United States Patent [19]

Cooley

[11] **3,792,740**

[45] **Feb. 19, 1974**

[54] HYDRAULIC POWERED HAMMER

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[22] Filed: **Apr. 5, 1972**

[21] Appl. No.: **241,121**

[52] **U.S. Cl.** **173/119,** 173/126, 173/139

[51] **Int. Cl.** **B25d 9/02**

[58] **Field of Search** ... 173/116, 119, 120, 121, 126, 173/139; 92/28, 84, 134, 27

[56] **References Cited**

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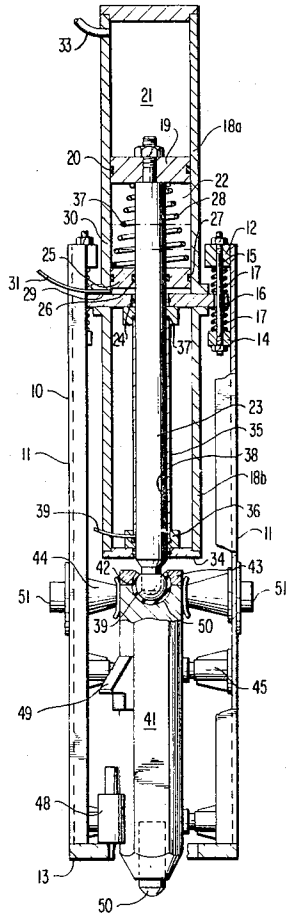
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Primary Examiner—Henry C. Sutherland
Assistant Examiner—William F. Pate, III
Attorney, Agent, or Firm—Edwin E. Greigg

[57] **ABSTRACT**

A gas spring-driven hydraulically cocked powered hammer wherein the hammer head is resiliently guided by liquid springs to absorb the energy of glancing blows and includes a ball joint disposed between the head and driving shaft, the driving piston being slidably fitted into a cylinder which is also resiliently mounted, the piston shaft and head being cocked by hydraulic pressure against a floating ring and being held in the cocked position by a friction lock while the ring is retracted, the friction lock then being released to fire the hammer with shock absorbers to engage the head and dissipate the energy should the head fail to strike its target.

14 Claims, 5 Drawing Figures



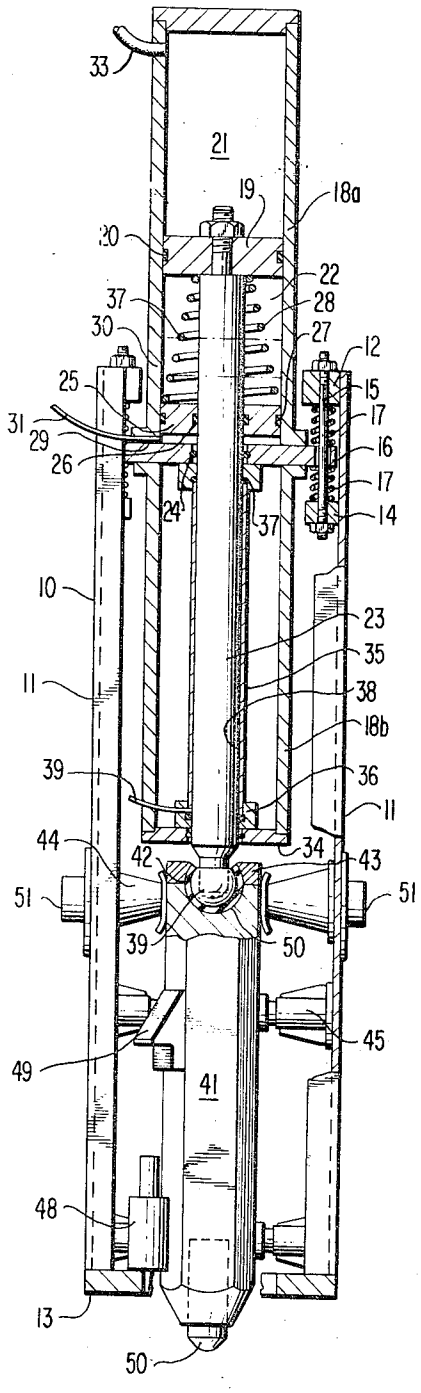

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FIG. 1

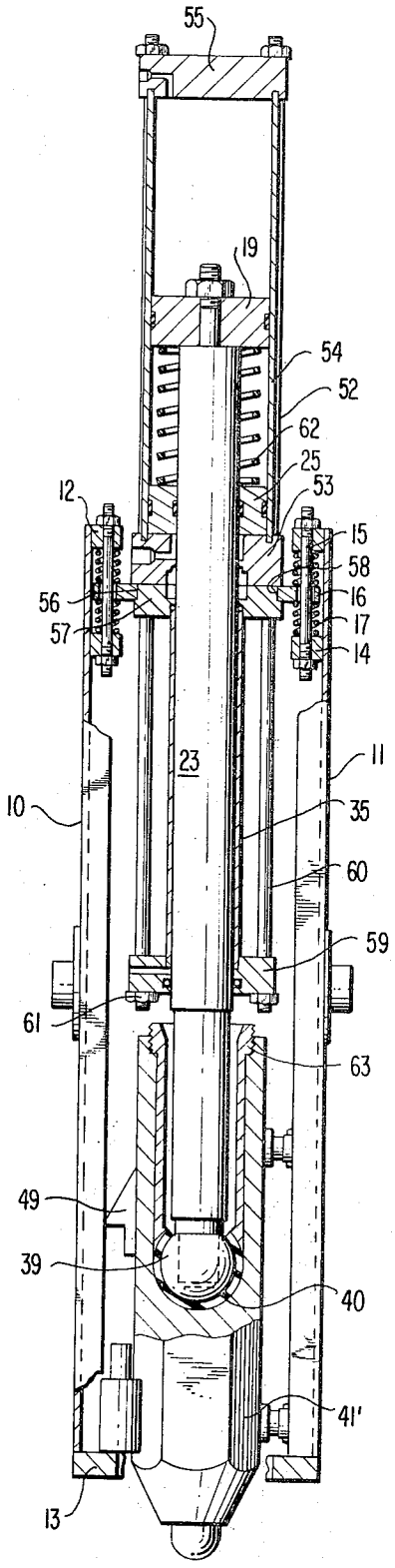

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FIG. 3

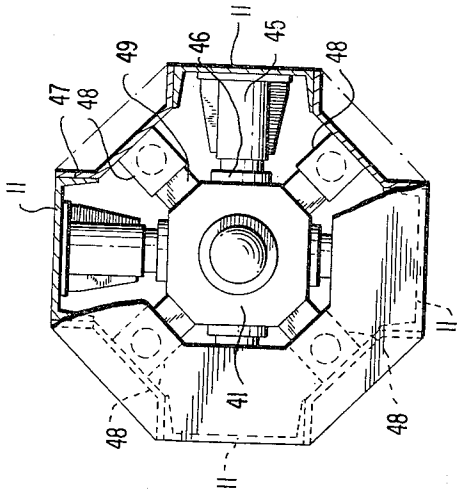


FIG 2

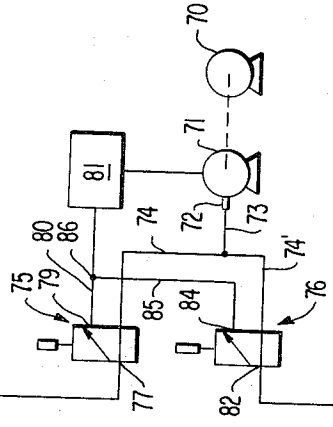


FIG 5

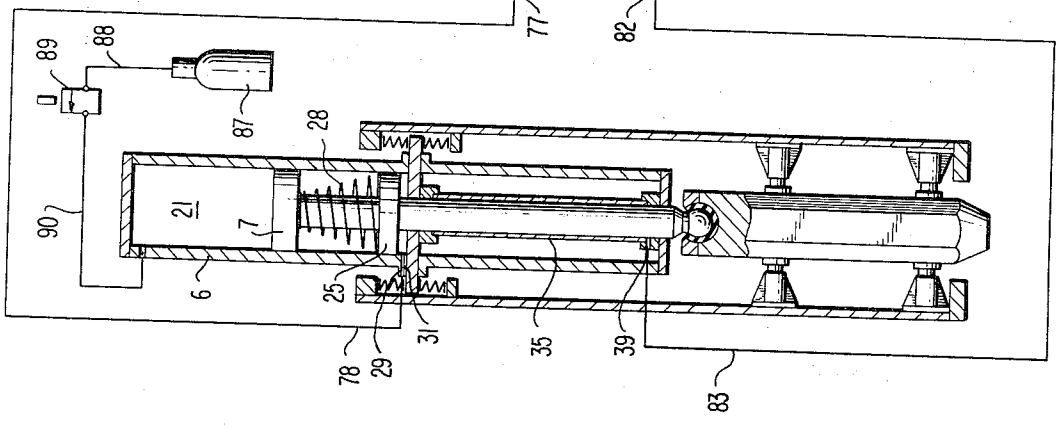
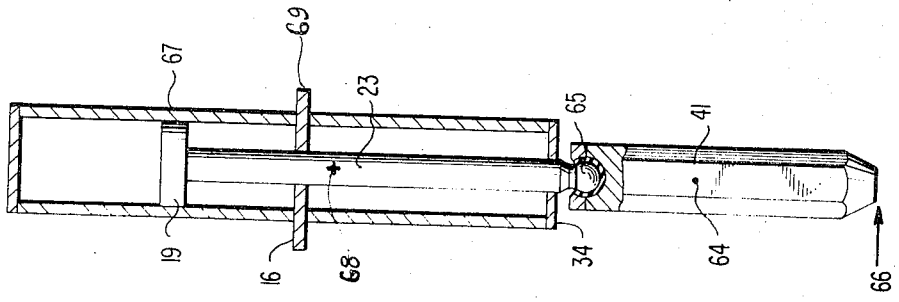


FIG 4



HYDRAULIC POWERED HAMMER

BACKGROUND OF THE INVENTION

Large hammers are required for tunneling, mining, quarrying and demolition work.

Hammers are now commonly used for breaking rock, concrete pavement, etc., and are operated in a variety of ways including the use of air, as well as hydraulic and electric motors for cocking the hammers and, in addition, air and mechanical springs for driving or imparting the energy to the head are also known as well. Most of the known hammers are limited in their size by failure of release mechanisms and other mechanical components. At the present time, those hammers which are available for commercial use are generally ineffective for mining, quarrying and heavy demolition work, particularly where a considerable amount of energy is required to rapidly break rock and other hard materials although there is already a need for hammers which are at least ten to twenty times the size of presently available commercial hammers. Designers have built, experimentally, larger hammers of the type for which there is a demand but they have had extreme difficulty in devising adequate release mechanisms and cocking mechanisms therefor, as well as have encountered difficulty in compensating in their structures for lateral shock which is ultimately transmitted to the hammer supports when the hammer strikes a glancing blow. Thus, short working life and reliability have been major obstacles which have yet to be overcome in order to produce a commercially feasible hammer design.

Several designs have evolved for cocking and driving large hammers which generally use high pressure oil to push a driving piston against a high pressure air spring, thereby storing energy in the spring which must then be released to obtain the energy to drive the hammer. This air spring has been found to be the most compact and efficient energy storing device, however, the oil used to cock and hold the driving piston must be throttled by passage through ports at high velocity to allow the piston to impart energy to the hammer head. This throttling of the hydraulic fluid wastes energy and therefore lowers the efficiency of the hammer, but in most cases the throttling of the fluid has been a necessity due to the lack of an adequate release mechanism. A large hammer for geophysical research has been devised utilizing a gas spring and a differential pressure release which worked well except that its reliability is dependent on selas and gas leakage which could inadvertently and unintentionally fire the hammer with potential danger.

Further, hammers have been devised with a connecting link non-rigidly connected to the head by a flexible joint and connected to the shaft of the driving piston by a similar joint, thereby providing a three-part system which permits lateral displacement of the head when it strikes a glancing blow, but also permits undue lateral displacement of the link during the driving phase, known as jack-knifing, resulting in undue side loads on the piston and driving shaft. To overcome this, Voitsekhovsky et al in U.S. Pat. No. 3,605,916, issued Sept. 20, 1971, use a single ball joint at the head and articulate the piston and its driving shaft thus eliminating the bearing support at the driving shaft to allow lateral displacement and further allowing the piston to rock within the cylinder. This three-part system, however, still has sealing difficulties and as a result the hammer must be removed from service to replace its seals.

The present invention overcomes the difficulties referred to herein by permitting the piston, cylinder and drive shaft to act as a single body thereby obtaining the advantages of a bearing support for the drive shaft as well as close fitting tolerances between the cylinder and piston and extended life for the seals used at these two critical points, while also retaining the two-part system to prevent jack-knifing during the power stroke. Angular rotation is permitted around the transverse axis of the cylinder and lateral displacement of the hammer head upon impact is also permitted thereby attenuating any destructive forces which otherwise may have been transmitted to the equipment and its supports.

It is therefore the primary object of this invention to provide a high energy gas spring-driven hydraulic hammer with high energy conversion efficiency and which is capable of providing long life.

It is a further object of this invention to provide a high energy, gas spring-driven hydraulic hammer which is capable of absorbing shocks and vibration which a high energy hammer must withstand without damage.

Still another object of this invention is to provide a high energy gas spring-driven hydraulic hammer which includes a lock to hold the hammer in a cocked position and release it without appreciable wear on the working surfaces.

A further object of this invention is to provide a high energy gas spring-driven hydraulic hammer which does not waste energy in throttling hydraulic fluid during the power stroke.

It is still another object of this invention to provide a high energy gas spring-driven hydraulic hammer in which the hammer head and piston drive shaft are flexibly connected and the drive cylinder system is flexibly mounted to the frame, so that lateral shock loads imposed on the hammer will be attenuated.

It is a further object of this invention to provide a high energy gas spring-driven hydraulic hammer with a flexible connection between the hammer head and the drive shaft which is located at the center of transverse rotation of the hammer head, associated with a center of percussion at the tip of the hammer head.

It is still another object of this invention to provide a high energy gas spring-driven hydraulic hammer in which the cylinder piston and drive shaft system is flexibly mounted to permit rotation in the frame around an axis which is located at the center of rotation associated with a center of percussion located at the point where the drive shaft is coupled to the hammer head.

It is a further object of this invention to provide a high energy hammer in which the mass of the head is supported perpendicularly to its axis by liquid springs to absorb side loads associated with glancing blows.

It is still another object of this invention to provide a high energy, gas spring-driven hydraulically cocked hammer which is cocked by a concentric floating ring.

These and other objects will become clear upon careful study of the following specification and drawings together with the appended claims.

DESCRIPTION OF THE DRAWINGS

FIG. 1 shows a partial elevational and partial cross sectional view through the axis of a preferred embodiment of the hammer;

FIG. 2 shows an end elevational view of the hammer with a portion of the housing removed to disclose the guides therefor;

FIG. 3 shows a partial elevational and partial cross sectional view of another embodiment of a hammer;

FIG. 4 shows schematically and in its simplest form the centers of percussion, gravity and rotation; and

FIG. 5 shows a cross sectional view of a hammer and a schematic of the manual control system therefor.

DESCRIPTION OF THE EMBODIMENTS

Referring now to FIG. 1, there is shown a support frame 10 comprising four metallic channel members 11 uniformly spaced around and rigidly affixed by welding or other means to an upper support ring 12 and lower perforated end plate 13 forming a rigid outer structure and through which the hammer components are mounted. Spaced from upper support ring 12 and welded to the internal surface of the channel members 11 is a spring support 14 which is perforated to receive guide bolt 15 which also extends through suitable perforations in the upper support ring 12 so that said guide bolt is supported parallel to the axis of frame 10. Centrally positioned between the upper support ring 12 and spring support 14 is a support plate 16 of the drive system that is suitably shaped with four perforated radially extending ears which are associated with channel members 11, the arrangement being such that the perforations are in alignment with guide bolt 15 and so arranged that the drive system support plate 16 is afforded a degree of rotational clearance. Slidably associated with guide bolt 15 and disposed between drive system support plate 16 and upper support ring 12 and between drive system support plate 16 and spring support 14 are compression springs 17 which retain the drive system support plate centered between the upper support ring 12 and the lower spring support 14. Extending upwardly from the drive system support plate 16 and rigidly affixed thereto is a cylinder 18 with a piston 19 which is provided with seal means 20 and slidably arranged in said cylinder and adapted to divide the cylinder into upper and lower chambers 21 and 22, respectively. Rigidly attached to and concentric with said piston 19 and extending downwardly therefrom is a shaft 23 which passes through and is slidably associated with a suitable perforation in the support plate 16 and sealed relative thereto by seal means 24. Concentric to and slidably associated with chamber 22 of cylinder 18 is a floating ring or follower 25 provided with seals 26 and 27, respectively. The ring 25 is normally held against the support plate 16 by a conical spring 28 interposed between the floating ring 25 and the piston 19. Cylinder 18 is associated with a hydraulic conduit 29 which is suitably sealed and extends through means defining an opening in wall 30 and thereby allows communication between the hydraulic line 31 and a space between the drive system support 16 and the floating ring 25. Thus, when a hydraulic pressure is applied, the floating ring 25 is forced upwardly against the resistance of spring 28 and if the piston 19 is in its lower position 32 with the spring under compression by reason of gas pressure that is later released, then the spring flattens completely, allowing the floating ring to raise the piston to its upper position, as shown. The cylinder 18 is tapped, as shown, and arranged to receive a conduit 33 to provide for introduction of high pressure gas into the upper chamber 21, this closed volume of gas functioning as the means for driving the piston 19.

Extending downwardly from the drive system support plate 16 and concentric with shaft 23 is a cylindrical housing 18*b* which is rigidly affixed to the drive system support plate 16 by bolts or other suitable means, not shown. Affixed to the lower end of the cylindrical housing 18*b* is an end closure means 34 which is concentrically perforated to allow shaft 23 to extend therefrom. Shaft 23 is arranged to receive telescopically a locking sleeve member 35 which is hydraulically actuated and the opposite ends of which are supported by seals positioned in ring members 36 and 37, respectively. The ring members are supported at their opposite ends by the support plate 16 and closure means 34. The hydraulic lock sleeve 35 is actuated by force introduced through the conduit to the fitting 39 this fluid communicating with tightly juxtaposed surfaces of shaft 23 and the surrounding sleeve 38 to thereby expand the sleeve when hydraulic pressure is applied and contract when pressure is relieved. This type of hydraulic lock structure is generally disclosed in U.S. Pat. No. 3,150,571 and marketed under the name of "Bear Loc." However, other types of hydraulically actuated mechanical locks which grip the cylindrical shaft may also be used.

Integral with the lower end of drive shaft 23 is a ball 39 which is received in socket 40 provided in the upper end of the hammer head 41. The socket 40 contains cushioning material 42, the ball 39 being axially restrained in hammer head 41 by a ring 43, which is bolted or otherwise attached to the upper end of hammer head 41. Firmly attached to the frame 10 and beneath the closure plate 34 are four rigid guide members 44 (see also FIG. 2) with which the upper end of the hammer head 41 is slidably associated and concentrically aligned with the drive shaft 23 when the piston is at its uppermost position as shown, thereby preventing jack-knifing upon initial acceleration of the head 41. Also, firmly attached to frame 10 and spaced to slidably support the hammer head 41 as it moves downwardly are liquid spring guide means 45 the function of which is known in the art; these guide means are adapted to provide support under high lateral loads. These guide means are better shown in the end view in FIG. 2.

Referring now to FIG. 2 there is shown an end elevational view of the hammer head 41 which is octagonal in configuration so that four alternate sides form flat surfaces which are adapted to be complemental with the shoe 46 of the aligned guide members 44 and 45, these guide members being attached to the center of each of the four frame channel elements 11, so as to restrain the hammer head in all lateral directions. Also clearly shown in this view and positioned between each of the guide members 45 and affixed to supporting members 47, these being welded or otherwise affixed to frame channel elements 11, are shock absorbers 48 which are provided at the lower end of frame 10, as best shown in FIG. 1. Also, positioned in equal spaced arrangement about the hammer, as shown at 49, are ears which are adapted to cooperate with shock absorbers 48 and to thereby absorb the shock near the end of the hammer stroke.

Referring back to FIG. 1, there is shown in this view in dotted outline, a bore provided in the end of hammer head 41, and into which is positioned a tungsten carbide tip 50 to increase resistance to wear when working

on hard materials. Also shown affixed to frame channel elements 11 are two trunnions 51, by means of which the hammer is supported, allowing it to be tilted from a vertical position during operation.

OPERATION

The hammer operates as follows: chamber 21 is pressurized with high pressure gas and acts as a spring. High pressure hydraulic fluid is admitted through conduit connection 29 to the lower side of floating ring 25 which is then forced upwardly against spring 28 and piston 19, thereby collapsing the spring 28 and raising the piston 19, shaft 23 and hammer head 41 to the cocked position, as shown in FIG. 1, thus increasing the gas pressure in chamber 21. During the cocking cycle high pressure oil is also admitted through conduit 39 to the hydraulic lock 35 to release it from engagement with the shaft and thereby allowing shaft 23 to slide freely upwardly. At the top of the stroke hydraulic pressure is removed from both the lock 35 and the cavity below floating ring 25 to allow the hydraulic lock 35 to grip the shaft 23, to thereby retain the hammer in its cocked position and to allow the spring 28 to expand, so as to force the floating ring 25 downward to its lowermost position, and to expel the oil from below it.

To fire the hammer, high pressure hydraulic oil is applied through hose 39 to hydraulic lock 35 to release it, allowing the energy stored in the compressed gas in chamber 21 to expand, driving the piston 19 downwardly and imparting energy through shaft 23 to the hammer head 41. As the piston is driven downwardly, it compresses the spring 28 and the gas in chamber 22, thereby dissipating some of its energy. To overcome this, the shaft 23 may be axially drilled to vent the chamber 22 to the outside, thus eliminating the compression of the gas that is otherwise trapped in chamber 22. Should the hammer head strike a glancing blow, it may deflect sideways with this resultant energy being absorbed by liquid springs 45, while the lateral force applied to the ball joint will cause the cylinder and drive system to rotationally deflect against the support springs 17 attenuating the shock load on the drive components.

Referring now to FIG. 3, there is shown a frame 10 constructed of four channels 11 joined together at their upper end by upper support ring 12 and at the lower end by lower end plate 13 as previously described in FIG. 1. The drive system 52 is supported by the drive system support plate 16 in frame 10 by support springs 17, guide bolts 15 and support block 14, as described before with the drive system support plate 16 being located at the center of rotation of the mass of the drive system including the cylinder, lock and drive shaft for purposes to be presently described. Extending upwardly from plate 16 is the lower cylinder sealing plate 53 which has an annular groove facing upwardly and into which the lower end of cylinder wall 54 is positioned and suitably sealed and is provided with a central perforation through which the shaft 23 is slidably positioned and sealed, the upper end of said cylinder being capped by a similar plate 55 with a similar annular groove and seal for the cylinder. Extending below the lower cylinder sealing plate 53 is a hydraulic lock 35 with an annular groove 56 which is machined into the upper outer corner of the hydraulic lock end plate 57 to fit into concentric opening 58 in drive system support plate 16 to thereby hold it rigidly between shoulder 56 and lower cylinder seal plate 53 when they are assembled. Lower hydraulic lock end plate 59 is perforated to receive a number of tie bolts 60 which extend upwardly through similar perforations, suitably aligned in the upper hydraulic lock end plate 57, lower cylinder seal plate 53 and upper cylinder end closure plate 55, the upper and lower ends of tie bolts 60 being threaded and fitted with nuts 61 to draw the whole assembly tightly together as tie bolts 60 are placed under tension. Slidably positioned within cylinder 54 is the piston 19 and the floating ring 25 as explained earlier with a square cross section spring 62 being disposed between piston 19, and the floating ring 25 to thereby urge the floating ring towards the lower cylinder seal plate 53. Shaft 23 extends downwardly from piston 19 through hydraulic lock 35 and into the hammer head 41 where it terminates with an integrated ball 39 that is received in a suitable socket 40 cushioned by shock absorbent material 42 and positioned at the transverse center of rotation of the hammer head 41. The ball 39 is retained in socket 40 by a sleeve nut 63 which is threaded into the upper end of hammer head 41, as shown. The support system for the hammer head is similar to that previously described in connection with FIG. 1 and therefore need not be repeated. The rigid slide supports 44 have not been shown in this view through they may be used for better alignment if necessary.

The operation of this embodiment is identical to that described in the first embodiment. When the floating ring 25 is moved upwardly to cock the hammer, it compresses the spring 62 and thereafter jacks the piston through the spring to provide a volume of low pressure gas between the floating ring 25 and the piston 19.

By placing the ball joint 39 in the hammer head 41 at the center of transverse rotation thereof and the drive system support plate 16 also at its center of transverse rotation, the lateral forces on the supports (due to a glancing blow) are minimized.

Referring now to FIG. 4, there is shown schematically a hammer head 41 with its center of gravity at point 64 and the pivotal spherical joint at point 65. The position of point 65 is determined by the relation

$$ab = K_g^2$$

where a is the axial distance from the tip of the hammer head where the transverse forces 66 are applied and the center of gravity point 64, and b is the axial distance from the center of gravity point 64 to the center of rotation point 65, and K_g is the radius of gyration of the hammer head 41 about its transverse axis through its center of gravity.

Attached at point 65 is the spherical ball joint which is rigidly affixed to the shaft 23, which is slidably fitted through the lower cylinder seal closure plate 34 of FIG. 1 and therefore able to transmit lateral loads to the entire drive system assembly which represents the entire mass of all components which are flexibly mounted to the frame.

In a similar manner, the center of gravity of the drive system is point 68 and its center of rotation is at point 69 which is the point at which the drive system support plate 16 must be located. The lateral load is applied at the spherical joint point 65. The position of the center of rotation is determined by the relation

$$ab = K_g^2$$

in which a in this case is the axial distance from the

spherical ball joint **65** to the center of gravity of the system point **68**, and *b* is the distance from the center of gravity point **68** to the center of rotation point **69**, and K_g is the radius of gyration of the whole system **67** including the extended shaft **23**.

The dimensions to be used are those that occur at the moment the target is encountered.

The control system required is relatively straight forward and is shown schematically in FIG. **5** where there is included a prime mover **70** driving high pressure hydraulic pump **71**. From the high pressure outlet **72** of pump **71** hydraulic line **73** divides, one line **74** communicating with the inlet port of cocking valve **75** and the other line **74'** communicating with the inlet port of fire valve **76**. The high pressure outlet port **77** of cocking valve **75** communicates through line **78** with conduit connection **29** of FIG. **1** which communicates with the lower side of the floating ring **25**. The return port **79** of cocking valve **75** communicates through line **80** with oil reservoir **81** to thereby return low pressure oil to be used again by the pump.

The high pressure outlet port **82** of fire valve **76** communicates through line **83** with conduit connection **39** of FIG. **1** which, in turn, communicates with the release surface of the hydraulic lock **35**. The return port **84** of the fire valve **76** communicates through line **85** with the return line **80** at point **86**, thus returning low pressure oil to the reservoir.

The high pressure gas spring is replenished when necessary through the use of high pressure nitrogen contained in bottle **86** which communicates through line **88** with replenishing valve **89** whose outlet port communicates with chamber **21** through line **90**, thus when the gas pressure in chamber **21** needs to be increased, valve **89** is manually opened.

It is believed from the foregoing to be clear to those skilled in the art that to cock and fire the hammer as previously described, first the fire valve **76** is operated to apply high pressure oil through line **83** to release lock **35**, then cocking valve **75** is operated to apply high pressure oil through line **78** to the floating ring **25** which pushes piston **19** upwardly to its cocked position, then the fire valve **76** is operated to vent oil from line **83** to line **85**, thus applying hydraulic lock **35** and then the cocking valve **74** is operated to vent oil from line **78** to line **80** releasing the pressure from below the floating ring **25** to thereby allow it to be returned to its lower position by the spring **28**. The hammer is now cocked and ready to fire. To fire the hammer, the fire valve **76** is operated to apply high pressure oil to line **83** to thereby release hydraulic lock **35** and to allow piston **19** to be driven downwardly by the high pressure gas in chamber **21**.

It is also believed apparent to those skilled in the art that the hammer control system may be made automatic by the use of limit switches or sensors to sense the position of the hammer head or piston and perform the appropriate actions as described above so that a series of repetitive blows may be delivered without manually operating the fire and cocking valves.

What is claimed is:

1. A hydraulic hammer comprising an elongated frame member having upper and lower extremities, guide means carried by the lower extremity of said frame serving to align a hammer head axially of said frame member, cylinder means, having upper and lower closure members, supported at the upper extremity of said frame member, piston means in said cylinder dividing said cylinder into upper and lower chambers, said piston means including shaft means extending from the lower closure member and terminating in a ball member seated in a socket member in said hammer head, lock means for controlling reciprocable movement of said shaft, resilient support means mounted to said frame member for resiliently and deflectably supporting said cylinder means; said cylinder means, said piston means, said shaft means and said lock means being displaceable as a unit about said socket member as a result of said resilient support means, and power means communicating with the lower chamber arranged for first advancing the piston into a cocked position and thence to release said power means to transmit force to said hammer head.

2. A hydraulic hammer as claimed in claim **1**, wherein resilient means are interposed between the ball and socket members.

3. A hydraulic hammer as claimed in claim **1**, wherein the socket member provided in said hammer head is disposed substantially medially of the extent of said hammer head and the ball member terminates therein.

4. A hydraulic hammer as claimed in claim **3**, wherein a sleeve member retains said ball member in said socket member.

5. A hydraulic hammer as claimed in claim **1**, wherein the frame member includes trunnion members.

6. A hydraulic hammer as claimed in claim **1**, wherein the lower closure member comprises a hydraulically actuatable follower.

7. A hydraulic hammer as claimed in claim **1**, wherein the lower closure member comprises a hydraulically actuatable follower and a spring means is interposed between said follower and said piston.

8. A hydraulic hammer as claimed in claim **1**, wherein said lock means is hydraulically actuated.

9. A hydraulic hammer as claimed in claim **8**, wherein the hydraulically actuated lock means for said shaft is disengaged therefrom when the piston is advanced into a hydraulically cocked position.

10. A hydraulic hammer as claimed in claim **1**, wherein said guide means are horizontally aligned and disposed in parallel planes.

11. A hydraulic hammer as claimed in claim **1**, wherein the ball and socket members intersect at the center of rotation of said hammer head.

12. A hydraulic hammer as claimed in claim **11**, wherein said center of rotation of said hammer head is associated with a center of percussion at the hammer tip.

13. A hydraulic hammer as claimed in claim **1**, wherein the cylinder support is placed at the center of rotation of a drive system including said cylinder means, said upper and lower closure members, said piston means, said shaft means and said lock means when a transverse shock load is applied to the ball and socket members.

14. A hydraulic hammer as claimed in claim **1**, wherein the guide means comprise liquid springs.

* * * * *