# United States Patent [19]

**Martini**

[11] **3,768,576**

[45] **Oct. 30, 1973**


[54] **PERCUSSION DRILLING SYSTEM**


[76] Inventor: **Leo A. Martini,** 5920 Sandhurst, Apt. 131, Dallas, Tex.



[22] Filed: **Oct. 7, 1971**

[21] Appl. No.: **187,299**

[52] **U.S. Cl.** ..... **173/73,** 173/136, 175/56, 175/92

[51] **Int. Cl.** ..... **E21b 1/06,** E21b 5/00

[58] **Field of Search** ..... 175/56, 92; 173/73, 173/80, 133



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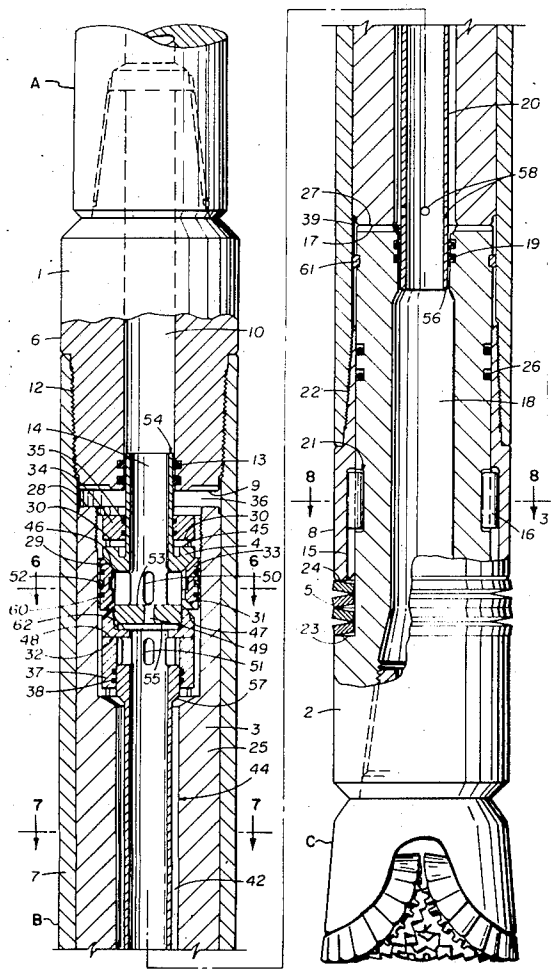
*Primary Examiner*—James A. Leppink
*Attorney*—D. Carl Richards et al.



[57] **ABSTRACT**

A unique, variably damped, steady state, broad range, fluid forced, force superposition, oscillatory, percussion drilling method and means, capable of resonant frequency operation and force magnification, is disclosed that is operable from liquid or gaseous fluid under pressure for rotary drilling of oil, gas, and water wells, geophysical holes, open strip mining blast holes, construction holes, and the like for greatly increasing the rate at which said bore holes are drilled. This comparatively simple prime mover produces sustained high frequency, high amplitude, longitudinal force spikes on a drill bit and synchronizes consumption of percussive force energy and drill collar weight force 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. In addition to force superposition an inphase energy consumption, this device when operating at or about resonant frequency will produce a phenomenon of force magnification, force reinforcement, and maximum transmissibility.


**45 Claims, 24 Drawing Figures**



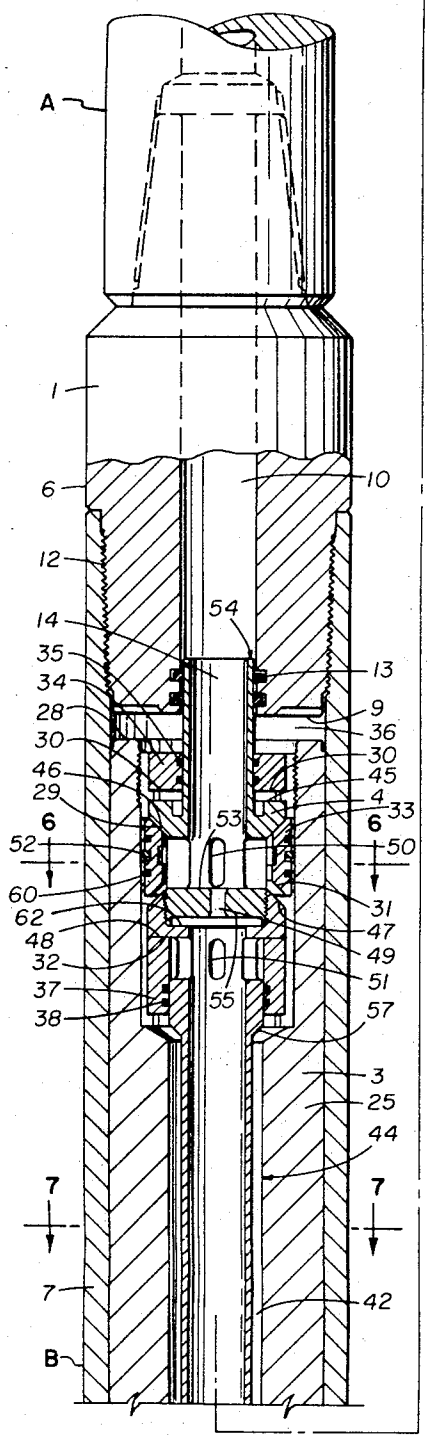



FIG. 1

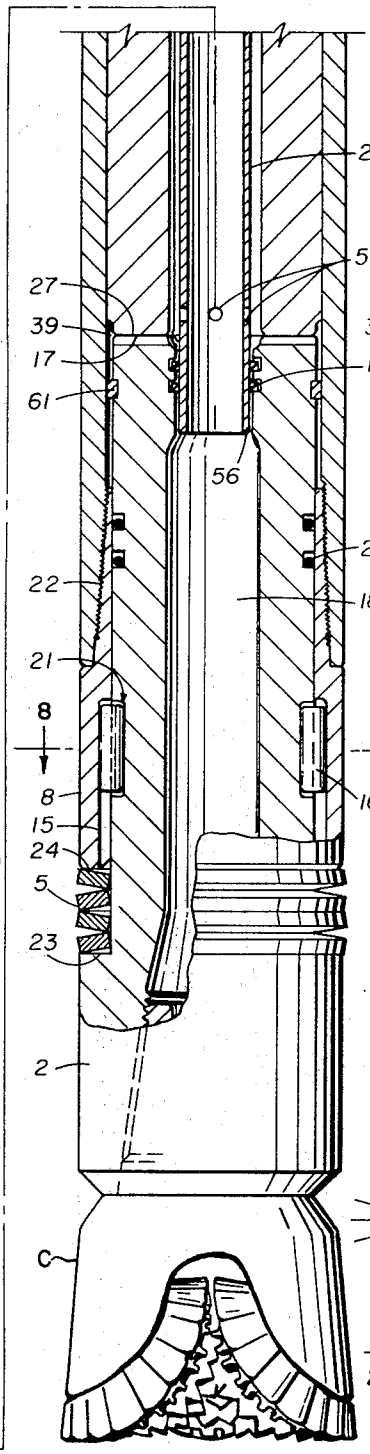



FIG. 2

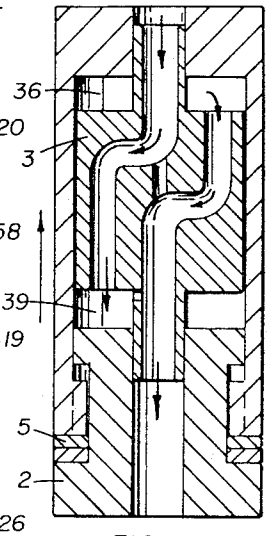



FIG. 3

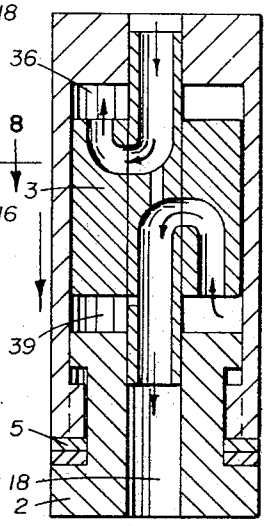



FIG. 4

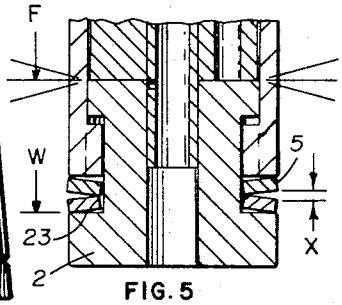



FIG. 5

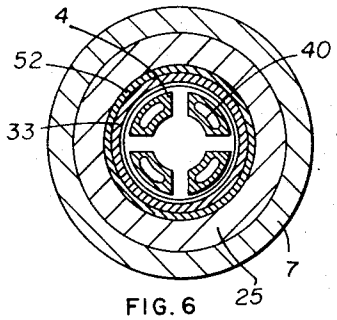



FIG. 6

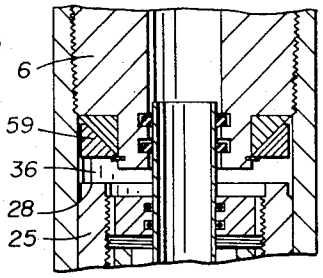



FIG. 12

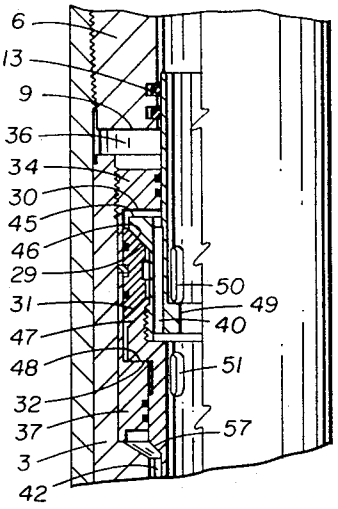



FIG. 10

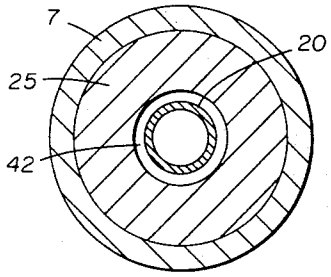



FIG. 7

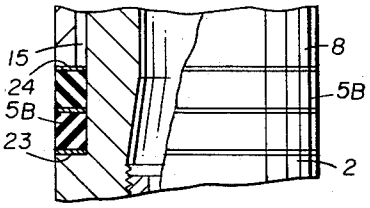



FIG. 9

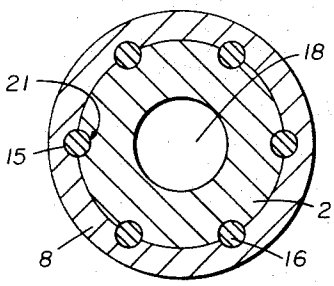



FIG. 8

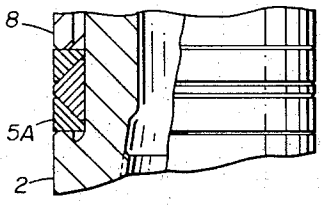



FIG. 14




FIG. 11

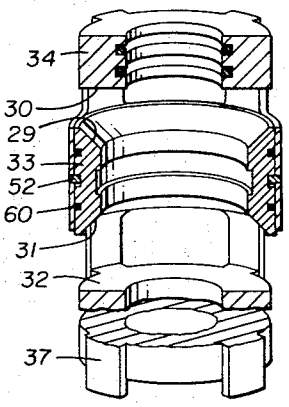



FIG. 16

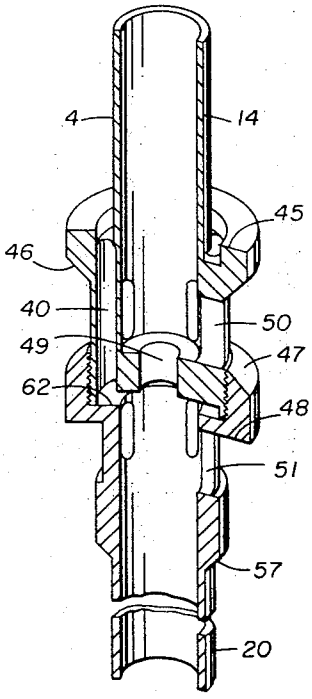



FIG. 15

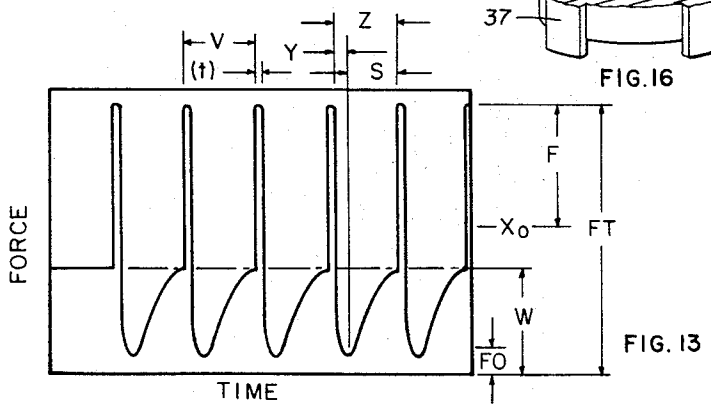



FIG. 13

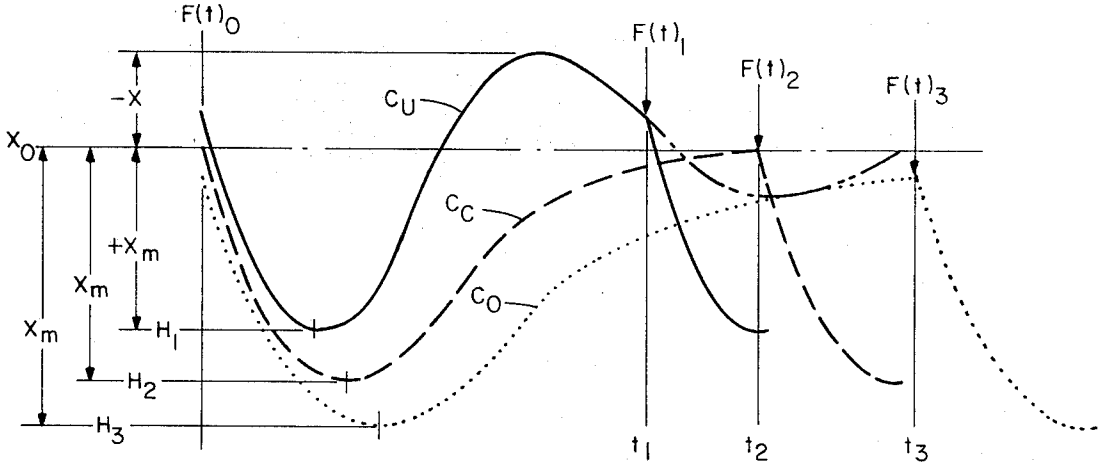



FIG. 18

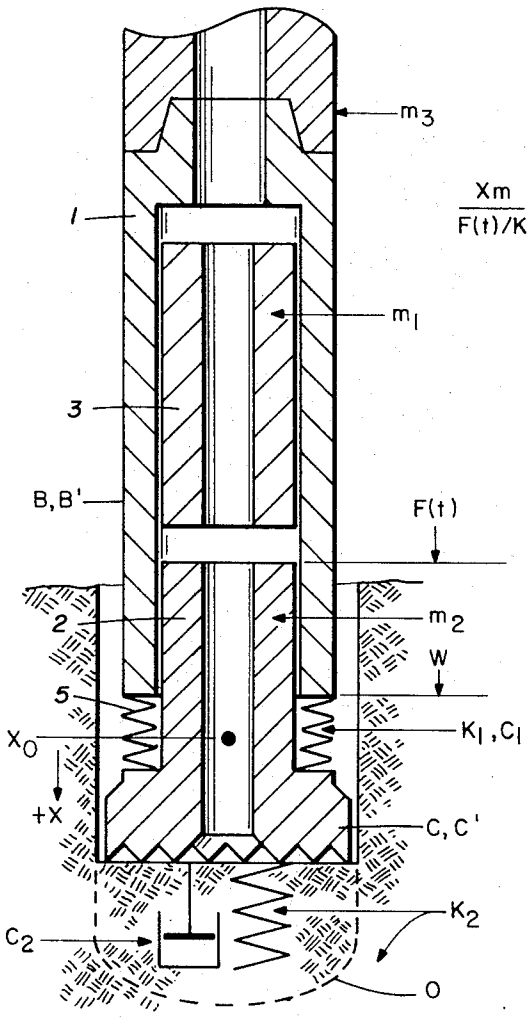



FIG. 17

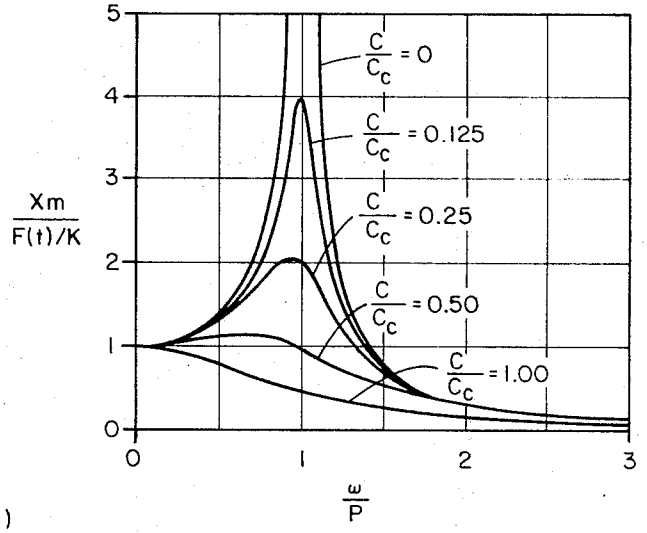



FIG. 19

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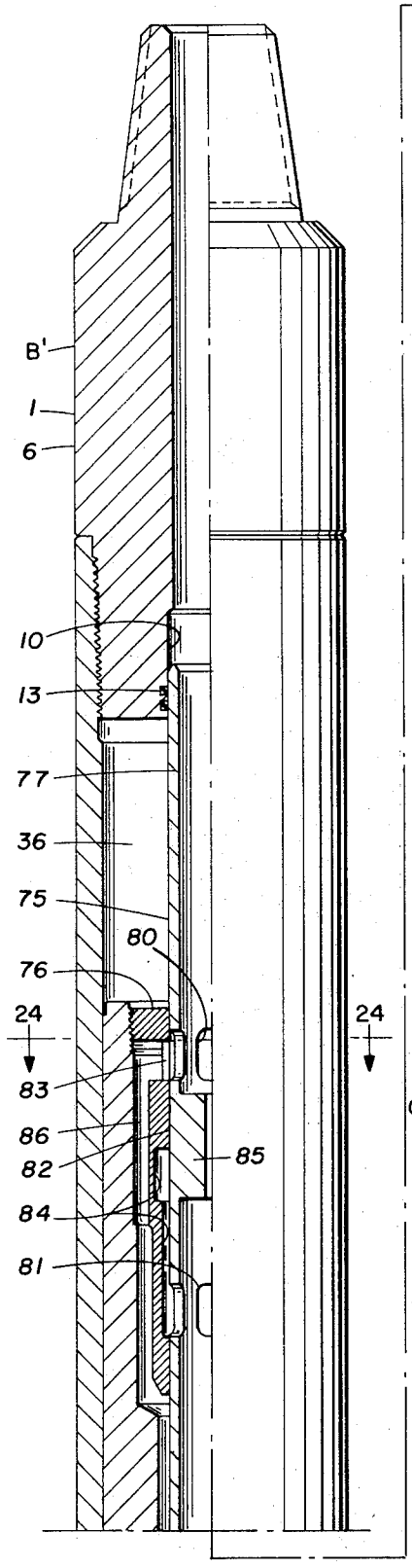



FIG. 21

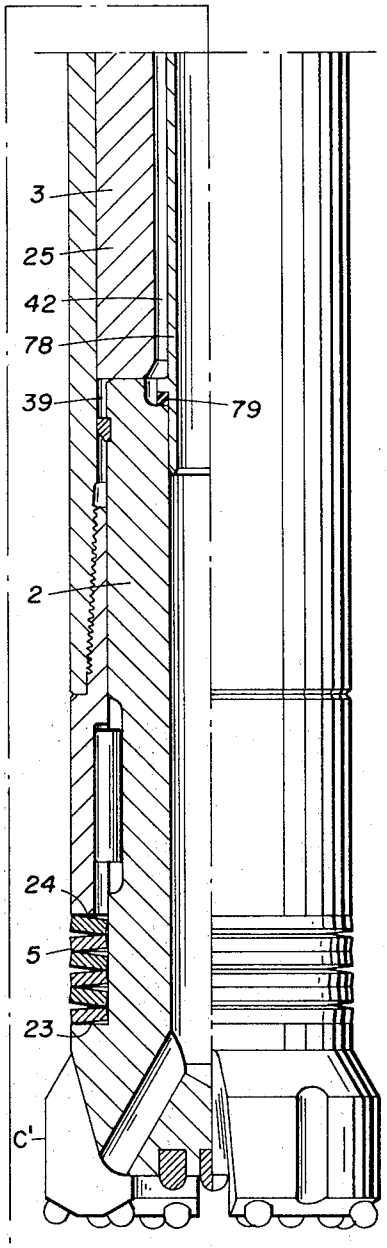



FIG. 22

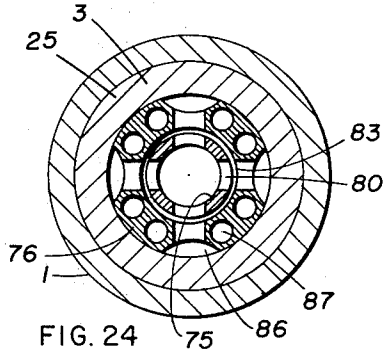



FIG. 24

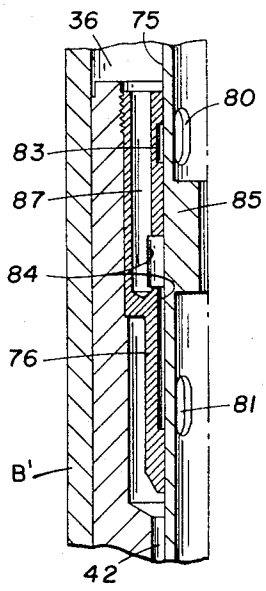



FIG. 23

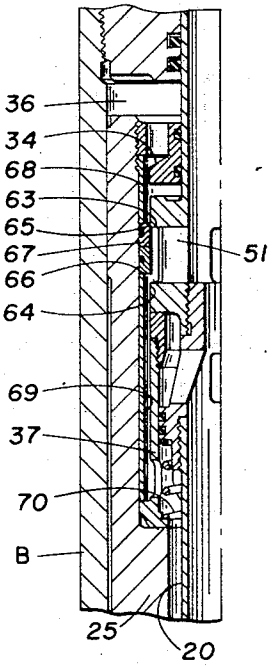



FIG. 20

INVENTOR:
LEO A. MARTINI

# PERCUSSION DRILLING SYSTEM

This application is a continuation-in-part of application Ser. No. 18,635, filed Mar. 11, 1970, now U.S. Pat. No. 3,612,191, granted Oct. 12, 1971.

The invention encompasses first a forced, damped-spring-mass oscillatory vibrational system formed in part by the percussion tool and formed in part by the formation being drilled and these parts cooperate to determine in part the operational characteristics of the drilling system. Secondly, the invention involves a system exciting fluid driven percussive motor which may use either of two automatic valving structures, one for liquid and the other for gaseous pressurized circulating mediums thereby providing for multiple industry application.

## 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 inpenetrable earth strata, it is imperative that improvements over existing devices be made since generally today they are still lacking in effectivity.

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 proven 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, compatability with existing equipment and systems, suitable tool operation, 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 wells. 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, well casing installation, 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 or 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 non-compressible 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. 1,892,517—Pennington—Dec. 27, 1952

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

U. S. Pat. No. 3,387,671—Collier—June 11, 1968

## SUMMARY OF THE INVENTION

This invention relates to a dual operational mode, fluid driven, forced frequency, impulsive force, steady state, variably damped oscillator capable of percussive force and superposition force production in the stiffness controlled operational mode and capable of producing percussive force, superposition force, and force magnification in the mass controlled resonant frequency mode.

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 time percussive impact force is applied, thereby adding or superimposing said forces, one on the other for more effective formation cleavage.

Another primary object of this invention is to provide a prime mover that will maintain drill collar weight force on a drill bit, superposition percussive blow force on said weight force and reinforce said forces with other generated inertial mass forces produced at resonant frequency system operation.

An object of this invention is to provide a stiffness controlled, forced, oscillator with critical or overdamping characteristics whereby superposition percussive blow forces are exerted in phase with other applied forces during nonresonant frequency operation.

Another important object of this invention is to provide a mass controlled resonant frequency oscillator that during operation will store or expend energy relative to the damping effect of the system output and will produce force magnification relative to the output work.

Still another important object of this invention is to provide an oscillatory percussion tool that will operate effectively in either mode of resonant and non-resonant frequencies and be capable of operational mode change.

Another object is to provide a device of this type that will generate more mechanical power output for a given power input by conserving system energy and a

device that will automatically adjust its operation according to the formation being drilled.

Another object is to provide a reciprocating impact prime mover particularly well suited to operation from a relatively incompressible pressure fluid 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 percussion drill 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.

An object of this invention is to make more effective use of drill collar weight when drilling with a percussion motor.

An object of this invention is to provide a drilling system that will cooperate with the formation being drilled to determine in part the operating characteristics of the system.

Yet another object of this invention is to provide two separate fluid valving constructions for the percussion motor which excites the vibrational system that will allow more optimized performance and economy for each of the fluid states, gaseous and liquid.

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.

In the drilling industry standard 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, this is generally assumed to be the best application of drilling technique. It, however, has proven to be less effective than anticipated with the 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 while 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. On conventional tools of this type it was found that at percussion of the hammer with the anvil-bit, the anvil-bit is impulsively driven forward relative to the weight force transferring surface of the drill string and out of engagement with same, and that during said disengagement the drill string weight force is released from the anvil-bit. In other words, drill collar weight will come to rest on the anvil before the 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. Further examination of tools in use today determined that the drill collar weight force energy application was approximately 180° out of phase with the percussive blow. That is, the percussive blow energy is applied and used from 0° to 180° with the maximum anvil displacement occurring at 180°. The following but slower drill collar load is then applied and comes to rest on the anvil-bit applying and using its energy after 180° and before 360° which indicates that it is completely out of phase to produce additive forces or be of much value in drilling. In fact this condition is considered highly undesirable. 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, because force applications are out of phase and tend to cancel each other. 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 zero and so uncoordinated, unsynchronized forces come into play producing a vibrating action of lesser effectivity.

To synchronize and substantially superposition percussive blow force and drill collar weight energy applications, extensive mechanical force storage and energy phase 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 ring springs store energy as they are compressed and release energy as they expand and thereby change the phase of drill collar energy usage from approximately 270° to 0° to 180° which is inphase with the percussive blow energy usage. This delayed force energy 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 and will achieve increased bit effectivity.

The anvil thrust rings may be thick, high force, low inertia, short reaction time frustro-conical or other type ring or coil 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 the 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 cancelled at percussion. Specifically in the case of a critically damped system, potential energy is stored in the anvil thrust rings after the maximum anvil-bit displacement which is 180° through the rest of the tool cycle back to 0° at which time another percussive blow occurs. At the percussion blow when threshold rock crushing forces are achieved by a combination of drill

collar and percussive load these energies are used and consumed together from 0° to 180°. Energy usage is then in phase and forces are additive. Drill collar weight force energy applied between percussions is stored for use at percussions, thus bringing these forces inphase to increase the force amplitude. Anvil acceleration forces at percussion can be greatly increased 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 tend to stabilize the torque required to rotate the drill stem and hence reduce dynamic torsional stress in the drill string.

The simple foregoing explanations are pertinent and relevant and form an important part of this specification but the invention encompasses more than the incorporation of the anvil thrust ring springs, and new valving structures. It is a new method and means for down the hole drilling system operation.

Due to the discovery that drill collar weight as being used was for the most part ineffective for increasing the rate of penetration when used with a percussive motor, an investigation of the drilling system as well as the formation being drilled was undertaken in an effort to improve system effectivity. Out of these studies came significant data which will surely advance drilling technology. Although much is yet to be learned about the visco-elastic properties of rock, formation elasticity under various hydrostatic pressures, rock fracture time versus elasticity, and overburden stress effects, certain data is known and form in part the basis of this invention. The research indicates that most all formations show greater compressive strengths with increased confining pressure. Under increased confining pressures formations exhibit large amounts of deformation and enhanced ductility prior to failure. Rock with brittle viscous behavior at atmospheric pressures will show remarkable elasticity under the forces of hydrostatic well pressures and heavy earth overburden and these physical characteristics of rock under stress will have a prounounced influence on the energy requirements for fracture by action of bit teeth and the resulting volume of dislodged formation. It was noted that the springlike condition of the formation was capable of storing a portion of the energy from a percussive blow and returning said energy to the anvil-bit and that this energy could be utilized in conjunction with the existing anvil thrust rings to increase the oscillation amplitude of the drilling bit and ultimately enhance system performance. This apparent natural obstacle of formation elasticity to drilling became an asset, was incorporated, and used to good advantage for conservation of system energy and by providing a unique tool that is responsive to formation conditions. It resulted in a drilling tool that will perform as well in deep bore holes as on the surface and in plastic, rubbery, tough, hard, elastic brittle formations, and one that can automatically adjust its force level accordingly.

To more fully explain the salient features embodied in this invention a technical and theoretical explanation of system operation is required. In this way a clear presentation of concepts and principles involved can be made and differences from existing systems can be viewed with respect to novelty and invention.

In a broader view this invention encompasses, first, the formation to be drilled, a mass that can be considered infinite, of varying hardness and elasticity, having certain energy absorbtion and spring rate characteristics and having certain damping qualities. Secondly, the invention involves a method and means of expediently penetrating the formation, having a part of comparatively large mass relative to the other parts and consists of the drill string. On the distal end of this drill string is an anvil-bit with acertain weight or mass and suitably constructed to mate with the drill string on its upper side and to engage and dislodge the formation on its lower side. Interposed between the anvil-bit and the drill string are the anvil thrust ring springs, having a suitable spring rate and certain small inherent damping qualities. The anvil-bit is mounted for longitudinal oscillation relative to the drill string and the formation as a whole. A reciprocal hammer mass motor located inside the drill string casing is provided to cause the anvil-bit oscillation in its one degree of freedom and is responsive to the pressurized fluid stream in said drill string. The system is shown schematically in FIG. **17.** This drilling system then embodies a vibrational mass under the influence of restoring forces that is adapted to be excited by a series of periodic impulses defining a forced oscillator and when certain conditions exist is a forced resonant frequency oscillator. These certain conditions are a particular relationship between the combined system restoring forces and the oscillatory mass and also having another particular relation with the periodic impulses. The forced stiffness controlled oscillator and the forced mass controlled resonant oscillator produce system output forces that are superpositioned on any preload such as drill collar weight that the system may have and the energy the oscillator produces is consumed simultaneously with the preload energy.

The formation effect on the drilling system described herebefore is descriptive of the inherent natural internal behavior of rock under various stress effects and although the elastic properties and potential energy storage therein contribute to the elastic restoring forces of the vibrational system, they make lesser overall contributions to the system than the greater governing physical laws of mechanics under which the system functions. These physical laws produce a system condition that can be called spring-like or as having the nature of an elastic member wherein a force is present, a displacement is present and a spring rate is produced in that there is a force change per unit displacement as well as potential energy storage relative to the resisting force. This is one aspect that in cooperation with the other parts allows this oscillatory and harmonic vibrational system to perform with elastic restoring forces on the anvil-bit and thus a true spring mass system even through the formation disintegration beneath the bit may be totally of a viscous damping nature.

The earth as a whole and the drill string that comes to bear on said earth have a mutual gravitational attrac-

tion relative to the mass of each. This said attraction causes equal and opposite forces at the lower end of the drill string on the bit face and the oppositely facing formation face in contact with the bit. The formation then is in effect pushing back against the bit as much as the bit is pushing against the formation. As the anvil-bit is driven forward by the percussive blow, damping occurs as the formation disintegrates before it but as the energy is consumed, increasing resistance builds up to the bit which is then momentarily halted to forward movement. But since the average rate of forward travel of the bit is the same as the travel of the drill string, when the anvil-bit stops its forward movement, it effectively makes a reverse stroke in that it moves closer to the drill string and thus a restoring force is apparent on the anvil-bit. It can be called an elastic restoring force because it is generated as formation disintegration occurs during the bit displacement and therefore has a rate of force buildup. This force is a resistance force and is independent from the energy consumed by damping and is actually the force that keeps the drill string from falling through the formation, yet in the operating system it can be called an elastic restoring force of the formation. Yet another restoring force is the mechanical action of the bit that tends to move the anvil-bit toward the drill string as it rotates and moves the bit cutting structure onto new formation. The elastic restoring force of the formation then may be potential energy returned from the formation, formation resistance to the bit load force, and forces due to the mechanical action of the rotating bit.

Due to gravitational effect or other forces, the drill string rests on the anvil thrust ring springs which in turn biases the anvil-bit which in turn exerts forces on the formation. These forces on the formation cause the portion immediately adjacent the bit to exhibit viscoelastic characteristics somewhat like a damped spring, according to its elasticity, its energy absorbtion rate and other factors. In this static situation with the anvil-bit elastically suspended., the position at which the anvil-bit mass center rests undisturbed will be considered its equilibrium position, $X_o$. $X_m$ will denote the amplitude or maximum displacement of the mass center from equilibrium. Although various other movements will take place during system operation, such as drill string rotation and its downward progression, only the theoretical aspects of the dynamic reciprocal steady-state motions will be considered. The direction of drill string progression which would normally be downward will be considered the positive direction and the opposite direction will be the negative direction. Damping as used throughout this specification is energy dissipation that would tend to diminish the amplitude of oscillations in the vibrational system, and although some energy reduction will be from ordinary operating machine losses most damping will occur as energy usage in formation disintegration.

The hammer mass $m_1$ provides the periodic impulsive exciting inertial forcing function $F(t)$ for anvil-bit mass excitation and is a part of the reciprocal fluid motor. $m_2$ is the anvil-bit mass, has a fixed value (weight) and is a part of the spring mass oscillatory sytem that reciprocates longitudinally under the influence of the elastic restoring forces K, the damping coefficient C, the percussive blow $F(t)$ of the hammer, and the force W applied by the drill string. $m_3$ is the mass of the drill string or other equivalent force means that provide the force W exerted on the spring mass system. These masses and forces are shown in FIG. 17, and determine in a large part the vibrational system performance. The anvil-bit mass, $m_2$, will behave according to the fundamental formula, $F=ma$ and modified for the specific conditions, the formula describing the motion of the center of mass, $m_2$, is $F(t)+W = m_2\ddot{X} + C_1\dot{X} + C_2\dot{X} + K_1X + K_2X$ where: F is the percussive blow force, $(t)$ is the transfer time of the blow force, and $F(t)$ may be called the system forcing function. W is the drill collar weight force applied to the anvil-bit. $\ddot{X}$ is acceleration in the X direction and $\dot{X}$ is a velocity in the X direction. $C_1$ is the damping coefficient of the anvil thrust rings, and $C_2$ is the damping coefficient of the formation adjacent the bit and combined make up the total damping coefficient, C. $K_1$ is the spring constant of the anvil thrust rings and $K_2$ is the spring constant of the formation adjacent the bit and combined make up the total spring constant, K. X is the displacement of the center of mass from the equilibrium position, $X_o$. FIG. 17 shows these factors affecting the anvil-bit mass. The use of the above equation allows predictable behavior of the anvil-bit mass and is the basis underlying this portion of system operation. Two desirable and distinct types of steady state vibrational system operation or modes can be achieved with the percussion tool. The mode is dependent on the exciting, elastic and damping characteristics of the system and will be described with specific limits but without assigning actual values to the controlling factors. An important aspect of each operational mode is the phase angle relationship of the power inputs to the receiprocating anvil-bit mass $m_2$. The driving or exciting motor mass $m_1$ percussive blow $F(t)$ is approximately 180° from the driven anvil bit mass $m_2$ in the stiffness controlled system operational mode while in the mass controlled system operational mode, $m_1$ leads $m_2$ by 90° with an allowable variation of plus or minus 90°. In addition to these primary power input phase angles with the percussive motor, the secondary power input phase angle is also of major significance since it may add or detract from the primary power input to the anvil-bit mass. The secondary power input resulting from the drill collar weight force W coincides with the downward anvil-bit travel as the anvil thrust rings expand and release their energy. This is to say that the energy from $F(t)$ and W added to the reciprocating $m_2$ is transferred and consumed inphase and thus these two power inputs compliment each other. This power consumption occurs from 0° to 180° in the stiffness controlled system and from −90° to +90° in the mass controlled system. Both power sources then add energy to the anvil-bit mass $m_2$ from top dead center to bottom dead center in the positive downward travel of the anvil-bit.

The first operational mode is the non-resonant frequency mode, $M_1$, and would normally be used in drilling the softer or more brittle formations where maximum energy for formation fracture would not be required and where a greater part of all energy would be directly used in formation disintegration. This operational mode is a critically or overdamped, stiffness controlled, forced, vibrational system in which the directly applied forces of drill collar weight, W, and percussive blow, $F(t)$, are of primary importance while the mass of the anvil-bit is a comparatively insignificant factor.

In operation the anvil-bit mass, $m_2$, is repeatedly displaced downardly, a distance, $X_m$, relating to the ap-

plied impulsive force, $F(t)$ and force, W, and returns to the equilibrium position, $X_o$, in a controlled manner due to a particular type of damping in the elastic restoring forces acting on it and specifically may be critically damped or over damped. This is to say that the combined restoring forces, comprised of the formation and the anvil thrust ring spring, returns the anvil-bit to its equilibrium position without causing it to "overshoot" or pass the equilibrium position between the percussive impulses. In the case of critical damping the return is done in the shortest possible time with the mass involved but in the overdamped condition the return time may be longer. The limit on this non-resonant operational mode damping factor $C/C_c$ is equal to or greater than one (1), where the damping factor $C/C_c$ is the ratio of actual combined damping to system critical damping, $C_c$, which is $\sqrt{4Km_2}$. This simply stated is that the combined damping is equal to $\sqrt{4Km_2}$ or greater. FIG. 18 graphically pictures the motions described by the center of mass in the criticaly damped condition by the curve designated $C_c$ and in the overdamped condition by the curve designated $C_o$. The motion described by the center of mass of the anvil-bit theoretically does not pass across the equilibrium position but only returns to it. The movements of the anvil-bit under actual conditions of operation may be very complex and the descriptions herein are not meant to emcompass the various minor movements caused by the rolling, climbing action of the bit or other factors but to generally describe the predominant characteristics of the damping factors and the elastic restoring forces as effects the anvil-bit.

Considering that a cycle starts at percussion which would be 0° and ends again at 0° or 360°, the next successive percussive blow, the maximum displacement, $X_m$, would be at 180° and from this point through the remainder of the cycle, the center of mass of $m_2$ returns to or approaches $X_o$ and does so exponentically or asymptotically with time depending on the particular degree of damping; but as before stated is $\sqrt{4Km_2}$ or greater. Then depending on the spacing of periodic impulses of the forcing function $F(t)$, the center of mass of $m_2$ is at or near $X_o$ at cycle start and end. Then it can be stated that weight or force energy, W, that has been stored over the time period, S, (FIG. 13) and used in doing work during time period, Y, is in phase with the primary power impulsive force, F, and will compliment forcing function energy usage. The fact that the center of mass of $m_2$ may not have reached $X_o$ between the impulse forces in the overdamped condition is of little consequence since $m_2$ has reached the major part of its negative travel to $X_o$ and in actuality $X_o$ will adjust downwardly to the center of mass of $m_2$. The important point is that the energy usage from the forcing function, $F(t)$, and the force energy, W, coincide and are used in phase. This is a considerable departure from the standard industry conditions where the percussive blow occurs, negates the drill collar weight, and then the percussive blow work is done without the assistance of the drill collar weight energy. Then somewhere about 180° out of phase with the percussive blow, the drill collar weight energy work is done and used without the assistance of the percussive blow tending to provide an uncoordinated system with little or no advantage over the conventional drilling system except in special cases where suitable drill collar weight or force cannot be applied to the system.

When the anvil-bit is driven downwardly on each cycle from its equilibrium position by the percussive blow, $F(t)$, the displacement, $X_m$, and force are positive and will be exerted on the formation in addition to the drill collar load forces, W, also exerted positively downardly on the formation. These forces F and W have like signs and are additive indicating superposition force. The following formula would express the mode forces exerted on the formation in operation of the system. $F_{M1} = F + W$.

The second operational mode, $M_2$, is resonant frequency system operation. This harmonic, underdamped vibrational system is said to be mass controlled and involves exciting the anvil-bit mass to a frequency substantially equal to its natural or fundamental frequency and thereby not only producing a percussive blow force superpositioned on drill collar weight force but also an additional force magnification, and maximum force transmissibility phenomena. Generation of this phenomena can be a worthwhile contribution to the effectivity of the system and would normally be applied in drilling the harder formations where the extra force output is needed and the greater formation elasticity is encountered. In this case the system operation can better be explained through successive tool cycles because the reinforcement phenomena occurs over a continuous series of cycles.

The periodic impulsive hammer percussive blow excites or displaces the anvil-bit mass downwardly from the equilibrium position imparting energy to it in the form of velocity. The anvil-bit then has momentum because of its mass and velocity and will produce formation forces relating to these factors. A portion of this kinetic energy will be used to "drill" formation and this is a form of damping designated as $C_2$. The remaining energy will be returned from the formation to the anvil-bit because of formation elasticity or spring rate, $K_2$, in the form of veloicty or rebound in the opposite or negative X direction. The inertial effect of the anvil-bit mass velocity causes it to "overshoot" the equilibrium position and causes anvil thrust ring spring compression, potential energy storage in them in addition to static equilibrium force, W, energy until the mass again slows down, reverses direction and accelerates in the positive direction. The elastic restoring forces in the formation and the anvil thrust ring springs always tend to return the anvil-bit mass center to equilibrium while the percussive blow continually disturbs this tendency. As the anvil-bit moves downwardly again another inphase positive impulse force, $F(t)$, is added to the positive anvil-bit motion, thereby adding more energy to the reciprocating mass, $m_2$, and reinforces the movement. The forces that now can be exerted on the formation are the static weight drill collar forces, percussive forces imparted to the anvil-bit by the hammer, and the forces supplied by the inertial kinetic energy in the anvil-bit and the potential energy of the anvil thrust ring springs. On each cycle as above described, the anvil-bit forces are such as to cause it to pass the equilibrium position and the system will be storing energy that is not used drilling the formation and over a series of cycles can develop steady-state forces that can be said to be magnified.

The anvil-bit has a natural undamped frequency, $p_u$, equal to $\sqrt{K/m_2}$, and a modification of this formula with the damping factor, $n = C/2m_2$, yeilds the system anvil-bit natural damped frequency $p = \sqrt{p_u^2 - n^2}$.

When the anvil-bit mass $m_2$, is excited by the forced frequency w, hammer forcing function F($t$), to equal its natural damped frequency, resonance occurs and the inphase impulse force conditions are met for force reinforcement and force magnification as described above. This is to point out that a first relationship between the anvil-bit mass $m_2$ and the combined system restoring forces K is such that the forces K must deliver the mass $m_2$ to certain advantageous positions within specified limits during steady-state oscillation, and a second relationship between the anvil-bit mass $m_2$ and the reciprocating hammer mass $m_1$ is such that percussion of the masses $m_1$ and $m_2$ must occur within said certain advantageous position limits.

Theoretically, if the forced frequency $w$ were equal to the undamped natural frequency $P_u$, the magnification factor and amplitude would be very large if no physical restraints were placed on the system in the form of damping. This can be seen from FIG. **19.** The system, however, has a fairly high range of damping but always less than $\sqrt{4Km_2}$ that is said to be variable because of the formation variable, $C_2$. This damping of the oscillating anvil-bit occurs mostly as energy usage in drilling the formation which may vary significantly and would contribute the major part of the damping coefficient, C, made up of $C_1$ and $C_2$. This variable, $C_2$, also will determine the numerical value of the ratio of actual damping, C, to the critical damping, $C_c$. The damping factor $C/C_c$ may range from near zero (0) for the hard elastic formations to unity (1) for the softer energy absorbing formations and because of this variation it is one of the major determinants of the actual values of the magnification factor, $X_m/F(t)/K$. Another major contribution to the magnification factor is the ratio of forced frequency to natural frequency. The effects associated with resonance are spread over a broad range of frequencies so specifically percussion should occur anytime during the positive travel of the anvil-bit mass. This means that from resonance the variation of forced frequency to natural damped frequency may arbitrarily range from no less than 0.5/1 to no more than $\sqrt{2}/1$ and within these limits magnification occurs.

The formula

$$\frac{X_m}{F(t)/K} = \frac{1}{\sqrt{[1-(w/p_u)^2]^2+[2(C/C_c)(w/p_u)]^2}}$$

expresses the magnification factor in terms of the frequency ratio, $w/p_u$, and the damping factor $C/C_c$. It may be used to determine the amplitude, A, of the steady state vibration produced by the impressed force, F($t$). This relationship of magnification factor, the frequency ratio, and the damping factor is also shown graphically in FIG. **19.** Thus it can be seen that considerable force magnification can be generated at or near system resonance in tough formations.

Considering the foregoing, the following formula would approximately express the system produced formation forces when operating in mode, $M_2$.

$$F_{M2} = W + \left(F \times \frac{X_m}{F(t)/K}\right)$$

showing that the forces exerted on the formation would be drill collar load forces plus a magnified percussive blow force,

As previously stated this tool is capable of two operational modes and both have been described herebefore. The tool design paramaters may be set so the system would always be a critically or overdamped stiffness controlled system and therefore would be assured of inphase superposition force operation. But each operational mode has a broad frequency range and may be made to overlap. Then it is also possible to provide design paramaters that would allow the tool to change operational modes when it encounters significant changes in formation characteristics. That is, if the tool design were set for some arbitrary condition of critical damping, any actual formation damping change could cause the damping to switch from the critical damping condition and allow mode change. Of cours when mode $M_2$ conditions existed, the forced frequency would have to substantially match the anvil-bit natural frequency. This could be easily adjusted by changing the fluid supply pressure if required, but the natural feed-back couple inherent in a conservative intertial interchange system would somewhat self-adjust or tune itself according to the hardness of the formation encountered and therefore adjust cycle frequency accordingly. Also, the resonant effects are spread over a broad variation range and would further tend to make it comparatively easy to operate the system within the specified limits described by the 0.5:1 and the $\sqrt{2}$:1 ratios. It is an object to maintain broad operating resonant frequency ranges in the mass controlled system so that the whole spectrum of tool operation can be covered without underdamped non-resonant tool operation. This specification sets specific limits on the invention with damping factors of one (unity) or more for operation in the non-resonant frequency mode stiffness controlled system and with damping factors of one (unity) or less for operation in the resonant frequency mode with 0.5/1 and $\sqrt{2}/1$ forced frequency to natural frequency ratio resonance limits in a mass controlled system.

No other known device of this type used for the same purposes exhibits the characteristics of this tool. The tools in use conform to standard industry practice and do not coodinate in like manner energy usage from the weight or force applied on the percussion drill and do not even remotely suggest the disclosures herein. Also, then this specification sets further limits that energy usage for formation disintegration of all operations in either mode be generally inpphase, coincide or substantially overlap.

## 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;

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 X at percussion and the reactionary period thereafter caused by a combination of percussive force F and anvil thrust ring force W;

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 an anvil thrust ring spring of an alternate construction;

FIG. 10 is a longitudinal cut-away view of the percussion tool in the area of the valves showing the components as in a portion of FIG. 1 except that cut-away sectional view taken is rotated 45° from that in FIG. 1 and shows the vertical passageway through the valve element as well as a 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 tool applied forces on the anvil-bit are plotted with respect to time during operation of the percussion tool and this graph shows the inter-relationship of these paramaters 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 cut-away view of the valve 4 showing the passages and ports in it, its four valve faces, and its overall configuration;

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

FIG. 17 is a schematic model diagram of the spring-mass vibrational system of the percussion tool and graphically represents the theoretical concepts involved in system function, the forces involved in system operation, and other factors involved in the invention;

FIG. 18 is a graphical comparison of the motion described by the center of mass, $m_2$, of the anvil-bit during steady state operation when the system is overdamped, critically damped, and underdamped;

FIG. 19 is a graph in which the amplitude or magnification factor has been plotted against the frequency ratio for the various values of the damping factor during resonant frequency or mode $M_2$ operation;

FIG. 20 is a longitudinal quarter sectional cutaway view in the area of the valves showing an alternate construction of the valve cage and valve element;

FIG. 21 is a transverse, vertical view of the upper portion of the percussion tool B' with a quarter sectional transverse cut-away view;

FIG. 22 is a continuation of FIG. 21 showing the lower portion of the percussion tool with a conventional solid button BUTTON bit, C', as an integral part of the anvil;

FIG. 23 is a longitudinal quarter sectional cutaway view similar to the view in FIG. 21 in the area of the valve cage but rotated so that a section is taken through the set of alternate passageways through the valve cage; and

FIG. 24 is a horizontal, cross-sectional view taken on line 24—24 of FIG. 21.

## DETAILED DESCRIPTION

Structure to carry out the objects set forth and the basic functional performance of two mode operation, superposition force, inphase energy consumption, force magnification, and formation cooperation with the drilling means are described hereafter. These constructions are for the most part the same but will vary to some extent because of existing equipment with which they must function, because of historical practices that came into being over the years, and because of difference in application which varies with the industry involved. While the oil and gas industry mostly employ liquids as the system operating fluid medium, the mining, blast hole, and construction industry employ mostly a gaseous fluid, primarily air, as the system circulating medium. The device must be capable of operation on a variety of fluids but fluids vary considerably in compressibility, viscosity and physical state such as gaseous or liquid. Therefore, although the basic ideas and constructions involved are primarily the same, a variation in valving means is included to provide optimum performance and optimum economy according to the fluid used in the system. Constructions shown in FIGS. 1, 2, 10, and 20 are specifically provided for relatively non-compressible fluids but could be used on compressible fluids while constructions shown in FIGS. 21, 22, and 23 are specifically for compressible fluids and will not operate on relatively non-compressible fluids. This alternate embodiment is limited to the valving means for hammer oscillation to provide tool simplicity and manufacturing economy when the tool is operated from a gaseous pressure source.

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 conventional tooth or button type, threadedly engages the lower end of the rotary percussion tool for drilling a borehole. Although a threaded detachable bit is shown, the bit may be made an integral part of the anvil and be of solid fixed design as shown in FIG. 22.

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 threadedly 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 screwthreaded, as indicated at 12, to threadedly 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 fluid-tight 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 the 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 a 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 threadedly 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 threadedly 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,** FIGS. **2** and **8,** 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 may threadedly engage 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 fluid-tight seal with the 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 snap ring **61** to assemble the 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 fluid-tight 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 confine anvil thrust rings **5.**

The anvil **2** may be a separate part from the bit as shown in FIG. **2** or may be an integral part with the bit as shown in FIG. **22** but in either case the anvil and bit together form the mass, $m_2$, of the spring mass system shown in FIG. **17.** The anvil **2** along with the bit is suspended by the elastic restoring forces of the system and is adapted to oscillate axially while the drilling tool rotates.

Anvil thrust rings **5** are extensive mechanical force storage and energy usage phase transfer means in the form of annular spring elements of various configurations and materials 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, frustro-conical annular springs as indicated in FIGS. **2, 11,** and **22** or radial ring spring shape as indicated in FIG. **14** by **5A.** Another anvil thrust ring spring design is indicated in FIG. **9** by **5B** in which flat annular metal rings are alternately stacked with an elastomer and having bonded interfaces. The function of the anvil thrust rings is to store various forces exerted on them and to return said forces to the confining shoulders. Specifically they store and transfer force energy exerted by the tool housing **1** to the anvil **2** and as indicated in FIGS. **5, 13,** and **17** by W. They also extend simultaneously with anvil **2** downward movement due to the percussive blow $F(t)$ and coact with said blow to simultaneously release the energy stored therein for superposition bit forces on the formation as used in mode $M_1$ tool operation. In this function and mode they also change the phase of drill collar weight energy consumption about 180°, depending on conditions, and puts said energy consumption in phase with the percussive blow for more effective energy usage. FIG. **13** graphically shows the phase alignment and hense superposition force. In addition to the above the anvil thrust rings **5** in mode $M_2$ operation provide the tool portion of the elastic restoring forces K and the damping coefficient C to the oscillating anvil-bit. This elastic restoring force is indicated in FIG. **17** by $K_1$ and the damping coefficient is indicated by $C_1$.

The formation, O, being drilled plays a large role in system operation as it cooperates with the system to provide damping, the energy consumption of formation disintegration, and a portion of the elastic restoring forces, the energy returned to the system by the formation. The formation damping coefficient is indicated in FIG. **17** by $C_2$ and the formation elastic restoring force is indicated by $K_2$. The formation then consumes system energy or returns it to the system in varying degrees depending on the makeup of the particular formation.

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 as 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.

The 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 screwthreaded 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 surfaces of valve **4** and is in sliding arrangement thereto and in fluid-tight 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 the 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 frustro-conical 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 fluid-tight 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 **4** which may be made of two or more joined pieces and screwthreaded 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 spool-like portion which joins the upper and lower tubular ends and has four valve faces **45, 46, 47,** and **48**, FIGS. **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**, FIGS. **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**, FIGS. **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 when valve **4** is shifted downwardly relative to hammer **3** that 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 **29**, 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 spool-like 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, inertia 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 area 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 sub 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 characteristics of the formation.

Lower tubular end 20 of the 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 passageway 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.

A valve and valve cage alternate construction is shown in FIG. 20. This design functions in the same manner as the valve shown in FIG. 1 but is of a simpler configuration made possible with the use of better erosion resistant materials and metalurgy. Also a spring has been incorporated that biases the valve upward and allows tool to begin cycling easier. This spring force aids initial valve shift when valve inertial forces are minimal.

An enlarged spool-like portion of valve 4 has external cylindrical and oppositely facing flat surfaces forming edges, 63 and 64, that in cooperation with internal and end face surfaces 65 and 66 of valve rings 67 control charge pressure fluid from passage 51. Cylindrical surfaces of 63 and 64 are slightly smaller than cylindrical surfaces of 65 and 66 of valve rings 67 and are positioned concentrically. The construction is such that when the fixed spaced edges 63 and 64 are shifted down relative to edges 65 and 66 pressure fluid can flow to the chamber 39 formed below piston 25, and pressure fluid is substantially blocked to chamber 36, formed above piston 26. Conversely, when valve 4 is shifted relatively up, coacting edges 63 and 64 move apart and allow charge pressure fluid flow therebetween while edges 65 and 66 move together and past each other essentially stopping the fluid flow between said edges. Valve rings may be generally rectangular in cross section and closely fitted to bore of piston 25 with a fluid seal therebetween. Valve rings are spaced in bore of piston by thin wall tubular spacers 68 and 69. Adapter 34 which is screwthreaded into mating thread at top of piston clamps spacers 68 and 69, valve rings 67 and adapter 37 in bore of piston 25. Spring 70 surrounds upper portion of lower tubular end 20 of valve 4 and its ends are nested against the enlarged surface 57 on the upper side, and against hammer 3 on the lower side. The spring as installed is somewhat compressed and maintains force between valve 4 and hammer 3.

The foregoing valve structures would be used for the most part on a liquid fluid driven tool while the following constructions shown in FIGS. 21, 22, 23 and 24 apply totally to a gaseous fluid driven tool. The liquid and gaseous fluid circuits are the same and only the circuit switching valve is of modified design. Also the housing 1, anvil 2, piston 25, and anvil thrust rings 5 are the same as before described. The valving means, although similar in some respects, has notable differences.

With more detailed reference to the Drawings, fluid switching valve means are comprised of the valve 75, and the valve block or cage 76. Valve 75 is a single element that conducts pressure fluid to and from the valve cage and in cooperation with the valve cage controls charging and discharging of chamber 36 above hammer 3 and chamber 39 below hammer 3. Hammer 3 in this embodiment is made up of piston 25 and valve cage 76. Valve 75 has an upper tubular end 77, extending into bore 10 of top sub 6 with a seal 13 therebetween, a lower tubular end 78 extending into and may rest on top portion of anvil 2 and in sealing arrangement therewith by seal 79. Although not shown, it is apparent that the valve 75 could equally well be fastened to the top sub 6 since these parts are stationary relative to each other and that this may be advantageous due to the fact that the valve 75 would be free from the rapid anvil 2 oscillations. Central portion 85 of valve 75 is solid or may have a passage therethrough that blocks or substantially restricts fluid flow from passage of upper tubular end 77 to passage of lower tubular end 78. Upper tubular end of valve conducts pressure fluid to the radially divergent side parts 80 above central restriction 85 while the lower tubular end conducts fluid from the radially divergent side parts 81 into the bore of anvil 2. Valve 75 is slidably mounted through central opening 82 of valve cage 76, and outside surface of valve 75 is a close sliding fit with opening 82 so that the surfaces in close proximity or contact form a seal or greatly restricts fluid flow therebetween, thus eliminating separate sealing means.

Valve cage **76** is a single element screwthreaded in piston **25** that has a central bore opening **82**, and two enlarged bores, **83** being the upper bore and **84** the lower two diameter bore, concentric with opening **82** and surrounding outside of valve **75**. Valve cage also has one or more passages **86** formed in part by valve cage **76** and piston **25** that allow fluid communication from space formed by bore **83** to passage **42** and one or more passages **87** that allows fluid communication from space formed by enlarged bore **84** to chamber **36** above hammer **3**.

Now it can be seen from FIGS. **21** and **22** that when hammer is in its lower position pressure fluid can flow down through upper tubular end **77** of valve **75**, through its side parts **80** into space formed by bore **83**, through passage **86** and **42** into chamber **39** to act on the underside of the hammer to raise same. Similarly it can be seen from FIGS. **21** and **23** that exhaust fluid can flow from chamber **36** through passage **87** and space formed by bores **84**, through valve side ports **81** and into lower tubular end **78** of valve **75** to allow fluid pressure dissipation above hammer **3** and thus a differential pressure across the hammer. Now when the hammer is in an up position, and valving is switched to power the hammer downwardly, the relative position of valve and cage parts have changed so that pressure fluid will flow from the upper tubular end of valve **75**, through its side parts **80** and into space formed by the two diameter bore **84**, through passages **87** in valve cage **76** into chamber **36**, thus providing force on hammer to drive it downward. Similarly it can be seen that since hammer is raised relative to valve ports **81** and thus exposing said port below lower end of valve cage **76** that exhaust fluid can flow from chamber **39** below hammer through passage **42**, through exposed part **81** into lower tubular end **78** of valve **75** and thus allows fluid exhaustion and pressure dissipation below hammer for creation of differential pressure across the hammer and movement of same downward for the percussion blow and anvil excitation. Since the valve is relatively stationary while the hammer moves, sliding port valving action occurs before the end of each up and down hammer stroke for alternate pressure fluid application up to hammer and reciprocation of same. This type valve action is permissible with a gaseous fluid and in fact highly desirable since the time required to bring hammer actuation pressure up to supply pressure is longer than when a liquid is used and thus in effect allows early fluid valving to compensate for the slower pressure buildup. Also the hammer is positively actuated in the opposite direction at each end of its stroke, thus eliminating "dead spots" at these positions. Fluid switching occurs intermediate the hammer stroke ends and momentum forces of the hammer are such as to carry out the valving action although the actuation fluid flow is momentarily halted when the involved ports do not allow fluid passage. The constant alteration of unbalanced pressures across the hammer cause rapid oscillations of the hammer and on each cycle imparts inertial forces to the anvil-bit according to the magnitude of the pressure source, the specific tool configuration and the formation being drilled. The object of course is to increase the rate of penetration of the bit through the formation and to accomplish this most expediently by exciting the bit in a certain fashion.

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 and anvil-bit formation forces.

FIG. **5** illustrates the forces and occurrences that transpire at percussion in mode $M_1$. W indicates the drill collar load applied to the bit by the anvil thrust rings before and at percussion. F indicates the percussive force applied to the anvil-bit. The combined forces of W and F cause anvil extension relative to the housing and is indicated by X.

FIG. **13** further graphically explains the sequence and nature of events before, during, and after percussion in a general and typical manner in operational mode $M_1$. On the graph, forces would be zero at the bottom of graph and would increase upwardly while time progresses from left to right. W indicates 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 S. 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 S 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 equal to drill collar weight load force W at percussion. F indicates the force applied to the anvil **2** and hence the bit by the percussive impact of the hammer on the anvil and, as shown, is synchronized for application at maximum drill collar load application W. Forces W and F coact and are added arithmetically to produce a relatively high instantaneous total load force spike, as indicated by FT, on the bit for formation crushing or cleavage by forcibly extending the anvil some distance, as indicated by X in FIG. **5**, said force spike exceeding formation resistance to contact areas of the bit. In the graph (*t*indicates time percussive force is applied to the anvil-bit and V indicates total tool cycle time. Bit loadings drop considerably after expenditure of forces F and W and anvil extension, and then rise to maximum again as the drill collars move downwardly and distort the anvil thrust rings while the hammer progresses through its up and down stroke to again deliver a percussive blow. FIG. **17** is a schematic model of the percussion tool on the end of a drill string in a borehole. The valving and pressure fluid have been omitted for clarity. This diagram, a particular configuration of the classical forced-damped-spring-mass system, indicates the factors affecting the anvil-bit, con-

trols its behavior, and further indicates conditions of the system. $m_1$ is the reciprocal hammer mass that provides the forcing function by delivering the percussion blow of F force for time duration ($t$). $m_2$ is the anvil-bit mass with its center of mass indicated at $X_o$, and its one degree of freedom indicated by an arrow and + or −X. It should be noted X is the displacement from $X_o$ and + or − indicate direction and position. For instance, X may be the displacement of $m_2$ in the minus direction but it may have a positive or negative velocity. $m_3$ is the mass of the drill string that exerts force W on the anvil thrust ring spring. O is the formation as a whole. $C_2$, the damping coefficient of the formation, and the spring constant, $K_2$, of the formation are graphical representations of formation conditions. It is thought that these variables, at least under some conditions, are velocity dependent and can be handled with greater accuracy when more knowledge of rock fracture versus time is available. $K_1$ and $C_1$ indicate the spring rate and the damping coefficient of the anvil thrust ring spring respectively and are design set to compliment other conditions of the system.

FIG. **18** shows in a comparative graphical way the anvil-bit motions for the conditions of $M_1$ and $M_2$ mode operation. The curve, $C_c$, correlates with the force-time graph of FIG. **13** but curve $C_u$, which is an example of mode $M_2$ operation, has no relation to FIG. **13.** This is because the forces of mode $M_2$ are not simply additive but additive and magnified. It should be pointed out, however, that the reinforced forces generated by mode $M_2$ operation could have been indicated as an increased F force in FIG. **13.** In steady state mode $M_2$ operation, the average energy input is equal to the average energy output and the total energy of the system is constant for constant conditions, but the system has stored energy in a "potential energy well" that allows force magnification. This energy is kinetic as the velocity of the anvil-bit or potential as force storage in the elastic restoring forces and these energies are constantly changing and interchanging during the reciprocating anvil-bit motions. These motions are described by the center of mass of the anvil-bit relative to time and conditions. X is the displacement of the center of mass of $m_2$ from $X_o$, $X_m$ being the maximum displacement. Time progresses from the left at $F(t)_o$ where time begins at the percussive blow and progresses to the right through the completion of one tool cycle of 360° for each mode operation motion curve. H indicates relative formation hardness and $H_2$ indicates the hardness at which critical damping occurs while $H_1$ is a harder formation and $H_3$ is a softer formation. Curve $C_c$, a dashed line, describes the motion of the anvil-bit when the system is critically damped and forms a limit or dividing line between mode $M_1$ and $M_2$ operation. Curve $C_o$, a dotted line, describes an overdamped system condition and is an example of mode $M_1$ operation. The total cycle time, $t_3$, is longer and the displacement is greater than that of $C_c$ indicating lower frequency and greater instantaneous penetration rate. The center of mass of $m_2$ only return, or attempts to return to $X_o$. In mode $M_2$ operation, indicated by a solid line and denoted $C_u$, the forcing function $F(t)$ displaces the mass, $m_2$, a distance, $X_m$, in formation hardness, $H_1$, in the same manner but the instantaneous penetration is less, and a certain amount of energy is conserved which returns the anvil-bit across position $X_o$ a distance, −X. Under the influence of the elastic restoring forces the motion is reversed and $m_2$ is in a minus position but begins positive motion. During the positive motion another percussive hammer impact, $F(t)$ occurs at time $t_1$, cycle end, and reinforces the motion of $m_2$. The force associated with this motion is substantially increased as compared with the force associated with that of mode $M_1$ operation because of the force reinforcement and resulting magnification. The force magnification factor described herebefore is shown graphically in FIG. **19.** Curve $C_o$ is descriptive of operational mode $M_1$ in that it is characterized by a damping factor of one or more and $C_u$ is descriptive of operational mode $M_2$ in that it is characterized by a damping factor of less than one and the percussive energy reinforces system dynamic stored energy. It should be noted that this specification does not cover underdamped non-resonant system frequency as this condition should be avoided. In FIG. **18** this condition would be shown as a percussive blow occuring during negative travel of $m_2$ on the solid line curve.

FIG. **19** is a graph in which the magnification factor has been plotted against the forced to natural frequency ratio for various values of the damping factor. It may be used to determine the system mode $M_2$ steady-state vibration amplitude of the anvil-bit. When damping is present and provided that it is less than critical the amplitude passes through a maximum as the forced frequency, w, is varied. Also it should be noted that as the damping factor, $C/C_c$, approaches unity that the resonant effects are spread over a wider range. If the system is critically damped as in mode $M_1$ operation as when $C/C_c = 1$, the mass, $m_2$, does not effect the system and amplitude tends to decrease with increased forced frequency and would indicate that a given tool designed to operate in both modes but operating in mode m, would be operated more advantageously at half or less than resonant frequency and at or near critical damping.

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 a 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 of torsion, shear, tension, and compression and are within the proportional stress-strain 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 for increased deflection and minimum load 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 frustro-conical 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 frustro-conical 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 the anvil is extended. Yet another configuration of anvil thrust ring spring is shown in FIG. **9.** This unit is made of ring-shaped flat washer metal plates spaced at intervals with an elastomer molded to and between the plates. This type anvil thrust ring possibly utilizing urethane or rubber would have application in the smaller portable mining blast hole drilling tools where the loads encountered would be lesser.

Although three preferred types of anvil thrust ring spring configurations are shown in the Drawings and indicated in FIGS. **2, 22,** and **11** by **5,** in FIG. **9** by **5B,** and in FIG. **14** by **5A,** it is anticipated that other spring designs such as coiled wire can be used to implement the baic 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, force magnification, inphase energy usage from two different sources, and the superposition of percussive blow force on drill collar weight applications has been explained throughout this specification, but the idea is not limited to use with dead weight forces caused by gravity. Most modern smaller drilling rigs as used for seismograph shot hole drilling, mining blast hole drilling, core drilling and quarry boring are equipped with automatic hydraulic pull down 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 the valve is positioned to allow fluid flow below the hammer.

Fluid is introduced to the liquid driven percussion tool, FIGS. **1** and **2,** 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 the 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 surface, 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 the 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 the 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.

The hammer and valve again move in unison downwardly. At the termination of hammer down stroke, hammer strikes anvil a percussive blow, said blow coacting with stored force in anvil thrust rings to instantaneously accelerate the anvil-bit, creating a high impulse force spike that is transferred through the bit to the formation for crushing same. At percussion and sudden deacceleration 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 fluid again flows as orginally directed below the hammer.

An operation cycle of the liquid tool mass motor 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 conversed 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 in this liquid driven motor. It insures 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 insures 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.

The gas driven motor valving structure effectively causes hammer oscillation and anvil-bit excitation just as the liquid valving structure does but the valving action functions somewhat differently because of the gaseous state of the fluid. Gas is introduced to the tool through the top central opening, flows into the upper tubular end of valve **75**, through its side ports **80**, counterbore **83**, passages **86** and **42**, and into chamber **39** below the hammer, tending to raise it from its lower position while lower pressure gas is exhausted from chamber **36** above the hammer which flows through passages **87** and bores **84** of valve cage, through side ports **81** and lower tubular end of valve **75** into the bore of the anvil and on through the system. It is apparent now that a differential pressure exist across the hammer and that it will move relatively up away from the anvil and gain velocity and momentum. The ports in valve **75** and the counterbores **83** and **84** in the valve cage are spaced so that after a certain valve cage travel all of the ports are blocked to fluid communication momentarily but because of hammer momentum and gas elasticity, the hammer continues to travel upwardly until valve ports have communication with the opposite chambers at the ends of the hammer. High pressure fluid now will flow from valve port **80**, into bores **84**, through passages **87**, into chamber **36** above hammer while fluid from chamber **39** exhausts through passage **42** into valve side port **81** exposed below valve cage and on through the system thus creating a differential pressure force that will drive the hammer downwardly relatively toward the anvil. Of course, because valving occurred on the hammer upstroke the hammer will travel upward a comparatively long distance before it is reversed by the high pressure fluid acting to accelerate it downward and bores **84** are rather long to provide full long stroke driving power for hammer acceleration and percussion with the anvil. Before percussion occurs fluid pressures are switched back as originally directed for another hammer upstroke. Valving is such that it switches intermediate the hammer stroke ends thus always creating a bias on the hammer and oscillation of same. This valving may be made simpler because gas is usually cleaner than liquids and is very tolerant to volume changes and is very resilient.

During steady-state drilling system operation the drill bit moves forward through the formation at an average rate that is equal to the rate of travel of the drill string that follows. The drill bit although having an average rate of progression is oscillating axially since the anvil-bit is periodically being driven forward by the percussive motor and is alternately returned by formation resistance to the bit. The center of mass of the anvil-bit then is oscillating relative to its average center of mass location and to the drill string. Also the anvil thrust rings are storing energy when the anvil-bit is moving closer to the drill string and are releasing said stored energy when the anvil bit is moving away from the drill string and are therefore adding energy to the forward bit thrusts and absorbing energy from the drill string as the bit slows its forward movement. The preload of the system which is the gravitational effect of the drill string or other equivalent force is continually pumping its energy into the oscillating system and is used in phase with the forward thrusts of the anvil bit to increase the amplitude of oscillation. Now depending on formation conditions which are the primary damping effects and to a large extent the elastic restoring force of the system, various conditions of operation are possible. If the system is overdamped or critically damped, superposition forces can be applied to the anvil-bit. This is to say that the percussive blow impulsive force and the anvil thrust ring spring forces are additive resulting in increased magnitude of forward thrusts of the bit. If the system is underdamped, magnified superposition forces can be applied for increased oscillation amplitude and increased formation disintegration.

The above describes the basic underlying operational parameters of the percussion tool. The key to this whole vibrational spring-mass drilling system lies in the anvil thrust ring elastic element(s) and their ability to store and return energy, their ability to change the phase of said energy usage, and their ability to coact with and reinforce the impulsive forces produced by the fluid driven inertial motor. The system then has the ability to store energy over comparatively long periods from two different sources, coordinate it and applies it in comparatively short period concentrated high energy bursts for increased effectivity in drilling resulting in an extraordinary drilling device.

These motors are so designed that they will not percuss against anvil when the lower extendable joint is fully extended. This permits full pressure bore hole 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 the 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.

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.** A percussion drilling apparatus comprising:
- a drilling mass having a fundamental damped vibrational frequency;
- means for elastically biasing the drilling mass into engagement with a formation to be drilled;
- driver means for reciprocating the drilling mass toward the formation at a rate substantially equal to its fundamental damped vibrational frequency; and
- fluid means for actuating the driver means to reciprocate the drilling mass.

**2.** A prime mover apparatus for drilling purposes comprising:
- a drilling mass;
- elastic restoring means for cooperation with the drilling mass to provide a natural damped period of oscillation of the drilling mass, and for biasing the drilling mass toward a formation to be drilled;
- fluid powered hammer means for reciprocating the drilling mass toward the formation at a frequency

substantially identical to the natural damped period of oscillation of the drilling mass; and

valving means for regulating the operation of the fluid powered hammer means.

3. A forced resonant harmonic prime mover apparatus for borehole drilling comprising:

a reciprocal drilling mass;

means for elastically restoring the mass to provide a natural damped period of oscillation of the mass and for biasing the drilling mass toward a formation to be drilled;

means for reciprocating the drilling mass relative to the formation at a frequency substantially equal to the natural period of oscillation of the mass; and

fluid means for actuating the means for reciprocating the drilling mass at a frequency substantially equal to the natural period of oscillation of the drilling mass and thereby reinforcing the movement of the mass to increase the amplitude of oscillation of the mass.

4. A forced resonant harmonic prime mover for borehole drilling according to claim 3 wherein the means for elastically restoring the drilling mass includes means for applying a separate force directed along the axis of oscillation of the drilling mass and thereby superimposing the oscillation forces of the drilling mass on the separate force.

5. The prime mover according to claim 4 wherein the means for reciprocating the drilling mass includes a reciprocal hammer powered in both directions of reciprocation by fluid under pressure.

6. A percussion drilling apparatus comprising:

a down hole reciprocatory percussion motor;

a drilling bit mounted for outward reciprocation by the percussion motor to form a borehole in the earth;

a drill string for supporting the percussion motor and the drilling bit and for applying an outwardly directed force; and

spring means mounted between the drill string and the drilling bit for applying the drill string force to the drilling bit during the outward reciprocation of the bit by the percussion motor so that the outwardly directed force of the percussion motor is superimposed on the outwardly directed force of the drill string to increase the drilling action of the bit.

7. The percussion drilling system according to claim 6 wherein the percussion motor reciprocates the drilling bit at a predetermined frequency and wherein the means for applying the force of the drill string to the drilling bit comprises at least one elastic element mounted between the drill string and the drilling bit for cooperation with the formation of the earth to oscillate the drilling bit at a frequency substantially identical to the predetermined frequency of operation of the percussion motor.

8. The percussion drilling system according to claim 7 wherein the percussion motor comprises an anvil slidably supported at the distal end of the drill string and having the drilling bit secured thereto, a hammer mounted for reciprocation into and out of engagement with the anvil, and valving means for alternately directing fluid pressure supplied through the drill string to the opposite sides of the hammer and thereby causing alternate inward and outward reciprocation of the hammer.

9. The percussion drilling system according to claim 8 wherein the valving means comprises a valving member mounted on the hammer for movement with respect thereto under its own inertia to a first valving position whenever the hammer reaches one extreme of its reciprocation and for movement under its own inertia to a second valving position whenever the hammer reaches the other extreme of its reciprocation, and means for moving the hammer toward the anvil whenever the valving member is in the first valving position and for moving the hammer away from the anvil whenever the valving member is in the second valving position.

10. The percussion drilling system according to claim 8 wherein the valving means comprises a member extending from the drill string through the hammer to the anvil and having pressurized fluid inlet and exhaust fluid outlet passages formed in it, and a pair of passageways formed through the hammer, the first extending to one end of the hammer and including a portion located for communication with the inlet passageway when the hammer is at one extreme of its reciprocation and for reciprocation with the outlet passageway when the hammer is at the other extreme of its reciprocation, and the second extending to the opposite end of the hammer and including a portion located for communication with the inlet passageway when the hammer is at one extreme of its reciprocation and a portion located for communication with the outlet passageway when the hammer is at the other extreme of its reciprocation.

11. A percussion drilling system comprising:

a drilling bit for penetrating a formation of the earth in a predetermined direction;

drill string means providing a force having the predetermined direction;

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

means mounted between the drill string means and the drilling bit for transferring the drill string force energy to the drilling bit inphase with the movement of the bit toward the formation under the action of the reciprocating means.

12. The percussion drilling system according to claim 11 wherein the force energy transferring means further comprises means for storing energy from the force of the drill string during movement of the bit toward the drill string.

13. The percussion drilling system according to claim 12 wherein the force energy transferring means comprises at least one spring mounted between the drill string means and the drilling bit and having such a spring constant that the formation of the earth and the spring cooperate to restore the drilling bit substantially inphase with the movement of the drilling bit reciprocating means.

14. The percussion drilling system according to claim 13 wherein the drilling bit reciprocating means comprises a percussion motor mounted in the distal end of the drill string and including a hammer which is reciprocated both inwardly and outwardly relative to the drill string means under the action of fluid pressure.

15. A system for drilling a borehole in a formation of the earth comprising:

a drill string;

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

means for cooperation with the formation being drilled and the anvil-bit mass to oscillate the anvil-bit mass at a natural damped period of oscillation; and

means for reciprocating the drilling anvil-bit mass outward during outward oscillation of the anvil-bit mass to reinforce the outward oscillation of the anvil-bit mass and thereby magnify the anvil-bit oscillation.

**16.** The system according to claim **15** wherein the means for cooperation with the formation being drilled and the anvil-bit mass includes means for applying the force of the drill string and thereby superimposing the anvil-bit mass oscillation forces on the force of the drill string.

**17.** The system according to claim **16** wherein the drill string provides an outwardly directed force and wherein the means for cooperation with the formation being drilled and the anvil-bit mass includes means for storing the force energy of the drill string during inward oscillation of the anvil-bit mass and transferring the force energy of the drill string to the anvil-bit mass simultaneously with the outward oscillation of the anvil-bit mass and thereby changing the phase of energy application from the drill string to the anvil-bit.

**18.** The percussion drilling system according to claim **15** wherein the means for reciprocating the anvil-bit mass includes a reciprocating hammer and wherein the anvil-bit mass oscillating means comprises at least one spring mounted between the drill string and the anvil-bit mass which in combination with the formation being drilled forms a combined elastic system that functions to substantially return the anvil-bit mass to a predetermined location in a prescribed period of time with a predetermined direction of movement.

**19.** The drilling system according to claim **18** further comprising an anvil as part of the bit mass for impact by the hammer and wherein the hammer is driven in both directions of its reciprocation by fluid pressure which is supplied through the drill string.

**20.** A percussion drilling system according to claim **19** wherein the drilling anvil-bit mass is retained in one directional longitudinal movement with the drill string by a retainer in the groove of the drilling mass having an outer surface larger than the reduced internal surface of the drill string and having a surface that coacts with the reduced internal surface of the drill string so that when said surfaces are forced together said surfaces will forcibly keep the retainer in the groove of the drilling anvil-bit mass.

**21.** The borehole drilling system according to claim **15** wherein the anvil-bit reciprocating means comprises a hammer mounted for reciprocation into and out of engagement with the anvil-bit and valving means for alternately directing fluid pressure supplied through the drill string to the opposite sides of the hammer and thereby causing alternate inward and outward reciprocation of the hammer.

**22.** The borehole drilling system according to claim **21** wherein the valving means comprises a valving member mounted on the hammer for movement with respect thereto under its own inertia to a first valving position whenever the hammer reaches one extreme of its reciprocation and for movement under its own inertia to a second valving position whenever the hammer reaches the other extreme of its reciprocation, and means for moving the hammer toward the anvil-bit whenever the valving member is in the first valving position and for moving the hammer away from the anvil-bit whenever the valving member is in the second valving position.

**23.** The borehole drilling system according to claim **21** wherein the valving means comprises a member extending from the drill string through the hammer to the anvil-bit and having pressurized fluid inlet and exhaust fluid outlet passages formed in it, and a pair of passageways formed through the hammer, the first extending to one end of the hammer and including a portion located for communication with the inlet passageway when the hammer is at one extreme of its reciprocation and for reciprocation with the outlet passageway when the hammer is at the other extreme of its reciprocation, and the second extending to the opposite end of the hammer and including a portion located for communication with the inlet passageway when the hammer is at one extreme of its reciprocation and a portion located for communication with the outlet passageway when the hammer is at the other extreme of its reciprocation.

**24.** A drilling prime mover consisting of:

an anvil-bit mass;

pipe means supporting the anvil-bit mass;

means mounted between the pipe means and the anvil-bit mass for cooperation with the formation being drilled and the anvil-bit mass to cause oscillation of the anvil-bit mass at its fundamental damped vibrational frequency; and

means for periodically impulsively forcing the anvil-bit in its direction of movement during oscillation to increase the amplitude of anvil-bit oscillation.

**25.** The prime mover according to claim **24** wherein the pipe means provides a force and the means mounted between the pipe means and the anvil-bit mass for cooperation with the formation being drilled and the anvil-bit mass includes means for transferring the force of the pipe means so that the oscillation forces of the anvil-bit are superimposed on the force of the pipe means.

**26.** The prime mover according to claim **24** wherein means mounted between the pipe means and the anvil-bit mass is an elastic member that biases the pipe means and the anvil-bit mass in opposite directions.

**27.** A method of forming a borehole by percussion drilling including the steps of:

storing energy in at least one spring positioned between a drill string and a drilling bit mounted at the distal end of the drill string;

periodically forcing the bit outwardly from the drill string and into engagement with a formation to be drilled; and

transferring the energy stored in the spring to the bit inphase with the outward movement of the bit and thereby increasing the drilling action of the bit.

**28.** The method according to claim **27** wherein the step of forcing the bit outwardly from the drill string is carried out by repeatedly actuating the bit with a percussion motor.

**29.** The method according to claim **28** wherein the step of forcing the drilling bit outwardly from the drill string is further characterized by repeatedly impacting an anvil connected to the drilling bit with a reciprocatory hammer which is actuated by fluid pressure supplied through the drill string.

30. The method according to claim 27 wherein the bit has a natural period of oscillation and further characterized by forcing the bit outwardly at a frequency substantially equal to the natural period of oscillation of the bit.

31. A method of percussion drilling comprising:
oscillating a drilling bit at a predetermined natural period of oscillation; and
impacting the drilling bit at a frequency which is substantially matched to the predetermined natural period of oscillation.

32. The method of percussion drilling according to claim 31 wherein the oscillating step is carried out by biasing the drilling bit with a spring which cooperates with a formation to be drilled to oscillate the drilling bit at said natural period of oscillation.

33. The method of percussion drilling according to claim 32 including the additional step of storing energy in said spring and transferring energy from said spring to the drilling bit during outward reciprocation of the drilling bit.

34. The method of percussion drilling according to claim 33 wherein the impacting step is carried out by impacting the bit with a reciprocating hammer which is driven both into and out of engagement with the drilling bit under the action of fluid pressure.

35. A method of drilling a borehole including the steps of:
storing energy in an elastically biased mass vibrational system having a natural period of vibration; and
periodically adding energy to the oscillating mass at a frequency substantially equal to the natural period of vibration of the mass to increase the energy of the mass.

36. The method of drilling a borehole according to claim 35 including the additional step of adding additional energy to the oscillating mass by forcing the mass along the axis of oscillation.

37. A method of drilling a borehole according to claim 36 wherein the step of periodically adding energy to the oscillating mass is carried out by the percussive impact of a reciprocal hammer driven in both directions of reciprocation by pressurized fluid.

38. A percussion drilling apparatus comprising:
a drill string forming a pressurized fluid inlet passageway;
an anvil-bit slidably supported at the distal end of the drill string and including a fluid discharge passageway;
means forming a cylinder adjacent the anvil-bit;
a piston mounted in the cylinder for reciprocation therein into and out of engagement with the anvil-bit;
said piston having at least two passageways formed through it, the first extending into communication with the portion of the cylinder beneath the piston and the second extending into communication with the portion of the cylinder above the piston;
fluid inlet means for directing pressurized fluid from the drill string to a pressurized fluid outlet opening; and
fluid outlet means for directing exhaust fluid from an exhaust fluid inlet opening to the fluid discharge passageway of the anvil-bit,
valving means responsive to positioning of the piston in the lower portion of the cylinder to connect the pressurized fluid outlet opening in communication with the first passageway in the piston to direct pressurized fluid to the portion of the cylinder beneath the piston and to connect the exhaust fluid inlet opening in communication with the second passageway in the piston to direct exhaust fluid from the portion of the cylinder above the piston into the fluid discharge passageway of the anvil-bit and responsive to positioning of the piston in the upper portion of the cylinder to connect the pressurized fluid outlet opening in communication with the second passageway in the piston to direct pressurized fluid to the portion of the piston above the cylinder and to connect the exhaust fluid inlet opening in communication with the first passageway in the piston to direct exhaust fluid from the portion of the cylinder beneath the piston into the fluid discharge passageway of the anvil-bit.

39. A percussion drilling system according to claim 38 wherein the fluid inlet means and the fluid outlet means comprise passageways formed in a member mounted in the cylinder and extending from the drill string through the hammer to the anvil-bit.

40. The percussion drilling system according to claim 39 further including a reduced diameter passageway extending through the member from the passageway in the member comprising the fluid inlet means to the passageway in the member comprising the fluid outlet means for providing continuous fluid flow through the fluid discharge passageway of the anvil-bit while maintaining a pressure differential between the fluid inlet passageway of the drill string and the fluid discharge passageway of the anvil-bit.

41. The percussion drilling system according to claim 40 wherein at least portions of the passageways in the piston are formed in a valve block mounted in the piston for reciprocation therewith.

42. In a percussion drilling system of the type including an anvil-bit having a fluid discharge passageway formed through it, a hammer mounted for reciprocation into and out of engagement with the anvil-bit, and a drill string for supporting the anvil-bit and the hammer and having a pressurized fluid inlet passageway formed through it, the improvement comprising:
a member extending from the drill string through the hammer to the anvil-bit, said member having a pressurized fluid passageway extending through it from the pressurized fluid inlet passageway in the drill string to a pressurized fluid outlet opening and having an exhaust fluid passageway extending through it from an exhaust fluid inlet opening to the fluid discharge passageway in the anvil-bit; and
two passageways formed through the hammer, the first extending to one end of the hammer and including a portion located for communication with the pressurized fluid outlet opening when the hammer is at one extreme of its reciprocation and another portion located for communication with the exhaust fluid inlet opening when the hammer is at the other extreme of its reciprocation, and the second extending to the opposite end of the hammer and including a portion located for communication with the pressurized fluid outlet opening when the hammer is at one extreme of its reciprocation and another portion located for communication with the exhaust fluid inlet opening when the hammer is at the other extreme of its reciprocation.

43. The improvement according to claim 42 wherein the pressurized fluid passageway and the exhaust fluid passageway of the member are of relatively large diameter and further including a relatively small diameter passageway extending through the member from the pressurized fluid passageway to the exhaust fluid passageway for permitting constant fluid flow through the exhaust fluid passageway to the fluid discharge passageway of the anvil-bit while maintaining a predetermined pressure differential between the pressurized fluid passageway and the exhaust fluid passageway.

44. The improvement according to claim 43 wherein the pressurized fluid outlet opening and the exhaust fluid inlet opening are outwardly facing and wherein the portions of the passageways of the hammer located for communication with the pressurized fluid inlet opening and the exhaust fluid outlet opening are inwardly facing so that sliding valve action is effected during reciprocation of the hammer.

45. For use in a percussion drilling apparatus, an anvil-bit having a mass which cooperates with elastic restoring forces to establish a natural period of oscillation of the anvil-bit and having a drilling bit on one end thereof, an anvil on the opposite end thereof, a shank diameter smaller than the bit extending from the anvil end, rotational drive surfaces on said shank, a retainer groove near said anvil end on said shank, fluid passageway through said anvil, shank, and bit and a substantially radially surface connecting the outside of the shank surface with an enlarged diameter of the anvil-bit mass extending from the drilling bit end toward the anvil end and adapted to receive the application of said elastic restoring forces.

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