United States Patent [19]

Cobb et al.

[11] **3,770,322**

[45] **Nov. 6, 1973**

[54] **APPARATUS FOR FRACTURE OF MATERIAL IN SITU WITH STORED INERTIAL ENERGY**

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[73] Assignee: **Caterpillar Tractor Co.,** Peoria, Ill.

[22] Filed: **Apr. 12, 1971**

[21] Appl. No.: **133,262**

[52] **U.S. Cl.** **299/37,** 37/DIG. 18, 172/40, 299/14

[51] **Int. Cl.** **A01b 35/00**

[58] **Field of Search** .... 172/40; 37/DIG. 18, 37/141 R, 141 T; 299/14, 37, 67; 198/10; 173/122

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Primary Examiner—Ernest R. Purser
Attorney—Charles M. Fryer et al.

[57] **ABSTRACT**

An earth working apparatus employs large amounts of stored energy which is cyclically delivered on demand to the work tool. The energy is stored in a large flywheel and delivered by suitable transmission means to the work tool.

41 Claims, 29 Drawing Figures

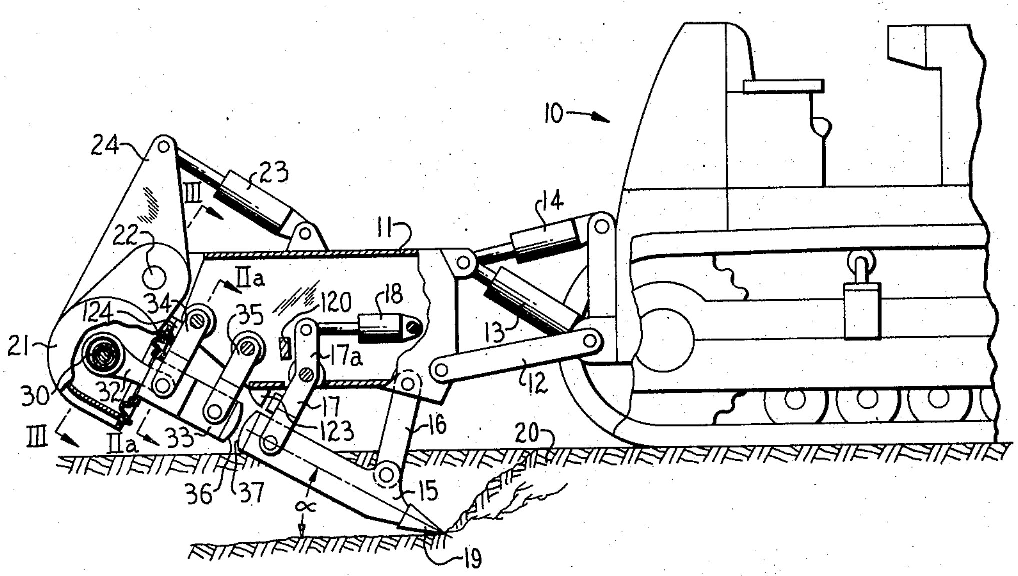
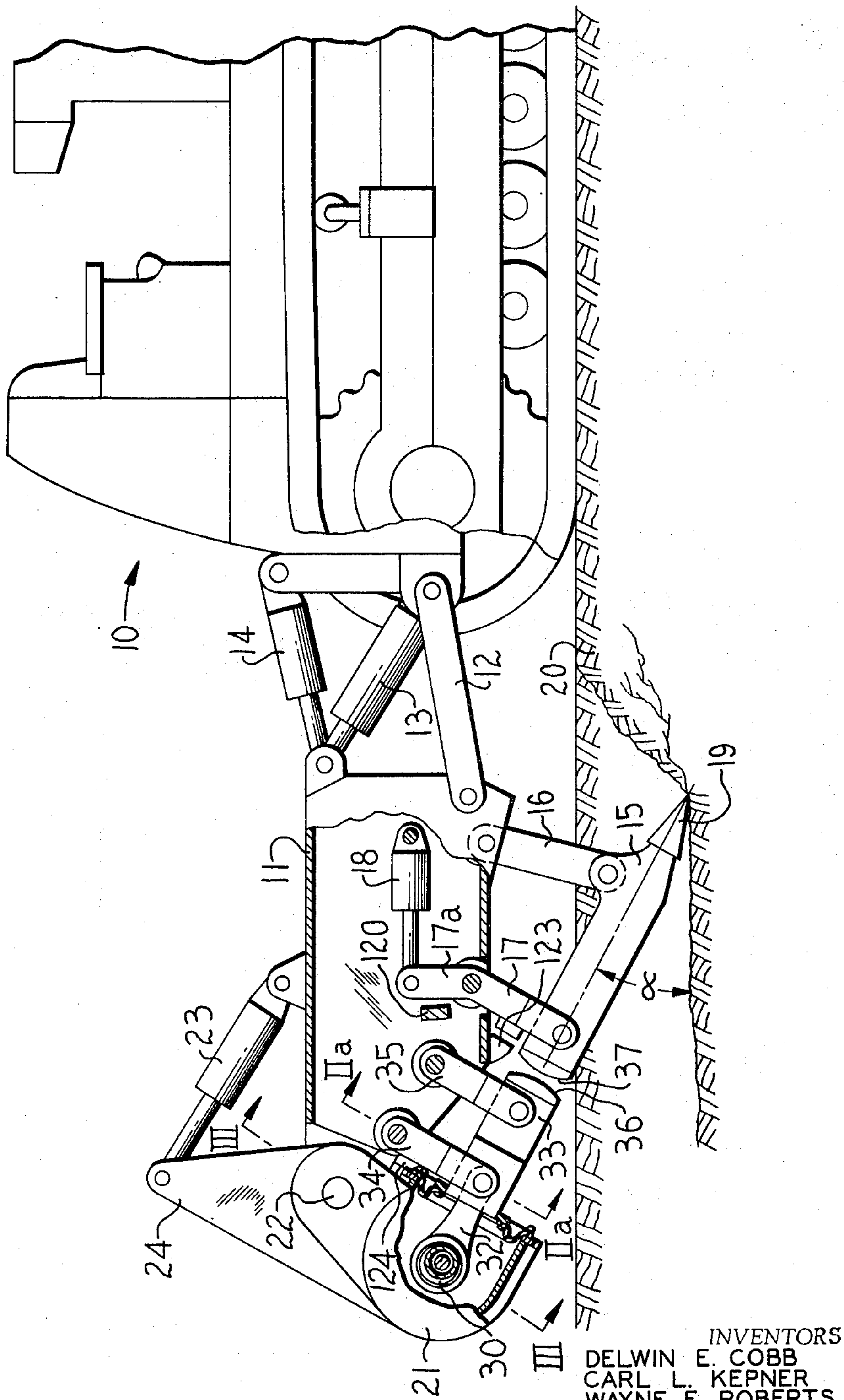


FIG. 1



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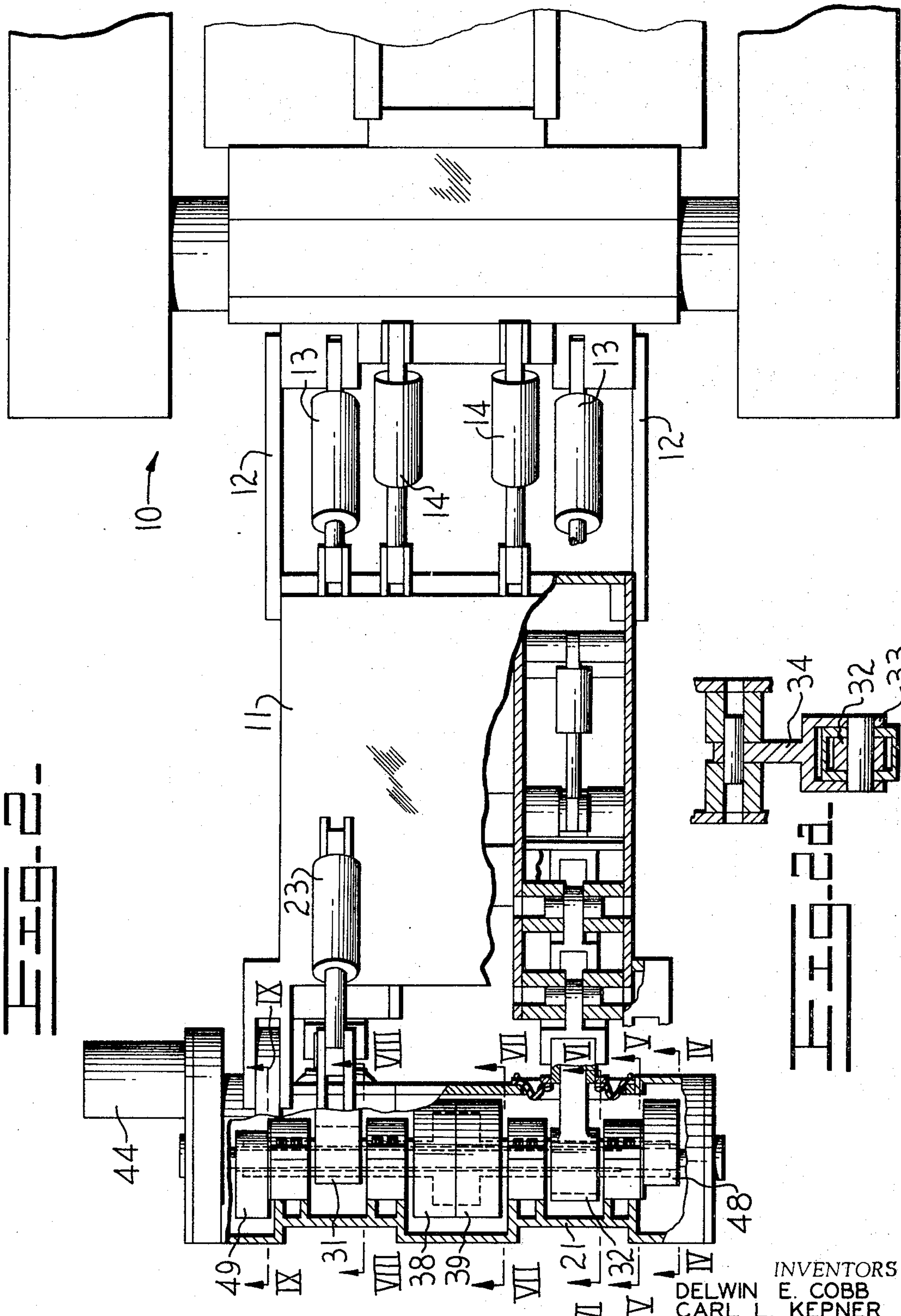

FIG. 2

FIG. 2a

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FIG. 3

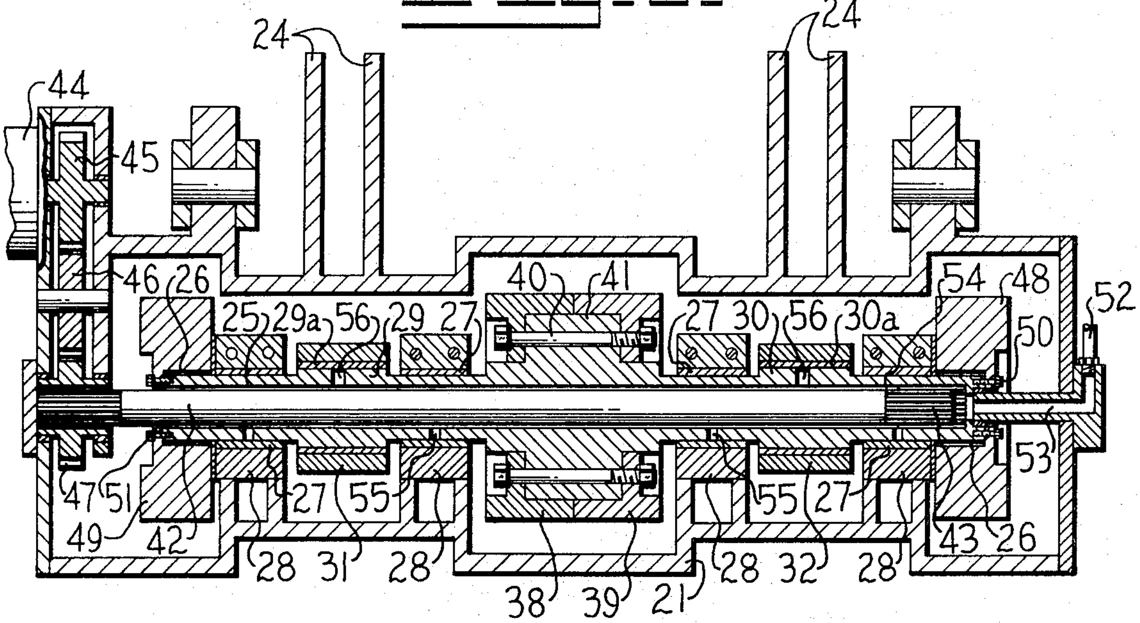

FIG. 4

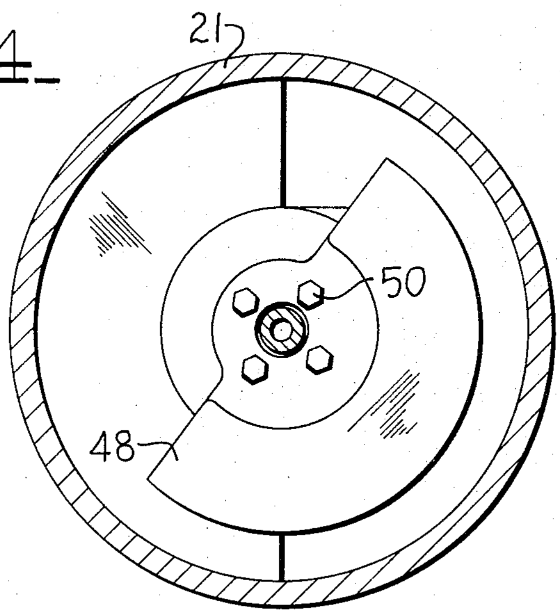

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FIG. 5.

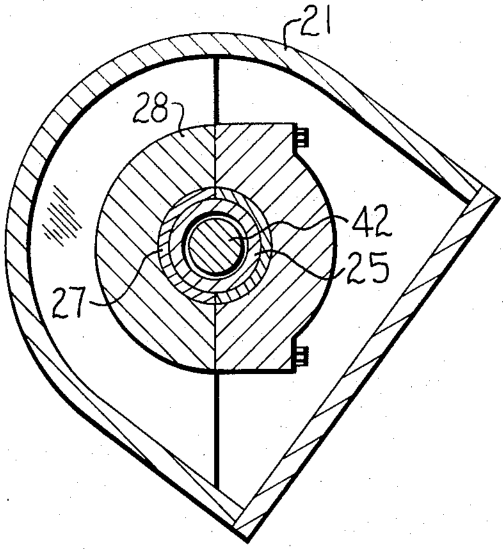

FIG. 6.

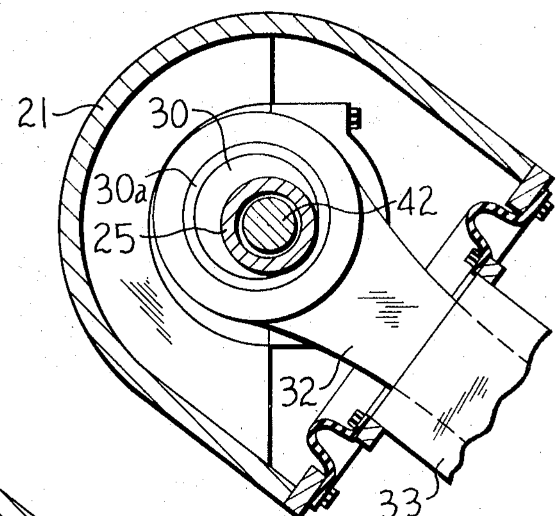

FIG. 7.

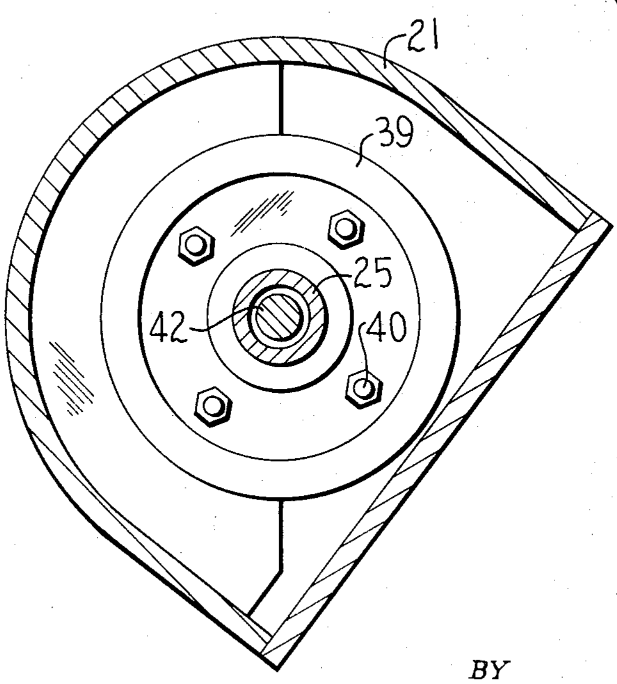

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FIG. 8

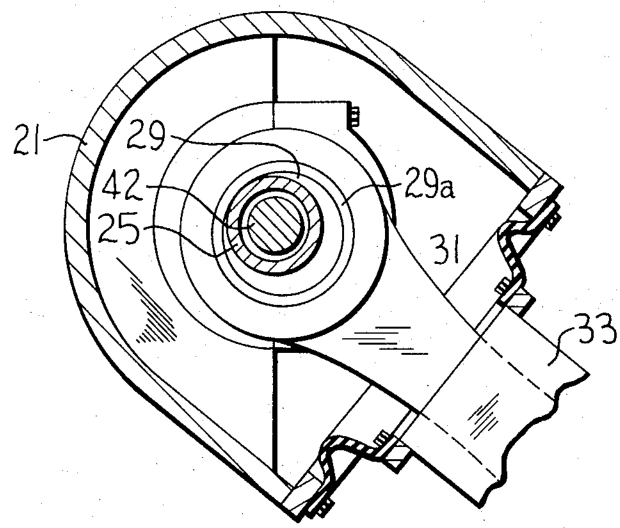
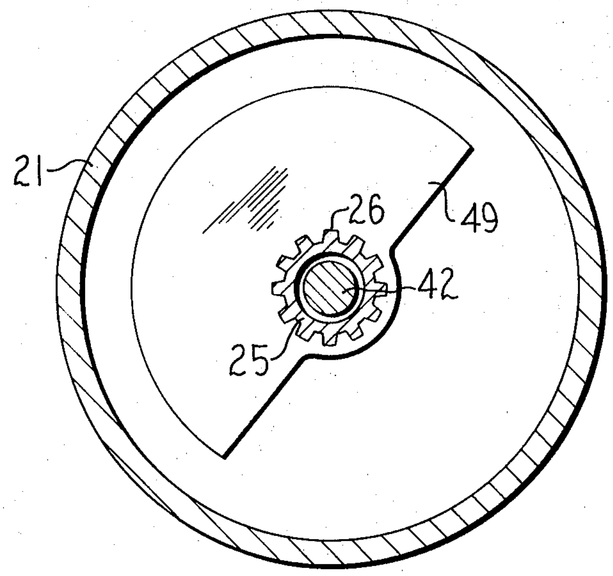


FIG. 9



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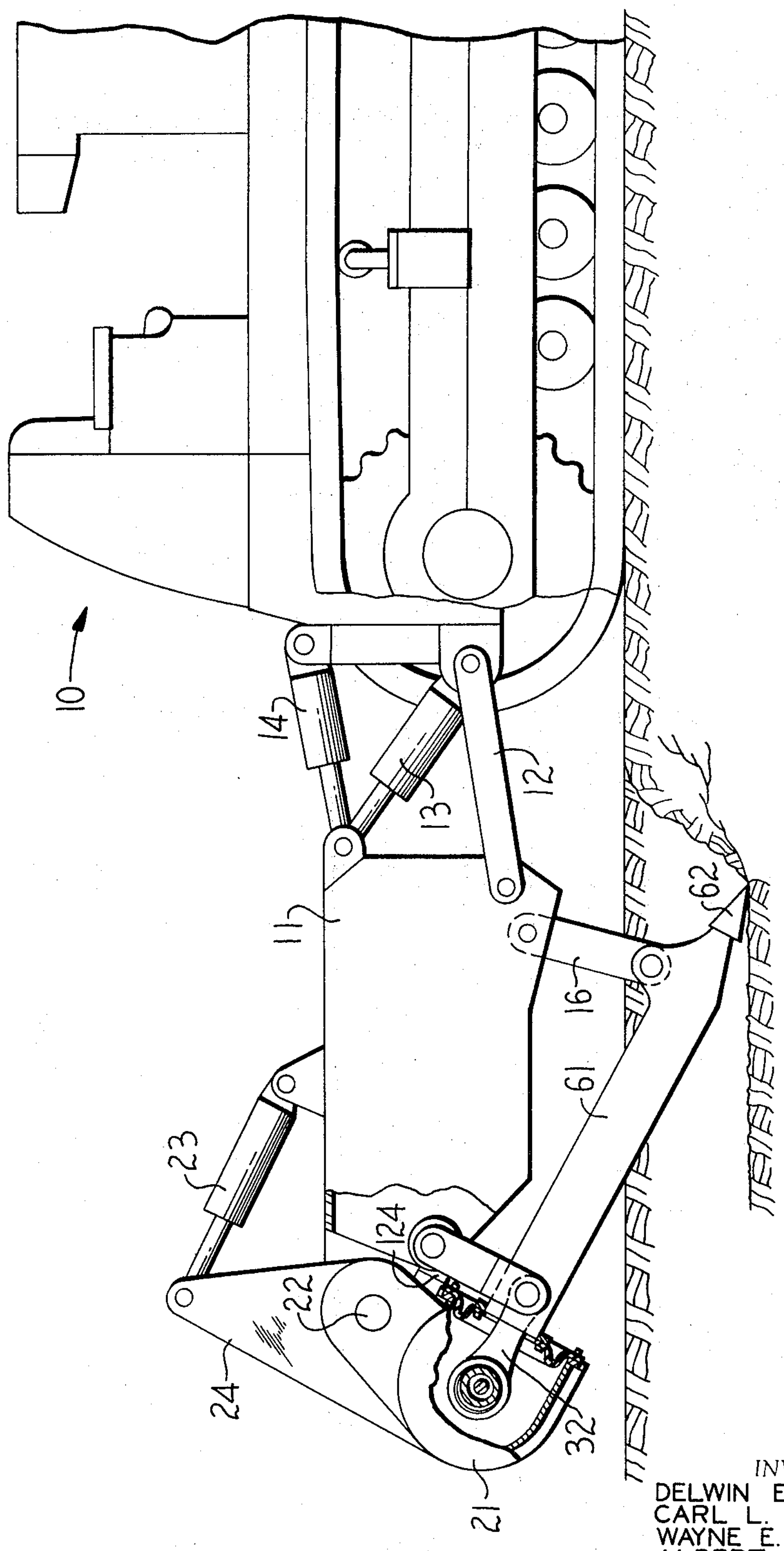

FIG. 10

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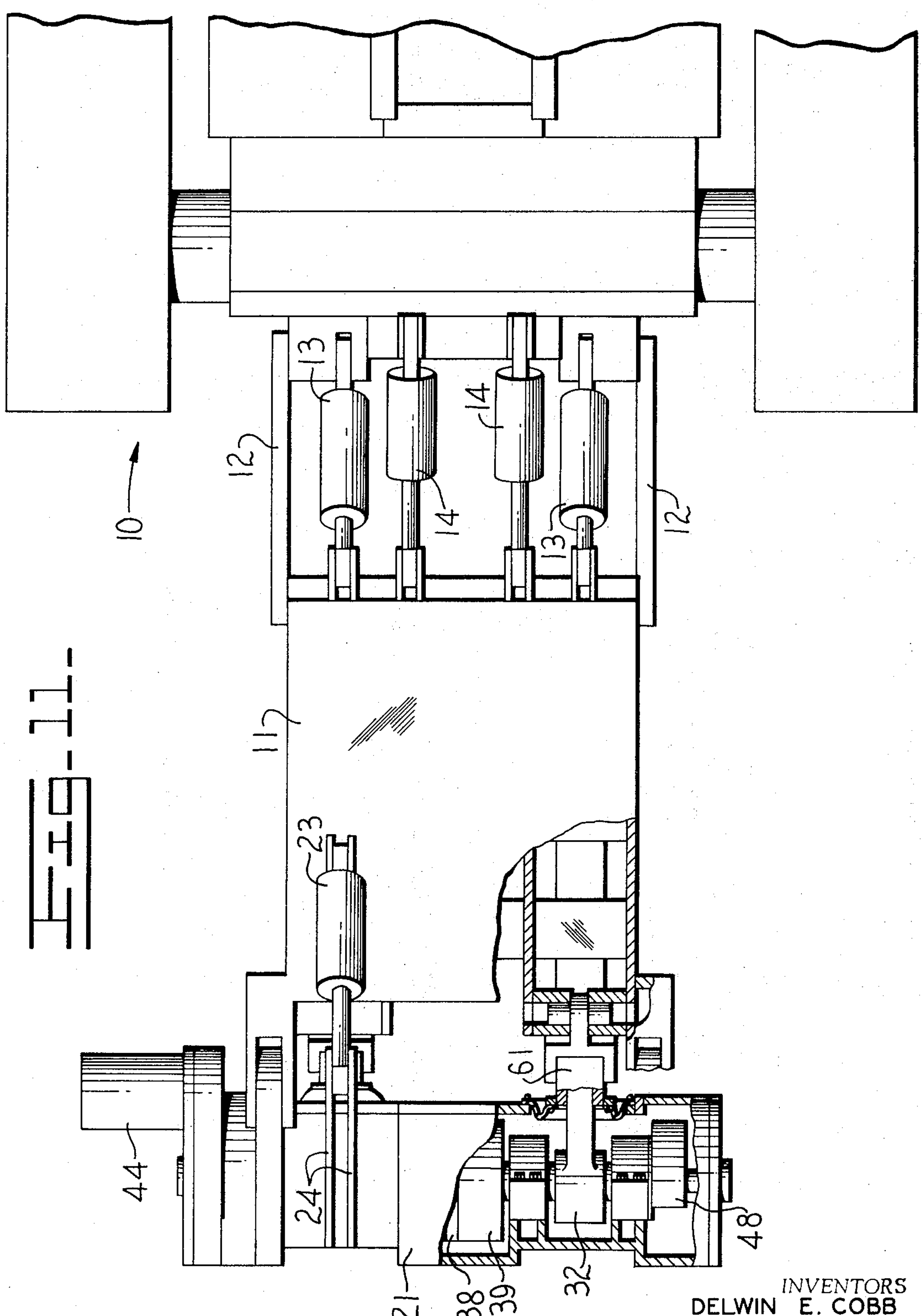

FIG. 11

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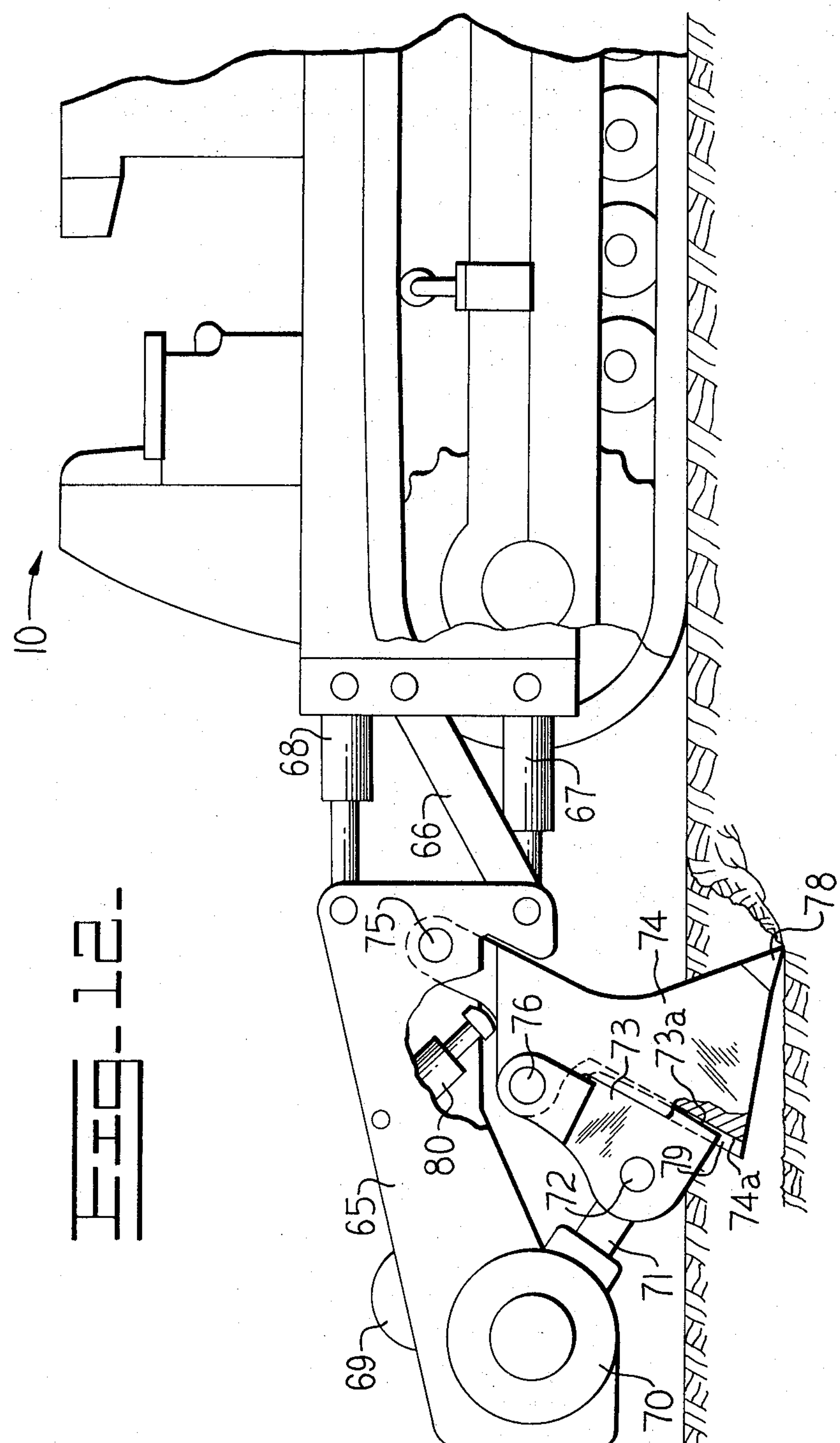

FIG. 12

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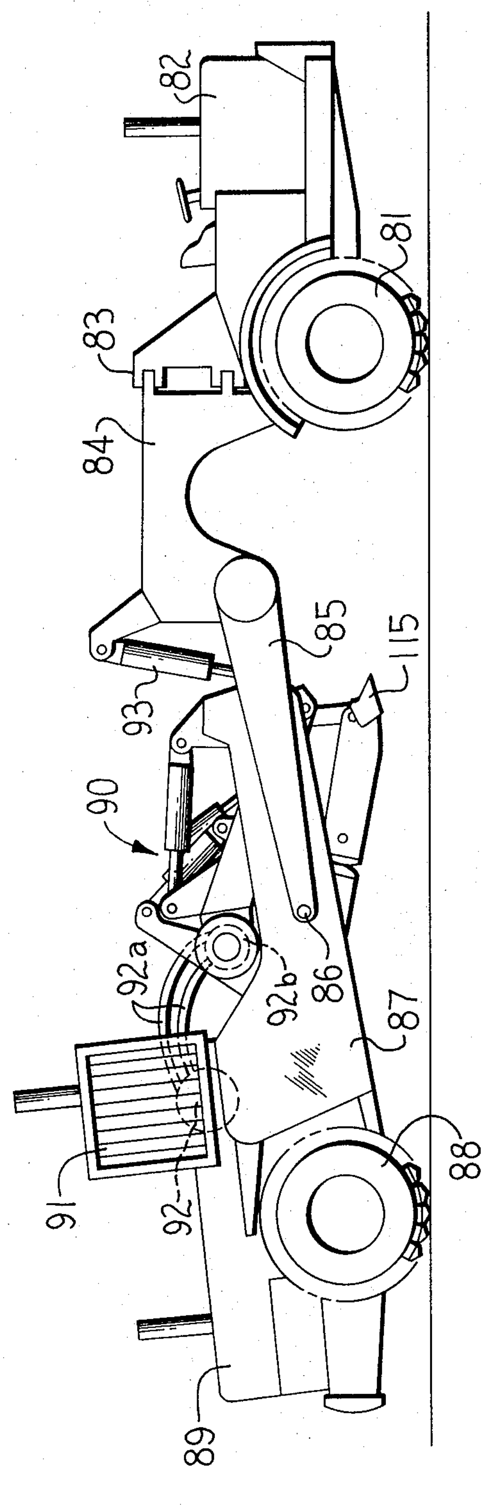
FIG. 13



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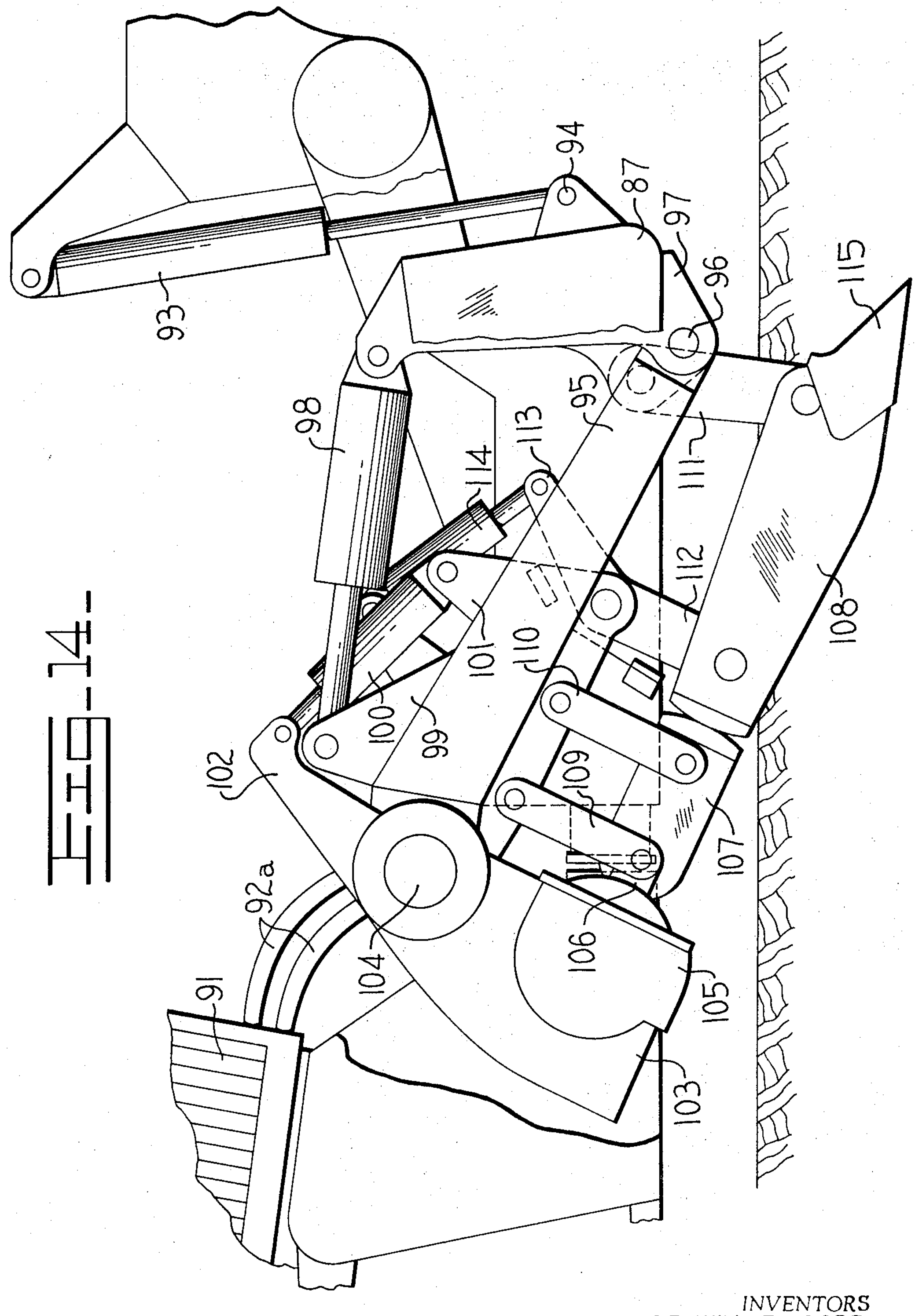

FIG. 14

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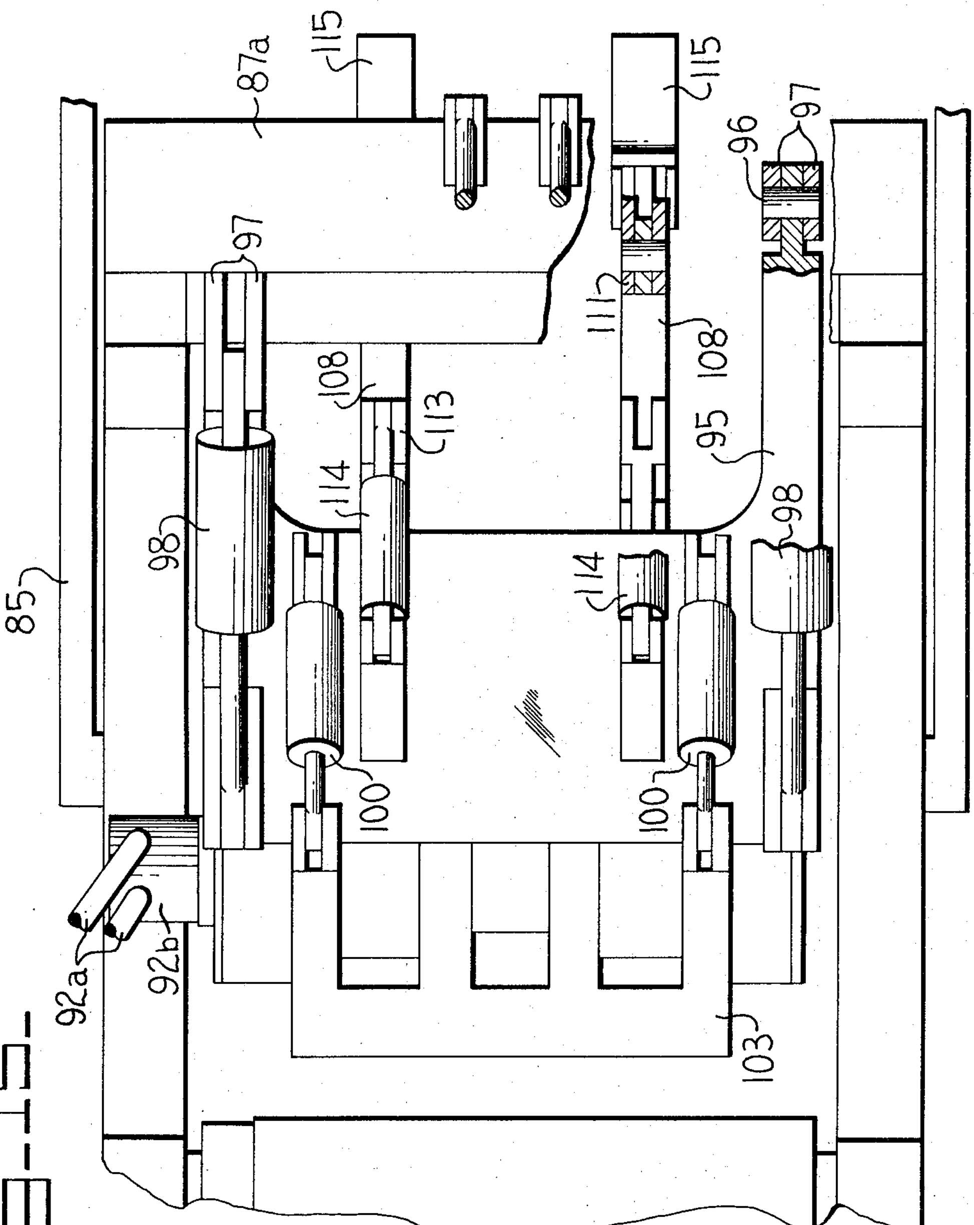

FIG. 15

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Fig. 16

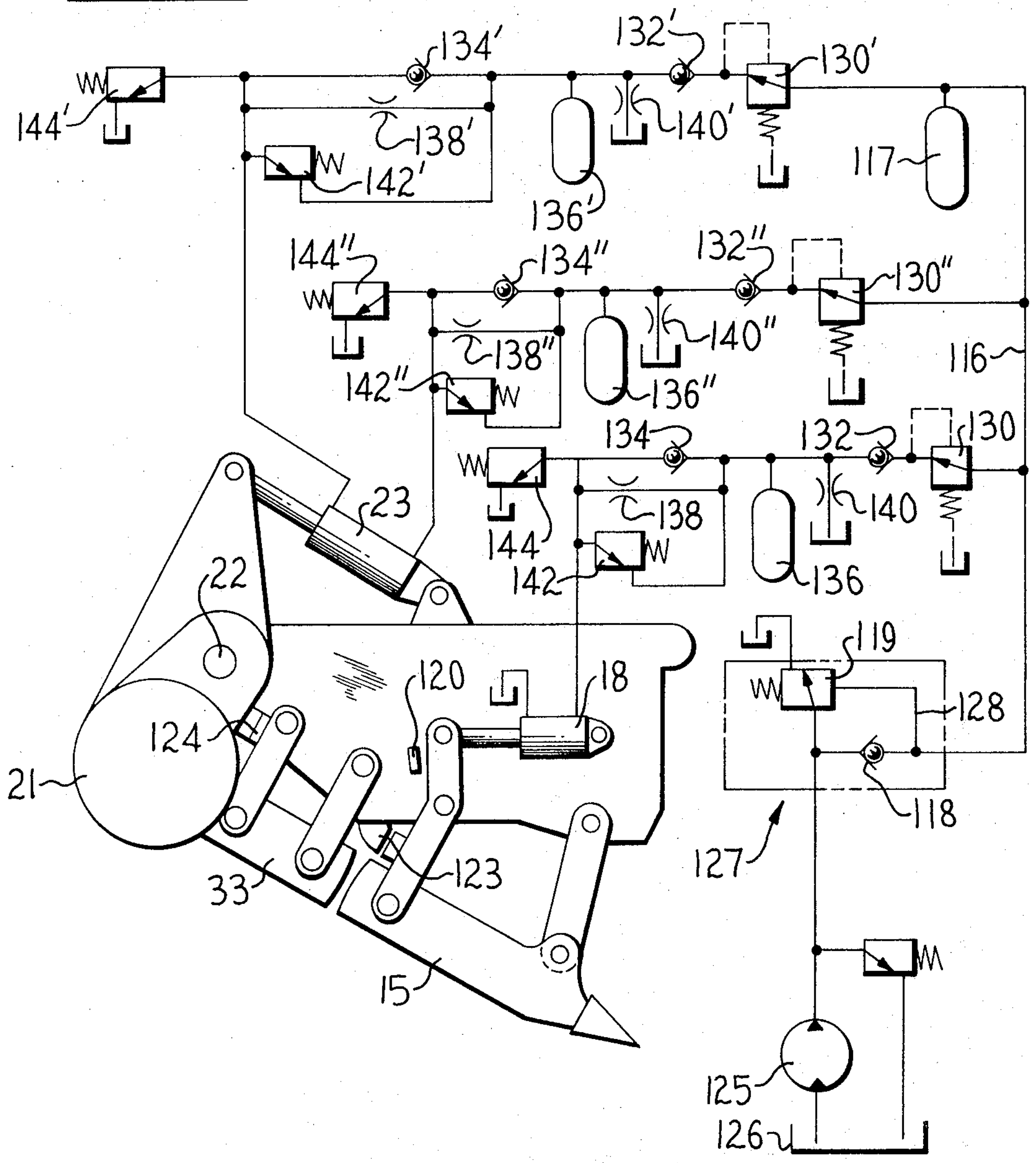

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Fig_17

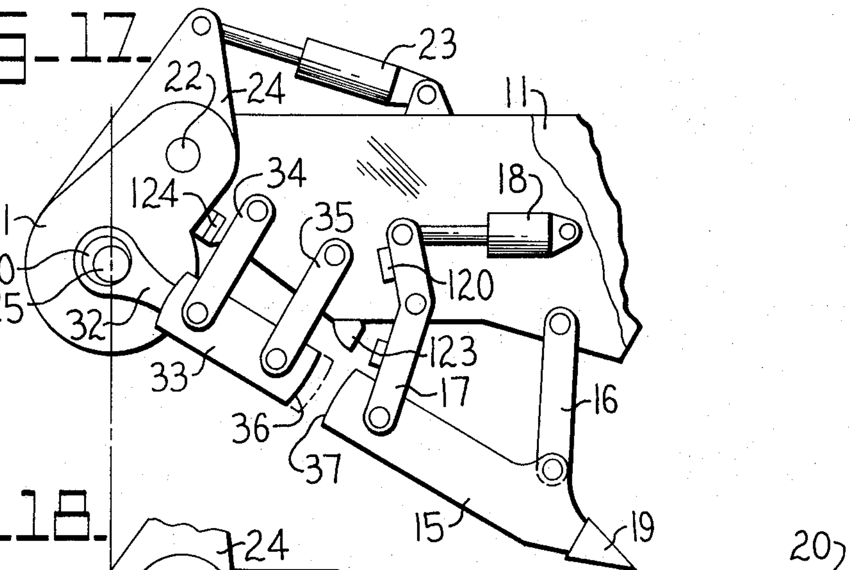

Fig_18

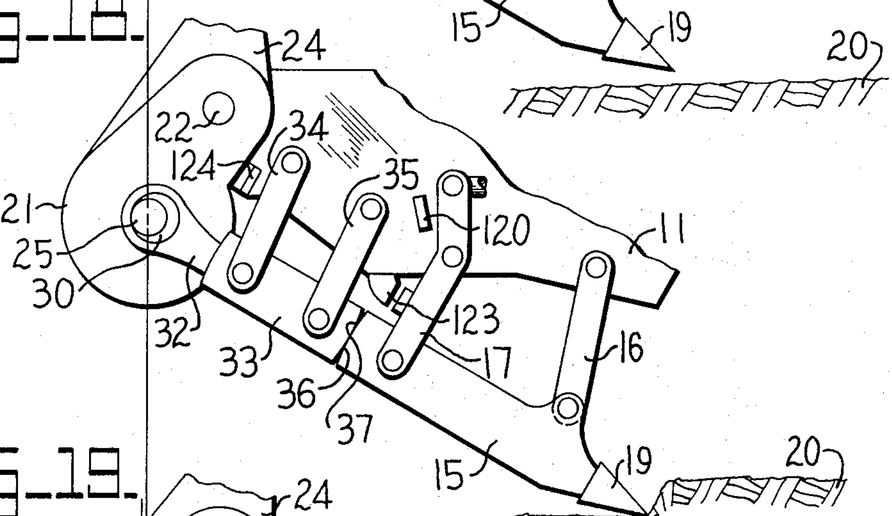

Fig_19

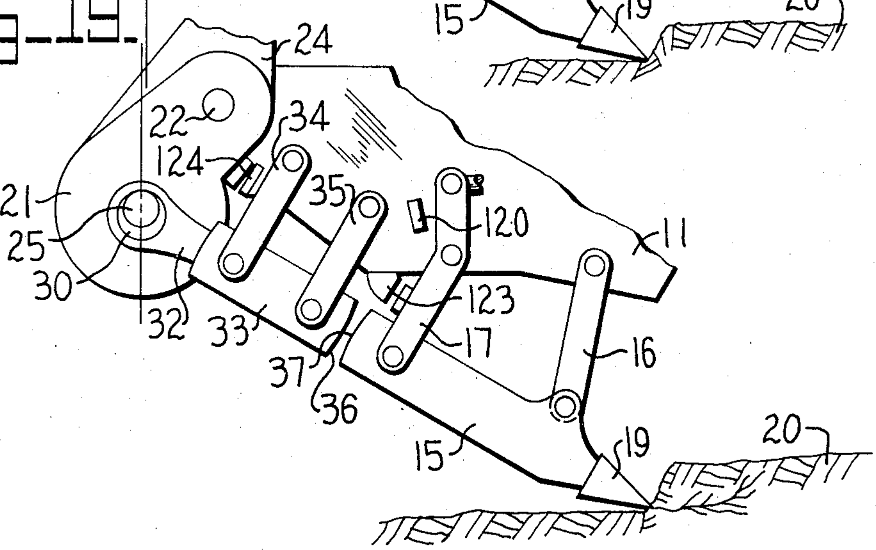

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Fig. 20

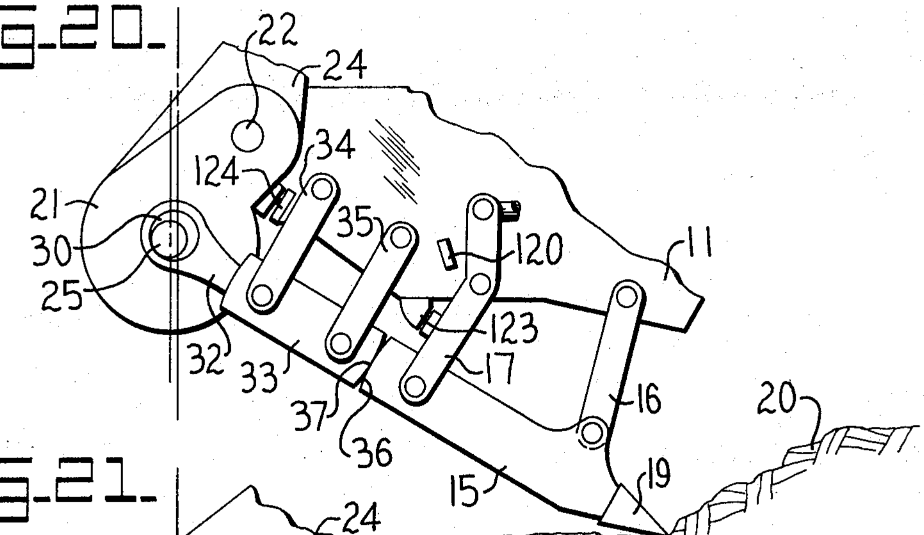

Fig. 21

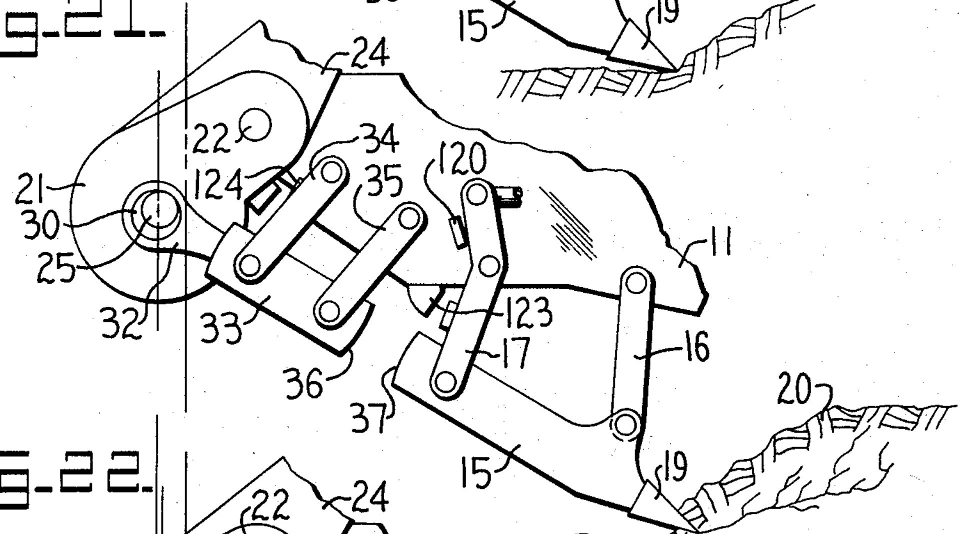

Fig. 22

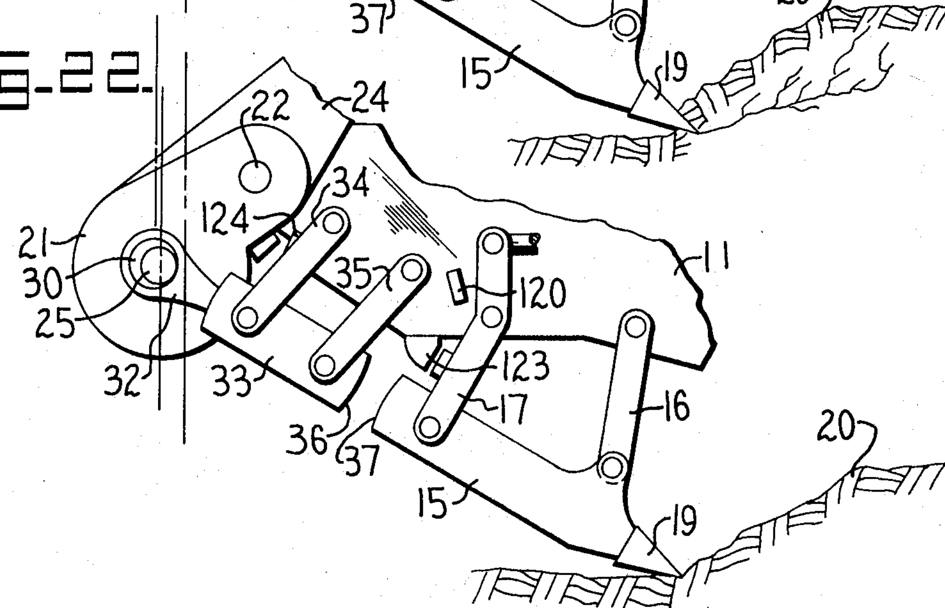

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Fig. 23

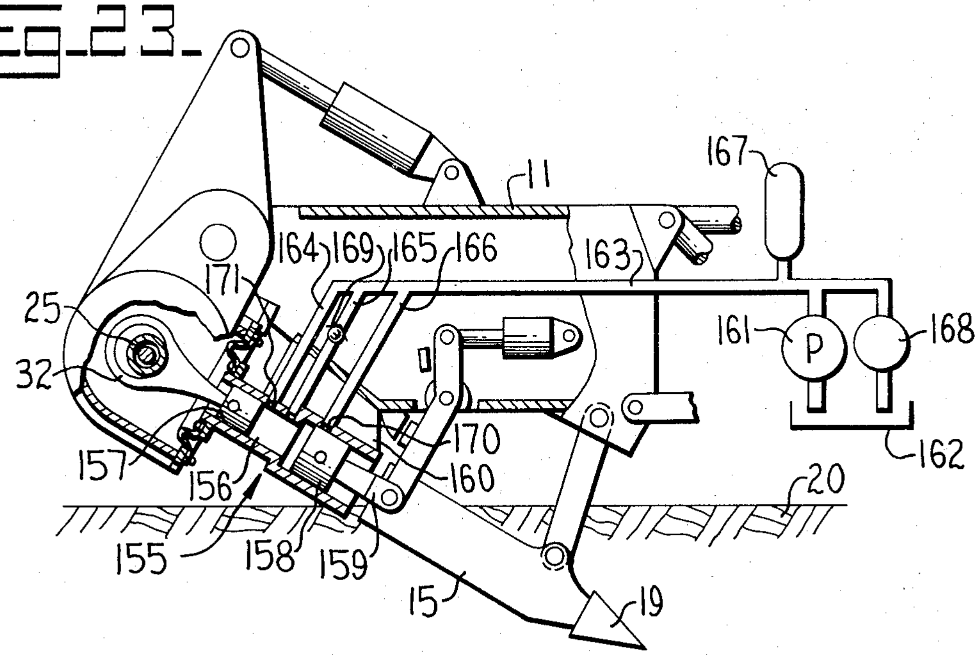

Fig. 24

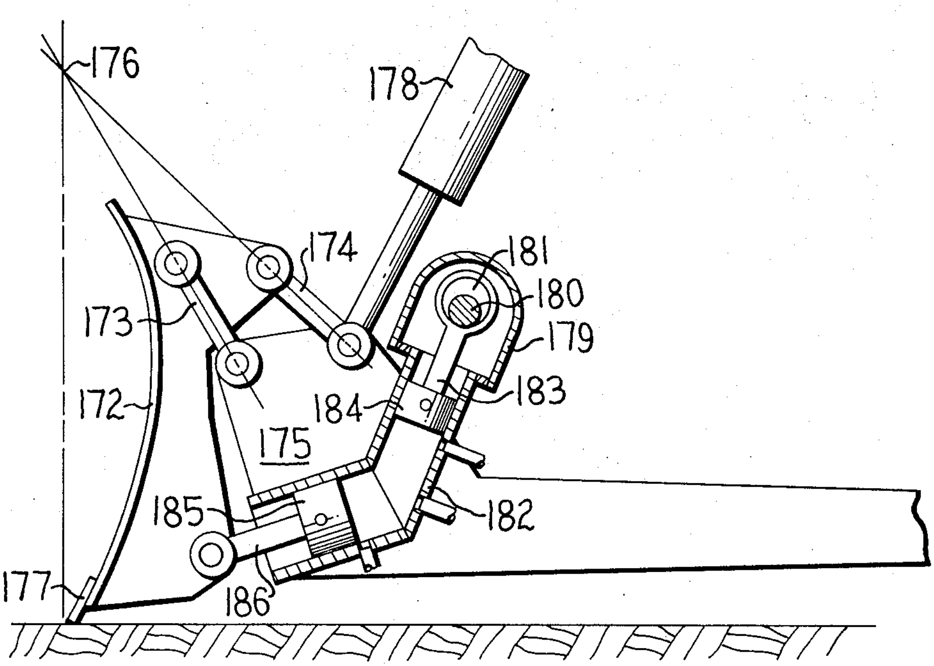

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Fig. 25.

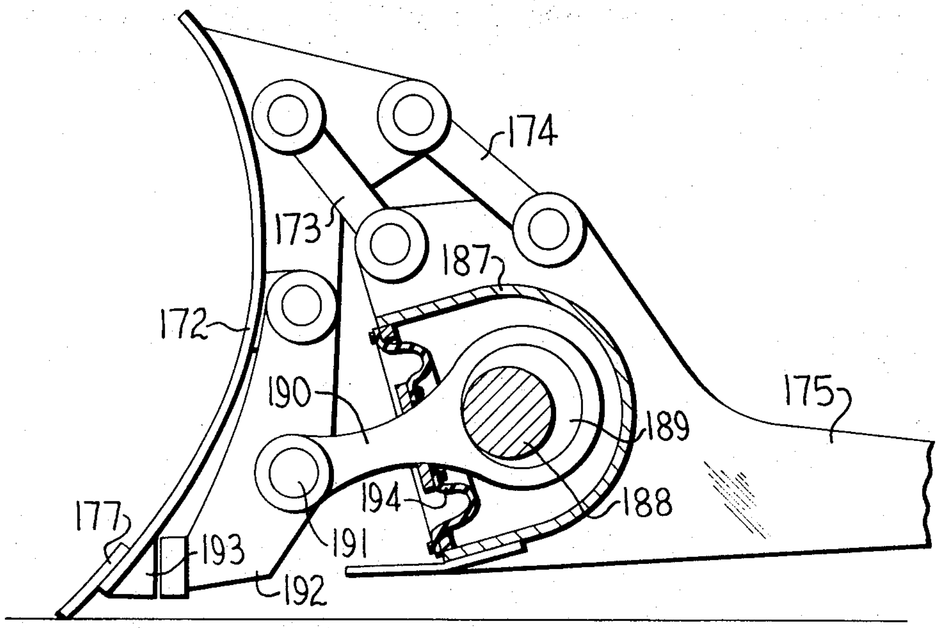

Fig. 26.

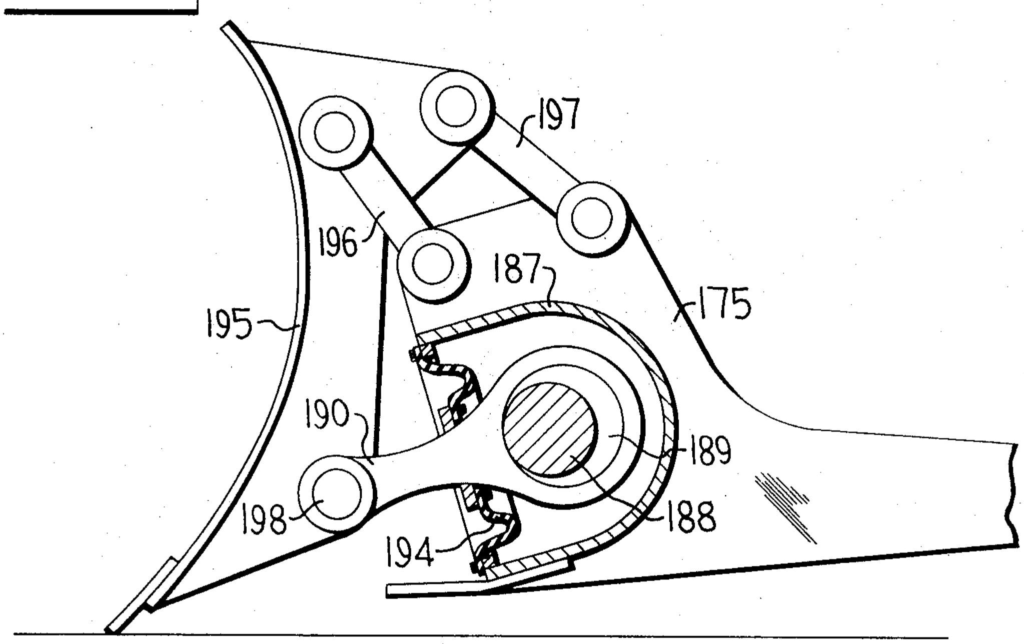

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Fig. 27

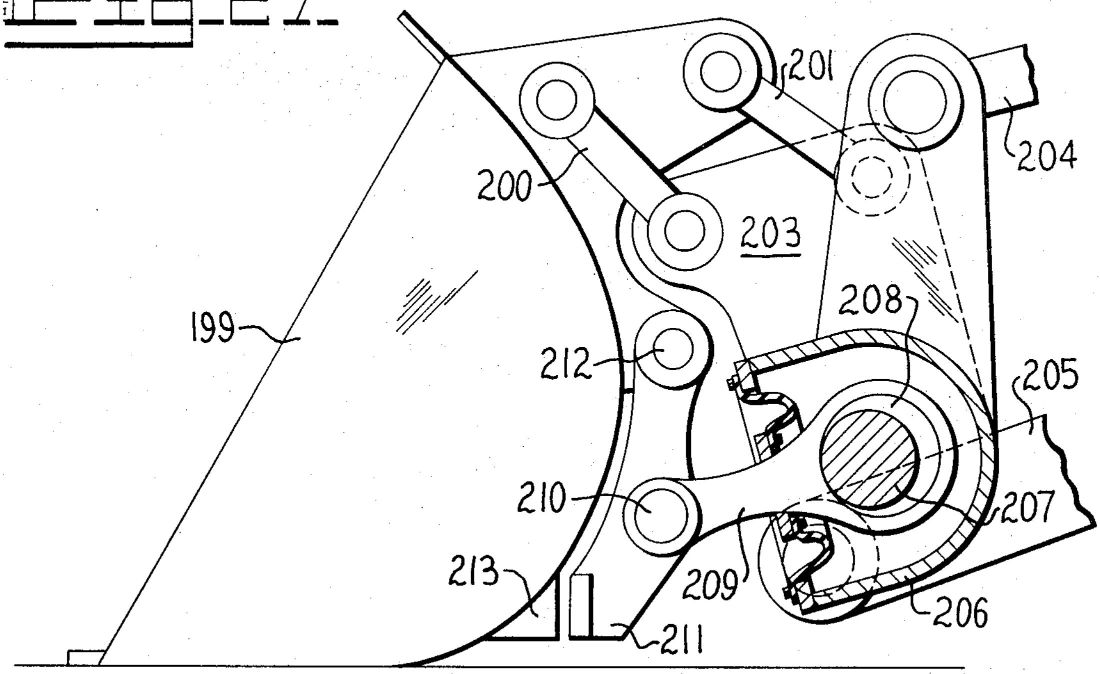
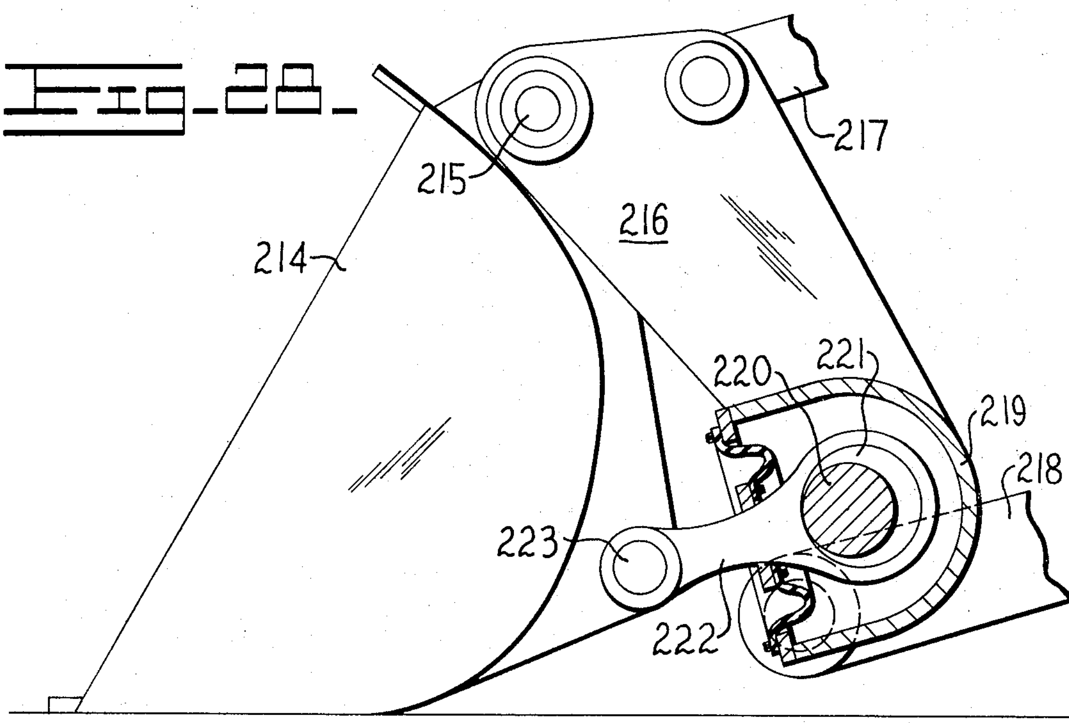


Fig. 28



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APPARATUS FOR FRACTURE OF MATERIAL IN SITU WITH STORED INERTIAL ENERGY

BACKGROUND OF THE INVENTION

This invention relates to earth moving and fracturing implements and pertains more particularly to a high energy mechanical apparatus for the storage and instantaneous delivery of high levels of energy to an implement for the fracturing or separation in situ of hard rock and other earth materials or the like.

A considerable amount of hard rock must be fractured yearly for the construction and mining industries. Most of this rock today is fractured by drilling with percussion or rotary drills and blasting with dynamite or ammonium nitrate. This technique is expensive, slow, noisy and dangerous.

Mechanical tractor drawn rippers have been developed which will operate efficiently in relatively soft, weathered, fissured, layered or previously blasted rock. However, such rippers will not operate in hard rocks.

One of the major problems with the use of rippers is the high forces that must be induced in rock and similar hard material to cause it to fracture. This necessitates the delivery of very high force and energy to the face of the rock or other material to be fractured or separated. Vehicles capable of delivering such forces statically would of necessity be enormous in size and cost and would thus be impractical.

Many exotic techniques have been proposed for fracturing earth formations. Such proposed techniques include sonic energy, electrical spark, water cannon, and others. Such techniques have shown the ability to fracture rock formations but have proven in most cases inefficient for commercial application.

One such sonic technique is the employment of a resonant vibratory system. This system stores vibratory energy in a spring which may take many forms. The energy is then cyclically delivered to a vibrating tool at the resonant frequency of the system. The major problem with this system is that a spring large enough to store adequate energy would be too large to be practical. Another problem is that frequency is critical and varies with load such that there is a major problem of control.

Other proposals have been made to apply vibratory energy to an earth working implement. Such proposals have generally met with failure for one reason or another.

Air and hydraulic hammers are impractical because of their low efficiency. Such large amounts of energy are required to vibrate a tool for breaking rock that unreasonably large amounts of input power would be required.

The present invention is based on the application of the theory that a dynamic system is desirable for cutting and breaking rock and other materials since large forces can be produced with a small average thrust. This is an important feature when the average thrust is limited by tractive effort and weight of a vehicle.

The average force will be proportional to the time the force is applied. For example, if 100,000 lb is applied for 1/10 of a second every second, the needed average force will be only 10,000 lb. The basic idea then is to put the desired force on and unload again as quickly as possible so that during most of the cycle the force is zero or small. If this can be done the average force, compared to the peak force, will be small. If it is assumed that the work done in breakng rock does not depend on the rate of loading, then the total work and average power will not be affected by this pulse type of loading.

The peak power requirements, however, will be large if the work is done in short pulses. This means that large amounts of energy have to be available to be released quickly when needed. For a flywheel-crank system most of the energy is stored as kinetic energy in the flywheel. For a vibrating mass-spring system, the energy cycles back and forth from potential energy in the spring to kinetic energy of the mass. In either case the force that can be developed by the tool is limited by the stored energy.

Large amounts of energy have to be available to produce large forces quickly. The reason for this is that the tool has to penetrate into the rock or soil before the needed force can be developed. If the rock or soil were rigid, little energy would be required. But since they act much like a spring, energy has to be released in order to develop the required force. As an example, assume that a tool in rock has a spring rate of 6×10^6 lb/in. and the breaking force is 6×10^5 pounds. In this case, 30,000 in-lb of energy will be needed to break out a chip. If this is done at a frequency of 20 times a second then 91 horsepower will be required.

In the subject invention as applied to a ripper, the work that the ripper performs is of an intermittent nature, taking place only during a small part of the time required for the driving shaft of the ripper to make a complete revolution. Since the material being fractured is extremely hard, the instantaneous demand of the rippers exceeds that of the drive motors. Therefore, a flywheel or flywheels are placed on the drive shaft to store sufficient energy to meet peak demands. During a greater part of the revolution of the driving shaft, the motor power is used to accelerate the speed of the flywheel. During the part of the revolution when the work is done, the energy thus stored up in the flywheel is given out at the expense of its velocity. As the velocity of the flywheel changes, the energy it will absorb or give up is proportional to the difference between the squares of its initial and final speeds and is equal to the difference between the energy which it would give out if brought to a full stop and that which is still stored in it at the reduced velocity.

Hence

$E = ½ I (n_1^2 - n_2^2)$

$n_1 = (2\pi N_1/60)\ (\pi N_1/30)$

$n_2 = \pi N_2/30$

$I = W K^2/g$

$E = ½ (W K^2/g)(\pi^2/30^2)(N_1^2 - N_2^2)$

$E = (W K^2/5873)(N_1^2 - N_2^2)$

E = Energy release (ft. — lbs)

I = Movement of inertia of rotating mass in lb. ft. sec.^{2}

W = Weight (lbs)

G = Gravity = 32.2 ft./sec./sec. at seal level

K = Radius of gyration in feet

N_1= Revolutions per minute (RPM) before any energy has been given out

N_2= Revolutions per minute (RPM) at end of period during which energy has been given out

n_1= Angular velocity radian/sec. before any energy is given out

n_2= Angular velocity radian/sec. at end of period during which energy has been given out

W K^2 is a measure of the energy potential of a flywheel system in lb.-ft.^{2} at a given RPM and can be determined by the formula

$$W K^2 = (E \cdot 5873)/RPM^2 = \text{lb.-ft.}^2$$

This formula is a derivative of the above formula for energy.

Extensive computer and soil bin model tests have been conducted on the subject concept as applied to a ripper. A model impact ripper has been built and tested in various rock materials to determine the feasibility of fracturing hard rock with an impact device constructed in accordance with the present invention. The performance criterion for the model was specific energy, and is defined as the amount of energy (in.-lbs.) required to fracture a unit volume (in.^{3}) of rock. Specific energy permits one to determine the amount of power necessary to obtain a given production (yd.^{3}/hr.) in a certain rock material. The specific energy of rocks vary. The specific energy is calculated according to the following equation:

$$S.E. = N\ E\ D/W$$

where S.E. = Specific energy (in.-lb./in.^{3})
N = Number of impacts during a run
D = Density of rock (1 ./in.^{3})
W = Weight of rock removed during a run (lbs)
E = Average Energy per blow (in.-lbs.)

It should be noted that the parameters must be considered collectively with a unique relationship existing between one another. The specific energy of the material being fractured is a significant factor.

With a steel spring of the type needed in a resonant system (a ripper shank, bar, spring), the maximum amount of potential energy that can be stored in a cubic inch of spring material is:

$$P.E. = S^2/2E$$

where S is the maximum axial stress that the material can endure and E is the modulus of elasticity. For a working stress of 40,000 psi of a spring material then, the stored energy will be $(4 \times 10^4)^2/(2)\ (30 \times 10^6) = 80/3 = 26.7$ in.lb./in.^{3}. In a vibrating system using a column of uniform cross section for example, the maximum stored energy will be one-half of this value since the column will not be uniformly fully stressed throughout its length.

In contrast, a flywheel, can store much more energy than this per cubic inch of steel. Use a thin annular ring rotating about its polar axis as an example. The tangential stress in the rotating ring is given by the equation:

$$S = w\, r^2\, n^2$$

In this case, w is the mass density of the ring, r is the radius, and n is the angular velocity. The kinetic energy in the ring due to its velocity will be:

$$K.E. = \tfrac{1}{2}\, I\, n^2 = \tfrac{1}{2}\, wVr^2n^2 = \tfrac{1}{2}\, VS$$

Where V is volume of ring. V = 1 in.^{3}
K.E. = ½ S = 20,000 in.lb. for S = 40,000 psi

The ratio between flywheel and spring energies is, therefore at least: 20,000/13.35 or 1500 to 1. In other words, 1500 times more energy can be stored in the flywheel than in an equivalent amount of steel spring for a given stress level. In addition, the stress in the spring is fully reversed each cycle, while the stress in the flywheel at most goes from zero to the maximum cycle. Reversing the stress in the spring energy cycle can adversely affect the fatigue life of the spring.

In order to work the above ring at 40,000 psi, the flywheel would have to be quite large or else rotate at a high speed. If the speed is limited to 1200 rpm with a wheel radius of 15 in. and a resulting stress of only 2600 psi, the energy in a cubic inch of the steel ring will be 1300 in.-lb. for an energy ratio of about 100. This is less than the 1500 ratio, but it is still a substantially significant advantage.

For breaking rock in situ in quantity, large forces and power are required. This means large energies per blow and many blows per minute. The prior art systems have been unable to meet these requirements.

The primary object of the present invention is to provide an earth working implement that is rugged and efficient and overcomes the above disadvantages of the prior art.

Another object of the present invention is to provide an earth working apparatus that is capable of storing large amounts of energy and selectively applying it to an earth working implement.

A further object of the present invention is to provide a dynamic system that is capable of delivering sufficient energy to a tool for earth working to be practical.

A still further object of the present invention is to provide a mechanical dynamic system that is capable of efficiently delivering high energy pulses to an earth working tool.

In accordance with the present invention large amounts of inertial energy is stored in a massive flywheel to be cyclically delivered to an earth working implement at peak power demand. Suitable transmission means including a crank and connecting rod is used to transmit the energy to an earth working implement such as a ripper.

BRIEF DESCRIPTION OF THE DRAWINGS

The above and other objects and advantages of the present invention will become apparent from the following description when read in conjunction with the accompanying drawings in which:

FIG. **1** is an elevational view partially in section of a rock ripper incorporating a preferred embodiment of the present invention;

FIG. **2** is a plan view partially in section of the embodiment of FIG. **1**;

FIG. **2*a*** is a section taken along lines II*a*—II*a* of FIG. **1**;

FIG. **3** is a sectional view taken along lines III—III of FIG. **1**;

FIG. **4** is a sectional view taken along lines IV—IV of FIG. **2**;

FIG. **5** is a sectional view taken along lines V—V of FIG. **2**;

FIG. **6** is a sectional view taken along lines VI—VI of FIG. **2**;

FIG. **7** is a sectional view taken along lines VII—VII of FIG. **2**;

FIG. **8** is a sectional view taken along lines VIII—VIII of FIG. **2**;

FIG. **9** is a sectional view taken along lines IX—IX of FIG. **2**;

FIG. **10** is an elevational view with portions broken away of an alternate embodiment of the invention;

FIG. **11** is a top view of the embodiment of FIG. **10**;

FIG. 12 is another embodiment of a rock ripper incorporating the present invention;

FIG. 13 is an elevational view of an embodiment of the present invention incorporated in a specialized vehicle;

FIG. 14 is a detailed view of the ripper apparatus of the embodiment of FIG. 13;

FIG. 15 is a plan view of the embodiment of FIG. 14;

FIG. 16 is a schematic of the control system of the present invention.

FIGS. 17 through 22 are side elevations illustrating the embodiment of FIG. 1 in various positions of operation;

FIG. 23 is an elevational view, partially in section, of a further embodiment of the present invention;

FIG. 24 is an elevational view of still another embodiment of the present invention;

FIG. 25 is an elevational view of an embodiment of the invention as applied to a dozer blade;

FIG. 26 is an elevational view of an alternate embodiment of the invention applied to a dozer blade;

FIG. 27 is an elevational view of an embodiment of the invention as applied to a loader bucket;

FIG. 28 is an elevational view of another embodiment of the present invention applied to a loader bucket.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

Referring now to the drawings, and more particularly to FIG. 1, there is illustrated an embodiment of the present invention incorporated in a rock ripper illustrated as being supported and towed by a track type vehicle generally designated by the numeral **10**. The ripping apparatus comprises a housing or support member **11**, supported or coupled to the back of tractor **10** by means of a link **12** and a plurality of upper hydraulic cylinders **13** and **14** which are operative to vary the dept of the ripper as well as the angular position of the ripping apparatus. A plurality of ripper shanks **15** are supported below the housing **11** by means of pairs of links **16** and **17** forming a quadrilateral linkage. The link **17** includes an arm **17*a*** pivotally connected to a hydraulic cylinder **18**. The cylinder **18** serves as damper and spring and is operative to maintain the shank **15** in forward position against the face of the formation and to limit rebound velocity of the shank. Note the hydraulic cylinders **13** and **14** may also function as spring and damper means for isolation of shock and vibration from the vehicle **10**. A suitable hydraulic circuit is shown and described later in FIG. 16. A hardened tip **19** is positioned to engage and fracture a rock or similar hard earth formation **20**.

A mechanism for storing and delivering high levels of energy to the ripper tip **19** comprises a housing means **21** pivotally connected at **22** to housing **11** and partially adjusted by means of a hydraulic cylinder **23** connected by means of a lever arm **24** extending from housing **21**. The remaining details may be best seen by referring alternately between the various FIGS. 1–9. A crank or cam shaft **25** having a pair of splines **26** at its ends is journaled for rotation in suitable bearings **27** carried in support blocks **28** carried in housing **21**. A pair of diametrically opposed eccentric throws **29** and **30** are formed on the crank or cam shaft. A pair of connecting rods **31** and **32** are journaled on throws **29** and **30** by a pair of suitable bearings **29*a*** and **30*a***. The connecting rods **31** and **32** are connected to a pair of hammers **33**, each of which are supported by a pair of links **34** and **35** from the housing **11**. The hammers **33** are adapted to deliver energy by means of intermittent blows between surfaces **36** and **37** to the ripper shank **15**. The surfaces **36** and **37** between members **15** and **33** are curved to reduce stress and to eliminate problems of alignment.

Returning now to the details as illustrated in FIG. 3, the main energy storage means comprises a massive flywheel comprising two parts **38** and **39** attached by means of bolts **40** to a flange **41** formed on crankshaft **25**. The flywheel is of sufficient means to store enough inertial energy to supply the peak demands of the system. The flywheel is preferably sufficiently large to supply the demands of the systems with no more than a 10 percent fluctuation in angular velocity. The crank or cam shaft **25** is driven by means of a quillshaft **42** extending concentric with the shaft **25** and splined at **43** to the crank or cam shaft. The use of the guillshaft **42** helps to isolate the drive motor and drive gears from intermittent rapid speed changes of the crank or cam shaft due to the resistance of the material being ripped being transmitted back through the shank, hammer, connecting rod, etc. Suitable prime mover means such as a fluid motor **44** is drivingly connected by means of a drive gear **45**, idler gear **46** and gear **47** to the quillshaft **42**.

A pair of counterweights **48** and **49** are connected such as by splines **26** and bolts **50** and **51** to the end of the crank. These counterweights **48** and **49** are mounted to counteract moments induced by the respective eccentric throws **29** and **30** and in acting in conjunction therewith essentially forms a concentric flywheel or additional storage means.

Suitable lubrication is provided for the crank or cam shaft bearings such as by means of a conduit **52** from a source (not shown) which communicates by way of passageway **53** to an annular passageway **54** formed between shaft **42** and shaft **25** which passageway conveys lubricant to bearings **27** by means of passageway **55** and to bearings **29*a*** and **30*a*** by means of passageway **56**.

The apparatus of the present invention operates to deliver high levels of energy to the ripper tip **19** wherein the energy as delivered in a concentrated point to the rock formation **20** for fracturing or chipping away the material of the formation. Large amounts of energy developed by motor **44** is stored as inertial energy in the flywheel means on the rotary shaft where the energy is then delivered at peak demand through the transmission means to the point of application. Rotation of the crank or cam shaft drives the hammers **33** through connecting rods **31** and **32** in an oscillatory manner such that when the hammers intermittently come into engagement with the shanks **15**, the energy is delivered thereto. The frequency of energy delivered to the shank **15** is of course controlled by the revolutions per minute of the crank or cam shaft **25**. The cutting point **19** is maintained in contact with the rock formation by forward motion of the vehicle **10** at a rate that is consistent with the cutting rate capabilities of the machine.

The depth of the cut and the angular position of the apparatus as pointed out above, are adjusted by manipulation of the hydraulic cylinders **13** and **14**. The angle of impact, in other words the angle at which the impact force is directed to the rock, is found to be critical. This angle, designated by Alpha (α) in FIG. 1, is measured from the rock surface to the axis of the cutting shank 15. When this angle is too large, the downward component will force the tool too deep. On the other hand, when this angle is too small, the tip will not penetrate and there will be excessive wear on the bottom surface of the tip. This angle must be adjustable to meet varying conditions and initiate penetration. For example, an angle of approximately 45 degrees has been found to be essential for penetration under many circumstances, but the angle must then be reduced to a lower angle to control depth of cut. A slightly steeper angle is normally needed after the tip wears. The angle of impact may be changed as desirable for the particular type rocks being fractured. Thus, as a result of these problems, the impact angle is made adjustable from approximately 20° to 55° and easy to change while the machine is in operation.

Another critical variable in this combination is the cutting edge tip motion. The tip motion should be substantially horizontal for normal operation. However, in order to maintain the desired depth of cut with a horizontal motion, the impact angle must have a downward component. The absence of such a downward component will cause wear and excessive heating in the tip, and loss of depth control. On the other hand, if the downward component is too large, energy will be wasted. The tip will be either forced into the rock farther than needed without a proportional increase in broken rock or the ripper frame will be forced to move up an excessive amount when the tip fails to go down. The tip motion depends to a large extent on how the shank is supported. The tip should be supported so that it will move forward without causing excessive movement of the mounting frame. If a single pivot point is used, it should be located ahead of the cutting tip to get the desired motion. If this type of mounting isn't convenient, then a four bar linkage, as illustrated in FIG. 1, can be used to obtain the desired effect.

The impact surfaces between the hammer and the shank 36 and 37 with an impact coupled design should be shaped to reduce the stresses in the system. In the four bar linkage embodiment as illustrated in FIG. 1, spherical impact surfaces are used. This shape provides a lower initial spring rate than flat surfaces and reduces metal stresses without having significant effect upon the rock forces.

The best included angle for the tip is found to be a compromise between performance and life. An angle of 30° is found to be more efficient in breaking rocks than one of a larger angle, but with available tool materials this angle is difficult to maintain and will wear too rapidly in harder rock material. The optimum included tip angle is found to be within the range of 30° and 60° with the best angle depending upon the rocks being fractured.

An important parameter for the impact coupled vibratory system is the ratio of peak vibratory velocity to the average travel speed. The larger this ratio the more the average force can be reduced. For large ratios the tool will be in contact with rock material for a small percentage of the time which produces a large ratio between peak force and average force. The peak velocity for sinusoidal vibration is directly proportional to the frequency and amplitude of vibration. Therefore, the larger the product of these two terms, the bigger the velocity ratio. Although the ratio can be increased by reducing travel speed, the production is also directly proportional to this variable. Thus it is undesirable to reduce travel speed just to obtain a large velocity ratio.

Referring now to FIGS. 10 and 11, there is illustrated an alternate embodiment of the present invention wherein identical elements are identified by the same reference numerals as in the previous embodiment. This embodiment differs from the previous embodiment in that it comprises wear may be referred to as a direct coupled linkage mechanism. That is, the output from the energy storage means is directly coupled by means of connecting rod 32 to the ripper shank 61. In this embodiment, instead of the energy being delivered by impact from a hammer to the tool which stays substantially in contact with the formation, the impact in this embodiment is of the tool directly against the formation at the point 62. This embodiment has the advantage of being less complex than the previous embodiment, however, it has the disadvantage of being more susceptible to tip waer because of the stroke of the tip. An optimum ratio of peak vibratory velocity to the average travel speed is also an important consideration for achieving optimum production as in the previous embodiment.

Referring now to FIG. 12, there is illustrated another embodiment of the present invention which is a modified form of the impact coupled concept. In this embodiment the apparatus comprises a frame 65 supported from the tractor 10 by means of a rigid link 66 and a pair of hydraulic cylinders 67 and 68, which cylinders are operative to adjust the depth of cut as well as the angle of attack. These cylinders also function as spring and damper to permit the drive system to float as in the previous embodiment. In this embodiment a hydraulic motor 69 delivers energy to an energy storage unit 70 which is identical to the previous embodiment's. The energy from the energy storage unit 70 is transmitted by transmission means which includes a connecting rod 71 which is pivotally connected at 72 to a hammer 73. The hammer 73 impacts against the ripper shank 74 which is pivotally supported at 75 from the frame 65. The hammer 73 is pivotally supported at 76 on shank 74. The pivot point 75 is located just ahead of the tip 78 so as to obtain the substantially horizontal forward motion of the tip 78 as discussed above. The impact surface 73*a* is shaped to reduce stresses. Impact surface 74*a* is located in a groove 79 which reduces the possibility of rock and other material from coming between the impact surfaces. In this case the impact surfaces would be shaped so that initial contact occurs near the center of rotation and then as the shank and hammer deform, the contact surface expands toward the lower edge as illustrated. This construction will reduce the impact stress by providing a cushioned impact and by reducing the initial impact velocity. A hydraulic cylinder 80 bears against the rear of shank 74 and serves as a damper and spring and is operative to maintain the shank in a forward position against the base of the formation and to limit rebound velocity of the shank. A suitable hydraulic circuit is shown and described later in FIG. 16.

Referring now to FIGS. 13 through 15, there is illustrated an embodiment of the present invention in which the vehicle arrangement or construction plays a significant role. One significance of this arrangement is that ripping points are contained or confined within the four corners of the machine as bounded by the driving wheels. This arrangement has the advantage of providing a more stable platform for the apparatus. This stability results from having driving wheels both fore and aft of the ripping shanks as well as having the reaction mass centered closer to the ripping tip. With this arrangement the mass of the machine can be used more efficiently to provide a downward tip force for better penetration and/or depth control.

This embodiment comprises a vehicle having a front end assembly having drive wheels **81** driven by an engine **82** and pivotally connected at **83** to a gooseneck **84**. The gooseneck **84** is connected to a pair of arms **85** which are pivotally connected at **86** to side walls **87** of the rear assembly. The rear assembly includes a pair of drive wheels **88** driven by an engine **89**. An impact ripping assembly similar to the previously disclosed embodiment is generally shown at **90** and is driven such as by a hydraulic motor driven by an engine **91** drivingly connected thereto such as by means of a suitable hydraulic drive system including a pump **92** supplying pressurized fluid via conduits **92a** to a fluid motor **92b**. The entire ripper assembly may be raised and lowered by means of hydraulic cylinder **93** pivotally connected at **94** to the cross beam **87a** of side assembly **87**. A frame **95** for the support of the ripper assembly is pivotally connected at **96** to brackets **97** rigidly connected to a cross beam **87a** extending between side walls **87**. A pair of hydraulic cylinders **98** are connected between the brackets **99** on frame **95** and to brackets **97** for permitting adjustment of the ripper mechanism for control of the shank angle of attack. A pair of hydraulic cylinders **100** are pivotally connected between brackets **101** on frame **95** and bracket or arm **102** on housing **103** combined to form or provide a spring and damping means for the housing **103**. A housing **103** is pivotally supported at **104** and houses the energy storage and transmission means. The energy storage and transmission means **105** is substantially as illustrated in the previous embodiment is connected by connecting rod **106** to a hammer **107** which is adapted to intermittently or cyclically deliver energy to the ripper shank **108**. The hammer **107** as in previous embodiments is supported by a pair of link members **109** and **110** for reciprocatory movement toward and away from the shank member **108**. The shank member **108** is pivotally supported by a quadrilateral linkage **111** and **112** with link **112** having an extended arm **113** which is connected to a pair of hydraulic cylinders **114** to provide spring and damping means for the ripper shank **108**. A suitable removable ripper tip **115** is provided for the ripper shank **108**.

Referring to FIG. **16**, a pump **125** supplies fluid from a tank **126** to an accumulator charging circuit comprising a charging valve **127**, a conduit **116**, and an accumulator **117**. The charging valve **127** is comprised of a check valve **118** and an unloading valve **119** responsive to a pressure in conduit **116**. Fluid from the charging valve **127** is directed to three branch circuits for cylinders **23** and **18** by conduit **116**. Accumulator **117** is charged to a predetermined pressure controlled by unloading valve **119**. Conduit **128** communicates the pressure in conduit **116** to open valve **119** permitting pump flow to be returned to tank **126** when the predetermined pressure has been reached. Check valve **118** prevents fluid flow back towards the pump.

The branch circuit for the head end of cylinder **18** includes a pressure reducing valve **130**, check valves **132** and **134**, accumulator **136**, restrictive orifices **138** and **140** and relief valves **142** and **144**. Pressure reducing valve **130** controls the pressure in the branch circuit to a pressure lower than that in the conduit **116**. The check valve **134** bypasses relief valve **142** and orifice **138** to communicate the accumulator **136** with the cylinder **18**. Restrictive orifice **138** controls the flow of fluid from cylinder **18** to accumulator **136**, thus restricting the rebound velocity of shank **15** thereby lessens the impact velocity between the shank and hammer **33**. Accumulator **136** provides a spring for shank **33** and provides for peak fluid demands during high velocity forward motion of the shank preventing cavitation of cylinder **18**. This prevents loss of rebound control of the shank due to cavitation since only orifice **138** controls rebound of the cylinder **18**. Forward movement of the shank **33** is not restricted in any way. Relief valve **142** limits the maximum damping provided by orifice **138**. Check valve **132** prevents flow of fluid back through valve **130**. Relief valve **144** protects the branch circuit of cylinder **18** from excessive pressure surges.

A small amount of fluid is constantly bled from the branch circuit for cooling purposes to prevent heat buildup with the flow controlled by restrictive orifice **140**. The resulting pressure drop in the branch circuit opens pressure reducing valve **130** whereupon the fluid is replenished from the accumulator **117**.

The hydraulic cylinder **23** is also connected by branch circuits to conduit **116**. Each branch circuit is identical to the branch circuit for the head end of the cylinder **18**, and like components have been marked with prime (') and ('') exponents. In a manner identical to the preceding branch circuit, restrictive orifices **138'** and **138''** control the flow of fluid from cylinder **123** to accumulators **136'** and **136''** and prevent rapid pivoting of the energy storage means **21**. The pressure in accumulator **136''** acts on the head end of cylinder **23** and along with the weight component of the energy source means establishes the average force for hammer **148**, a spring for energy storing means **150** and prevents cavitation of the cylinder. The pressure accumulator **136'** mainly acts as a volume storage and replenishing means preventing cavitation of the rod end of the cylinder **123** with only a very small spring rate provided. The remaining components of the branch circuits function identical to those of the branch circuit for cylinder **18**. The pressure setting of each pressure reducing valve **130, 130'**, and **130''** determines the inital pressure in each branch circuit with the spring rate determined by the gas volume of the accumulators.

The action of the impact coupled embodiment of FIG. **16** for example, is illustrated in FIGS. **17** through **22**. These figures schematically illustrate the position of the various members of the system under different conditions of operation and at different stages in the operation. FIG. **17** illustrates the state of the apparatus while under power but prior to engagement of the ripping shank with the upper surface of the formation **20**. in this position, the ripping shank **15** is in its forward most position with the upper end of lever or link **17** against stop **120** with the lower end away from stop **123**. In this position, housing **21** has rotated about pivot **22** and has come against stop **124**, where even in this most forward position with the crank or cam shaft **25** rotating, the hammer **33** oscillates as shown in phantom without contact with the end of shank **15.** Thus the system is in essentially a neutral or idle position without any energy being transmitted to the ripping point.

FIG. **18** shows the apparatus wherein the hammer **33** has come into light contact with the shank **15** which has just been lowered into engagement with the formation **20.** Sufficient forward bias has been applied to the shank **15** by the transport vehicle to draw the upper end of link **17** away from stop **120** and not yet into engagement with stop **123** while at the same time a light blow by hammer **33** is insufficient to drive the housing **21** and energy storage means away from stop **124.**

Contact of the hammer **33** with shank **15** has driven the shank or ripping tip **19** forward into the formation as shown in FIG. **19** causing a fracture in the material. The impact against the shank **15** under load has caused pivoting of housing mechanism **21** about pivot **22** such that it has drawn away from contact with stop means **124.** Continued operation of the apparatus together with a consistent progressive movement into engagement with the formation will result in the system coming into a dynamic equilibrium such that the energy demanded of the system will be continuously delivered thereto as a result of blows by hammers **33** against shank **15.** The rearward motion of the shank **15** and movement of the housing **21** will be simultaneously cushioned by means of cylinders **18** and **23,** respectively.

As the ripping apparatus penetrates deeper into the formation, continued forward movement of the supporting vehicle will result in the ripping shank being pivoted further toward the rear. This will result in the hammer contacting the rear of the ripper shank sooner or even possibly prior to the engagement of link **17** with stop **123.** Thus, in effect, the greater the resistence encountered the further to the rear the shank **15** will pivot resulting in greater engagement of hammer **33** and consequently greater amount of energy delivered through the system to the ripper point **19.** This is illustrated in FIG. **20** with the shank forced backwards such that link **17** is in imminent contact with stop means **123.** At the same time, with housing **21** almost in the forward position against stop **24** the hammer **33** has come into engagement with the shank **15** at approximately a quarter turn of eccentric **30.** This results in a shank **15** being driven to a forward position as shown in FIG. **21** and the housing **21** being pivoted due to rebound to the rear as shown in this same figure. If the load on shank **15** has been too great, the housing **21** may have been caused to move sufficiently toward the rear that the hammer will not come into engagement with the shank **15** on the next forward stroke. This situation may be further aggravated by the fact that adjacent ripper shanks have experienced a similar concentrated blow pivoting the housing rearwardly. As the housing then approaches a state of equilibrium it moves back towards the shank, the hammer again contacts the rear of the ripper shank and repeats the above cycle. FIG. **22** illustrates the apparatus with the housing **21** pivoted to an extreme position to the rear such that, even with forward motion of the vehicle forcing shank **15** to the rear, the hammer **33** may not make contact with the shank **15.**

A proper arrangement of the components of this system can result in optimum performance of the system. For example, the mass of the components carried in housing **21** making up the energy storage means serves along with the housing **21** as the reaction mass for the blows struck by the hammer **33** against the shank **15.** Further reaction is imposed by means of the hydraulic system as applied to the hydraulic rams **18** and **23.** These forces together with an optimum frequency of operation and forward motion of the vehicle may be set for a given material such that blows are struck by hammer **33** on every cycle at an optimum frequency such that the maximum amount of energy is imparted to the ripper tip **19** for fracturing the formation. Thus with proper selection and control of parameters the system can achieve an equilibrium position automatically during operation by pivoting of housing **21** about pivot point **22** and pivoting of shank **15** about its support structure such that optimum engagement and delivery of energy for fracturing of materials is achieved.

The above described operation illustrates a significant feature of the present invention which permits it to be successful as a mechanical means of fracturing rock and other earth materials where other such mechanical systems have failed. This feature comprises a massive rotating flywheel incorporated in a housing **21** which has been referred to as the energy storage means and includes means for converting this rotary energy into reciprocatory energy which is applied ultimately to the ripper shank **15** only when demanded and in proportion to the demand. This can be readily observed from an inspection of FIGS. **17** through **22.** The mass of the apparatus contained in housing **21** constitutes reaction mass when energy is delivered to the ripper shank **15.** This reaction force increases as the housing **21** is forced to pivot further about pivot point **22** and thus increases the reaction and consequently the equal and opposite average force that is applied along the axis of shank **15** by means of hammer **33.** Thus it is clear that this system provides an automatic self regulating energy delivery system which delivers the amount of energy demanded along the shank **15** to the ripper tip **19** for fracturing materials such as formation **20.**

Referring now to FIG. **23,** there is illustrated a further embodiment of the present invention wherein like elements are identified by like numbers. In this embodiment, the energy is delivered to the ripper shank **15** by means of a hydraulic coupling or link **155.** This sytem performs much like a mechanical impact coupled system since the ripper shank is pulsed in only one direction when respective hydraulic ports are closed. This hydraulic link comprises a chamber **156** having a first piston **157** to which link **32** is coupled and a second piston **158** which is coupled by means of a link **159** to the shank **15.** The chamber or barrel **156** may be suitably supported such as by means of a bracket **160** from the housing **11.** Hydraulic fluid contained within the chamber between the pistons **157** and **158** transmits forces from piston **157** to piston **158.** The forces transmitted to the ripper shank will be in a unidirectional manner similar to that of the impact coupling arrangement described above. The hydraulic link means includes a hydraulic replenishing circuit which comprises a pump **161** for supplying fluid from tank **162** to chamber **156** by way of the conduits **163, 164, 165** and **166.** The circuit includes an accumulator **167,** a relief valve **168** and a check valve **169.** Relief valve **168** protects the circuit from high pressure surges.

When the shank **15** is lowered into engagement with the formation **20,** sufficient forward bias is applied to the shank by the transport vehicle to move the shank rearward. Rearward movement of the shank **15** moves the attached piston **158** rearward in chamber **156**, closing port **170** from conduit **166**. High pressure in chamber **156** will develop as crankshaft **25** rotates, moving piston **157** forward, closing port **171** from conduit **164**. Check valve **169** prevents fluid from returning to tank **162** by way of conduit **165**. The shank will be forced forward until port **170** is opened again. On the return stroke of piston **157**, fluid will flow back into chamber **156** through conduits **164**, **165** and **166**. Accumulator **167** provides for peak fluid demands.

Referring now to the embodiment of FIG. **24**, there is illustrated the hydraulic delivery concept as applied to an earth working tool such as a grader or dozer blade. In this embodiment a grader or dozer blade **172** is pivotally supported by means of a pair of links **173** and **174** from a support member **175** which is suitably mounted or carried by a vehicle (not shown). The instantaneous center **176** of links **173** and **174** becomes the effective pivot point for blade **172**. Links **173** and **174** are located such that the instantaneous center **176** is slightly ahead of the cutting edge **177**, resulting in essentially horizontal motion of cutting edge **177**. A hydraulic cylinder **178** may support the blade and frame **175** for suitable depth control. Suitable energy storage means are carried in a housing **179** and comprise a shift **180** carrying eccentric **181**. A hydraulic link comprising a cylinder **182** containing suitable hydraulic fluid is utilized to couple the energy being transmitted to the dozer blade **172**. This hydraulic link includes a link **183** journaled on eccentric **181** and coupled to a piston **184** which transmits force to the fluid in cylinder **182**. The fluid transmits the force to a second piston **185** which in turn is connected by a link **186** for delivery of energy to the blade **172**.

FIG. **25** illustrates an alternate embodiment of this apparatus wherein like elements are indicated by identical numbers. In this embodiment suitable energy storage means as in the previous embodiments are enclosed in a housing **187** and include a rotary shaft **188** coupled by means of an eccentric **189** to a link **190**. The link **190** is then coupled such as by pin **191** to the hammer **192**. In this arrangement the rotary energy stored in massive flywheels in housing **187** is converted to oscillatory motion which is transmitted by hammer **192** which is then transmitted by intermittent blows to anvil **193** and thereby to blade **172**. A suitable seal **194** seals the housing **187** around the link **190**.

Referring now to FIG. **26**, there is illustrated an alternate embodiment in which the energy storage means is coupled directly to a dozer blade **195**. This blade **195** is pivotally supported by means of links **196** and **197** to a support member **175**. In this embodiment the rotary shaft **188** is coupled as in the previous embodiment by means of an eccentric **189** to link **190** which is pivotally connected directly at **198** to the blade **195** to thereby transmit energy from shaft **188** in the form of oscillatory motion directly to the entire blade **195**.

These embodiments of the invention as applied particularly to a bulldozing blade has shown the ability to significantly improve the productivity of a given size tractor particularly under low or poor tractor effort conditions.

Referring now to FIG. **27**, there is illustrated an embodiment of the present invention as applied to the bucket ot a loader. In this embodiment a bucket **199** such as for a loader is pivotally supported as by links **200** and **201** to a support member **203** which is carried by control links **204** and **205** from a suitable vehicle (now shown). A housing **206** contains energy storage means as previously described which comprises a rotary shaft **207** connected by means of eccentric **208** to a link **209** which in turn is pivotally coupled at **210** to a hammer **211**. The hammer **211** is pivotally connected at **212** to the bucket **199** or suitable support means therefor and intermittently engages an anvil **213** attached to bucket **199** and for intermittently delivering vibratory energy thereto.

The embodiment illustrated in FIG. **28** comprises a suitable loader bucket **214** pivotally connected at **215** to a support member **215** which is carried by suitable control links **217** and **218** which are carried by a suitable vehicle (not shown). A suitable housing **219** contains suitable energy storage means including shaft **220** and eccentric **221** connected by link **222** by a pivot point **223** to the bucket **214**. In this embodiment the entire bucket **214** is oscillated about pivot point **215** by the application of oscillatory forces through link **222**.

While the present invention has been described and illustrated with respect to specific embodiments to which the present invention applies, it is to be understood that many modifications and changes may be made in the illustrated embodiments without department from the spirit and scope of the present invention as defined in the appended claims.

What is claimed is:

1. A mechanical system for imparting energy intermittently to an earth working implement, said system comprising:

an implement for separating portions from an earth formation;

means for supporting and manipulating said implement;

inertial energy storage means including a massive balanced flywheel system mounted on rotatable eccentric shaft means for storing high levels of inertial energy sufficient to meet the peak demands of said systems;

a prime mover for continuously delivering energy to said energy storage means; and

transmission means including an impact device and a substantially rigid element establishing a positive connection of said eccentric shaft means to said impact device for converting and intermittently transmitting said stored inertial energy to said earth working implement on demand.

2. The apparatus of claim **1** wherein said implement is a rock fracturing tool;

said tool is supported on a mobile vehicle for travel with the point thereof in engagement with a rock formation beneath the surface thereof to continuously separate portions from said formation along a path of travel of said vehicle; and,

said impact device is adapted to intermittently engage said tool at an angle from the surface of said formation.

3. The apparatus of claim **2** wherein said tool is supported so that the angle of attack with respect to said formation may be selectively varied.

4. The apparatus of claim **3** wherein said angle of attack is variable between 20° and 55°.

5. The apparatus as defined in claim **2** wherein said frequency of said impact is variable.

6. The apparatus as defined in claim 4 wherein said impact device is supported by a pair of links.

7. The apparatus of claim 6 wherein said links form a quadrilateral.

8. The apparatus of claim 6 wherein said implement is supported by means of a pair of links.

9. The apparatus as defined in claim 1 wherein said implement is a dozer blade.

10. The apparatus of claim 9 wherein the angle of attack of said blade is variable.

11. The invention of claim 9 wherein the vibratory energy is imparted to said blade at an angle from the surface of the formation.

12. The invention of claim 1 comprising movable reaction means including a housing pivotally supporting said flywheel and crankshaft on said frame for permitting said transmission means to move to an equilibrium position in response to load demands on said implement.

13. The invention of claim 12 wherein said reaction means is pivotable for movement of said transmission means and said energy storage means substantially along the path of stroke of said implement.

14. The invention of claim 12 including means for biasing said transmission means toward said implement.

15. The invention of claim 12 wherein the pivotal support of said housing to said frame is disposed upward and forward of the axis of said crankshaft so that said impact device swings toward said implement under the influence of the weight of said flywheel and crankshaft so that said reaction means varies automatically in response to variable load on said implement.

16. The invention of claim 15 wherein said reaction means comprises hydraulic control means.

17. The invention of claim 1 wherein said implement is supported centrally on said vehicle within boundaries defined by four main support wheels of said vehicles so that substantially the entire weight of said vehicle may be imposed on said implement.

18. The invention of claim 17 wherein said wheels are driving wheels.

19. The invention of claim 1 wherein the r.p.m. of said prime mover means and the frequency of said transmission means are variable to maintain a high velocity ratio of said implement to the forward speed of said vehicle.

20. The invention of claim 19 wherein said implement is supported for vibratory movement by means of a quadrilateral linkage such that the instantaneous center of pivoting for said implement is slightly ahead of the cutting edge thereof.

21. The invention of claim 20 wherein said implement is a rock fracturing shank.

22. The invention of claim 20 wherein said transmission means includes an impact coupling.

23. The apparatus as defined in claim 1 wherein said implement is pivotally supported to move into and out of position for engagement by said impact device under force reaction from said formation; and,

stop means operative to limit the movement of said implement into engagement with said impact device so that said eccentric means rotates at least one quarter of a revolution before engagement of said impact device with said implement.

24. The apparatus as defined in claim 23 wherein said implement is a rock ripping shank pivotally mounted on said supporting means and including a chisel shaped tip extending in a forward direction;

said impact means comprises a hammer pivotally mounted directly behind said shank for intermittent engagement there-with; and,

said rigid element comprises a rigid link journaled at one end on said eccentric shaft means and pivotally connected at the other end to said hammer.

25. The apparatus as defined in claim 24 wherein said hammer is pivotally mounted directly to said shank.

26. A dynamic mechanical system for applying vibratory energy to an implement for earth working, said system comprising:

an implement for separating portions from an earth formation;

a frame mounted on a mobile vehicle;

said implement being pivotally supported to said frame and movable to a position for engaging said earth formation;

stop means on said frame to engage and limit the pivotal movement of said implement away from said formation;

means including a massive balanced flywheel mounted on a rotary crankshaft for storing sufficiently high levels of inertial energy to meet the peak power demand of said system;

housing means pivotally mounted on said frame and rotatably mounting said flywheel and said crankshaft, said housing being pivoted to said frame on an axis above said crankshaft, and pivotal under the weight of said flywheel, said crankshaft, and said housing toward said implement to oppose bias of said tool away from said formation and thereby define reaction means;

a prime mover for continuously delivering energy to said energy storing means; and

transmission means including impact means operatively connected by a rigid link journaled to said crankshaft for converting and cyclically transmitting said stored energy to said earth working implement on demand to drive said implement toward said formation.

27. The invention of claim 26 wherein said energy storage means stores sufficient inertial energy to prevent more than a ten per cent variation in angular velocity of the crankshaft.

28. The apparatus of claim 26 wherein said implement is supported so that the angle of attack with respect to said formation may be selectively varied.

29. The apparatus as defined in claim 26 wherein the frequency of said impact is variable.

30. The apparatus as defined in claim 29 wherein said impact device and said implement are each supported by a pair of links.

31. The apparatus of claim 30 wherein each of said pairs links form a quadrilateral with the respective one of said impact device and said implement.

32. A mechanical system for imparting vibratory energy to an earth working implement, said system comprising:

an implement for separating portions from an earth formation;

means for supporting and manipulating said implement;

inertial energy storage means including a massive balanced flywheel mounted on a rotatable crankshaft for storing high levels of inertial energy sufficient to meet the peak demands of said systems;

a prime mover for continuously delivering energy to said energy storage means;

transmission means including an impact device operatively connected to said crankshaft for converting and intermittently transmitting said stored inertial energy to said earth working implement on demand;

movable reaction means including a housing pivotally supporting said flywheel and crankshaft on said frame; and

a pressurized fluid circuit operatively connected to bias said housing toward said implement for permitting said transmission means to move to an equilibrium position in response to load demands on said implement.

33. The invention of claim **32** wherein said circuit includes an accumulator.

34. A mechanical system for imparting vibratory energy to an earth working implement, said system comprising:

an implement for separating portions from an earth formation;

means for supporting and manipulating said implement;

inertial energy storage means including a massive balanced flywheel mounted on a rotatable crankshaft for storing high levels of inertial energy sufficient to meet the peak demands of said systems;

a prime mover for continuously delivering energy to said energy storage means;

transmission means including an impact device operatively connected to said crankshaft for converting and intermittently transmitting said stored inertial energy to said earth working implement on demand; and

movable reaction means including a fluid circuit and a housing pivotally supporting said flywheel and crankshaft on said frame for permitting said transmission means to move substantially along the path of stroke of said implement to an equilibrium position in response to load demands on said implement.

35. The invention of claim **34** wherein said circuit includes an accumulator circuit.

36. A mechanical system for imparting vibratory energy to an earth working implement, said system comprising:

an implement for separating portions from an earth formation;

means for supporting and manipulating said implement;

inertial energy storage means including a massive balanced flywheel mounted on a rotatable crankshaft for storing high levels of inertial energy sufficient to meet the peak demands of said systems;

a prime mover for continuously delivering energy to said energy storage means;

transmission means including an impact device operatively connected to said crankshaft for converting and intermittently transmitting said stored inertial energy to said earth working implement on demand;

movable reaction means including a housing pivotally supporting said flywheel and crankshaft on said frame for permitting said transmission means to move to an equilibrium position in response to load demands on said implement; and

means including a fluid circuit for biasing said transmission means toward said implement.

37. The invention of claim **36** wherein said circuit includes an accumulator.

38. A dynamic mechanical system for applying intermittent unidirectional energy to a rock fracturing implement, said system comprising:

an elongated fracturing shank;

means including a support frame for pivotally supporting said fracturing shank for vibratory movement;

a rotatable shaft including eccentric means;

means including a massive balanced flywheel mounted on said rotatable shaft for storing substantial amounts of rotative inertial energy; and

impact means comprising a rigid link journaled to said eccentric means to directly connect said eccentric means and said impact means to intermittently engage said shank for transmitting said energy from said flywheel to said shank in the form of high energy unidirectional blows.

39. The invention of claim **38** wherein the point of impact of said energy transmitting means with said shank includes a curved surface.

40. The invention of claim **39** comprising means to shield said impact area from foreign matter.

41. The apparatus as defined in claim **40** wherein said hammer means is pivotally mounted directly on said shank.

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