United States Patent

[11] **3,544,075**

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[54] **VIBRATOR SYSTEMS**
13 Claims, 13 Drawing Figs.

[52] **U.S. Cl.** **259/1,** 175/55, 299/34, 299/86
[51] **Int. Cl.** **B28d 1/24,** E21c 31/02
[50] **Field of Search** ..... 259/1, (Fluid Digest); 175/55, 56; 60/(Inquire)

[56] **References Cited**

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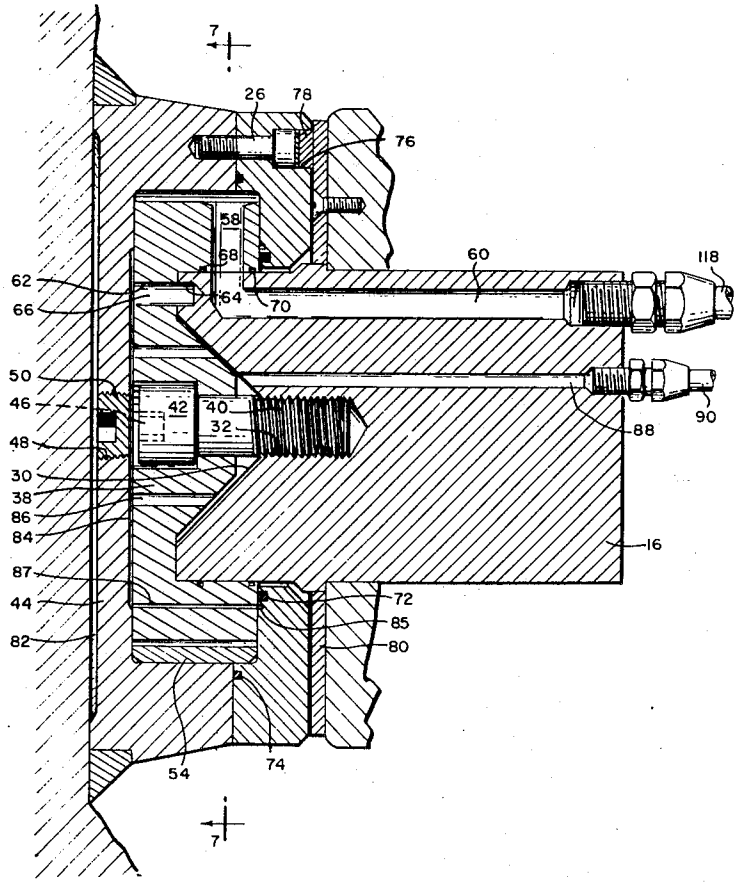
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Primary Examiner—Ernest R. Purser
Attorney—Graybeal, Cole & Barnard

ABSTRACT: A first member includes an inner cavity containing a second member which diametrically is slightly smaller than the cavity and axially is snugly received in the cavity. One of the members is connected to a large mass backup structure and the other is free for orbiting radial shifting movement within the limits of the radial gap between the members. Radially movable seals carried by one of the members divide the gap into a plurality of pulse chambers. A separate bidirectional flow passageway leads into each pulse chamber. Fluid pressure pulses are distributed around the gap in repetitive cyclic fashion from pulse chamber to pulse chamber. A radial piston type pulse generator divided into two axially spaced sets of pistons moved radially by separate rotary cams offset on opposite sides of a cam shaft. A cutting edge on such a first member, and means for moving the backup structure to advance the cutting edge into an earth formation.



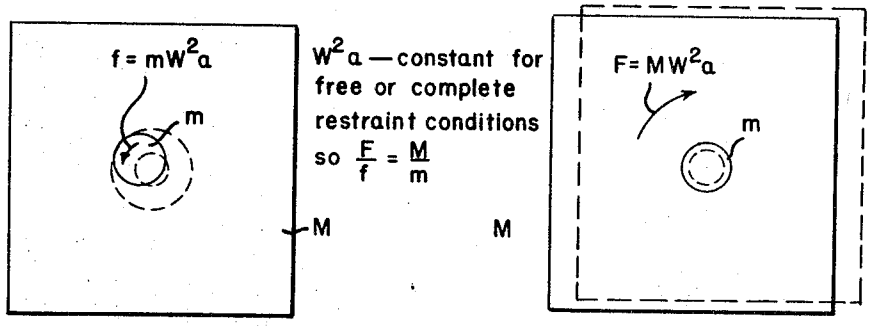
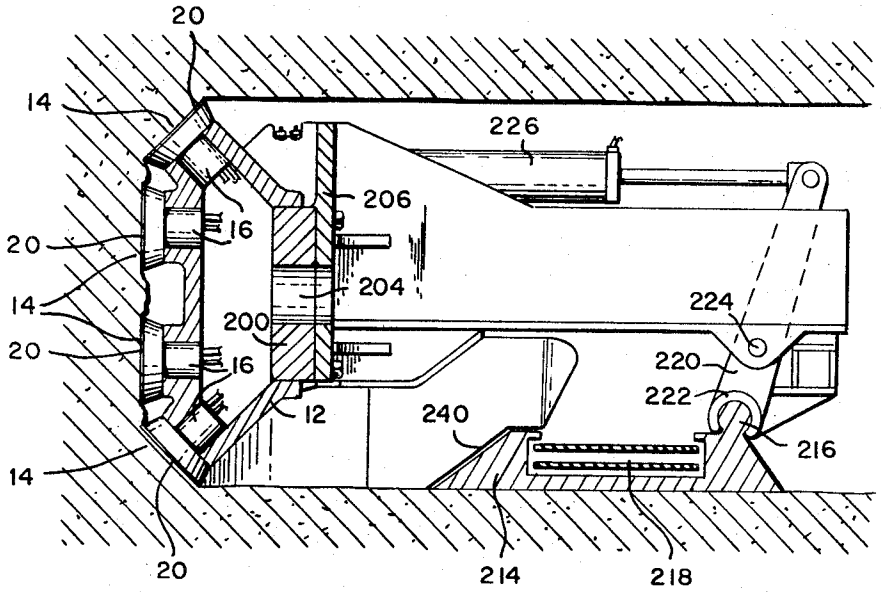


FIG. 1 FIG. 2

FIG. 3



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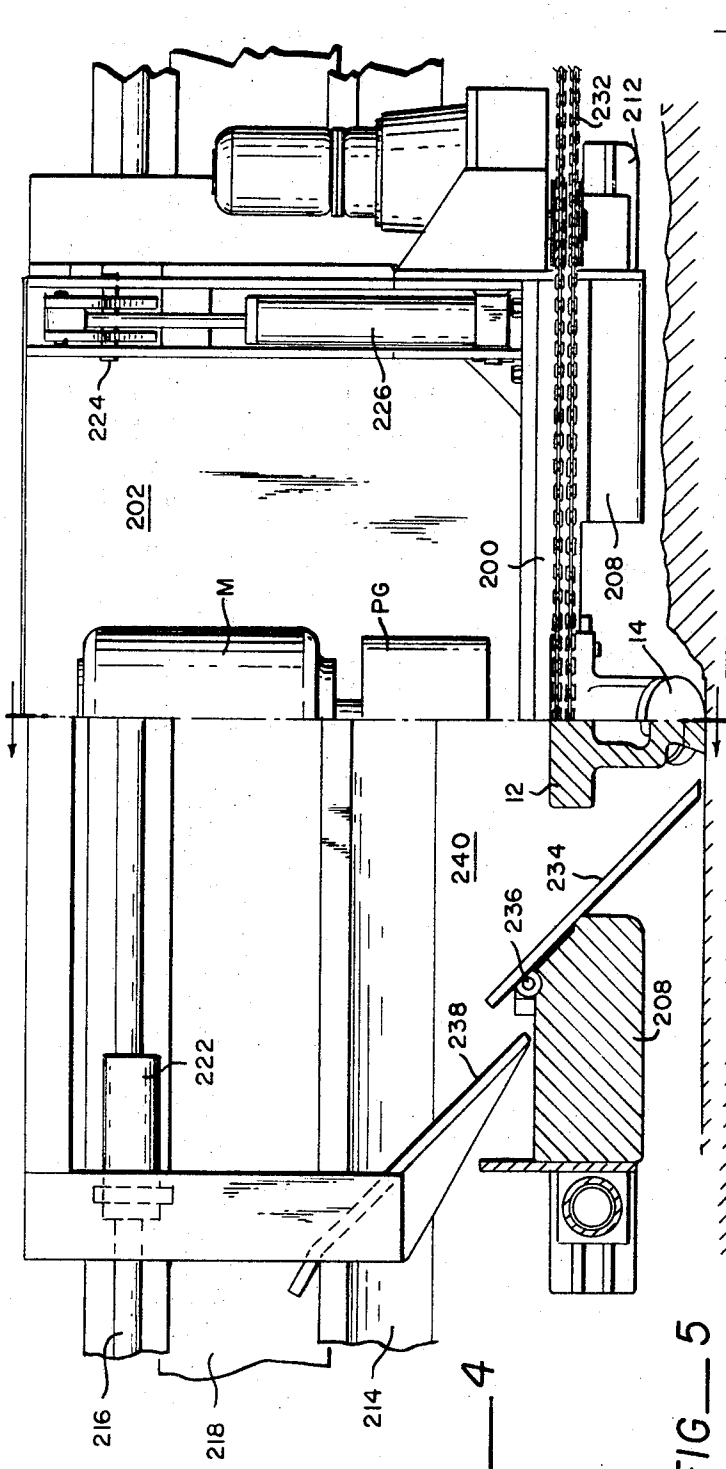


FIG. 4

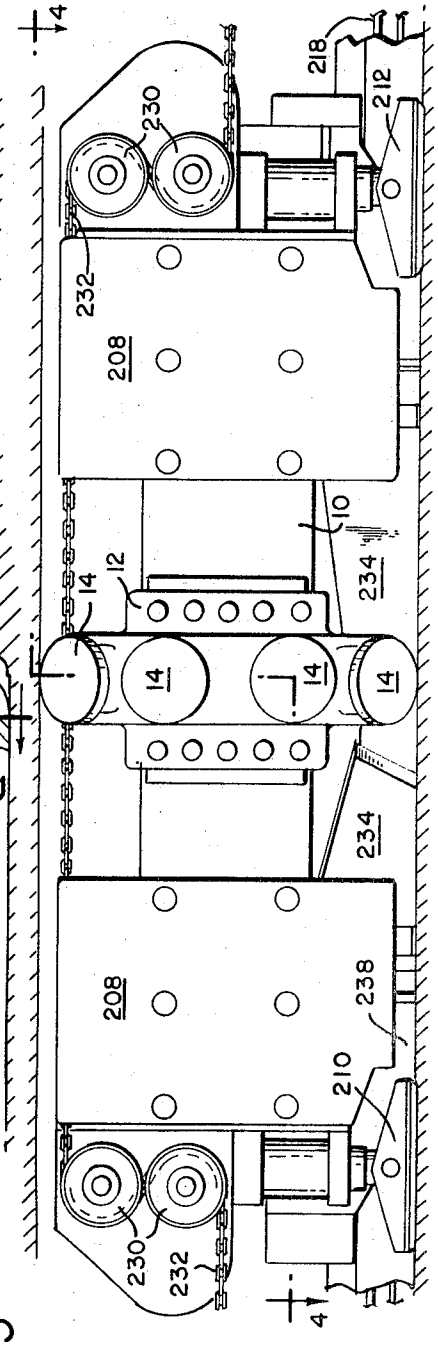
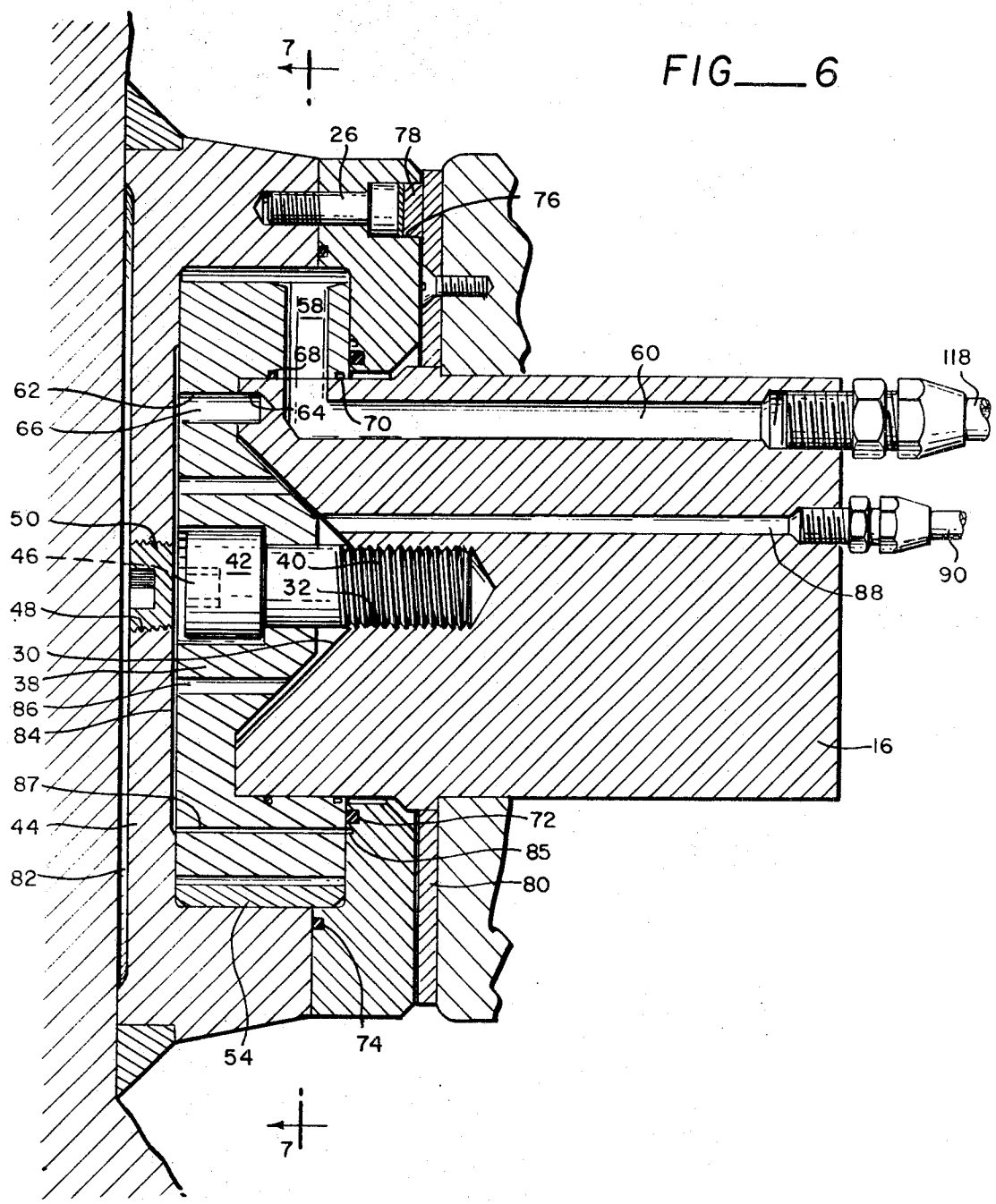


FIG. 5

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FIG. 6



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FIG. 7

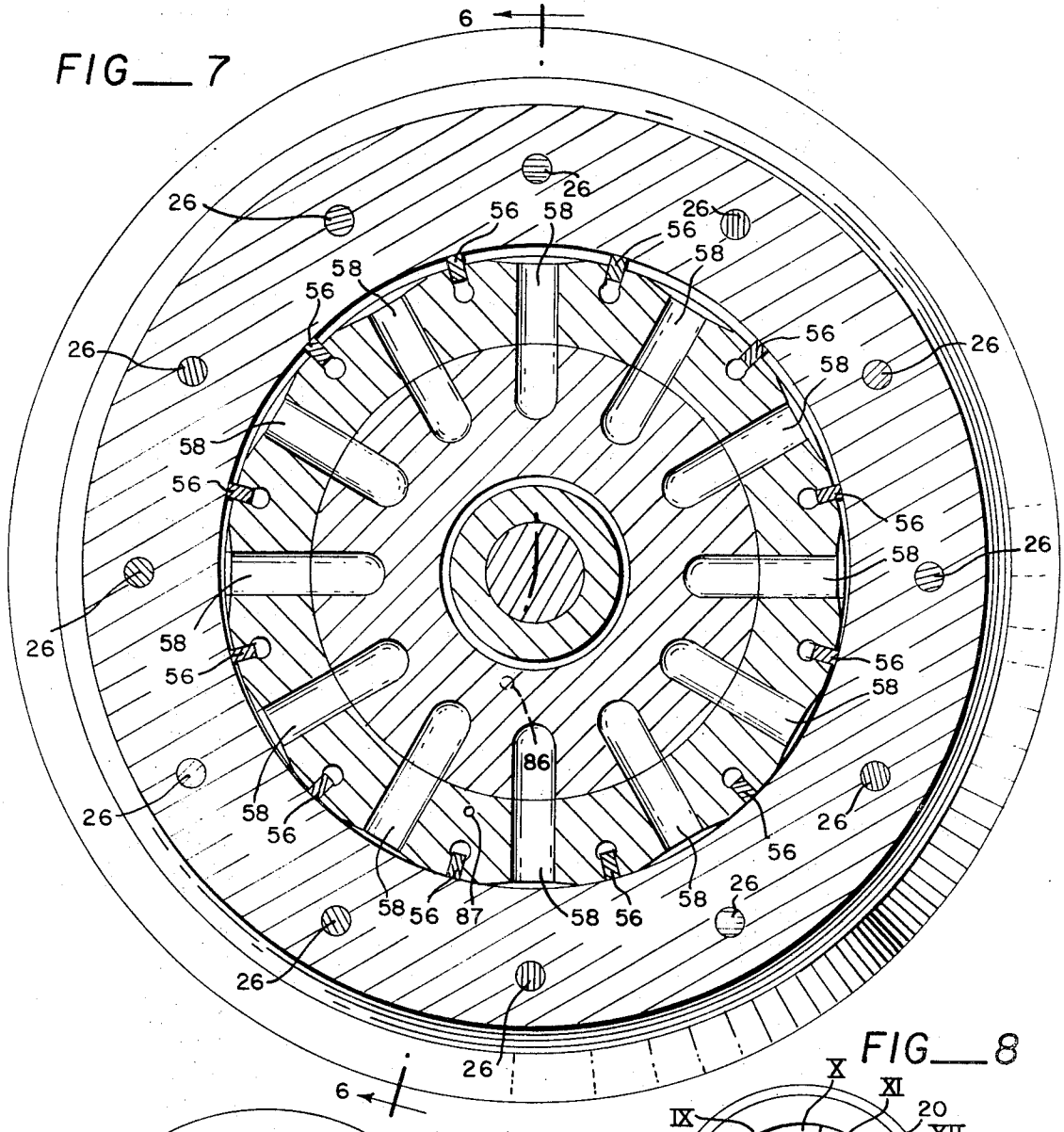
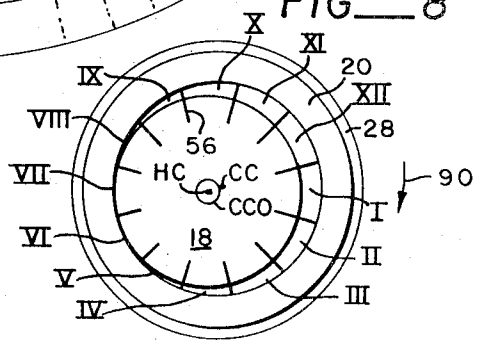


FIG. 8



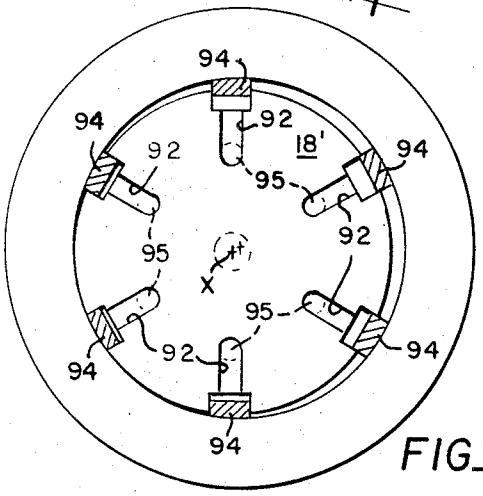


FIG. 9

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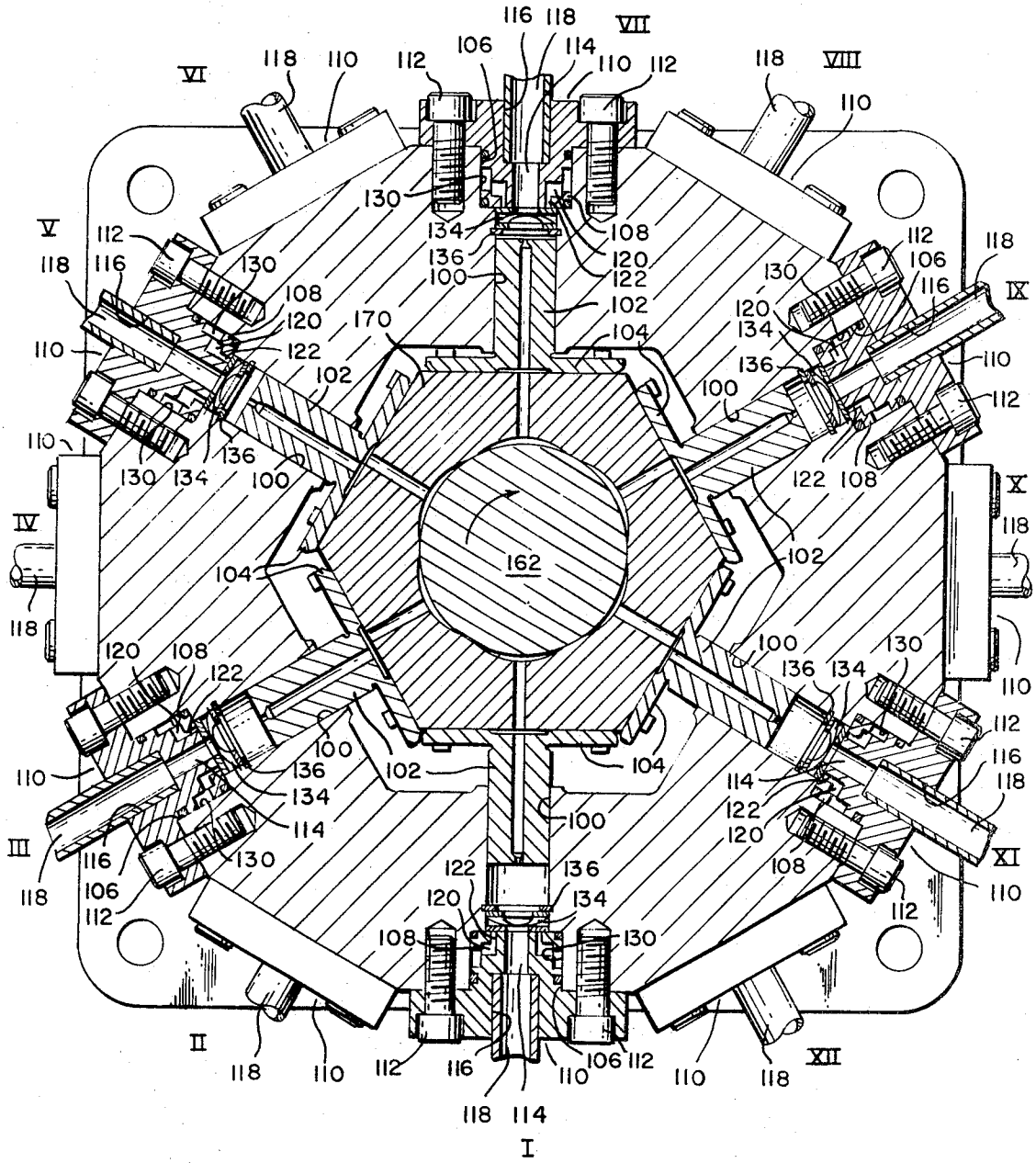
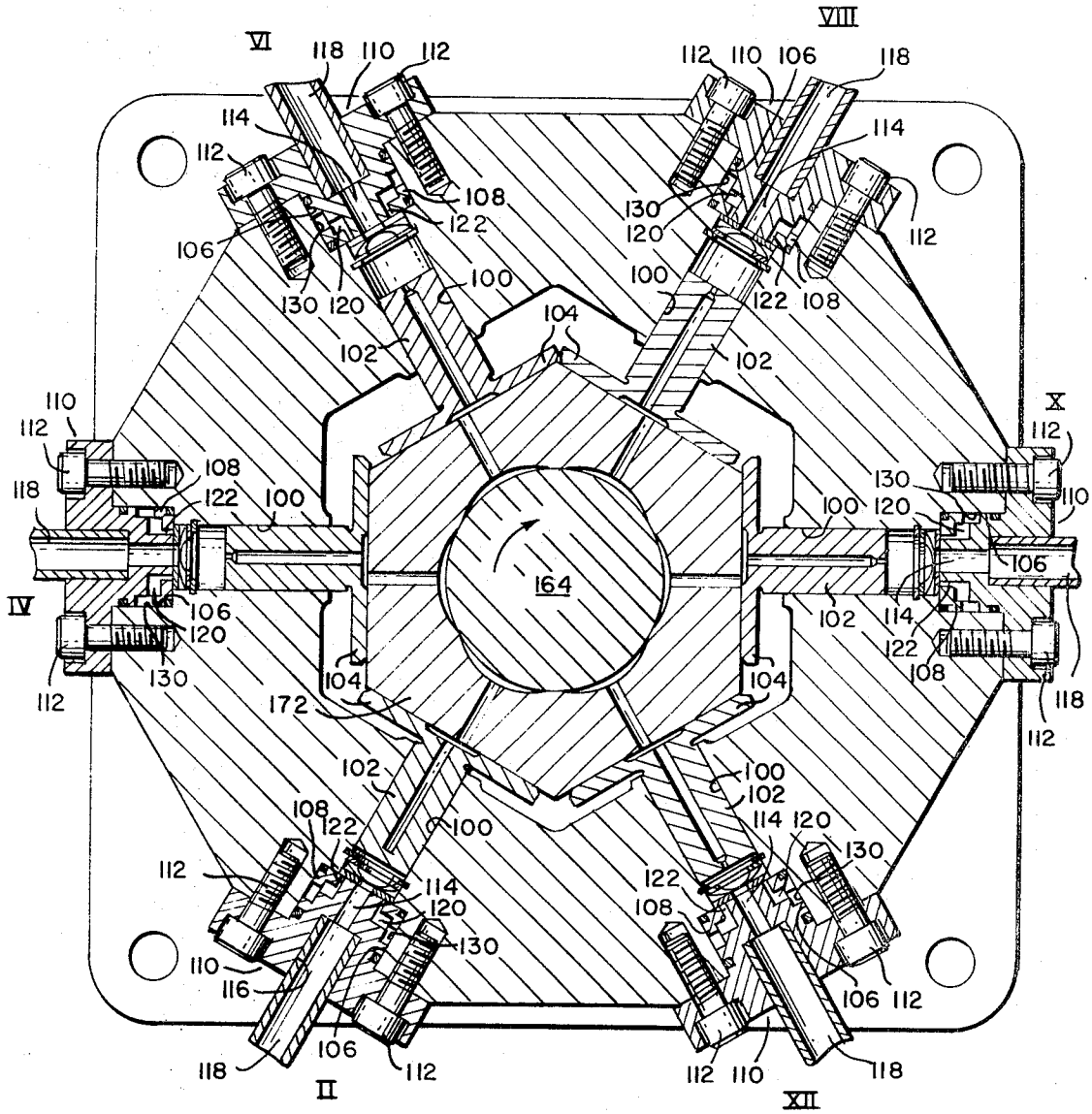


FIG. 10

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FIG. 11



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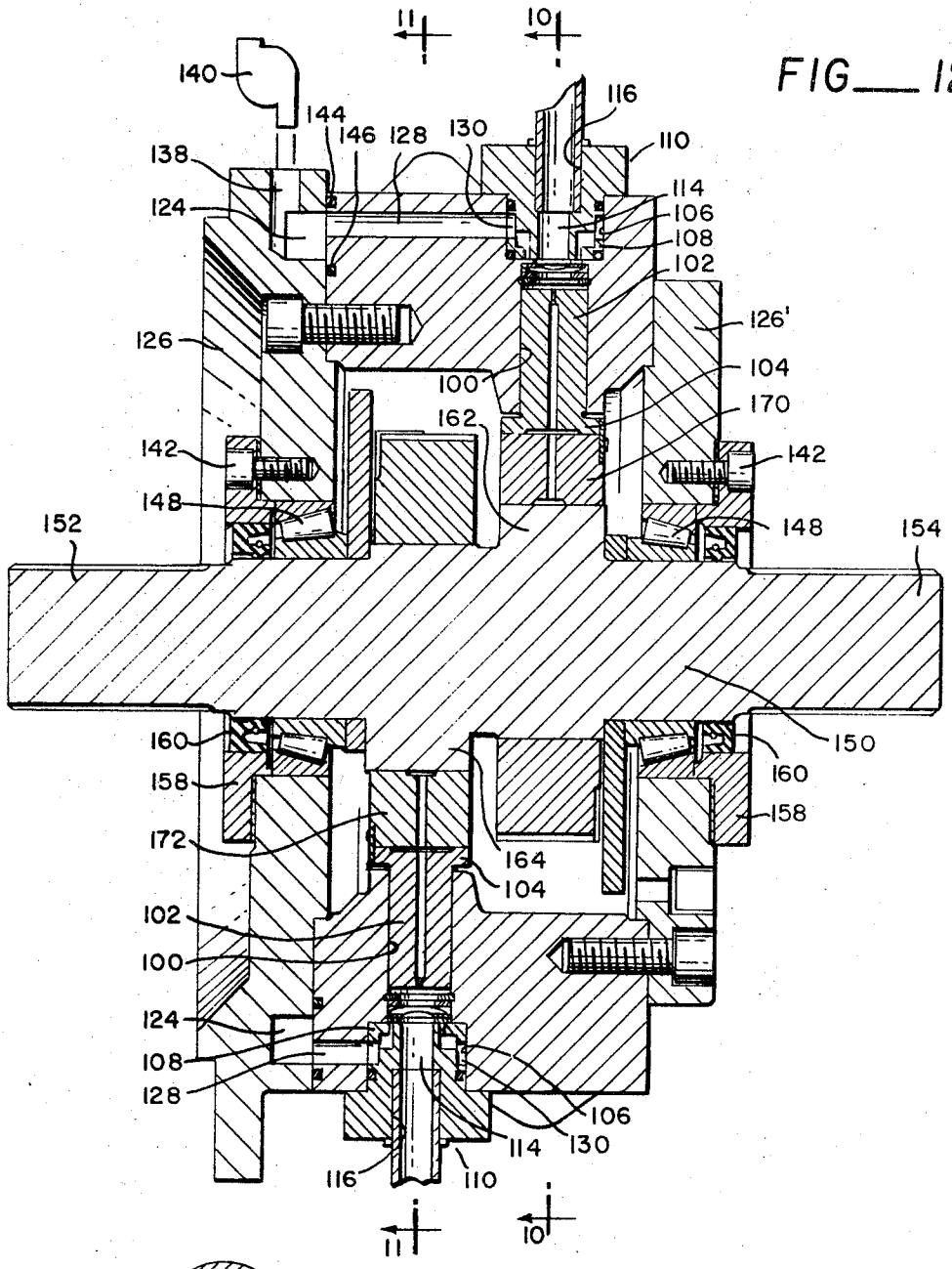


FIG. 12

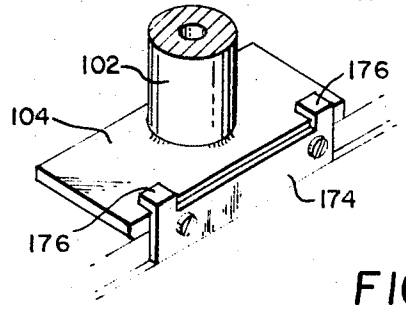


FIG. 13

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VIBRATOR SYSTEMS

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to an orbiting radial force pulse type vibrator, to specific fluid pulsator systems (including specific pulse generators and specific pulse receivers) for creating the force field, and to rock cutting type utilization mechanisms.

2. Description of the Prior Art

Others have proposed making rock cutters vibrate for the purpose of enhancing their cutting action. However, no earth mining machine is known which comprises small mass cutter wheels which normally orbit and a large mass backup structure which orbits whenever cutting wheel movement is restrained by the material being cut, and in so doing establishes high order inertial forces which are transmitted through the stalled cutter(s) to the earth material.

U.S. Pat. No. 2,659,585, issued Nov. 17, 1963 to Rober A. McCallum discloses a technique of digging or breaking earth material by use of a rotary digging head which contains an out-of-balance weight. Rotation of the assembly results in a simultaneous rotation and vibratory movement of digging elements carried by the head. In this system the vibratory movement is caused solely by the rotating weight, and it is necessary for such weight to be relatively large if its addition is to materially enhance the digging action. The digging wheel and the weight are supported by a relatively husky drive shaft which in turn is supported by several apparently large capacity bearings.

U.S. Pat. No. 3,235,311 relates to mining machines comprising cutter bits which are normally forced into the material which they cut by machine or cutterhead advancement alone, but are adapted to reciprocate in the manner of a percussion bit whenever they encounter above normal resistance from the material being cut.

U.S. Pat. No. 3,193,256, issued July 6, 1965 to George L. Malan, discloses a fluid powered vibrator which includes a ring shaped rotor which orbits about the inside of a cylindrical casing, and is moved by the output of a continuous delivery pump, which output is divided into pressure pulses by a system of valves.

SUMMARY OF THE INVENTION

In basic concept this invention relates to a vibrator type mechanism characterized primarily by an orbiting small mass member supported on a relatively large mass backup structure, which is adapted to itself orbit whenever the small mass member is prevented from moving by a restraining load, and to transmit a high inertial force created by its movement through the small mass member to whatever load is tending to restrain such small mass member against movement.

According to the invention the large mass backup structure includes a mounting member for the small mass member. The small mass member is free for 360° of limited radial movement relative to its mounting member. Means are provided for generating short duration radial force pulses between the members, and for cycling the pulses throughout the 360°, This causes an orbital movement of the small mass member when it is not restrained. However, when the small mass member is loaded to the extent it is at least partially restrained from moving, the cycling force pulses operate to orbit the backup structure.

The vibrator system of this invention has general utility. However, the present principal interest in the system is its adaptability to earth mining operations, and in particular to hard rock cutting. According to one embodiment of the invention the small mass member may take the form of a cutter wheel having a peripheral cutting edge, with the large mass backup structure being an earth mining or boring machine, e.g. the type shown by FIG. 3 of the aforementioned U.S. Pat. No. 3,235,311; the type disclosed by U.S. Pat. No. 3,232,670, issued Feb. 1, 1967 to Richard J. Robbins et al. or the type disclosed by U.S. Pat. No. 3,200,494, issued Nov. 30, 1965 to Robert E. Cannon et al.

Preferably, the rotating force field is established by a pulsator type fluid system in which the orbiting cutter wheel and its mounting member are principal components of the pulse receiver, and such pulse receiver includes a ring of variable volume fluid chambers which receive fluid pressure pulses from a pulse generator having a like number of variable volume chambers, each of which is coupled to one of the chambers of the pulse receiver. In operation, columns of a substantially incompressible fluid are moved through interconnecting back and forth conduits between the variable volume chambers of the generator and the corresponding variable volume chambers of the receiver, primarily by an energy input into the generator. However, one principal advantage of the pulsator type system in a rock cutting mechanism is that energy transmitted to rock which does not break but rather reacts elastically is transmitted by the rock back into the system for moving the fluid columns back to the generator.

BRIEF DESCRIPTION OF THE DRAWING

Reference is now made to the accompanying illustrations of certain embodiments of the invention, wherein like numerals refer to like parts, and wherein:

FIG. 1 is a diagram depicting a large backup mass still and a small mass orbiting;

FIG. 2 is a diagram similar to FIG. 1 but depicting the small mass restrained and the large mass orbiting;

FIG. 3 is a transverse sectional view taken through the mid-portion of a rock planing machine typifying one aspect of the invention, the section being taken substantially along line 3–3 of FIG. 4;

FIG. 4 is a top plan view of the planing machine of FIG. 3, showing a portion of the machine in section, such view being taken substantially from the aspect indicated by line 4–4 of FIg. 5;

FIG. 5 is a side elevational view of the mining machine looking toward the cutterhead side thereof;

FIG. 6 is an axial sectional view of one of the cutter assemblies, taken substantially along the line 6–6 of FIG. 7;

FIG. 7 is a cross-sectional view taken through the cutter assembly, substantially along line 7–7 of FIG. 6;

FIG. 8 is a diagram showing the fixed true center of the mounting head in relationship to the circular path of travel of the true center of the cutter wheel;

FIG. 9 is a view similar to FIG. 7 but of a modified form vibrator;

FIG. 10 is a cross-sectional view of a pressure pulse generator typifying another aspect of the invention, taken substantially along line 10–10 of FIG. 12;

FIG. 11 is a cross-sectional view of the pressure generator, taken substantially along line 11–11 of FIG. 12;

FIG. 12 is an axial sectional view, taken substantially along line 12–12 of FIG. 10; and

FIG. 13 is a fragmentary perspective view taken in the region of contact between the base portion of one of the pistons and the associated flat peripheral surface of the piston operator, showing the guide element which prevents rotation of the piston.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Herein the term "orbiting" means the center of the orbiting member travels a circular path with little or no actual rotation of the member about its center. FIG. 1 diagrammatically shows a small mass member m (e.g. a cutter wheel) orbiting relative to a stationary large mass backup structure M. FIG. 2 diagrammatically shows the small mass member m still (as when restrained) and the large mass member M orbiting. In each case the inertial force f, F is a product of the mass m, M and the angular acceleration, $\Omega^2 a$. Since the angular acceleration is constant (because the displacement is constant) the ratio of the inertial forces equals the ratio of the masses. Thus, for example, if the large mass M is one thousand times as great as the small mass m, the inertia force F of the large mass M,

when the small mass in entirely still, is one thousand times the inertia force f of the small mass m. Accordingly, when the small mass m tends to stall the large mass M begins to work, and the greater the tendency to stall the greater the contribution of the large mass M.

The planar shown by FIGS. 3—5 is especially adapted for planing operations of the type described in U.S. Pat. No. 3,310,346, issued Mar. 21, 1967 to Willie Heyer. It is shown to comprise a main frame **10** which carries a side placed cutterhead **12** intermediate its length. By way of typical and therefore nonlimitive example, the cutterhead **12** is shown to carry four cutter assemblies **14**. The intermediate pair of cutter assemblies **14** are vertically oriented, and the upper and lower cutter assemblies **14** are diagonally oriented so that they cut on about a 45° angle extending between the vertical side wall being cut and the roof and floor of the room being formed.

Referring to FIGS. 6 and 7, the mounting means for each cutterhead includes a short stub shaft **16** which is rigidly attached to the cutterhead **12**. Each cutter assembly **14** includes a mounting head **18** which is attached to the end of its stub shaft **16**, in a manner to be hereinafter described in detail, and a cutter wheel **20**.

In the illustrated embodiment each cutter wheel **20** comprises a major part **22** and a minor part **24**, both of which are formed to include a cuplike recess in its axial inner portion. The minor part **24** includes a central opening for receiving the outer end portion of the stub shaft **16**. The cuplike recess of the major part **22** and the cuplike recess of the minor part **24** together form an inner cavity for receiving the mounting head **18**. The two parts **22, 24** may be secured together about the head by means of a circular array of cap screws **26** which in the illustrated embodiment are shown to extend through and be countersunk in the minor part **24**, and to anchor in the major part **22**.

The cutting edge may be formed on the major part **22**, but preferably is formed on a separate annular cutting ring **28** which is firmly secured onto a peripheral portion of the cutting wheel **20**. This latter construction makes it possible to fabricate the cutting edge from a harder and usually more expensive material (e.g. tungsten carbide) than the rest of the cutter wheel. It also makes it possible to replace the cutting ring when it wears out and there is still life left in the cutter wheel proper.

In preferred form a conical socket **30** is formed in the outer end of shaft **16**, and a tapped bore extends axially inwardly through the shaft **16** beyond the bottom of the socket **30**. The mounting head comprises an annular peripheral portion **34**, a radially extending outer wall **36**, and an axially inwardly, generally conical boss **38** sized to relatively closely fit within the socket **30**. As clearly shown by FIG. **6**, a side recess is formed in the mounting head **18** radially between the peripheral portion **34** and the boss **38** and is sized to snugly receive and accommodate the outer end portion of the shaft **16**. A slight clearance is allowed between the conical wall of the boss **38** and the conical wall of the recess **30**, and such clearance forms a part of a fluid path which will hereinafter be described.

The mounting head **18** is secured to the shaft **16** by a single cap screw **40** which extends through the center of boss **38** and threads into the tapped bore **32**. Cap screw **40** is countersunk into the mounting head **18** so that its head portion **42** does not project outwardly beyond the outer radial surface of the mounting head **18**. The circular end wall **44** of the cutter wheel includes a tapped opening at its center, coaxial with the wrench socket **46** formed in the head **42** of the cap screw **40**. Opening **48** is provided with a flush cap **50** which includes a wrench socket **52**. By way of typical and therefore nonlimitive example, the wrench sockets **46, 52** may be hexagonal in shape and operable by a hex key wrench.

An annular insert **54** made from a hard bearing material is fitted into the inner cavity of the cutting wheel **20** to form a liner therefor. This also facilitates manufacturing a square corner between the straight side wall and the cylindrical peripheral wall of the inner chamber.

As clearly shown by FIG. **6**, the axial length of the mounting head is approximately equal to the axial dimension of the inner cavity of the cutter wheel **20**, i.e. close clearances exist between the radial surfaces of the mounting head and the radial surfaces of the inner cavity. However, a gap (e.g. 0.050 inches) is intentionally provided radially between the mounting head **18** and the surrounding cylindrical surface of the cutter wheel. This gap is divided into a plurality of pulse chambers by means of a plurality of radially movable seals, shown in the form of radially reciprocating vanes set within radial slots or pockets formed in the mounting head **18**. Each pulse chamber is completely closed except for a single induction-eduction port (i.e. a bidirectional flow port), and whatever leakage that may occur.

In the illustrated embodiment a separate bidirectional flow passageway extends to each port **58**, first axially and then radially through the fixed shaft **16**.

Since the mounting head **18** is attached to the shaft **16** by a single cap screw **40**, alineable openings **62, 64** are provided in the mounting head **18** and the shaft **16**, respectively for receiving an alinement pin **66**. When the openings **62, 64** are in alinement the radial ports **58** are in correct alinement with the radially extending outer portions of the passageways **60**.

The cutter assembly may be assembled in the following manner. Starting with the cutter wheel **20** apart, the bearing ring **54** is inserted into the major part **22**. The cap screw **40** is inserted into the mounting head **18**, and then the mounting head is inserted into the outer part **22**. Next the seals **56**, the alinement pin **66** and the O-rings **68, 70** are installed. Then the O-rings **72, 74** are set in place in grooves formed for them in the minor part **24**, and the minor part **24** is set into place and secured to the major part **22**, by means of the cap screws **26**. The heads of the cap screws **26** are countersunk into the minor part **24** a sufficient distance to provide room for an annular groove. An annular dirt seal, shown in the form of an annular metal backed resilient member, with a circumferentially grooved exposed surface, is press fitted into the channel.

As should now be evident, the mounting head **18**, the cutter wheel **20** and the dirt seal **78** all together constitute a single unit which is attachable to the fixed shaft **16** by means of a single fastener, i.e., the cap screw **40**. It is installed by merely pointing the inner end of the cap screw **40** into the conical socket **30**, turning the unit somewhat, if necessary, to aline the projecting end portion of pin **66** with the opening or socket **64**. Next a hex key wrench is inserted through the access opening **48** and used to tighten the cap screw. The plug **50** is then installed in opening **48**.

Preferably, an annular wear collar **80** is attached to the cutterhead **12** in the region thereof surrounding each shaft **16**, and the projecting ridges of the dirt seal **78** contact the outer surface of the member **80**.

A recess is formed in both the outer and inner sides of the cutter wheel wall **44**. The outer recess **82** merely provides a clearance where contact between the cutter wheel **20** and the rock or other material being cut is not wanted. The inner recess **84** is provided for generally the same purpose, i.e. to provide clearance where contact between the cutter wheel wall **44** and the mounting head **18** is undesirable. In addition, clearance **84** serves to collect fluid that has leaked from the pulse chambers and directs it to axial ports **86** which communicate recess **84** with the clearance which exists between boss **38** and the wall of socket **30**. An annular groove **85** in the radial wall portion of part **24** serves to collect leakage on the opposite side of the head **18**, and an axial bore **87** through head **18** directs such leakage to the recess **84**. The fluid leaves this latter space via a collection passageway **88** extending axially through the shaft **16** to a fluid conduit **90** provided for carrying such fluid away.

For clarity of illustration, in FIG. **8** only one seal **56** has been referenced and the bidirectional flow passageways have been omitted. The pulse chambers have been numbered I—XII and an arrow **90** is provided to show the direction of pulse travel or "phase rotation". In operation of the cutter, fluid pressure pulses are rapidly distributed about the gap in

cyclic fashion, from pulse chamber to pulse chamber. This rapid stepping of the pressure pulses about the gap tends to cause an orbital shifting movement of the cutter wheel **20** relative to the head **16**, and such movement of the cutter wheel **20** actually occurs unless the cutter wheel **20** is restrained against movement by the substance into which the cutter wheel **20** is being moved. In FIG. **8** the head center is denoted HC and the orbital path of the true center of the cutter wheel is designated CCO. In the condition illustrated pulse chamber VII is substantially closed and pulse chamber I is at maximum volume. An instant later pulse chamber II will be at maximum volume, pulse chambers II—VII will increase somewhat in volume and pulse chambers VIII—I will decrease somewhat in volume, and the true center CC of the cutter wheel will shift in position somewhat on its orbit CCO.

As will be evident, when the cutter wheel **20** is restrained from movement by the material into which it is being advanced, the mounting head **12** tends to shift rather than the cutter wheel **20**. The forces tending to cause orbital shifting movement of the mounting head **12** are transmitted through the cutterhead **12** to the frame, which as will be later described in some detail is intentionally constructed to contain considerable mass. The fluid forces transmitted through the mounting head **18** cause the frame to orbit, and as it moves a relatively high order inertial force is transmitted by it back through the mounting head **18** to the cutter wheel **20** and from the cutter wheel **20** to the rock. The fluid system is regenerative during the periods the rock is acting elastically, in that the rock transfers the energy it receives back into the system. During operation of the cutter the energy input into the hydraulic system equals the energy withdrawn plus losses. During the period of rock elasticity very little energy is withdrawn. Hence, the energy input is relatively low until the rock actually breaks, at which time the strain energy of the rock is lost and an equivalent amount of energy is withdrawn from the system.

Owing to the geometry of the system the force experienced by the cutter is determinate and can be limited to a finite maximum. Further, the fluid, though basically incompressible, provides a considerable degree of cushioning under shock loading. If a mechanical drive system were employed, there would essentially be an unlimited upper limit on the force experienced by a cutter when shock loaded. In designing the system of this invention, a maximum force may be chosen which is below the fracturing force of many hard and brittle materials, such as tungsten carbide, for example. Hence, the controllability of the maximum loading on the cutter makes it possible to successfully employ these materials for the cutting parts.

The four cutter assemblies are driven out of phase with each other, i.e. in a manner such that there is a phase difference between the pulses in each cutter, so that the combined resultant of the restraining force on all the cutters is kept at a minimum. This results in the largest possible force being transmitted to the working cutter by the smallest possible backup structure, since the mass of the backup structure is not being divided equally between all of the cutters but rather is being concentrated on only one of the cutters at a time.

In the fluid pulse receiving mechanism of this invention the sealing problem is not particularly serious, because there is provided a small phase difference between pressure pulses in adjacent pulse chambers I—XII, and as a result the pressure differential between adjacent pulse chambers is low. This feature makes it possible to employ higher maximum pressures in the receiver than would be possible if the pressure differential were greater.

FIG. **9** shows a modified construction of the pressure receiver, usable, for example, as a concrete form vibrator. It is shown to comprise a mounting head **18'** including a plurality of radially extending piston chambers **92**, each containing a piston **94**. Pressure pulses are cycled through bidirectional flow passageways **95** into the spaces radially inwardly of the pistons **94** in the same manner the pulsing is done in the pulse receiver of FIGS. **6** and **7**. As will be evident, movement of the pistons will normally cause the small mass wheel **96** to orbit about the true center X of the mounting member **18'**.

A preferred form of pulse generator is shown by FIGS. **10—12** to comprise a main body in the nature of an open centered block **98** having formed therein two axially spaced banks of radial piston chambers, each preferably in the form of a cylindrical bore **100**. Each chamber **100** is equipped with a piston having a cylindrical portion **102** snugly received in its bore or chamber **100**, and a flanged base **104**. The pistons **102** are fitted into the bores **100** from the space within the body. A countersink **106** is provided in the body in the region of the outlet from each piston chamber **100**. A tubular inner portion **108** of an inlet-outlet fitting **110** is received in the countersink **106**, and the fitting **110** is secured to the body by means of a pair of cap screws **112**. A small bore **114** extends axially through the inner end portion of the fitting and a slightly larger diameter bore **116** continues therefrom and extends through the remaining portion of the fitting. The end of a conduit **118** which communicates the piston chamber with its associated pulse chamber in the pulse generator fits into each outer bore **116**. An annular chamber **120** is formed in the inner portion of the fitting about the inner bore **114**, and a relatively narrow gap passageway **122** extends inwardly from each said chamber **120** and terminates in an annular outlet which is generally flush with the inner end of the fitting, and is bordered on each side by a flat end surface of the fitting.

Referring now to FIG. **12**, an annular manifold **124** is formed in the left (as pictured) end plate **126**. A fluid delivery port, shown in the nature of a bore **128** formed axially through the radially outer portion of the body, communicates the annular manifold with an annular passageway **130** surrounding the inner end portion of the fitting, and such annular passageway **130** is in communication with the annular chamber **120**.

The annular outlet from each passageway **122** is normally closed by a flat washer valve **132** which is normally biased outwardly by a washer like spring **134**. The spring **134** is sandwiched between the movable washer valve **132** and a fixed washer **136** which is firmly anchored to the body **98** near the outlet of its piston chamber. As shown, the spring **134** is in the nature of a bowed washer. It is resilient enough that it will flatten out when subjected to enough pressure by the washer valve, permitting the valve to move inwardly and unblock the annular passageway. This type of valve is quick to respond. It has to be quick because the time period for delivery of makeup fluid is very short.

The annular manifold **124** in end plate **126** receives makeup fluid via an inlet port **138**, from a booster pump **140** shown only schematically in FIG. **12**.

The two end plates **126**, **126'** are secured to the body **98** each by means of a circular array of cap screws **142** or the like. O-rings **144**, **146** are inset into the body on the radial inner and outer sides of the annular manifold **124**. The end plates **126**, **126'** have open centers for receiving roller bearings **148** or the like which surroundingly engage and support for rotation a drive shaft **150**. The drive shaft **150** is shown to include two outwardly extending, splined end portions **152**, **154**, one of which is connected to an electric motor or other prime mover (not shown). The opposite end of the shaft may be secured to one end of a similar pulse generator, so that both of the generators can be powered by a single drive mechanism. In the usual fashion, a seal assembly **158**, including an annular seal member **160** which surroundingly engages the shaft **150** immediately outwardly of the bearing, is secured to each end plate **126**, **126'**.

Interiorly of the body the shaft includes a pair of circular cams **162**, **164** which are eccentrically related to the shaft proper. As FIG. **12** and a comparison of FIGS. **10** and **11** will show, the cams **162**, **164** are offset from each other generally on opposite sides of the shaft, so that the radial loadings on the shaft during operation are as close to being balanced as is possible. In the illustrated embodiment each bank of pistons

comprises six pistons. The twelve pistons are phased thirty degrees (30°) apart and are operated in the sequence depicted in FIG. 11, using for each piston the number of its corresponding pulse chamber in FIG. 8.

Referring to FIG. 12, a plate type counterweight 166, 168 is positioned axially outwardly of each cam 163, 164 and extends radially outwardly from the shaft 150 on the side thereof opposite the cam. The weights 166, 168 are sized and weighted appropriately to substantially counterbalance the centrifugal forces generated by the rotating and reciprocating parts.

As clearly shown by FIGS. 10 and 11, the inner cavity of the body 98 is in the shape of a polygon and has the same number of sides as there are pistons. In the illustrated embodiment the cavity for each bank of pistons is hexagonal. An operator member 170, 172 which has the same cross-sectional shape as the cavity, but diametrically is somewhat smaller, snugly surrounds each cam 162, 164 and is moved thereby. The flat inner end surfaces of the pistons abut the flat peripheral surfaces of the operators 170, 172. The cavities and the shaft 150 have the same center, but the center of the cams 162, 164 is offset from the center of rotation. As a result, the cams 162, 164 shift the operators 170, 172 around the periphery of the cavity as the shaft 150 rotates. As the operators shift they sequentially displace the pistons outwardly and permit sequential return of the pistons.

Guide clips 174 (FIG. 13) may be secured to the side surfaces of the operators closely adjacent the bases of the pistons, such clips 174 each including a pair of guide fingers 176 which loosely project over outer surface portions of the flanged bases. These clips 174 prevent unwanted rotation of the pistons but at the same time permit the relative sliding motion between the piston bases 104 and the operators 170, 172 which wants to occur while the operator is shifting. The fingers 176 also help to return the pistons during the phases the operator movement is radially inwardly.

As illustrated in FIGS. 10 and 11, cavities are provided in the operators 170, 172 between them and the cams 162, 164, and in the inner end surfaces of the pistons, between them and the operators 170, 172. These cavities are in constant communication with the pressure chambers, via meter orifice containing ports 180, 182, so that a balancing fluid cushion exists where linear sliding occurs between the pistons and the operators 170, 172, and where rotational movement occurs between the cams 162, 164 and the operators 170, 172.

Referring now again to FIGS. 3—5, the cutterhead 12 is shown to be bolted to a husky side beam 200 constituting part of the main frame 10. The pressure pulse generators PG and their drive motors M are mounted on a deck 202, laterally inwardly of the cutter head 12. As shown by FIG. 3, the cutterhead 12 is hollow, and the fluid conduits 90, 118 fan out within such hollow interior and lead outwardly to the ports 60 in the fixed shafts 16. From within the cutterhead 12 the conduits extend inwardly of the machine through a transverse opening 204 extending through both the side beam 200 and a side plate portion 206 of the main frame 10 to connections on the pulse generators.

Large and heavy pieces of steel 208 are secured to the main frame 10 both forwardly and rearwardly of the cutterhead 12 to increase the mass of the machine to the required amount.

Adjustable supporting shoes 210, 212 are supported on the cutterhead side of the machine, near the front and rear ends of the machine, respectively, and generally in line with the heavy weights. The shoes 210, 212 are pivotally attached to the lower ends of pistons which project downwardly from fluid operated adjustment rams, the cylinders of which are rigidly secured to the main frame 10.

The side of the machine opposite the cutterhead 12 travels over a floor beam 214 including a guide rail 216 and a power conveyor 218 of the type disclosed by U.S. Pat. No. 2,745,651, and U.S. Pat. No. 2,691,514, for example. The machine is guided during movement by a pair of lever arms 220, one at the front and one at the rear of the machine, having claws 222 at their lower ends which engage the floor rail 216. The levers 220 are pivoted intermediate their ends at 224 to the frame 12, and are connected at their upper ends to horizontally disposed fluid actuators 226. By virtue of this arrangement the horizontal distance between the claws and the side wall being mined can be adjusted by merely varying the stroke of the actuators 226, and the ground rail 216 does not have to be moved after each cut taken by the machine. Further, the actuators 226 can be moved differentially so that the lead angle, or angle of cut, can be adjusted. The machine may be moved longitudinally by means of traction wheels 230 turning on a traction chain 232 that is firmly anchored at each of its ends. The traction wheels 230 are reversible so that the machine can be moved in either direction.

A deflector-scraper member 234 is hinge connected to the frame on each side of the cutterhead within the line of travel. These members 234 are hinge connected to the frame at 236, near the ends thereof which are farthest from the cutterhead. As a result, when the machine is moved in one direction the leading member 234 automatically closes when contacted by the floor material and the trailing member 234 automatically opens and functions to direct the cuttings on the floor rearwardly to a fixed inner deflector 238 which in turn directs the cuttings up a ramplike apron 240 and then off the apron onto the conveyor belt.

I claim:

1. A vibrator system comprising: a relatively large backup mass; a mounting member rigidly connected to said backup mass; a relatively small mass member, with one of said members surrounding the other, and with the small mass member being free for 360° of limited radial movement relative to said mounting member unless it is restrained against such movement; means for generating short duration radial force pulses between the members comprising a ring of variable volume fluid chambers which are expansible and contractible radially and are closed except for a single fluid passageway for each said chamber, serving as both a fluid supply and exhaust passageway for the chamber; and means for distributing fluid pressure pulses from passageway to passageway and in that manner from chamber to chamber in repetitive cyclic fashion around said ring, to cause cyclic oscillating of the fluid in the passageways and chambers for producing an orbiting radial shifting movement of said small mass member when it is unrestrained, and a counterdirectional orbital movement of said backup mass when said small mass member is restrained, attended by a transfer of the inertial force of the moving backup mass to said relatively small mass member.

2. A vibrator system according to claim 1, wherein a radial gap is formed between said members and said fluid chambers are formed by and between circumferentially spaced seals carried by at least one of said members, with the fluid pulses in said chambers pressing both radially inwardly against the surrounded member and radially outwardly against the surrounding member.

3. A vibrator system according to claim 1, wherein said fluid chambers are radially disposed piston chambers formed in one of said members, and said chambers each contain a piston having a first end forming a wall of the chamber and an opposite end in force transmitting contact with the other member.

4. A vibrator system according to claim 1, wherein said means for distributing fluid pressure pulses from passageway to passageway comprises a pressure generator including means forming a plurality of variable volume chambers, each of which is in communication with a related one of the vibrator chambers via its supply-exhaust passageway, and means for harmoniously expanding and contracting said generator chambers, to cause a harmonious shifting in position of the columns of fluid in said passageways back-and-forth between said generator and vibrator chambers.

5. A vibrator system according to claim 1, wherein said small mass member surrounds said mounting member, and said passageways include first portions extending axially and

second portions extending radially therefrom outwardly to the variable volume chambers of the vibrator.

6. A vibrator system according to claim 1, wherein said small mass member surrounds said mounting member and is exposed.

7. The vibrator system of claim 6, further including a fixed shaft having an opening in the end thereof, wherein said mounting member fits against said end and includes an opening therein alineable with the opening in said fixed shaft, and a removable fastener extends through both of said openings for securing the mounting member to the fixed shaft, and wherein said small mass member is of split construction and includes parts assemblable from opposite sides of the mounting member, and connecter means for connecting said parts together about the mounting member, whereby the mounting member with said small mass member attached thereto may be assembled onto and removed from the fixed shaft as a single unit by manipulation of said fastener.

8. The vibrator system of claim 7, wherein a first portion of each supply-exhaust passageway extends through the fixed shaft and a second portion extends through the mounting member, and wherein said system also includes an alinement pin and an alinement passageway extending partially through the mounting member and partially through the fixed shaft, and oriented so that when the alinement pin is inserted into the two parts thereof the portions of the supply-exhaust passageways in the mounting member are correctly alined with the portions of such passageways in the fixed shaft.

9. A vibrator system according to claim 7, wherein said small mass member includes a wall which is disposed outwardly of and covers the portion of the mounting member through which the fastener extends, and said wall includes an access opening in alinement with said fastener.

10. A vibrator system comprising; a relatively large backup mass; a mounting member rigidly connected to said backup mass; a relatively small mass member surrounding said mounting member, said small mass member including an exposed exterior portion and said member being free for 360° of limited radial movement relative to said mounting member unless it is restrained against such movement by an object in contact with said exposed exterior portion; and means for generating short duration radial force pulses between the members, and for cycling said pulses throughout the 360°, to cause an orbital movement of said small mass member when it is unrestrained, and a counterdirectional orbital movement of said backup mass when said small mass member is restrained, attended by a transfer of the inertial force of the moving backup mass to said relatively small mass member and from said small mass member to the object.

11. A vibrator according to claim 10, wherein said means for generating short duration force pulses comprises a ring of variable volume fluid chambers and means for distributing fluid pressure pulses from chamber to chamber in repetitive cyclic fashion around said ring.

12. A vibrator according to claim 11, wherein a radial gap is formed between said members and said fluid chambers are formed by and between circumferentially spaced seals carried by at least one of said members, with the fluid pulses in said chambers pressing both radially inwardly against the surrounded member and radially outwardly against the surrounding member.

13. A vibrator according to claim 11, wherein said fluid chambers are radially disposed piston chambers formed in one of said members, and said chambers each contain a piston having a first end forming a wall of the chamber and an opposite end in force transmitting contact with the other member.