June 1, 1965 J. A. ROLL 3,186,498

IMPACT TOOL

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Fig. 1

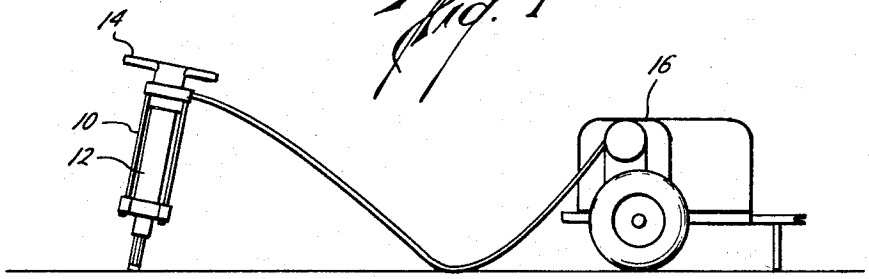


Fig. 9

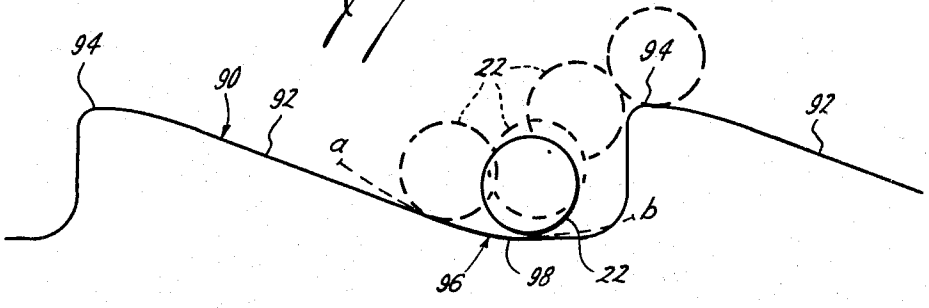
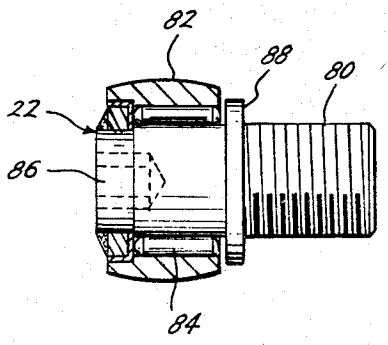


Fig. 10



Jack A. Roll
INVENTOR.

BY James F. Weiler
Jefferson O. Gilker
William A. Stout
Paul L. DeVorter II
ATTORNEYS

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IMPACT TOOL

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Fig. 2

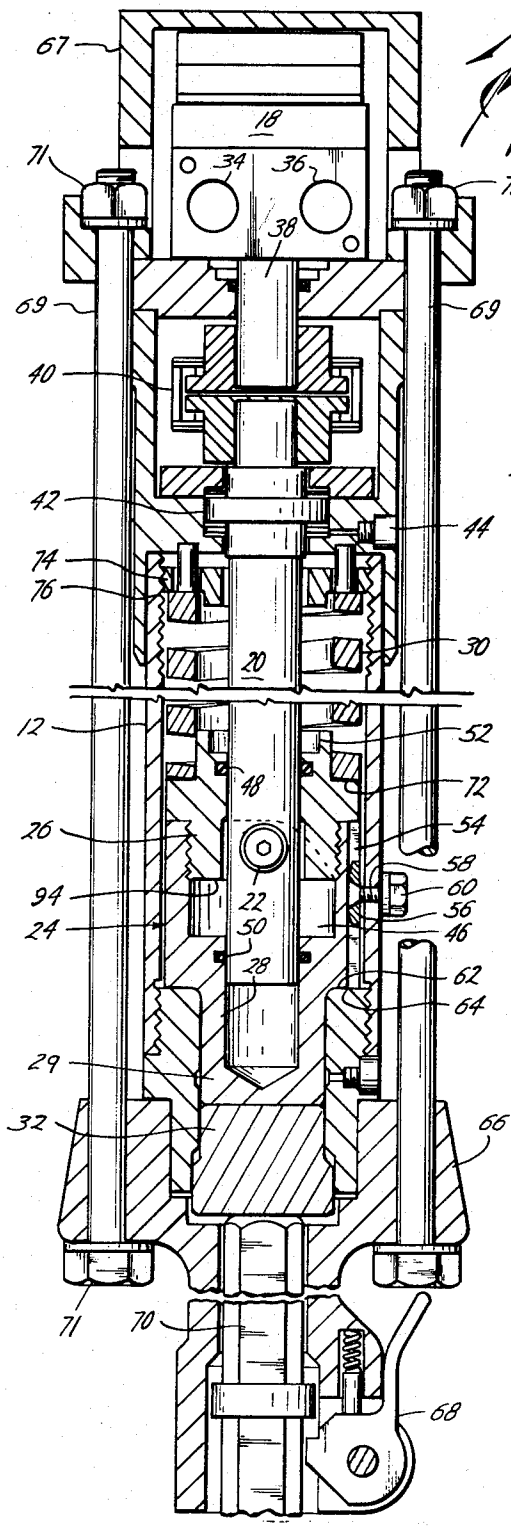
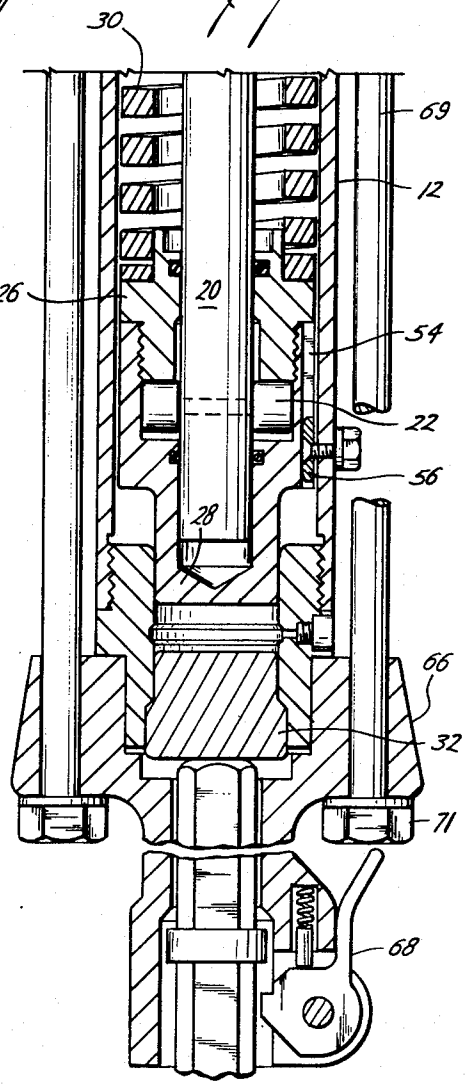


Fig. 3



Jack A. Roll
INVENTOR.

BY James F. Weiler
Jefferson D. Geller
William A. Stout
Paul L. DeVenter II
ATTORNEYS

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IMPACT TOOL

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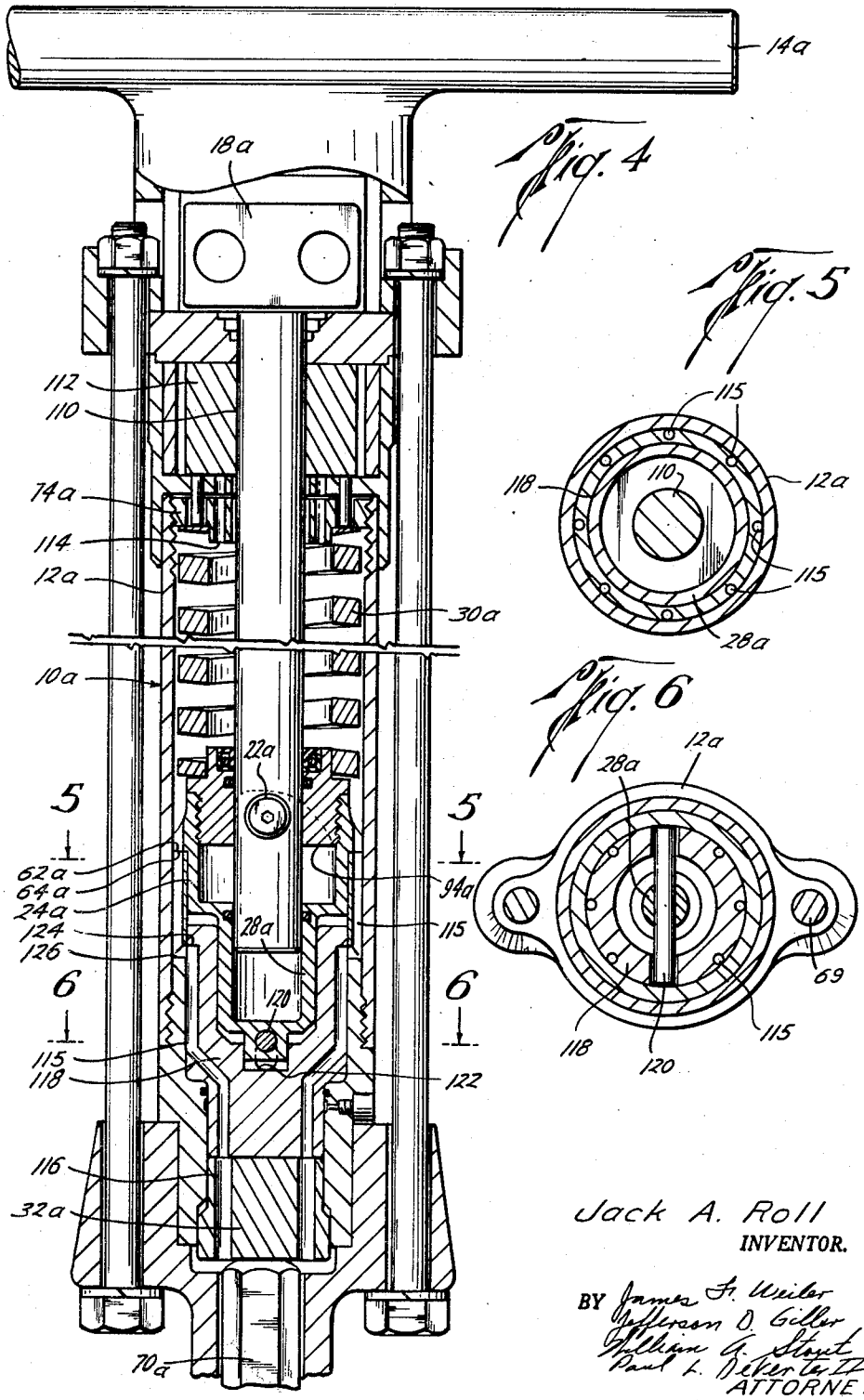


Fig. 4

Fig. 5

Fig. 6

Jack A. Roll
INVENTOR.

BY James F. Weiler
Jefferson O. Giller
William G. Stout
Paul L. Dekertegg
ATTORNEYS

June 1, 1965 J. A. ROLL 3,186,498

IMPACT TOOL

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Fig. 7

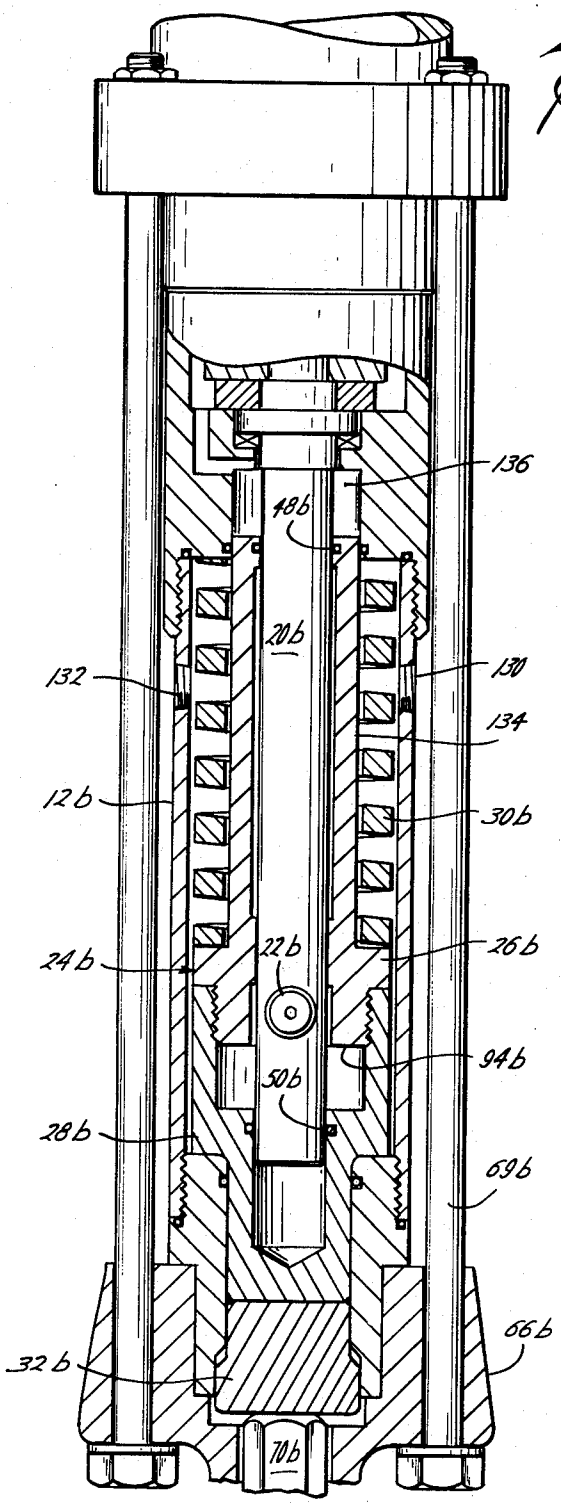
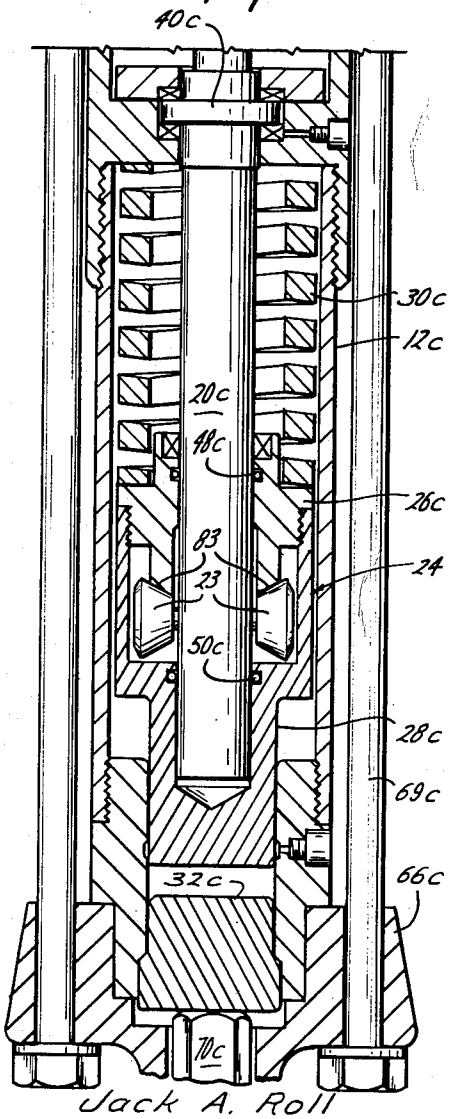


Fig. 8



Jack A. Roll
INVENTOR.

BY James F. Weiler
Jefferson D. Gilley
William G. Stout
Carl L. DeVerter II
ATTORNEYS

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IMPACT TOOL

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Fig. 11

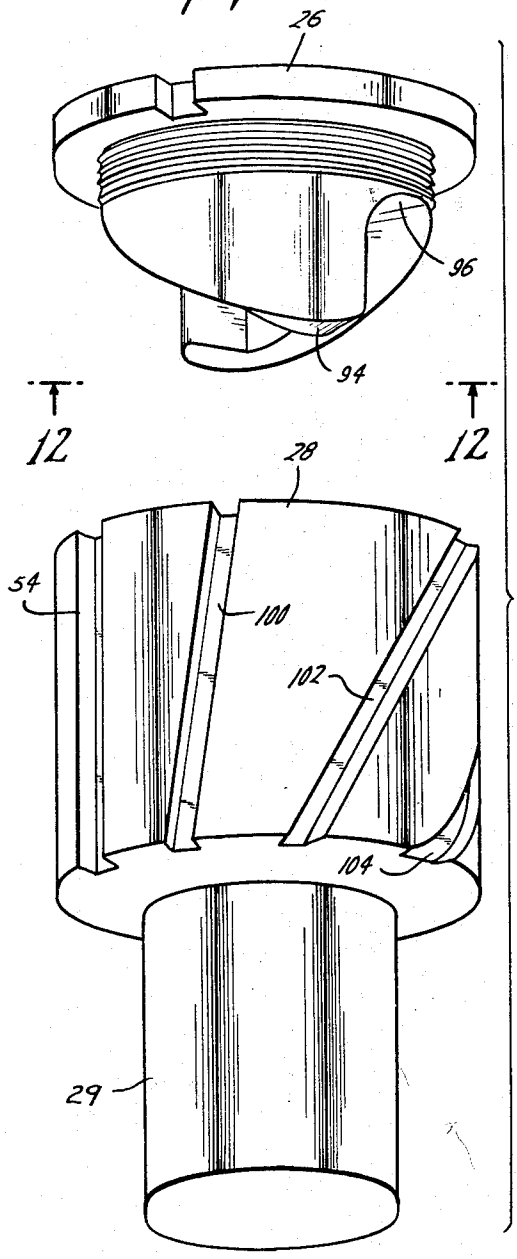
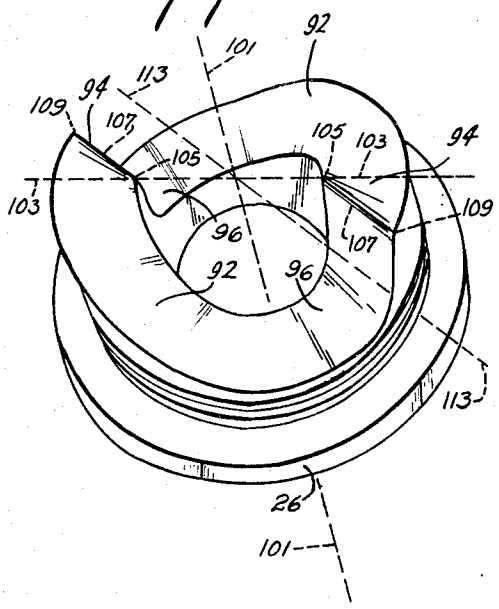


Fig. 12



Jack A. Roll
INVENTOR.

BY James H. Miller
Jefferson O. Gilley
William A. Stout
Paul L. DeVerter II
ATTORNEYS

June 1, 1965 J. A. ROLL 3,186,498

IMPACT TOOL

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Fig. 13

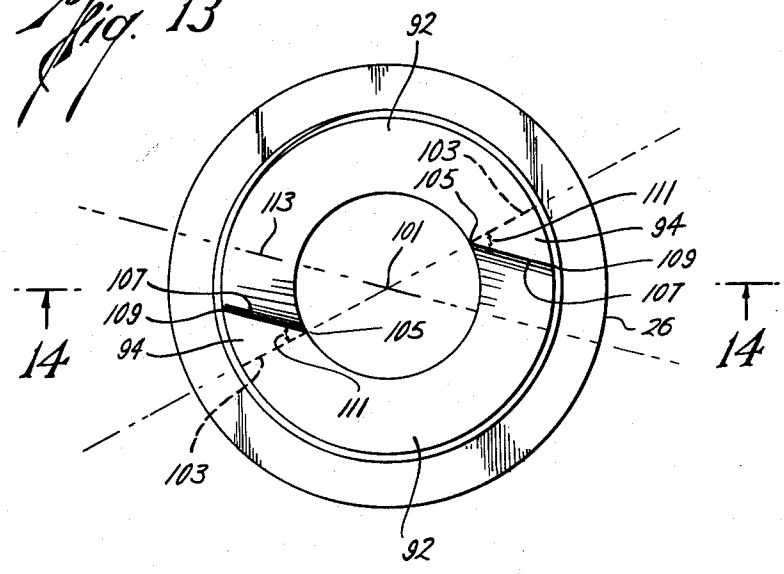
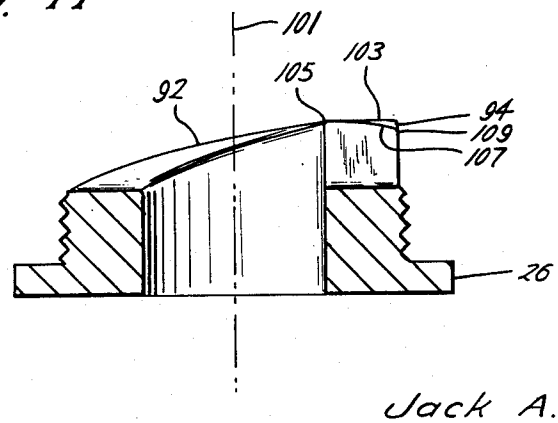


Fig. 14



Jack A. Roll
INVENTOR.

BY James F. Weiler
Jefferson D. Gilley
William A. Stout
Paul L. DeVerter III
ATTORNEYS

United States Patent Office 3,186,498 Patented June 1, 1965

3,186,498

IMPACT TOOL

Jack A. Roll, Bryan, Tex., assignor to Albritton Engineering Corporation, Bryan, Tex., a corporation of Texas

Filed Oct. 2, 1961, Ser. No. 142,393
6 Claims. (Cl. 173—123)

The present invention relates to improvements in an impact tool, and more particularly, relates to a power rotary actuated hammer using a spring and cam mechanism to convert the rotary motion to an impact motion.

Most impact power tools of the heavy duty type available to the trade today are of the pneumatic type. The pneumatic air hammers are disadvantageous in that they are relatively large and complicated in design, have a high noise level, are limited by design to constant speed and impact blow, are highly inefficient in that the air under pressure is vented to the atmosphere in use, and the auxiliary equipment required to supply a sufficient amount of air pressure, for example, the air compressors, are quite expensive.

The principal of using a spring and cam mechanism to convert rotary motion to that of a longitudinal impact motion has been used in the past, such as electric motor driven impact drills and the like. However, this type of impact hammer has been restricted to applications requiring little force and has not been suitable for impacting and breaking up hard formations, such a rock, pavement, concrete walks and the like.

It would be highly advantageous to provide an impact tool for heavy duty use in which considerable destructive force is produced, which is highly efficient and which is of a comparatively small initial and operative cost.

Therefore, it is a general object of this invention to provide an impact tool which is operated by a rotary motor to provide a sufficiently powerful and efficient force for heavy duty use, for example, for breaking up very hard formations such as rock, concrete, paving, asphalt, and the like as well as for drilling, chipping, impact driving and the like.

It is therefore an object of the present invention to provide an improved power impact tool which overcomes the previously mentioned disadvantages of other power impact hammers by providing a rotating hydraulic motor which drives a cam spring impact mechanism to provide a longitudinal impact of considerable magnitude.

However, in the past, the use of the cam spring impact mechanism has not been used for heavy duty impacting as there was no structure capable of a prolonged and rapid transmission of sufficient energy to perform heavy duty work.

Therefore, a still further object of the present invention is to provide an improved power impact tool which utilizes heavy load bearing antifriction roller bearing cam followers in order to provide a high-speed heavy duty structure capable of indefinite usage without appreciable wear or breakdown.

Yet a further object of the present invention is the provision of a cam surface which prevents shock loads and/or point pressure contact from being applied to the roller bearings followers by insuring that the cam surface provides and maintains a complete full line contact between the outer race of the bearing followers and the cam surface at all times when they are in contact and under load.

Still a further object of the present invention is the provision of an improved impact tool having a cam spring mechanism wherein shock to the bearing followers is further prevented by providing a cam incline surface which picks up and contacts the bearing race as gradually as possible and which has sufficient rotary clearance to insure that the bearing followers do not strike the cam on downward impact of the hammer.

Still a further object of the present invention is the provision of an impact spring cam mechanism whereby tendency to wear between the bearing followers and the cam surfaces is reduced by crowning the outer race of the bearing followers.

Yet a further object of the present invention is the provision of an impact tool which eliminates slippage and possible wear between the cam surfaces and bearing followers by providing a tapered outer bearing race and a corresponding matching tapered cam slope.

Yet a further object of the present invention is the provision of a hydraulic impact tool in which the critical moving mechanism is sealed in a bath of lubricant to provide efficient lubrication for the moving parts.

Still a further object of the present invention is the provision of a power impact tool which is provided with an adjustable spring collar to preload the spring so as to suitably adjust the impact of the tool.

Yet a still further object of the present invention is the provision of an improved power impact tool in which the cam may be rotated a desired amount to compensate for the continued rotation of the bearing followers and thus prevent the bearing followers from striking the cam surface even at high speeds after passing over a cam shoulder, which will permit rapid blows with a light force if desired as it is not necessary to depend entirely upon the spring to return the hammer rapidly enough for the followers to return above the clearance surface of the cams.

Yet a still further object of the present invention is the provision of an improved impact tool wherein a hydraulic balance within the housing is provided so that a flow of hydraulic fluid under pressure may be passed through the housing without any pressure effect on the hammer thereby providing cooling and/or lubrication for the entire mechanism.

A still further object of the present invention is the provision of an improved power impact tool having an air pump which provides an air supply which is desirable in certain applications to blow out dust and provide cooling for a unit in operation.

Still a further object of the present invention is the provision of an improved impact tool which is provided with a free fall hammer in which the inertia of a movable hammer element provides a striking blow independent of the spring and cam mechanism thereby permitting ideal cam design so as to minimize load pickup stress to the bearing followers.

Other and further objects, features and advantages will be apparent from the following description of presently preferred embodiments of the invention, taken in conjunction with the drawings, in which like character references designate like parts throughout the several views, and where:

FIGURE 1 is an elevational view illustrating a hydraulic impact tool according to the present invention,

FIGURE 2 is an elevational view, in cross-section, illustrating the impact tool of FIGURE 1 in the position of actual impact,

FIGURE 3 is a fragmentary elevational view, in section, illustrating the device of FIGURE 2 in which the tool is in a cocked position,

FIGURE 4 is an elevational view, in cross-section, illustrating a modified impact tool of the present invention utilizing a free fall hammer and having an air pump,

FIGURE 5 is a cross-sectional view taken along the line 5—5 of FIGURE 4,

FIGURE 6 is a cross-sectional view taken along the line 6—6 of FIGURE 4,

FIGURE 7 is a fragmentary elevational view, partly in section, illustrating the impact tool of the present invention being hydraulically balanced and provided with hydraulic cooling inlet and outlet,

FIGURE 8 is a fragmentary elevational view, partly in cross-section, illustrating a modified hydraulic impact tool in which the bearing followers and cam surfaces are tapered,

FIGURE 9 is a layout of the cam design showing the action of the bearing followers therein,

FIGURE 10 is an elevational view, partly in cross-section, illustrating the construction of one of the bearing followers,

FIGURE 11 is an exploded view of the hammer assembly of the impact tool of the present invention,

FIGURE 12 is a perspective view taken along the line 12—12 of FIGURE 11.

FIGURE 13 is a plan view of the cam of the present invention, and FIGURE 14 is a cross-sectional view taken along the line 14—14 of FIGURE 13.

Referring now to the drawings, and particularly to FIGURE 1, the impact tool of the present invention is generally designated by the reference 10 and includes a housing 12. The tool 10 may be powered by any convenient rotary acting motor, but preferably a hydraulic motor is used as will be presently described. In this event the handles 14 which are secured to the housing 12 may include a conventional valve control (not shown) for controlling the rate of flow of hydraulic fluid to the hydraulic motor. The hydraulic impact tool 10 is connected to a suitable source of hydraulic power for actuating the hammer, here shown as a conventional gasoline engine driven hydraulic pump 16, although the hydraulic impact tool 10 can be suitably connected to any available hydraulic energy source such as provided on conventional tractors, gradalls, backhoes, and the like.

Referring now to FIGURE 2, the housing 12 generally encloses, protects, and connects the various working components of the impact tool. Basically, the impact tool includes a rotary motor 18, preferably hydraulic, which rotates a driving shaft 20 and which includes a pair of roller bearing cam followers 22. An axially movable hammer assembly 24 includes an impact or hammer element 28 and a cam element 26 which coacts with the followers 22. A spring 30 acts against the hammer assembly 24 to urge assembly 24 downwardly into impact with the tappet member 32. Thus, as the driving shaft 20 is rotated the followers 22 roll along the cam surface, which includes an inclined surface and axially aligned shoulder for each follower 22 and alternately moves the hammer assembly 24 against and compressing spring 30 and subsequently releasing it thereby providing a longitudinal impact.

The motor 18 is preferably a conventional rotary hydraulic motor and includes suitable inlet and outlet connections 34 and 36, respectively, for connection to any suitable source of hydraulic energy such as the gasoline actuated hydraulic pump unit 16 (FIGURE 1). The motor shaft 38 is connected through a conventional coupling 40 and then to the driving shaft 20 through a conventional thrust bearing assembly 42. Thus, the rotation of the motor 18 rotates its shaft 38, the coupling 40 and the driving shaft 20 through the antifriction thrust assembly 42. A suitable grease fitting 44 is provided to properly lubricate the roller and thrust bearings in the antifriction assembly 42.

Referring now to FIGURES 2, 3 and 11, the hammer assembly 24, as previously mentioned, includes the cam member 26 and the hammer element 28 which are preferably threadably secured together and form a lubricating chamber 46 (FIGURE 2) which may be filled with a suitable lubricant to enclose the critcal portion for the moving mechanism, the cam followers 22 and the cam surface in a bath of lubricant. Suitable sealing O rings 48 and 50 are provided to seal the chamber 46 between

the shaft 20 and the cam member 26 and also between the shaft 20 and the hammer element 28, respectively. It has been found in practice that these sealing rings provide an effective seal for the combination rotary and reciprocating action between the shaft 20 and the hammer assembly 24, and the churning action of the moving cam followers 22 in the lubricating chamber 46 effectively coats all exposed surfaces within this chamber with lubricant. If desired, a suitable wiper ring 52 may be provided to prevent any foreign materials from reaching and injuring O ring 48.

The hammer assembly 24 is positioned in the housing 12 for limited axial movement relative to the shaft 20 and in some cases as will be more fully described hereinafter for limited rotational movement relative to the housing 12. That is, the compression spring 30 yieldably acts against the hammer assembly 24 to urge it in a downward direction. As the cam followers 22 ride upon the cam surface of the cam element 26 the hammer assembly 24 is moved upwardly against the spring 30 until the followers 22 move off of the cam shoulders, which will be more fully described hereinafter, thereby allowing the hammer assembly 24 to again move downwardly and provide an impact blow. The longitudinal reciprocation of the hammer assembly 24 may be controlled by providing a slot 54 which may be axially aligned in the outer surface of the hammer assembly 24 and which coacts with a guide member 56 which may be suitably secured to the housing 12 such as by bolt 58 and nut 60. Thus, the hammer assembly 24 may move longitudinally but is restricted in its radial movement by the action of the guide 56 in the assembly slot 54. In addition, the axial movement of the hammer assembly 24 away from the compression spring 30 is limited by the mating of shoulder 62 on the hammer element 28 against a stop shoulder 64 which is secured to the housing or may form an integral part of the housing 12.

The hammer element 28 is positioned to strike a tappet 32 as it is accelerated downwardly by the action of the compression spring 30 when the hammer assembly 24 is released to provide the longitudinal output impact. A nose cone 66 forms part of the housing 12 and encloses the lower end of the assembly thereby locking the tappet member 32 in place. Bit holding means 68 may be provided to releasably hold any suitable bit 70 within the nose cone 66 of the impact hammer and in position whereby the tappet 32 is directed against the head of the bit 70. Of course, the tappet 32 may be omitted and the bit 70 may be directly actuated by the hammer element 28. While the housing 12, its closure members, nose cone 66 and motor enclosure 67 may be connected in any suitable manner, it has been convenient to provide bolts 69 and nuts 71 to perform this function.

The compression spring 30 is positioned within the housing 12 and is used to store and suddenly release the energy needed for the rapid impact in the hammer. To provide this function the spring acts against a shoulder 72 of the hammer assembly 24 and is supported at the other end by the housing 12. If desired, an adjustable preload collar 74 may be provided to initially compress and thus preload the spring thereby providing a predetermined load on the spring consequently adjusting the impact of the hammer. For example, if a 1,000 lb. per inch spring is installed so that the preload collar permits the spring to be at its free length at the conclusion of the impact stroke, an average driving force of 550 lbs. will be provided assuming a one inch hammer stroke. This, of course, is the average between zero and 1,000 lbs. However, if the preload collar is adjusted so that a 550 lb. force is provided at the conclusion of the impact stroke (a ½ inch preload) an average driving force of 1,100 lbs. will be provided with the same hammer stroke. To provide this adjustable feature, the collar 74 may be threadably adjustable with reference to the threads 76 formed on the interior

of the housing 12. Thus, the adjustable collar 74 may be adjustably threaded with reference to the housing thereby determining the desired amount of compression or preload of the spring at the end of its hammer stroke.

As previously mentioned, the previous cam and spring driven impact tools have been unable to provide a heavy duty longlife structure. Applicant has provided such a structure that will operate with at least a 1,100 lb. spring at a speed of 800 r.p.m. of the driving shaft 20. Attempts have been made in the past to use various spring cam impact mechanisms such as abrupt shoulder cam means or such as a simple roller against a cam. Such designs resulted in a theoretical infinite pressure when the cams or cam followers reached the drop-off point on the cam shoulder and consequently resulted in a device having a life of only a few operating cycles. Also, these designs were subject to high sliding friction which could not be overcome sufficiently to utilize energy of the magnitude necessary for heavy duty impact tools. The problem of sliding friction has been overcome in the present device by the provision of the antifriction bearing followers 22. As best seen in FIGURE 10, the heavy-load bearing followers 22 have a shaft 80 for suitably securing the followers 22 to the shaft 20 such as by threads or by force fitting. The followers 22 include an outer race 82 which revolves on a plurality of bearings 84 around the body of the bearing 22. Suitable stop members 86 and 88 are provided on the followers to suitably position and hold the bearings 84 and race 82 in position. Preferably stop 88 is formed integral with the shaft 80 to increase the strength of the bearing 22 at its weakest point, the shaft. Member 86 is then staked to shaft 80. The use of such followers has overcome the problem of sliding friction between the cam and cam followers and/or between the elements of simple rollers thus permitting the design of a structure capable of producing heavy duty high speed impacts for a prolonged life of the structure.

However, in view of the heavy impact forces desired to be exerted by the present impact tool 10 it is necessary that the impact forces generated by the mechanism not be transmitted to the followers, so as not to shock and damage the followers 22.

Referring now to FIGURE 9, the cam surface 90 of the cam element 26 is shown in layout form and generally includes an inclined surface 92 and an axially aligned shoulder 94 for contact with each follower 22. It has been found in previous impact tools of the spring cam mechanism that a theoretical infinite point pressure is encountered between the cam shoulder 94 and the race 82 of the cam follower at the drop-off point. That is, the portion of the race 82 furtherest from the shaft 20 in these tools leaves the conventional cam shoulder before the portion of the race 82 nearest the shaft 20; therefore, the load on the follower 22 is during the drop-off being transferred from a line of contact between the entire cam surface and the cam follower to a lesser contact between the portion of the race nearer the shaft 20 and the portion of the shoulder nearer the shaft 20. This results in a higher stress on these parts especially in view of the fact that the spring is at that point under its maximum compression.

Therefore, it is an important feature of the present invention that the followers 22 maintain a complete full line contact between the outer race 82 of the followers 22 and the cam surfaces of the cam element 26 at all times while they are in contact and under load. This complete full line contact may be easily attained on the inclined surface of the cam; however, it becomes more difficult at the transition from rise to fall portion of the cam which is also the most important since it encompasses the point of maximum load. The line of contact must remain in a plane with the axis of the cam follower perpendicular to the cam surface while the plane moves successively from a position off parallel to the axis of the shaft 20, by the degrees of cam rise, to a position perpendicular to the axis of the shaft 20 as the shaft 20 rotates revolving the follower about the axis of the shaft 20. This cam surface just described is more fully shown in FIGURES 11 and 12. That is, the cam surface is formed to move successively from a surface perpendicular to a plane passing through the line of contact and the axis of the bearing 22 and off the perpendicular to the shaft 20 axis by the angle of the cam rise to a plane which is parallel to the shaft 20 axis.

That is, the transition between the inclined surface 92 and the abrupt termination of the shoulder 94 at 107 must at all times be such that the cam followers 22 will maintain a tangential full line contact with the cam surface. Referring now to FIGURES 12, 13, and 14, the exact shape of the cam surface to provide full line contact is best seen. Line 101 denotes the longitudinal axis of the cam and radial lines are drawn through the points 105 which are the starting point at the interior periphery of the cam at which the drop-off begins. It is noted that a line 103 is a radial line on the cam surface and forms the high point of the cam surface. However, as previously stated, if the cam shoulder terminated at the line 103 the inner periphery shoulder point 105 would encounter a theoretical infinite point pressure as the cam followers 22 rotated off the line 103 and the followers 22 would be entirely supported on the points 105. Therefore, the cam shoulder 94 terminates in a line 107 which leads beyond the point 105 at the outer periphery of the cam and terminates at a point 109 which radially leads the inner cam periphery point 105. Thus, while the cam followers 22 will contact the cam shoulder 94 at a full line contact along radially directed lines 103, they will when further rotated maintain a full line contact with the cam shoulder 94 and be positioned to contact shoulder line 107 at the drop-off point thereby maintaining the desired full line contact. The abrupt drop-off shoulder termination line 107 forms an angle 111 with the radial line 103, the magnitude of angle 111 being determined by the diameter of the cam follower and its distance from the cam axis 101. The abrupt drop-off line 107 is in a plane parallel to a radial plane passing through radial line 113 and is spaced from radial line 113 at a distance of one-half of the outer diameter of the cam follower 22. The surface of the cam from the high-line 103 to the drop-off line 107 is curved from a minimum path length at the inner radius of the cam cylinder to a maximum path length at the outer radius of the cam cylinder such that the resultant surface between line 103 and line 107 will have a shape that will allow full line contact between the cam surface and the cam follower 22 whose axis is always normal to the axis 101 of the cam cylinder as the follower's axis rotates about the cam cylinder axis 101.

Impact to the antifriction bearing followers 22 is prevented by insuring that the mating shoulders 62 and 64 (FIGURE 2) meet prior to the contact of the cam element 26 with the bearings 22. That is, referring to FIGURES 2 and 9, it is necessary that the shoulder element 62 on the hammer assembly 24 contact the stop shoulder 64 prior to the time that the bearing follower 22 contacts the cam surface 90 after passing the cam shoulder 94. Furthermore, the cam surface includes a pickup portion 96 whereby the cam surface is designed to pick up and contact the bearing 22 as gradually as possible. Therefore, sufficient rotary clearance is designed into the cam surface to insure that the cam followers 22 do not strike the cam incline surface 90 on the downward movement of the hammer. It is therefore preferable that the cam surface 96 be theoretically designed to pick up and contact the bearing 22 at a point that is close as possible to a point 98, which should theoretically contact the follower 22 in a vertical plane passing through the center or axis of the follower 22. However, it is to be noted that

the position of the hammer and thus, the cam element 26 is dependent on whether the hammer element 28 and the other impact members, tappet 32 and the bit 70, encounter an immovable or movable object. The solid bearing 22 in FIGURE 9 shows the position of the bearing 22 when the hammer element and bit meet no resistance. In this event the stop shoulder 62 and 64 determine the nearest point of movement of the follower 22 with reference to the cam surface 96 after the follower 22 rolls over the shoulder 94. The dotted follower 22 in FIGURE 9 shows the path of travel of the follower 22 when the hammer element and the bit 70 encounter an immovable object which is being impacted. Therefore, it is important that the pickup area 96 of the cam surface be modified along the curve *a–b* so as to pick up the load on the follower 22 as gradually as possible regardless of whether the hammer elements and bit are running free of or are meeting an immovable object. Surface 96 must be designed to provide a gradual vertical pickup for both possible conditions. This pickup surface 96 is gradually merged and meets with the inclined surface 92.

Thus, as the shaft 20 is rotated by the hydraulic motor 18, the bearing followers 22 similarly rotate and move along the cam surfaces 96, 92 and 94 of the cam element 26 and alternately compress the hammer assembly 24 against the spring 30 and release the hammer assembly 24 as the bearing followers 22 pass over the cam shoulders 94 to allow the compression spring 30 to drive the hammer assembly 24 downwardly with an impact thereby providing an impacting force against the tappet 32 and consequently the bit 70.

It is apparent that the race 82 of the cam follower is subjected to an unusual condition in that the outer periphery of the race 82 at the point most distant from the center of the shaft 20 has a further distance to travel than has the inner periphery of the same race 82 at a point closer to the center of the shaft 20 for each revolution of the shaft 20. This fact causes slippage between the race 82 and the cam surface 90 and tends to cause wearing of the surfaces. This slippage may be minimized by slightly crowning the race 82 of the cam follower 22 as shown in FIGURE 10. The slight crowning, while not enough to fatigue the race 82, is enough to have increased pressure at the center of the race causing half slippage in one direction and half in the other and thereby preventing the slippage and tendency to wear from being concentrated.

As previously mentioned, and as shown in FIGURES 2 and 3 the radial movement of hammer assembly 24 as it is reciprocated longitudinally in the housing 12 may be controlled by means of the slot 54 and guide 56. However, and referring to FIGURE 11, the hammer assembly 24 may include a plurality of slots in addition to slot 54 such as slots 100, 102 and 104 which are designed to slightly rotate the hammer assembly 24 during the striking blow. Thus, the slots 100, 102 and 104 are positioned at angles to the axis of the hammer assembly 24. The hammer assembly will, as it encounters the guide 56 be rotated a predetermined amount, depending upon the angle of the engaging slots, during its longitudinal movement. One purpose of providing such rotational movement is to compensate for the continued rotation of the bearing followers as they leave the cam surfaces 94 under rapid rotation so as to prevent the followers from picking up the incline surface 90 at an unfavorable angle. Thus, referring again to FIGURE 9, the hammer assembly 24 including the cam element 26 is caused to rotate by the angularly positioned slots 100, 102 and 104 to cause the cam element 26 and its cam surfaces as shown in FIGURE 9 to rotationally follow, but at a lesser and to a limited extent, the roller bearings 22 thereby preventing the bearings 22 during high speeds from moving too far past the shoulders 94 and striking the inclined cam surfaces 92.

In addition, the angularly spaced engaging slots 100, 102 and 104 may be used to give a somewhat radial action to the hammer which may be in turn transmitted to a cutting tool, such as a star drill for instance, wherein rotation of the cutting edge is desirable. Therefore, on assembling the tool the guide 56 may be assembled in the desired hammer assembly slot depending upon the type of operating conditions being encountered.

Of course, various modifications and various additional features may be utilized with the present invention. FIGURES 4–8 illustrate modifications and additions of the impact hammer according to the present invention, the letters "*a*," "*b*," and "*c*," being applied to the parts corresponding to those in the previously discussed figures for convenience of reference.

FIGURES 4–6 illustrate a hydraulically actuated impact tool wherein the motor shaft and driving shaft are made an integral shaft 110 for convenience and an air pump 112 is provided and connected to shaft 110 for actuation. It may be desirable in some applications to have an air supply to blow out dust, for example in star drill work or perhaps to cool a tool in operation under extreme speeds and/or unusual conditions. For this purpose a conventional vane type air pump 112 may be suitably attached to the shaft 110 and enclosed within the housing 12*a*.

In order to transmit the air output from the pump 112 a series of suitable passages must be provided to direct the air output to the desired location. For instance, air passageways 114 may be provided in the preload collar 74*a*, and suitable passageways 115 may be provided through the hammer assembly 24*a*, and still further air passageways 116 may be provided through the tappet 32*a* to direct the air out and around the bit 70 thereby blowing out the dust around the bit and also providing a cooling action as the air passes through the interior of the tool 10*a*.

In addition, the hammer element 28 as shown in FIGURES 2 and 3, may be of the free fall type which permits a striking blow to be delivered independent of the spring and cam mechanism. Such a feature is shown in FIGURES 4–6. The free fall hammer is obtained through the use of a free fall hammer element 118 linked to the hammer assembly 24*a* by slidable connections such as a pin 120 and slot 122 connection, one of which is connected to the element 118 and the other of which is rigidly connected to the hammer element 28*a*. In this structure, the shoulder 62*a* and 64*a* meet prior to the time the striking blow is delivered by the free fall hammer element 118. The contact of shoulder 62*a* with shoulder 64*a* stops the downward movement of hammer element 28*a*, but the pin 120 and slot 122 connection allows the free fall hammer element 118 to continue downwardly with inertia force to strike the tappet 32*a*. This structure has the advantage that the cam followers 22*a* are positioned at the same lateral or axial position relative to the cam surfaces at each cycle regardless of whether the bit 70 encounters resistance or not. Thus, referring to FIGURE 9, the cam follower 22*a* of FIGURE 4 would reach the solid figure path after leaving the shoulder 94*a* during each cycle thereby allowing the design of the cam pickup surface 96 to be provided for all operating positions with a single and better designed pickup contour. The free fall hammer thereby permits ideal cam design for the most effective method of converting the rotary energy provided by the motor into the spring for eliminating the load pickup stress to the cam followers 22*a*. The hammer element 118, after the element 28*a* has been stopped by the face 64*a* continues on with inertia force to strike the tappet 32*a* thereby transferring the blow to the bit 70. A secondary stop shoulder 124 and 126 is provided on the member 118 and the housing 12*a*, respectively, to absorb the force of the hammer blow should it not be transmitted to the bit 70 for any reason thereby preventing any damage to the mechanism.

Referring now to FIGURE 7, the structure therein shown permits the flow of hydraulic fluid to be passed through a portion of the housing 12*b* when the impact tool 10*b* is operating under conditions where it might be de-

sirable to use the hydraulic fluid from the hydraulic motor circuits to circulate through the housing structure for cooling and/or lubricating purposes. Thus, an inlet port **130** and an outlet port **132** is provided in the housing **12***b* whereby suitable hydraulic connections such as from the hydraulic motor **18** may be provided to permit the flow of hydraulic fluid under pressure through this structure. However, in order to provide a hydraulic balance in the tool and prevent any pressure effect on the hammer by the hydraulic fluid, a hammer assembly extension **134** is provided which extends around the shaft **20***b* and telescopes within a working chamber **136** in the housing **12***b*. The outer diameter of extension **134** is sized the same as the outer dimension of the hammer element striking shaft **29** thereby providing a hydraulic balance in the tool as the hammer assembly **24***b* is reciprocated in the housing. Both the extension **134** and the hammer element **28***b* are sealed from the atmosphere by the conventional use of O rings. That is, a hydraulic balance is provided in the tool in that the fluid volume in the housing **12***b* which is to receive the cooling hydraulic fluid is to remain constant; otherwise, a hydraulic lock would occur if the fluid space in the housing **12***b* changed as the hammer assembly **24***b* reciprocated in the housing. Of course, the operation of the reciprocating hammer assembly **24***b* in the hydraulic fluid will result in some loss of power as the hydraulic fluid flows back and forth around the hammer assembly **24***b* during operation, but this loss is not serious as compared with the advantage of obtaining cooling of the moving parts.

Referring now to FIGURE 8, a further modification shows the provision of a tapered cam surfaces and tapered bearings **23**. An unusual condition encountered by the cam followers **22** in the mechanisms of FIGURES 2, 3, and 10 is that the race **82** if unrestrained would attempt to roll in a straight line. However, we restrain this outer race through attachment to the shaft **20**. Here again the components of the opposing forces result in slippage. While through actual tests it has been found that the use of proper lubricants prevents any detrimental results, in the sizes of units so far experimented with, the slippage previously discussed as well as the above discussed slippage effect can be entirely eliminated by means of tapering the outer follower race **83** and providing a corresponding matching slope of the cam surfaces of the cam elements **26***c*. Providing such a structure will result in a force component perpendicular to and away from the axis of the shaft. Both the slope and the percentage of the force factor decrease with increased cam diameter. The use of a tapered outer race **83** and cam surface may be desirable in certain sized impact tools. The outward force on the bearings **23** can be effectively controlled by incorporating thrust rollers (not shown) in the bearings **23**.

In operation, a suitable hydraulic source of power is connected to the hydraulic motor **18** such as by the portble gasoline operated hydraulic pump trailer **16** or by a hydraulic energy source such as is readily available on tractors, gradalls, backhoes and the like. The use of the rotary hydraulically motorized unit is advantageous over a pneumatically actuated air hammer in that both the prime investment cost and the operating cost of the energy supplying unit are approximately 20% of the pneumatic cost. The rate of flow of the hydraulic fluid which regulates the speed of the motor **18** is controlled by a conventional regulator valve (not shown). Referring to FIGURES 2 and 3, the rotation of the motor **18** rotates the motor shaft **38**, the flexible connection **40**, and the hammer driving shaft **20**.

The rotation of the shaft **20** consequently rotates the bearing followers **22**. Assuming that the cam surface contour is directed as shown in FIGURE 11, the driving shaft **20** and consequently the bearing followers **22** are rotated in a counter-clockwise direction looking from the top of the impact hammer in FIGURES 2 and 3 in order that the roller bearings roll up the cam surfaces **96** and **92** and over the cam shoulders **94**. During the rotation of the driving shaft **20**, the bearing followers **22** which are fixed in a longitudinal direction engage the cam element **26** thereby moving it longitudinally and compressing the spring **30**. As the bearing followers **22** rotate over the axially aligned cam shoulders **94** (FIGURES 9 and 11) the entire hammer assembly **24** is released and the energy built up in spring **30** accelerates the hammer assembly **24** downward to provide an impacting blow. Thus, the hammer element **28** is driven against the tappet **32** which in turn contacts and impacts the bit **70**. FIGURE 2 shows the relative position of the elements at the time of impact. FIGURE 3 shows the position of the elements when the hammer is cocked and the bearing followers **22** are positioned on the shoulders **94** of the cam members **26**.

It is noted that the use of the antifriction bearings **22** as shown in detail in FIGURE 10 eliminates the high sliding friction between the cam surfaces **90** and the cam followers **22** which otherwise would result from the use of the high-pressure spring **30** which in turn is necessary to provide a sufficient energy of the magnitude necessary for an effective impact tool. However, it is necessary that the structure of the impact hammer be designed to prevent any shock stress of any sizeable magnitude to the bearings **22** in order to prevent their breakage. In this regard, the cam surfaces **96**, **92** and **94** are designed to provide and maintain a complete full line contact between the outer race **82** of the cam followers **22** and these cam surfaces at all times while they are in contact and under load. And while this cam contour is relatively simple on the rising incline surface **90**, it is necessary adjacent the cam shoulders **94** (since this encompasses the point of maximum spring load) that the cam shoulder surfaces **94** must be formed such that the line of contact between the outer race **82** of the cam followers **22** and the cam surface **94** moves, successively from a plane through the axis of rollers **22**, off parallel to the shaft axis **20** by the degree of cam rise to a plane through the axis of the rollers perpendicular to the shaft axis **20**. Such a surface maintains a complete full line contact between the outer race **82** of the cam followers **22** and the cam surface **94** during the drop off portion of the cycle.

In providing additional protection against shock to the roller bearings **22**, it is noted in FIGURE 2 that the shoulder **62** on the hammer assembly **24** will contact and stop the downward axial movement of the hammer assembly **24** when the shoulder **62** contacts shoulder **64** on the housing **12**. This prevents the roller bearing followers **22** from impacting on the surface **90** of the cam element **26** after passing the cam shoulders **94** and damaging the followers **22**. Shock to the bearing followers **22** is further avoided wherein the pick up surface **96** is designed to contact and pick up the bearing contact as gradually and smoothly as possible.

Furthermore, if desired, the outer race **82** (FIGURE 10) of the bearing followers **22** may be slightly crowned at the center in order to even out the slippage between the bearings **22** and the cam surface **90** as previously described. In addition, if desired, the bearing and cam surfaces may be tapered as illustrated in FIGURE 8 to avoid the slippage effects caused between the bearings **23** and the cam **26***c*. This slippage is due to the fact that the outer race of the cam followers generally attempt to roll in a straight line and when restrained, as they revolve around the center of the driving shaft, result in a slight slippage and wear between the bearing followers and the cam surfaces.

Referring to FIGURES 2, 3, 4, 7 and 8, it is noted that a closed lubricating chamber such as chamber **46** in FIGURES 2 and 3 is provided so that the critical portions of the moving mechanism of the present impact hammer, the cam surfaces **90** and the bearing followers **22** are actually sealed in a bath of lubricant.

Of course, the impact of the present hammer may be adjusted to meet the operating conditions that are encountered. Thus, the adjustable pre-load collar **74** (FIG-

URES 2 and 3) may be adjusted to provide for any desired preload to be placed upon the compression spring 30 and thereby to increase or decrease the impact stroke that is desired from the hammer.

Referring now to FIGURES 4–6, it is noted that the hydraulic motor driving shaft may be integrated with the driving shaft to form a common shaft 110 which is particularly advantageous when size and weight are important factors in small impact tools. This feature is also advantageous if the use of an air supply is desired as in that case a conventional air pump 112 may be connected to the shaft 110 to provide an air supply which is passed through the air passages 114, 115 and 116, thereby providing an air supply around the bit 70 to blow out dust and to cool the impact tool 10*a*.

A free fall hammer is illustrated in FIGURES 4–6 which may be suitably incorporated within any of the modifications herein shown. This hammer structure permits the striking blow of the hammer to be delivered independent of the spring and cam mechanism. The movable hammer element 118 is linked to the fixed hammer element 28*a* by a suitable pin and slot arrangement 120 and 122 which permit the hammer element 118 to continue on with inertia force to strike the tappet 32*a* after the fixed hammer element 28*a* has been stopped by the contact of the shoulder 62*a* on 64*a*. The effect of such a free fall hammer is to position the cam roller followers 22*a* at the exact lateral position relative to the cam pick up surfaces (regardless of whether the bit strikes a movable or immovable object) at each cycle and thereby permits an ideal cam design to eliminate load pick up shock to the cam roller bearing followers. A secondary anvil face is provided wherein shoulder 124 on the movable hammer element 118 is provided to mate with and contact stop shoulder 126.

Referring now to FIGURE 7, the modified structure therein shown provides a hydraulic balance within the housing 12*b* for cooling purposes. The hydraulic balance is obtained by providing a hammer assembly extension 134 which extends into an opening 136 and is of the same diameter as the hammer element 29. This structure thereby permits, even when the hammer assembly 24*b* is reciprocating within the housing, the flow of hydraulic fluid under pressure through the structure without pressure effect on the hammer and without surge effect to the hydraulic fluid throughout the cycle. A suitable inlet 130 and outlet 132 are provided for the flow of hydraulic fluid through the housing structure 12*b*. The use of this structure is used when operating under conditions where it is desirable to use the hydraulic fluid from the hydraulic motor to circulate through the housing structure 12*b* for cooling and/or lubricating purposes.

Referring now to FIGURES 2, 3 and 11, it is noted that the hammer assembly 24 reciprocates longitudinally within the housing 12. As shown, a suitable straight slot 54 may be provided in the exterior of the hammer assembly 24 which coacts with a guide 56, which is rigidly connected to the housing 12, to restrain and limit radial movement of the hammer assembly as it is reciprocated. However, if desired, suitable angularly positioned slots such as 100, 102 and 104 may be provided to slightly rotate the hammer assembly 24 during the striking blow for several purposes. First, the hammer assembly 24 and consequently the cam element 26 may be rotated in a direction so as to follow the rotation of the roller followers 22 in order to compensate for the continued rotation of the roller followers after they leave the cam shoulders 94 and thus prevent the roller followers 22 from striking the inclined cam surface 92.

Furthermore, limited rotation of the hammer assembly 24 may be provided to permit rapid blows with light force if desired as it is not necessary in that event to rely on the spring alone to return the hammer rapidly enough for the followers to return above the clearance surface of the cams. In addition, the limited radial movement of the hammer assembly 24 may be used to give a somewhat

radial action to the hammer element 28 which in turn may be suitably, such as frictionally transmitted to a cutting tool, such as a star drill for instance, wherein rotation of the cutting edge is desirable.

Therefore, to secure the desired amount of rotation of the hammer assembly 24 it is only necessary to assemble the impact hammer with the guide 56 in the proper slot.

The impact hammer of the present invention is therefore well suited and adapted to attain the objects and ends and has the advantages and features mentioned as well as others inherent therein.

While presently preferred embodiments of the invention have been given for the purpose of disclosure, along with representative and typical uses thereof, other uses thereof will occur to those in the various arts in which the invention may be used, and changes in details and arrangement of parts may be made which are within the spirit of the invention and the scope of the appended claims.

What is claimed is:

1. In an impact tool having a housing, a driving shaft mounted in the housing for rotation relative to said housing, a hammer assembly in the housing and positioned about the shaft for axial movement relative to said shaft, a cam in the housing assembly, at least one roller bearing connected to said shaft and positioned to mate with said cam, a spring in said housing and around the shaft urging said hammer assembly in one direction to provide an impact, the improvement in the cam comprising,

said cam having an inclined surface and an axially aligned shoulder surface permitting sudden axial travel of the hammer assembly in said one direction,

the axially aligned shoulder being in a nonradial plane with the longitudinal cam axis with the outer edge of the cam surface leading the inner edge of the cam surface whereby said cam surface forms a substantially constant line of contact with the outer surface of the bearing and the cam surface during the entire time of contact of said bearing with the cam surface.

2. In an impact tool having a housing, a driving shaft mounted in the housing for rotation relative to said housing, a hammer assembly in the housing and positioned about the shaft for axial movement relative to said shaft, a cam in the housing assembly, at least one roller bearing connected to said shaft in position to mate with said cam, a spring in said housing and around the shaft urging said hammer assembly in one direction to provide an impact, the improvement in the cam comprising,

said cam having an inclined surface and an axially aligned shoulder surface permitting sudden axial travel of the hammer assembly in said one direction,

the cam surface between the inclined surface and the axially aligned shoulder forming a substantially constant line of contact between the outer surface of the bearing and the cam surface as the bearing moves from the inclined surface to the axially aligned shoulder.

3. An impact tool comprising, a housing, a driving shaft mounted in the housing for rotation relative to said housing, a hammer assembly in the housing and positioned about the shaft for axial movement relative to said shaft, a cam in the hammer assembly, two rollers bearings connected to said shaft and positioned to mate with said cam, a spring in said housing and around the shaft urging said hammer assembly in one direction, said cam including an inclined surface and axially aligned shoulder surface permitting sudden axial travel of the hammer assembly in said one direction, the improvement in the cam comprising,

said cam surface adjacent the aligned shoulder surface including a radially directed line of contact at the high point of the cam surface, and an abrupt drop-off shoulder line of contact at an angle to said radial line with the outer periphery of said abrupt drop-off line leading the inner periphery of said abrupt drop-off line,

the cam surface between said radially directed line and the abrupt drop-off line forming a substantially constant line of contact between the outer surface of the bearing and the cam surface as the bearing moves from the line of high point to the abrupt drop-off line.

4. In an impact tool having a housing, a driving shaft mounted in the housing for rotation relative to said housing, a hammer assembly in the housing and positioned about the shaft for axial movement relative to said shaft, a circular cam in the housing assembly positioned about the shaft, two roller bearings connected to said shaft in position to mate with said cam, a spring in said housing and around the shaft urging said hammer assembly in one direction to provide an impact, said cam having an inclined surface and an axially aligned shoulder surface permitting sudden axial travel of the hammer assembly in said one direction, the improvement in the cam comprising,

the high point of the cam surface being along a radial line from the cam axis and positioned between the cam inclined surface and the axially aligned shoulder surface,

the axially aligned shoulder surface forming a plane at an angle to the radial line of contact with the outer periphery of the cam leading the inner periphery of the cam at the axially aligned shoulder,

the cam surface between the radial line and the axially aligned shoulder surface tapering downwardly from the inner periphery of the cam to the outer periphery of the cam to form a substantially constant line of contact between the outer surface of the bearing and the cam surface during the entire time of contact of said bearing with the cam surface.

5. In an impact tool having a housing, a driving shaft mounted in the housing for rotation relative to said housing, a hammer assembly in the housing and positioned about the shaft for axial movement relative to said shaft, a cam in the housing assembly, two roller bearings connected to said shaft and positioned to mate with said cam, a spring in said housing and around the shaft urging said hammer assembly in one direction to provide an impace hammer the improvement in the hammer assembly comprising,

a first member rigidly connected to the cam member,

a second member being positioned on the side of the first member for travel in said one direction, said second member movably connected to the first member,

a first mating shoulders on the first member and on the housing positioned to mate with each other and stop the movement of said first member in said one direction, and

second mating shoulders on the second member and on the housing positioned to mate with each other after the first set of mating shoulders have made contact.

6. In an impact tool, having a housing, a driving shaft mounted in the housing for rotation relative to said housing, a hammer assembly in the housing and positioned about the shaft for axial movement relative to said shaft, a cam in the housing assembly, two roller bearings connected to said shaft and positioned to mate with said cam, a spring in said housing and around the shaft urging said hammer assembly in one direction to provide an impact, the cam having an inclined surface and an axially aligned shoulder surface permitting sudden axial travel of the hammer assembly in said one direction, the improvement in the cam comprising,

said cam surface adjacent to the aligned shoulder surface including a radially directed line of contact at the high point of the cam surface and an abrupt drop-off shoulder line of contact at an angle to said radial line with the outer periphery of said abrupt drop-off line leading the inner periphery of said abrupt drop-off line,

the abrupt drop-off shoulder line being in a plane parallel to a second radial line from the cam axis and spaced from said second radial line a distance of one-half of the bearing diameter, and

the cam surface between the first radially directed line and the abrupt drop-off line forming a substantially constant line of contact between the outer surface of the bearing and the cam surface as the bearing moves from the line of high point to the abrupt drop-off line.

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BROUGHTON G. DURHAM, *Primary Examiner.*

CHARLES E. O'CONNELL, *Examiner.*