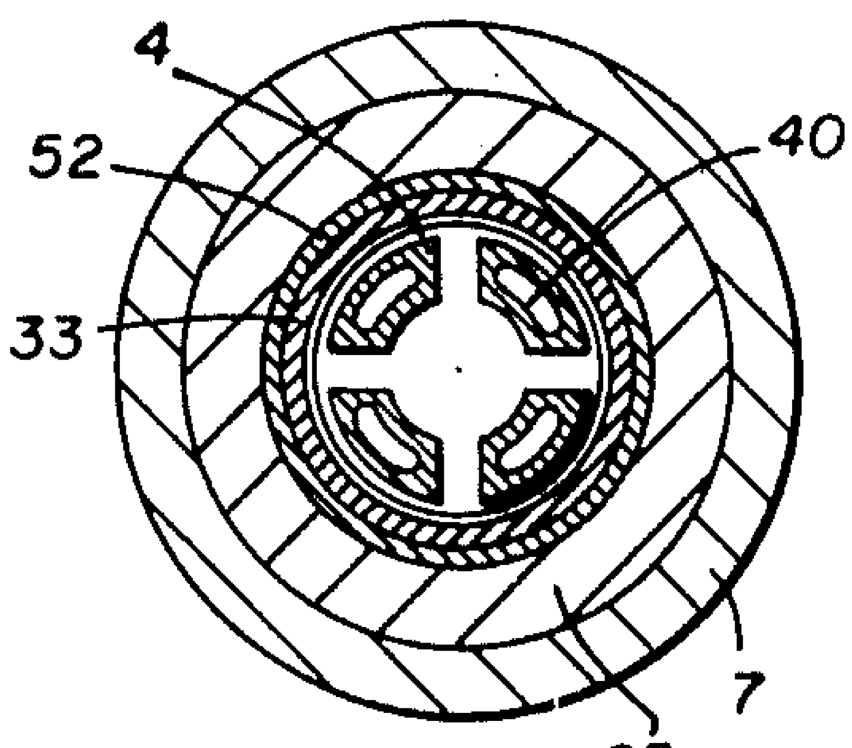

FIG. 6

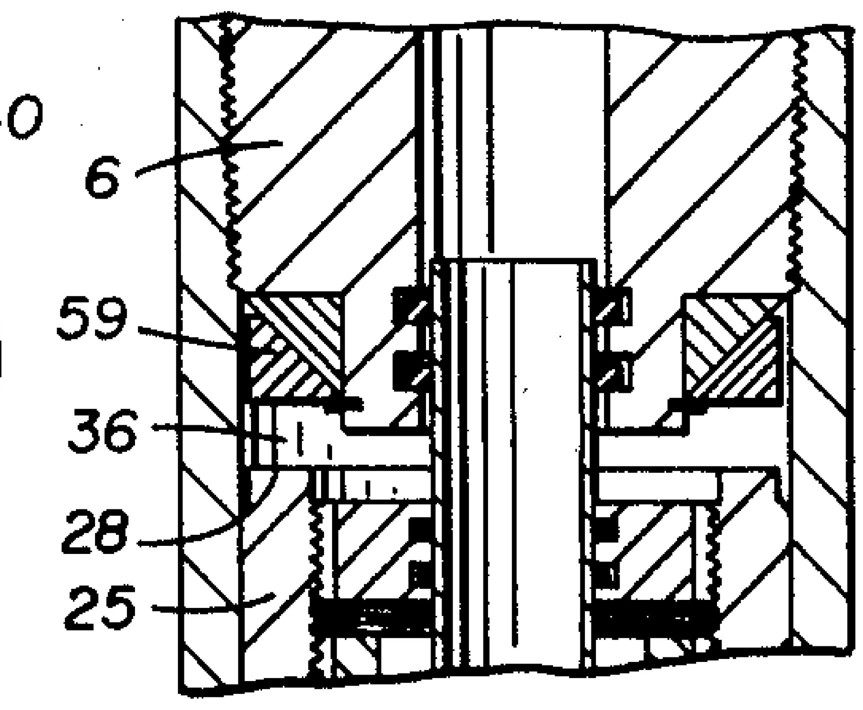

FIG. 12

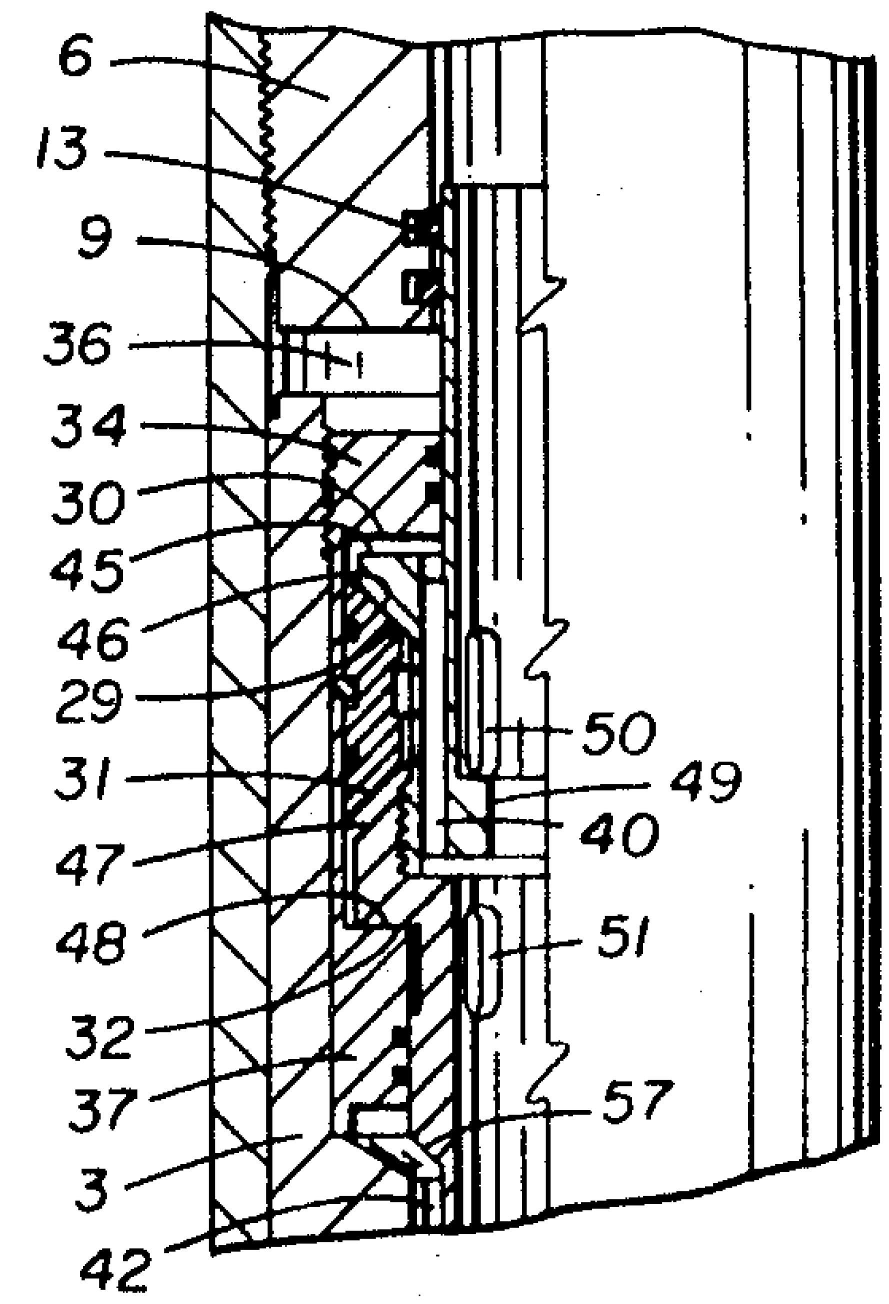

FIG. 10

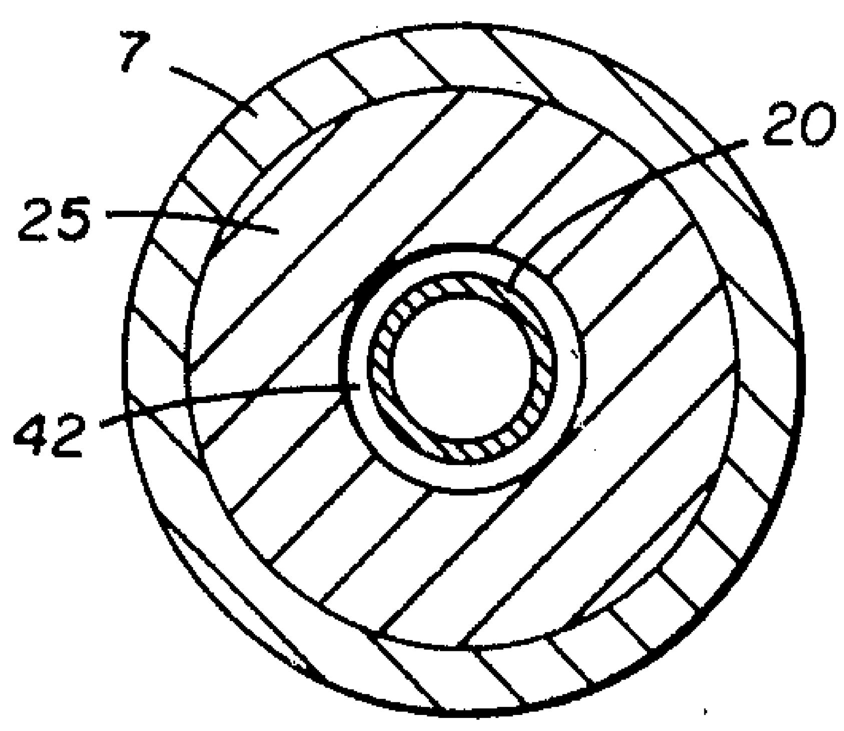

FIG. 7

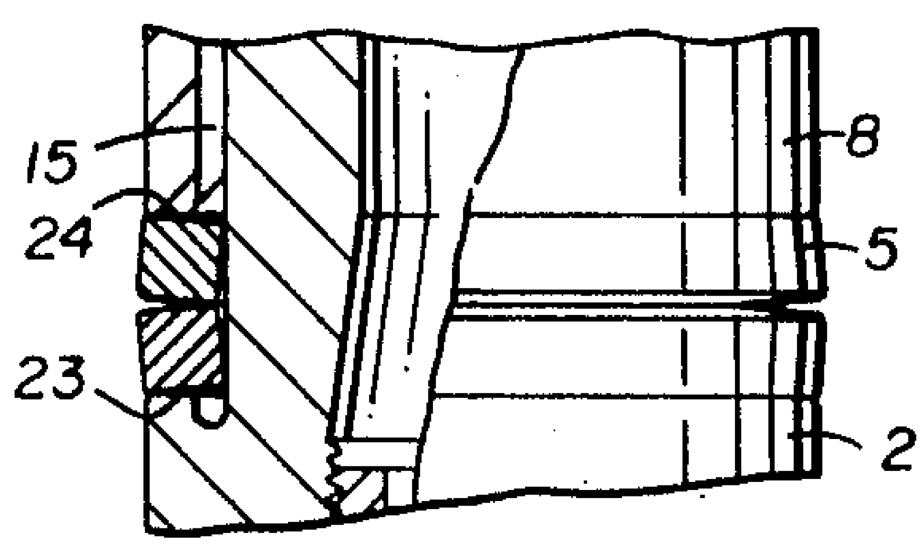

FIG. 9

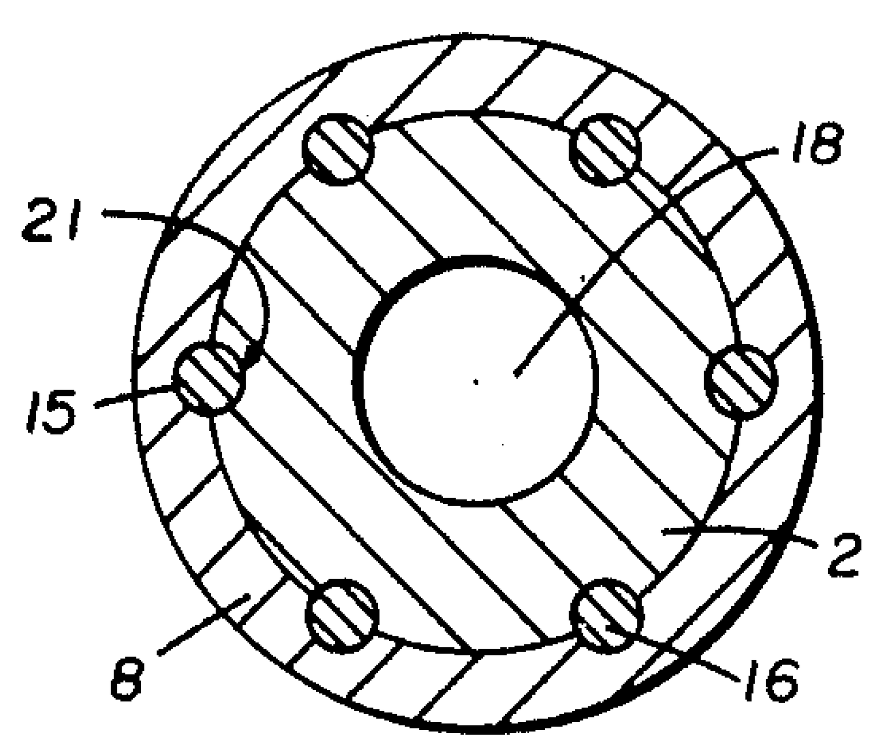

FIG. 8

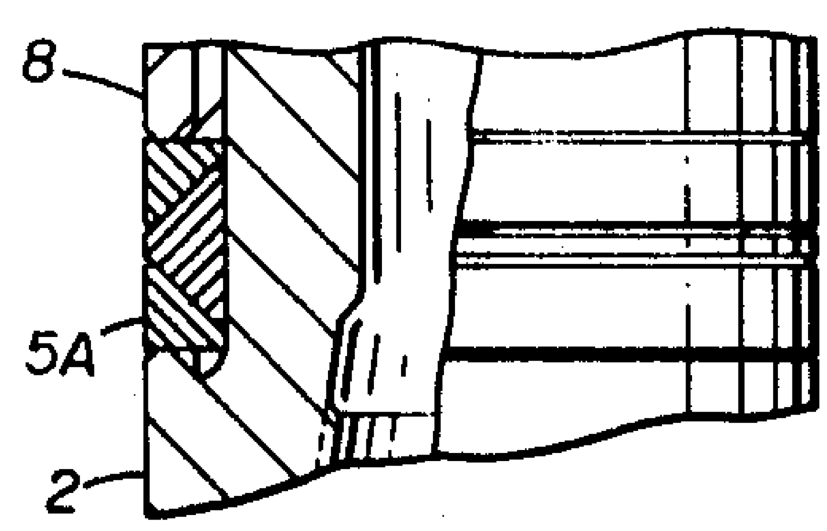

FIG. 14

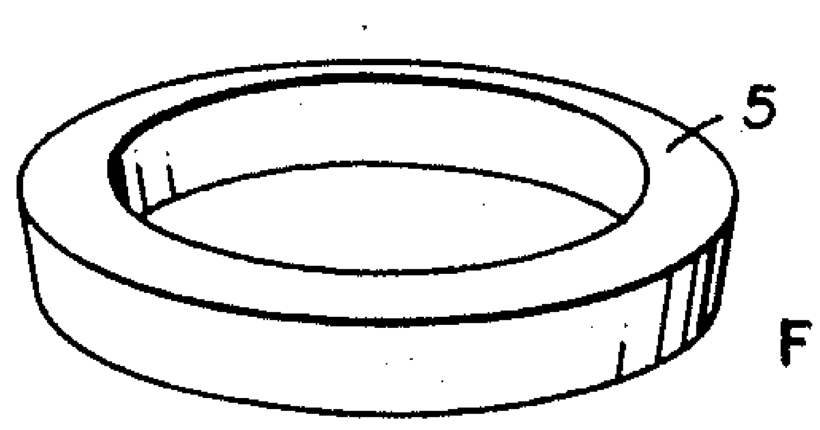

FIG. 11

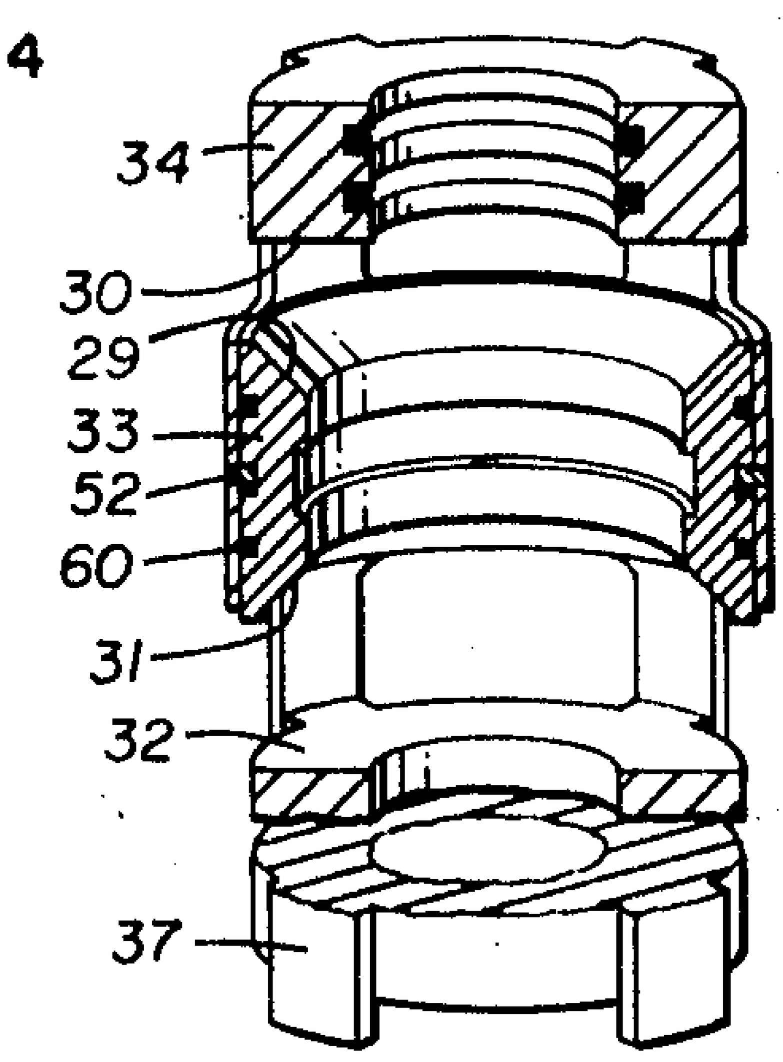

FIG. 16

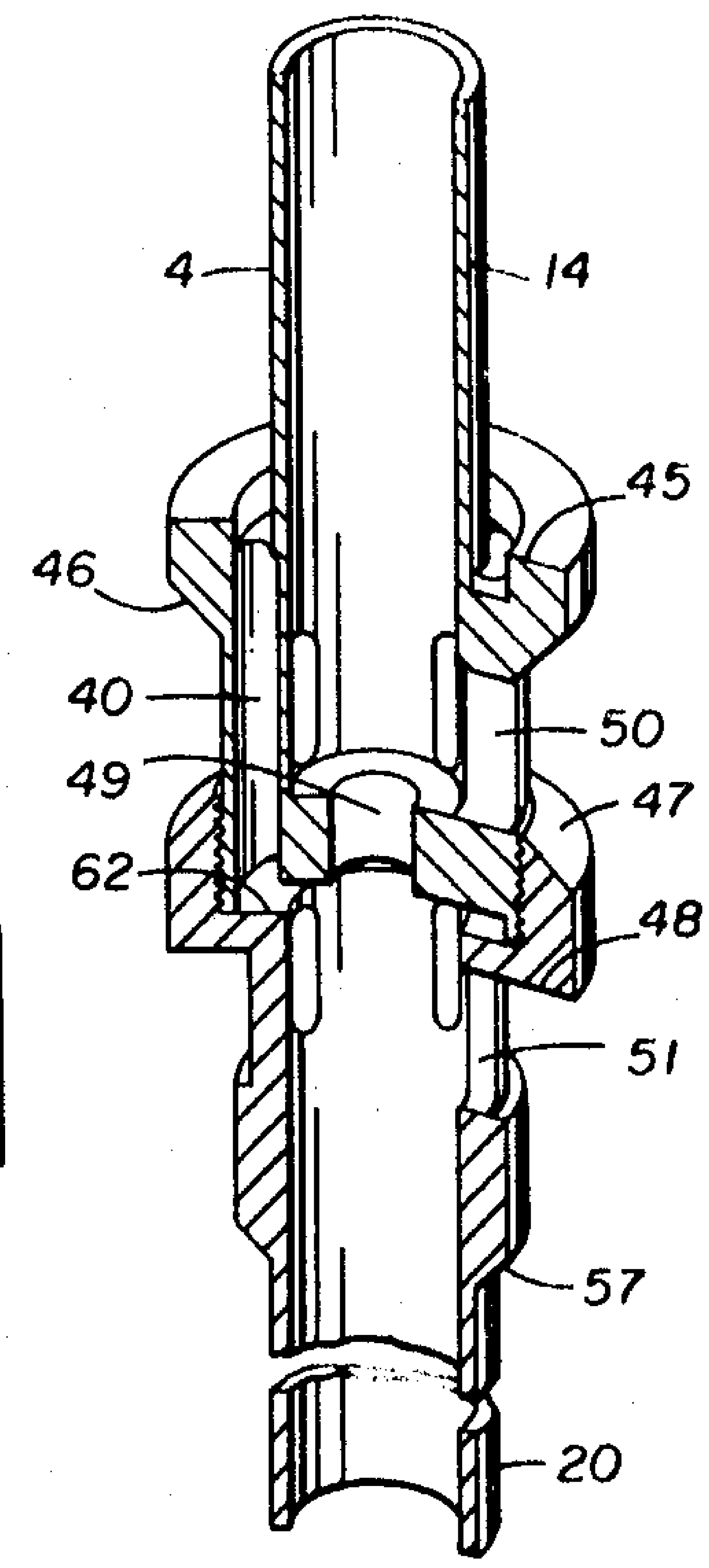

FIG. 15

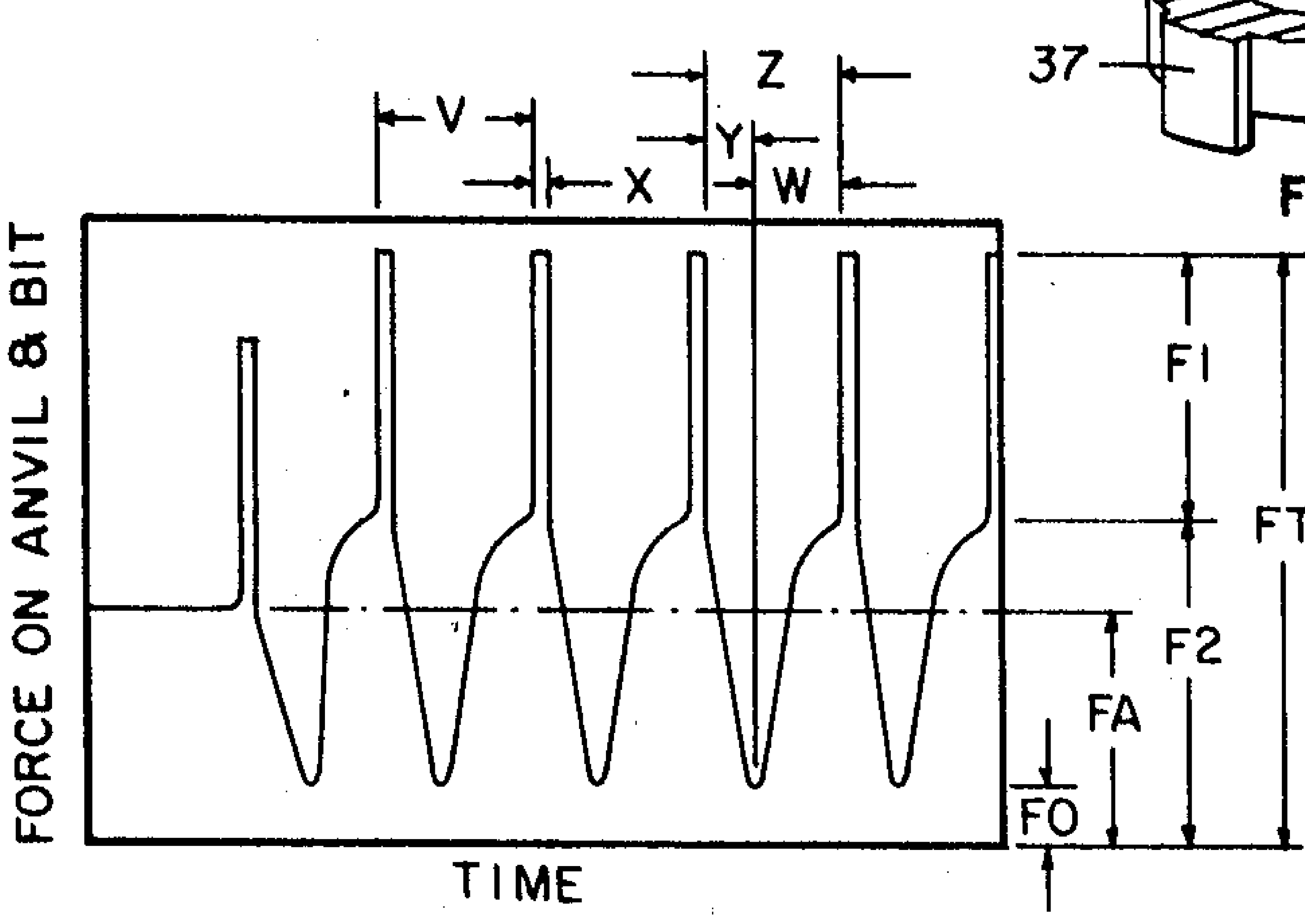

FIG. 13

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PERCUSSION DRILLING TOOL

BACKGROUND OF THE INVENTION

Although the concept of percussion drilling is old, the need exists today more than ever before for a tool that satisfies the rigorous requirements of a truly effective and successful tool. With wells being drilled deeper and into hard, virtually impenetrable earth strata, it is imperative that improvements over existing devices be made since generally today they are still lacking in effectivity and are short lived.

Despite many frustrating failures of man and machine in this area, more intense liquid percussion tool development has spanned this decade, and has progressed substantially as new insights occurred, and new materials, processes, and techniques were proved and adopted. These devices are progressing toward market and promise to be in general usage in a few years.

In the past, emphasis has been on basic approaches to rock attack, compatibility with existing equipment and systems, suitable tool operations, valve erosion prevention and other fundamental design parameters. With many of these things having been determined, the effort has shifted to increasing tool effectivity, increasing service life of major as well as minor components, making tools field serviceable and reducing manufacturing costs to make them economically justifiable.

The trend today is to power generation for drilling at or near the bit where the work is to be done because of several factors. One is due to power losses and considerable drill pipe wear incurred mostly from the pipe string rubbing the borehole wall between the bit and the generally remote surface locations, ranging today up to and above 20,000 feet deep as in oil and gas well. Another factor in this trend is that bit life per foot of hole drilled can be increased when drilling with a percussion tool, thereby reducing the number of times the drill pipe has to be pulled out of the borehole to replace the bit. Another contributing factor is that straighter boreholes are possible, thus diminishing the chances of troubles with stuck or twisted-off drill pipe, casing and well-pumping unit problems. The major factor, however, influencing the endeavor to produce more effective percussion tools by today's standards, is the promise of well completion in less time with the associated reduction of labor, material, and equipment costs.

Since over 90 percent of the oil and gas wells being drilled today use liquid, usually water of an oil-based mud, as the system-circulating medium, tool operation on said liquid is naturally the area of greatest concentrated effort but also harder to achieve because of the inelasticity or noncompressible characteristics of the liquid and the circulated solids that pass through the tool causing erosion and wear. Generally the operating conditions of this type tool are severe, considered from every aspect, and historically have not lived up to the required standards of being reasonably long lasting, having the desired effectivity, and being comparatively simple in construction.

AMONG PRIOR ART CONSIDERED

U.S. Pat. No. 2,661,928, **Topanelian, Jr., Dec.** 8, 1953;
U.S. Pat. No. 2,774,334, **Cline, Jr., Dec.** 18, 1956;
U.S. Pat. No. 2,859,733, Bassinger et al. Nov. 11, 1958;
U.S. Pat. No. 3,410,353, Martini, Nov. 12, 1968;
U.S. Pat. No. 3,327,790, Vincent et al. June 18, 1967.

SUMMARY OF THE INVENTION

This invention relates to prime movers of the reciprocating impact type and, more particularly, to a fluid-actuated percussion drilling tool capable of converting fluid energy to mechanical energy and to a means and method of increasing the effectivity and power generation of same and having improved construction and operating characteristics.

A primary object of this invention is to provide a prime mover of the reciprocating type that will maintain drill collar weight force on a drill bit during and after the percussive impact force is applied thereby adding or superimposing said forces, one on the other for more effective formation cleavage.

In the drilling industry an arrangement where a percussion tool is installed in the drill string with a bit attached at its lower end and drill collars attached above the tool is generally assumed to be the best application of drilling technique. This, however, has proved to be untrue with percussion tools known today. With use of better test instruments and greater insight into what truly happens during operation of the drilling system, it has been found that all known percussion tools produce a vibratory movement or oscillate axially and while some worthwhile increase in penetration rate is produced, it is far from being the most advantageous. It was found that at percussion and with the slightest percussion tool anvil extension, drill collar weight force is relieved from the bit for the length of time of said anvil and bit extension. In other words, drill collar weight rests on the anvil between percussive blows but at percussion, the anvil, due to the percussive force, is driven out from under the drill collar weight. This is due to the fact that the percussive blow is very fast, on the order of 0.0002 second, and in this increment of time the anvil can achieve an instantaneous acceleration rate many times the gravitational acceleration rate of a free-falling body, such as that of the drill collar weight regardless of its mass and, thus, cannot maintain its weight force on said anvil. Drill collar weight then does not enhance percussion tool operation as it has been applied, but merely produces an interchange of percussive force and drill collar weight alternately applied to the bit with little advantage. Percussive blow force actually offsets or negates the drill collar weight that would normally be applied to the bit. It should be noted that the drill collar weight is not maintained on the anvil and the bit, and, particularly, that no anvil and bit accelerations are possible due to drill collar weight during percussion in the situation described. When percussive blow force is maximum on the bit, drill collar weight force on the bit is essentially zero and when drill collar weight force on the bit is maximum percussive blow force on the bit is essentially zero and so uncoordinated, unsynchronized forces come into play producing a vibrating action of little consequence.

To synchronize percussive blow force and drill collar weight energy applications, extensive mechanical force storage and transfer means in the form of anvil thrust rings have been provided on the percussion tool between the driver sub and the anvil shoulder. These anvil thrust rings transfer and maintain drill collar weight force over a certain distance and serve to accelerate the anvil downward when bit penetration forces are achieved. This transferred force can follow and add to instantaneous percussive force and anvil accelerations after percussion. This arrangement prevents cancellation of percussive force by reduction of drill collar weight force applied after the most minute anvil extension and will achieve the increased bit effectivity.

The anvil thrust rings are thick, high-force, low-inertia, short-reaction-time frustoconical or other type ring springs as manufactured and will exert forces in resistance to loadings tending to flatten, deflect, or distort them and will exert said forces through their shape change. They compress, flatten or distort under drill collar load and transfer said load to the bit. In the situation as before percussion, force on the bit is drill collar load force, but at percussion, force on bit is drill collar weight force as applied by anvil thrust rings plus percussive blow force. It is the objective here to stack these forces one on the other at percussion to make them more useful rather than let them be canceled at percussion. Drill collar weight force applied between percussions is stored for use at percussions. Anvil acceleration forces at percussion can easily be doubled since drill collar weight energy stored in the anvil thrust rings is also applied at percussion and will produce high-magnitude rock-crushing forces. The instantaneous high forces become very worthwhile and effective when drilling the harder formations and crushing forces over a distance need to be applied for good bit tooth penetrations.

The anvil thrust rings also provide other desirable features such as allowing the drill collars to move downward more smoothly and uniformly, tending to cushion their movement as energy is absorbed by the anvil thrust rings and thereby isolating the drill string above from rapid anvil movements. They also prevent any negative bit-bearing loadings and tend to stabilize the torque required to rotate the drill stem and hence reduce dynamic torsional stress in the drill string.

Another object is to provide a device of this type that will generate more mechanical power output for a given hydraulic power input by conserving system energy by use of the rebound of tough elastic masses and a device that will automatically adjust its operation according to the formation being drilled.

To increase the output energy of this percussion tool, rebound or bounce of tough elastic masses are used. Strike surfaces are provided on the upper surface of the hammer and the lower surface of the top sub and arranged for proper coaction. The hammer at the termination of its upstroke will impact and rebound downwardly so that instantly it has reversed direction and attained appreciable reverse velocity. The hydraulic horsepower used to accelerate the hammer upwardly is not lost or wasted. The kinetic energy of the hammer is conserved or retained therein and since the hammer has an instantaneous initial downward velocity, the pressure fluid application above hammer will raise its velocity from its rebound velocity and thereby achieve a greater terminal velocity at end of the hammer downstroke for a percussive blow of greater magnitude. Likewise, at the end of the downward hammer travel, collision with the anvil occurs and a percussive blow is transferred. Now if the formation is relatively soft and large bit tooth penetrations are achieved, most percussive blow force is used to crush formation and hence little or no hammer rebound off anvil. But, if formation is very hard, with little or no bit tooth formation penetrations, hammer rebounds off anvil thereby conserving at least part of its kinetic energy and has an instantaneous reverse velocity to begin its upstroke. Now, allowing for frictions and various other energy losses, hammer, through successive rebounds off top sub and anvil together with pressure-fluid-pulsed accelerations in between, can achieve considerable velocity. Horsepower increases by the square of the hammer velocity as does the kinetic energy of a percussive blow, and since the anvil serves as the percussion tool energy output all input hydraulic horsepower will be used on each percussive blow as in softer formation or retained in the tool to increase frequency of operation and percussive force and expended this way tending to make the toolwork harder in the harder formations. Basically, the idea conceives of a mass bouncing between two opposed masses as in the case of the hardest formations with velocity being added to the reciprocating mass between impacts to raise the magnitude and frequency of percussive blows and naturally the energy output. In this way the tool adjusts itself to the formation being drilled and makes the most effective use of the pressure fluid supply.

Another object is to provide a reciprocating impact prime mover particularly well suited to operation from a relatively incompressible pressure liquid that incorporates among other desirable features an improved valving means that lies within the hammer, is rapidly shiftable therewith precisely at termination of hammer stroke, and has one shiftable valve member that simultaneously opens one set of valve surfaces while closing another set of valve surfaces, and cooperates with the hammer to provide a true "on demand" cycling mechanical hydraulic system.

Another primary object of this invention is to provide a reciprocating percussion motor that is comparatively small in size, is comparatively economical to manufacture, has increased drilling effectivity, and has sufficient durability that it can be used in most all drilling operations, thereby reducing well completion time with the associated reduction of labor, material and equipment costs.

Yet another object of this invention is to provide a special application device that is capable of drilling out well control tools such as cementing apparatus, bridge plugs, and packers that may be made of aluminum, magnesium, cast iron, or other materials.

The foregoing objects, together with other objects, will be more fully apparent from the descriptions that follow.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a transverse, vertical, sectional view of the upper portion of the percussion tool B showing it connected to the drill collar A. The hammer location is shown at percussion and the central valve element shifted relatively downward as it would be momentarily after percussion and valves are positioned for piston upstroke;

FIG. 2 is a continuation of FIG. 1 showing the lower portion of the percussion tool with a conventional rotary drill bit C attached at its lowermost end. The anvil thrust rings are shown flattened as may occur under drill collar load;

FIG. 3 is a reduced diagrammatic longitudinal sectional view illustrating fluid flow to below the hammer from the drill stem and from above the hammer into the central anvil bore while piston is moving upwardly;

FIG. 4 is a reduced diagrammatic view similar to FIG. 3 illustrating fluid flow into the chamber above the hammer and exhaust fluid flow from the chamber below the hammer while the hammer is on the downstroke;

FIG. 5 is a reduced diagrammatic longitudinal sectional view of the lower portion of the percussion tool illustrating anvil extension distance D at percussion and the reactionary period thereafter caused by a combination of percussive force F1 and anvil thrust ring force F2;

FIG. 6 is a horizontal, cross-sectional view taken on line 6—6 of FIG. 1;

FIG. 7 is a horizontal, cross-sectional view taken on line 7—7 of FIG. 1;

FIG. 8 is a horizontal, cross-sectional view taken on line 8—8 of FIG. 2;

FIG. 9 is a fragmentary view similar to a portion of FIG. 2 except illustrating the anvil thrust ring configuration and the anvil spacing from the lower end of the housing when the anvil thrust rings are unloaded;

FIG. 10 is a longitudinal cutaway view of the percussion tool in the area of the valves showing the components as in a portion of FIG. 1 except cutaway sectional view taken is rotated 45° from that in FIG. 1 and shows the vertical passageway through the valve element as well as means of support for valve seat members;

FIG. 11 is a perspective view of one design configuration of an anvil thrust ring;

FIG. 12 is a traverse vertical sectional view similar to a portion of FIG. 1 in the area of the lower end of top sub illustrating an alternate construction that embodies spring bumper rings on the top sub;

FIG. 13 is a graph in which the forces on the anvil and the bit are plotted with respect to time during use of the percussion tool and it shows the interrelationship of these parameters in a typical manner;

FIG. 14 is a view similar to FIG. 9 showing an anvil thrust ring spring of different design;

FIG. 15 is an enlarged perspective cutaway view of the valve 4 showing the passages and ports in it, its four valve faces, and its overall configuration; and

FIG. 16 is an enlarged cutaway perspective view of adapters 34 and 37, valve seat ring 33, and locator ring 52 as they are positioned relative to each in assembly.

DESCRIPTION

With more detailed reference to the drawings, the letter A designates the lowermost end of the drill collars in a string of rotary drill pipe which supplies the torque for turning the bit, weight loadings for the bit and system circulating fluid and on which a percussion drilling tool B is mounted. A rotary drill bit C, which may be of the conventional tooth or button type, threadably engages the lower end of the rotary percussion tool for drilling a borehole.

The percussion tool B generally is made up of the housing **1**, anvil **2**, hammer **3**, valve **4**, and anvil thrust rings **5**, each of which will be described in detail.

The housing **1** is comprised of the top sub **6**, barrel **7**, and driver sub **8** and serves, among other things, as a casing for the hammer, guide for the anvil and a means of attachment to the lower end of the drill string as well as a means of fluid containment and conductance. The top sub **6** threadably engages the lower end of the drill collars A and has a lower face **9** and an axial bore **10** for passage of drilling fluid used to power the percussion tool. The lower end of top sub **6** is screw threaded, as indicated at **12**, to threadably engage the upper end of tubular barrel **7**. The lower end of top sub **6** has packing elements **13** surrounding the bore **10** thereof to form a fluidtight seal with the upper tubular end **14** of valve **4** extending into bore **10**. The lower end face **9** may have a plurality of radial grooves to allow unrestricted pressure fluid flow in chamber formed above hammer. A design variation of lower end face **9** is shown in FIG. **12** and may be a desirable feature in some applications. The configuration incorporates a spring element **59** mounted on lower end face **9** for coaction with upper end face **28**, of piston **25** for reversing hammer **3** at the termination of its upstroke.

The barrel **7** is a cylindrical casing threadably attached to top sub **6** on its upper end and to the driver sub **8** on its lower end, as indicated at **22** and surrounds the hammer **3** and upper end of anvil **2**.

The driver sub **8** is generally tubular in shape and is threadably attached to the lower end of barrel **7** and forms the lower end of housing **1**. The driver sub **8** has a lower face **24** and an internal bore that forms a bushing guide for the anvil **2**. The driver sub **8** also has a plurality of internal grooves formed longitudinally thereof, as indicated at **15**, FIG. **2** and **9**, to receive pins **16** in sliding relation thereto.

The anvil **2** is slidably connected to housing **1** and rotatable therewith and extends through driver sub **8** and into lower end of barrel **7** and on its lowermost end threadably engages drill bit C. the anvil **2** has an upper end face **17** that may have a plurality of radial grooves to facilitate fluid flow between said face and oppositely facing hammer face. An axial bore **18** is formed centrally of anvil **2** for passage of drilling fluid from the lower tubular end **20** of valve **4** to the bit. The upper portion of bore **18** has packing elements **19** surrounding the bore **18** thereof to form a fluidtight seal with lower tubular end **20** of valve **4** extending into bore **18**.

The anvil **2** has a plurality of recesses **21** on its shank complimentary to grooves **15** of driver sub **8** to receive a portion of the pins **16**, the arrangement being such that a splined joint is formed allowing longitudinal movement of the anvil **2** relative to the housing **1** but no appreciable rotational movement between said members. The shank of anvil **2** near the upper end is grooved to receive a split annular retainer or snapring **61** to assemble anvil with the driver sub **8** and provides a means to limit the longitudinal movement between said anvil **2** and driver sub **8**. Packings **26** are interposed in fluidtight sealing arrangement between driver sub **8** and anvil **2** to prevent fluid flow therebetween. Anvil **2** intermediate its ends has a shoulder **23** that together with lower face **24** of driver sub **8** and shank of anvil confines anvil thrust rings **5**.

Anvil thrust rings **5** are extensive mechanical force storage and transfer means in the form of spring elements of various configurations that in usage provide a resilient bias between shoulder **23** of anvil **2** and lower face **24** of driver sub **8**. The anvil thrust rings may be thick, high-force, low-inertia, short-reaction-time, frustoconical-shaped springs as indicated in FIG. **9** or **11** or ring spring shape as indicated in FIG. **14** by **5A**. The anvil thrust rings **5** store and transfer forces that are applied to housing **1** to the anvil **2** and coact with forces also applied to anvil **2** by the percussive impact of the hammer **3** to instantaneously accelerate the anvil.

Hammer **3** consists primarily of piston **25**, adapters **34** and **37**, valve seat ring **33** and locator ring **52**. Piston **25**, adapters **34** and **37**, and valve seat ring **33** also form a valve cage internally in hammer **3** intermediate its ends and is defined by the chamber formed between said parts and enlarged portion of valve **4**.

Piston **25** of hammer **3** is reciprocally mounted in barrel **7** of housing **1** for fluid biased reciprocation therein for imparting percussive blows to the anvil **2**. Outside diameter of piston **25** is in fluid tight sealing arrangement with bore of barrel **7** and has a lower end face **27** oppositely facing to upper end face **17** of anvil **2** and adapted for coaction therewith. Piston **25** also has an upper end face **28** facing opposite to end face **9** of top sub **6** and said faces may be used for coaction. Piston **25** has a stepped internal bore extending centrally from end to end and is used for fluid passages and mountings for the various parts forming the valve cage.

Valve cage is formed on its top side by adapter **34** which has a valve seat **30**, a central bore surrounding upper tubular end **14** of valve **4** and is in sliding relation thereto and in sealing arrangement therewith by seals **35**, and has a concentrically spaced passage formed in part by its outer sides and radially divergent ports through its downwardly depending skirt to allow fluid flow between chamber **36** above hammer and valve cage. Adapter **34** may be screw-threaded in piston **25** or otherwise rigidly mounted thereto and serves to rigidly fix locator ring **52** and adapter **37** in enlarged bore of piston **25**.

Adapter **37** has a valve seat **32**, a central bore surrounding enlarged mating surface of valve **4** and is in sliding arrangement thereto and in fluidtight sealing arrangement therewith by seals **38**, concentrically spaced passages formed in part on its outer surface, an upwardly depending ported skirt for spaced relation with adapter **34** and locator ring **52**, and downwardly extending legs for spaced relation with reduced piston **25** bore diameter. The adapter **37** skirt, ports, passages and legs being such as to allow fluid communication between valve cage, passage **42** between piston bore **44** and outer surface of lower tubular end **20** of valve **4** and chamber **39** below hammer.

Locator ring **52** may be rectangular in cross section and annular in shape and split one place to allow expansion assembly in external groove of valve seat ring **33** and is rigidly fixed in location by upwardly depending ported skirt of adapter **37** and the downwardly depending ported skirt of adapter **34**. Locator ring **52** serves to limit the axial sliding movement of valve seat ring **33** relative to adapters **34** and **37**.

Annular valve seat ring **33** has two seat surfaces **29** and **31** which may be frustoconical in shape and has an outside centermost groove complimentary to but wider than locator ring **52** for limited sliding arrangement therewith. Outside surface of valve seat ring **33** is mounted for axial limited sliding movement in skirts of adapters **34** and **37** and is in fluidtight sealing arrangement therewith by seals **60**, provided in grooves in the outside diameter.

Valve **4** is a single element that conducts pressure fluid to and from the valve cage and controls charging and discharging of chamber **36** above hammer **3** and chamber **39** below hammer **3**, thus creating differential pressures alternately across the hammer thereby causing reciprocation of same. Valve which may be made of two or more joined pieces and screw threaded or otherwise fixed together is mounted in hammer **3** for limited longitudinal reciprocation therewith and is adapted to shift relative to hammer at termination of each up and down hammer stroke to redirect pressure fluid. Valve **4** has an upper tubular end **14** extending through adapter **34** and into bore **10** of top sub **6** and a lower tubular end **20** extending downwardly through adapter **37**, bore **44** of piston **25** and into upper end of bore **18** of anvil **2**. Valve **4** has an enlarged spoollike portion which joins the upper and lower tubular ends and has four valve faces **45**, **46**, **47**, and **48**, FIG. **1** and **15**, formed to generally compliment internal shape of valve cage and are in fixed spaced relation to each other. Valve faces **45**, **46**, **47**, and **48** mate and coact with valve seats **30**, 29, 31, and 32, FIG. 1 and 16, but the spacing of said valve seats and valve faces are such that when valve 4 is shifted downwardly relative to hammer 3 valve face 48 and valve seat 32 are closed and valve face 46 and valve seat 29 are closed. Conversely, when valve 4 is shifted upwardly with respect to hammer 3, valve face 45 is closed with valve seat 30 and valve face 47 is closed with valve seat 31. The valve face and seat arrangement is such that when two pressure fluid passages between a valve face and its respective valve seat are closed, two pressure fluid passages between a valve face and its respective valve seat are open.

Valve 4 has a restrictive fluid passageway 49 communicating with bore of upper tubular end 14 and lower tubular end 20 to provide bypass fluid through the tool. Passageway 49 may be designed to be easily replaceable for size adjustment to accommodate various flows as may be required for well conditions and associated equipment used, but must be restrictive to create a differential fluid pressure between bore of upper tubular end and lower tubular end of valve 4 for the percussion tool to operate.

Upper tubular end 14 of valve 4 above restrictive passageway 49 has a multiplicity of radially divergent passages 50 allowing pressure fluid flow from bore of upper tubular end 14 to valve cage space between valve faces 46 and 47 of valve 4. Lower tubular end 20 of valve 4 below passageway 49 has a multiplicity of radially divergent passages 51 allowing fluid communication with bore of lower tubular end 20 and valve cage space inside adapter 37. Enlarged portion of valve 4 also has a multiplicity of concentrically spaced passages 40, FIG. 10, 15, and 6, longitudinally therethrough that allows fluid communication with lower tubular end 20 and valve cage space formed by a recess inside valve face 45.

Now it is apparent that when valve 4 is shifted downwardly relative to hammer 3 pressure fluid can flow from bore 10 through upper tubular end 14 and passages 50, between valve face 47 and valve seat 31, through ports and passage of adapter 37, through passage 42 and into chamber 39 below hammer to apply pressure force to act on underside of hammer 3 and, simultaneously, exhaust fluid can flow from chamber 36 above hammer 3 through passages formed in part by adapter 34, between valve face 45 and valve seat 30, through passages 40 and lower tubular end 20 allowing fluid pressure dissipation above hammer and acceleration of same upwardly. This is shown diagrammatically in FIG. 3. It is also apparent that when valve 4 is shifted upwardly relative to hammer 3 that pressure fluid can flow from bore 10 through upper tubular end 14 and passages 50, between valve face 46 and valve seat 19, through ports and passages of adapter 34 into chamber 36 above hammer 3 to exert fluid pressure thereon and exhaust fluid can flow from chamber 39 below hammer 3 through passage 42, around and through passages formed in part by adapter 37, between open valve face 48 and valve seat 32, through passages 51 and lower tubular end 20 of valve 4 allowing fluid pressure dissipation below hammer 3 and downward acceleration of same due to differential pressure across it. This is shown diagrammatically in FIG. 4. Thus, it can be seen that two valve faces 46 and 47 control and direct the higher pressure fluid to chambers 36 and 39 while two valve faces 45 and 48 control and direct the lower pressure exhaust fluid from chambers 36 and 39 and that a change in position of valve 4 which carries said valve faces would switch differential fluid pressures across hammer 3.

Valve 4 has upwardly facing surfaces 53 and 54 constantly exposed to supply pressure fluid and downwardly facing surfaces 55 and 56 constantly exposed to exhaust fluid and since supply fluid pressure is always greater than exhaust fluid pressure, valve 4 has a continual downward bias from fluid acting on said surfaces. Valve 4 also has a generally downward facing surface 57 constantly exposed to pressure fluid within passage 42 and chamber 39 below piston and an upwardly facing surface area 62 equal to area of surface 57 constantly exposed to pressure fluid in lower tubular end 20 of valve 4. When supply pressure fluid is valved to chamber 39 below hammer 3, fluid acts on surface 57 and opposes downward bias caused by pressure acting on surfaces 53 and 54, and surface 57 may be sized to provide a net bias on valve 4 in the upward direction when pressure fluid is acting on lower surface 27 of hammer 3 to raise it. Valve 4 then is biased upwardly when hammer 3 is on its upstroke and biased downwardly when hammer is on its downstroke. Pressure fluid on other surfaces of valve 4 may be balanced vertically and radially or unbalanced in the vertical direction to some degree to aid valve shift. This can be done by varying areas exposed to pressure fluid on spoollike flanges of valve 4 and seat ring 33 which is fluid biased in the direction that would tend to unseat closed exhaust valve faces 45 or 48. Although valve 4 is biased in the direction of hammer 3 travel, inertial holds the valve positioned to continue feeding pressure fluid for acceleration of said hammer movement until hammer is rapidly decelerated as when it strikes the anvil 2 or top sub 6 at which time the inertia of valve 4 and the pressure bias in the direction of travel will shift the valve very rapidly, and precisely at termination of hammer stroke. It is, therefore, important to maintain certain relations of weight, bias, and acceleration of valve 4 to hammer 3. Valve operation is "on demand" and is responsive and cooperative to hammer accelerations and velocities.

It should be noted here that by proper sizing of fluid bias areas acting on valve 4 relative to mass of hammer 3 and speed of same, this tool can be made operational without the hammer striking an upper internal end face such as face 9 of top such 6 as in some cases may be desirable. This can be done by providing an upward bias on valve 4 in the hammer 3 upstroke that will cause said valve to accelerate upwardly faster than the hammer acceleration at some point in the upward travel of said hammer such as the hammer 3 leveling off to a constant velocity and a zero rate of acceleration due to limited pressure fluid supply. This arrangement would allow hammer upstroke length variation to be determined by supply fluid pressure, terminal velocity of hammer on its downstroke and relative masses of hammer and anvil as well as hardness of the formation being drilled and other factors. Of course, in any given device with constant fluid supply pressure, the only appreciable variable would be the formation being drilled and the tool would tune itself by stroke length adjustment and associated frequency, energy output, etc., to the hardness of the formation.

Lower tubular end 20 of valve 4 may have one or more ports 58 through its wall constantly communicating with the bore of lower tubular end 20 and passage 42 for fluid passage therethrough. These ports can provide a means for controlling the hammer upstroke velocity by reducing the fluid pressure applied to raise the hammer. They also provide better fluid dissipation below hammer on its downstroke.

FIG. 3 shows diagrammatically the charge pressure fluid flow from the bore of the drill string into chamber 39 below hammer and the simultaneous discharge pressure fluid flow from chamber 36 above hammer into the anvil bore and the resultant hammer upstroke caused by differential pressure across the hammer. FIG. 4 depicts diagrammatically the charge pressure fluid flow from the bore of the drill string into chamber 36 above hammer and the simultaneous discharge of pressure fluid flow from chamber 39 below hammer into the anvil bore and the direction of resultant hammer downstroke caused by differential pressure across the hammer. During the hammer upstroke and downstroke, the anvil thrust rings are also releasing and then storing energy provided by drill collar weight loadings.

FIG. 5 illustrates the forces and occurrences that transpire at percussion. F2 indicates the drill collar load applied to the bit by the anvil thrust rings immediately before and at percussion. F1 indicates the percussive force applied to the bit. The combined forces of F1 and F2 cause anvil extension relative to the housing and is indicated by D.

FIG. 13 further graphically explains the sequence and nature of events before, during, and after percussion in a general and typical manner. On the graph, forces would be zero at the bottom of graph and would increase upwardly while time progresses from left to right. FA indicates average drill collar load and is shown on the left part of the graph as before the percussion tool is operating. Operation is indicated by the variable wave curves and indicates forces that come into play during operation and are plotted versus time. Z indicates the anvil thrust ring energy cycle between percussive blows and consists of a first part energy usage time Y and a second part energy storage time W. During energy usage time Y the anvil thrust rings **5** coact with the percussive blow to extend the anvil **2** relative to the housing **1**, thus expending all or part of their stored energy. During energy storage time W the anvil thrust rings **5** absorb energy because the drill collar load moves downwardly tending to compress them. Drill collar load force applied to the bit as transferred by the anvil thrust rings may range from a minimum as indicated by FO between percussive blows to a maximum greater than average drill collar weight load force FA at percussion and indicated by F2 due to inertia of the drill collars downward movement. F1 indicates the forces applied to the anvil **2** and hence the attached bit by the percussive impact of the hammer on the anvil and, as shown, is synchronized for application at maximum drill collar load application F2. Forces F1 and F2 coact and are added arithmetically to produce a relatively high instantaneous total load force spoke as indicated by FT on the bit for formation crushing or cleavage by forcibly extending the anvil some distance, as indicated by D in FIG. **5**, said force spoke exceeding formation resistance to contact areas of the bit. In the graph X indicates time percussive force is applied to the bit and V indicates total tool cycle time. Bit loadings drop considerably after expenditure of forces F1 and F2 and anvil extension, and then rise to maximum again as the drill collars move downwardly and distort the anvil thrust rings while hammer progresses through its up and down stroke to again deliver a percussive blow.

A preferred configuration of anvil thrust ring is shown in FIG. **11**. This annular single ring spring element is generally rectangular in section and the surfaces are defined as a portion of a cone. It could be considered a modified Belleville dish-shaped ring spring of high volumetric efficiency. This shape spring has a relatively large load capacity, small deflection and small height. Under axial load the unit deflects or distorts in resistance to the load applied and, thus, absorbs and stores the load energy. Distortion stresses are a combination or torsion, shear, tension, and compression and are within the elastic limits of the material used. A single spring element is shown in FIG. **11**, but any number of units may be stacked either in series as shown in FIG. **9** for increased deflection and minimum deflection or with surfaces parallel for increased load and minimum deflection or in parallel series for increased load and increased deflection. Also, it is anticipated that instead of frustoconical upper and lower surfaces as indicated in FIG. **11**, these surfaces may be spherically curved and that this shape tends to add a stiffening effect and may be used to vary the spring rate. Another type of anvil thrust ring is shown in FIG. **14**. These thrust rings would have a frustoconical or spherical coacting interface that when loaded axially cause radial forces that load the inner ring in compression and the outer ring in tension causing distortion of said rings and, therefore, energy storage that can be recovered in the axial direction when the rings return to their reduced stress conditions as when anvil is extended.

Although two preferred types of anvil thrust ring spring configurations are shown in the drawings and indicated in FIGS. **2**, **9** and **11** by **5** and in FIG. **14** by **5A**, it is anticipated that other spring designs can be used to implement the basic idea involved. The above considers configuration or shape, materials, heat treatment, number of units stacked, method of stacking, total loads, rate of load and deflection, etc. There would be many different requirements of the anvil thrust rings in the numerous applications and, therefore, many designs, but all essentially fulfilling the requirements as before stated.

The basic idea involved in the use of the anvil thrust rings and the synchronizing of percussive blow force with drill collar weight has been explained throughout this specification, but the idea is not limited to use with dead weight forces caused by gravity. Most modern smaller drillings rigs as used for seismograph shothole drilling, mining blasthole drilling, core drilling and quarry boring are equipped with automatic hydraulic pulldown units on the rig that can apply force on the bit should dead weight not be used, and the same principles are applicable as described herebefore. Also, the basic idea is not limited to vertical application, but may be used in any direction as long as the drill stem is forced in the direction of drilling.

The percussion tool is normally installed in the rotary drill string above the bit denoted by C and below the drill collars denoted by A. It is driven by and operable from the drilling fluid normally circulated in the system for cleaning formation cuttings from the borehole and for subterranean formation and pressure control. The drilling fluid under pressure is forced down through the inside of the drill string, through the percussion tool for operation of same and exhausts through the bit and flows up the borehole, outside the drill string to the surface.

As installed in the drill string, lower extendable joint is closed or retracted due to formation resistance to the bit and drill collar load applied on the tool housing, said load being transferred through the housing, anvil thrust rings, anvil, and bit to the formation. Hammer is in its lower position resting on the anvil and valve is in its lowermost position relative to said hammer.

Fluid is introduced to the percussion tool through the central bore in the top sub of the housing, flows into the upper tubular end of the valve, through its radially divergent side ports, past the open annular valve seat and valve face **47**, down between outside of lower tubular valve end and hammer bore to the substantially closed chamber below hammer, formed in part by the lower surface of the hammer. Pressure fluid acts on the downward facing exposed hammer surface causing it to lift and accelerate the hammer relatively away from anvil. At the same time, second valve seat and valve face **45** is open and allows exhaust of fluid and pressure dissipation from the substantially closed chamber above hammer, formed in part by upwardly facing surface of hammer. This exhaust fluid travels from said chamber above hammer through concentrically spaced passages in hammer, through openly spaced valve and seat surfaces, through radially divergent side ports below pressure restriction orifice into lower tubular end of valve, into bore of anvil and on through the system.

Hammer and valve move upwardly away from anvil in unison until hammer strikes upper inside surface of housing at which time hammer rebounds downwardly and valve, due to its inertia, continues upwardly until it reseats in its alternate position relative to hammer. Passages formed by valve faces and seats that were open are now closed and alternate passages also formed between annular valve faces **46** and **48** and their respective coacting seats are open. This opening of one set of valve surfaces and the closing of the alternate set of valve surfaces redirect the supply and exhaust fluid to and from substantially closed chambers on opposite ends of hammer. This is to say that pressure fluid is now redirected to top chamber to act on top surface of hammer and drive it downwardly relatively toward anvil while exhausting chamber below hammer, allowing pressure and fluid dissipation therein. The same chambers and communicating passages are used for fluid redirection but different valve face and seat surfaces allow passage of fluid or stoppage of fluid flow.

Hammer and valve again move in unison downwardly. At the termination of hammer downstroke, hammer strikes anvil a percussive blow, said blow coacting with stored force in anvil thrust rings to instantaneously accelerate anvil, creating a high-impulse force spike that is transferred through the bit to the formation for crushing same. At percussion and sudden deceleration or rebound of hammer, valve, due to its inertia, shifts to original lowermost downward position relative to hammer and hammer is again accelerated upwardly as before as fluid again flows as originally directed below hammer.

An operational cycle is now complete and the tool continues sustained high-frequency operation responding to pressure fluid supplied, formation hardness, etc., while being rotated conventionally.

Valve shifts from one position to its alternate position are unusually fast because of the inertia tending to maintain valve speed at terminal hammer velocity and also because of conserved energy in the hammer, causing it to rebound in the direction opposite of valve inertial travel. Hammer and valve each move a portion of the short-shift travel distance. This quick shift tends to minimize valve erosion. It is also very important because total hammer stroke is short and, therefore, relatively less total hammer stroke is used for valve shift. This means that a greater percentage of total stroke is used for hammer acceleration and corresponding increase in cycle frequency.

Also, the fact that the hammer is accelerated to collision before any valve shift occurs is of major importance. It insured maximum terminal velocity of hammer at percussion, since hammer inertia in heavy fluid, as drilling fluid often is, can be reduced considerably in very short distances. This not only adds to cycle frequency, but insured maximum production and transfer of percussive energy. Close-dimensional control of hammer, hammer stroke, housing length, and anvil end location is also eliminated since end of hammer stroke determines valve shift and hammer reversal.

This tool has the most positive displacement of any tool known at this time, discounting bypass fluid. It has a predictable cubic inch hammer displacement for each cubic inch of supply fluid and, therefore, can be said to produce the most efficient use of horsepower supplied to it. Because of this high efficiency, it will operate on a comparatively small pressure drop across it (supply pressure versus exhaust pressure). This is advantageous for several reasons. One is that more pressure drop can be utilized at the bit for hole bore cleaning. Another is that less fluid erosion of tool parts will occur and at this time said erosion is the factor that most limits the life of this type tool. It also means that comparatively more fluid can pass through the tool chambers and less will be directly bypassed with direct loss of hydraulic horsepower.

This tool is so designed that it will not percuss against its anvil when the lower extendable joint is fully extended. This permits full-pressure borehole flushing but no percussion tool operation. When drill string weight is lifted allowing bit and anvil to extend, hammer and valve are also lowered relative to housing. Before full joint extension top tubular end of valve is withdrawn from engagement with small central upper bore of housing allowing fluid to flow therebetween into top chamber through hammer and valve and on through the tool. As long as valve is out of engagement with upper bore of housing, piston will not lift, yet allows full fluid passage.

Surface monitoring of this tool and other desirable downhole information is possible by its self-produced sound waves traveling up through the drill string. These sound waves are produced when hammer at termination of its upstroke strikes the housing as will occur on each operational cycle. By comparing sound wave characteristics produced with predetermined frequency and other known data, unusual tool operation, formation change such as hardness and energy absorption rate, and other general data can be determined.

Although specific embodiments of the invention are illustrated in the drawings and described herein, it will be understood that the invention is not limited to the embodiments disclosed but is capable of rearrangement, modification and substitution of parts and elements without departing from the spirit of the invention.

What is claimed is:

1. In a percussion drilling tool:

a hammer mounted for movement between a first position and a second position;

valving means responsive to movement of the hammer to the first position for movement under its own inertia to a first valving position and responsive to movement of the hammer to the second position for movement under its own inertia to a second valving position, and

means responsive to movement of the valving means to the first valving position for moving the hammer to the second position and responsive to movement of the valving means to the second valving position for moving the hammer to the first position.

2. The tool according to claim 1 wherein:

the valving means is mounted on the hammer for movement with respect thereto between the first valving position and the second valving position, and

the hammer-moving means includes means for directing a fluid under pressure to the hammer and cooperating means on the hammer and the valving means for causing the fluid under pressure to urge the hammer toward the second position when the valving means is in the first valving position and for causing the fluid under pressure to move the hammer toward the first position when the valving means is in the second valving position.

3. The tool according to claim 1 wherein the direction of movement of the hammer is reversed each time the hammer arrives at one of its positions and wherein the valving means moves under its own inertia from one valving position to the other each time the direction of movement of the hammer is reversed.

4. The tool according to claim 1 wherein:

the hammer is mounted in a drill string for movement between a stop and an anvil and includes a pair of pressure-receiving surfaces, one facing the stop and the other facing the anvil;

the valving means is mounted in the hammer for movement with respect thereto under its own inertia from the first valving position to the second valving position whenever the hammer engages the stop and for movement with respect thereto under its own inertia from the second valving position to the first valving position whenever the hammer engages the anvil, and

the hammer-moving means includes means for directing a fluid under the pressure through the drill string to the hammer and cooperating means on the valving means and hammer for directing the fluid under pressure into engagement with the one pressure-receiving surface of the hammer whenever the valving means is in the first valving position and for directing the fluid under pressure into engagement with the other pressure-receiving surface of the hammer whenever the valving means is in the second valving position.

5. The tool according to claim 1 wherein the hammer-moving means also alternately biases the valving means toward the first valving position and toward the second valving position.

6. The tool according to claim 5 wherein the hammer-moving means imposes a bias force on the valving means that tends to move the valving means to the second valving position whenever the valving means is in the first valving position and imposes a bias force on the valving means that tends to move the valving means to the fist valving position whenever the valving means is in the second valving position.

7. A percussion drilling tool comprising;

a drill string;

a drilling bit mounted at the distal end of the drill string;

means for reciprocating the drilling bit relative to the drill string, and

at least one spring mounted between the drill string and the bit for applying the force of the drill string to the bit as the bit is reciprocated and thereby increasing the force that is applied to the bit during each reciprocation, said spring comprising a frustoconical-shaped spring that is distorted into a right circular cylinder configuration by the force of the drill string and that returns to the frustoconical configuration as the bit is moved away from the drill string.

8. The percussion drilling tool according to claim 7 wherein the spring is a thrust ring which is normally distorted by the force of the drill string and which returns to its undistorted condition as the bit is moved away from the drill string by the reciprocating means.

9. The percussion drilling tool according to claim 7 wherein the thrust ring is positioned between the pair of facing surfaces, one connected to the drill string and the other connected to the drilling bit.

10. A percussion drilling tool comprising:

an elongate drill string;

an anvil mounted at the distal end of the drill string for longitudinal movement with respect thereto;

an anvil-actuating hammer mounted within the drill string for movement between the anvil and a portion of the drill string;

a valving member mounted on the hammer for movement with respect thereto under its own inertia to a first valving position whenever the hammer engages the anvil and for movement under its own inertia to a second valving position whenever the hammer engages the portion of the drill string;

means for moving the hammer toward the portion of the drill string whenever the valving member is in the first valving position and for moving the hammer toward the anvil whenever the valving member is in the second valving position, and

at least one compressive spring positioned between the drill string and the anvil for continuously urging the anvil away from the drill string.

11. The percussion drilling tool according to claim 10 wherein the drill string includes a surface that faces the anvil, wherein the anvil includes a surface that faces the drill string, and wherein the spring comprises a thrust ring positioned between the surfaces on the drill string and on the anvil.

12. The percussion drilling tool according to claim 10 wherein the hammer-moving means includes means for alternately directing a pressurized fluid into a chamber formed between the drill string and the hammer and into a chamber formed between the anvil and the hammer.