United States Patent

[11] 3,533,479

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[45] Patented **Oct. 13, 1970**

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[54] IMPACT MECHANISM WITH IMPROVED HAMMER AND HAMMER FRAME ASSEMBLY THEREFOR

16 Claims, 30 Drawing Figs.

[52] **U.S. Cl.** **173/93.5**
[51] **Int. Cl.** **B25d 15/02**
[50] **Field of Search** 173/93,93.5

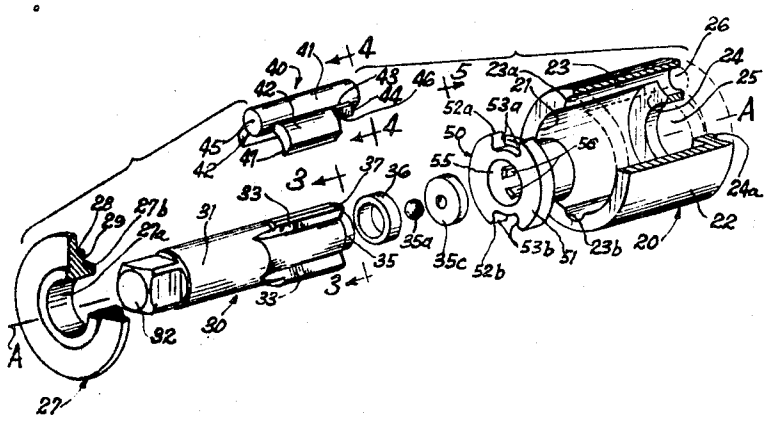
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Primary Examiner—Ernest R. Purser
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ABSTRACT: A rotary impact mechanism in which the driven hammer mass is divided between a hammer frame which rotates about an axis and one or more hammer dogs which are cam controlled to oscillate relative to the hammer frame to effect delivery of a succession of impact blows to an anvil, and wherein, in addition to providing for balance in the rotary assembly, the stresses in relatively movable hammer parts are distributed and wear reduced by the provision of extended and integrally formed bearing surfaces.



Patented Oct. 13, 1970 3,533,479

FIG. 1

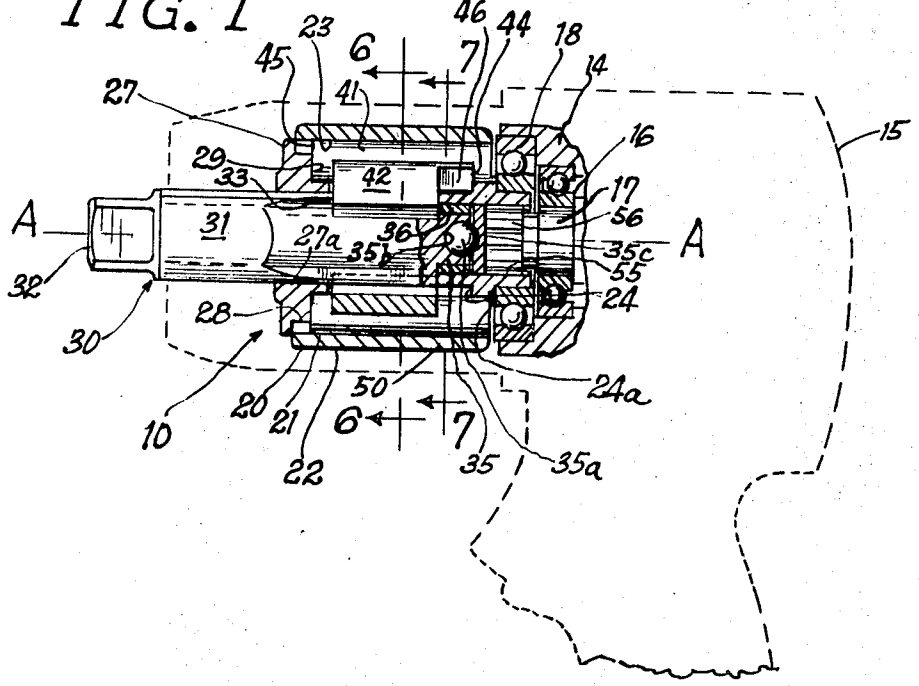


FIG. 2

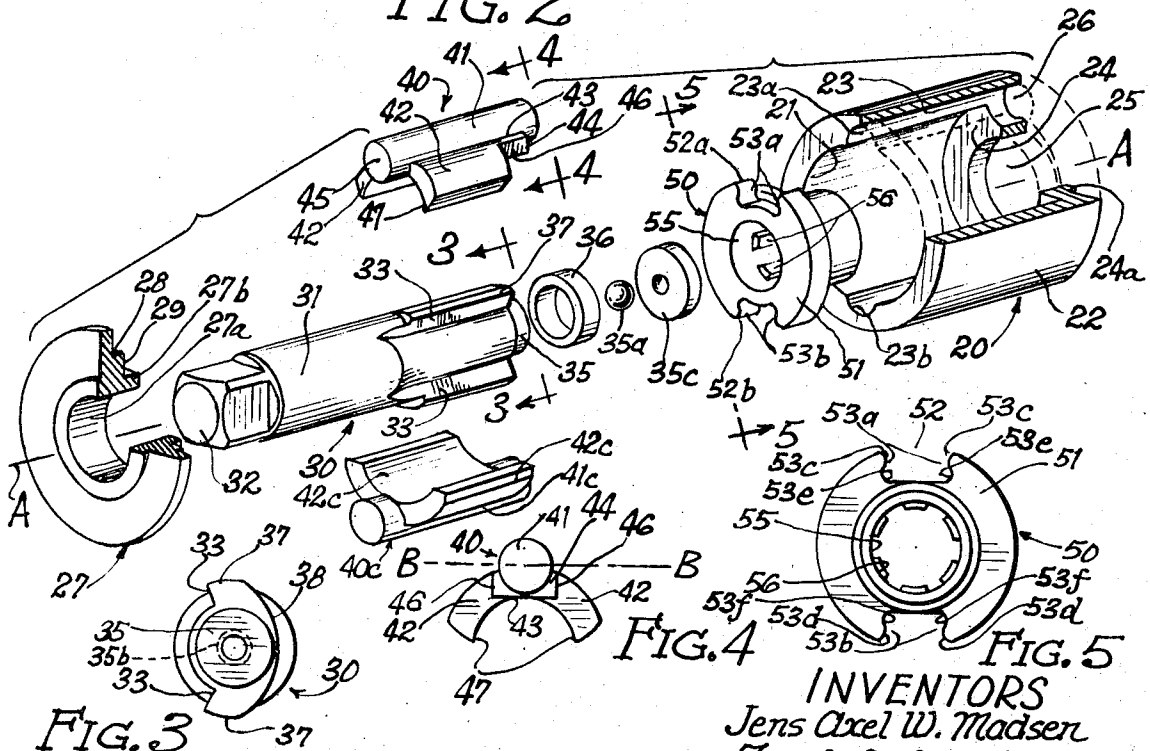


FIG. 3

FIG. 4

FIG. 5

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and Brugman Attys

FIG. 6(a)

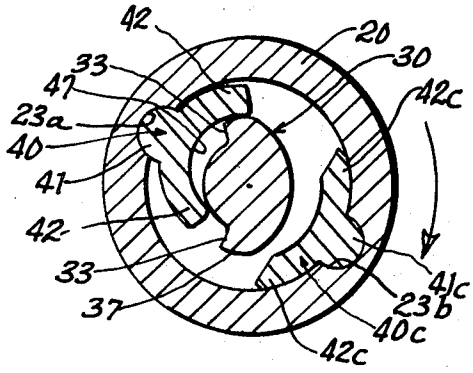


FIG. 6(b)

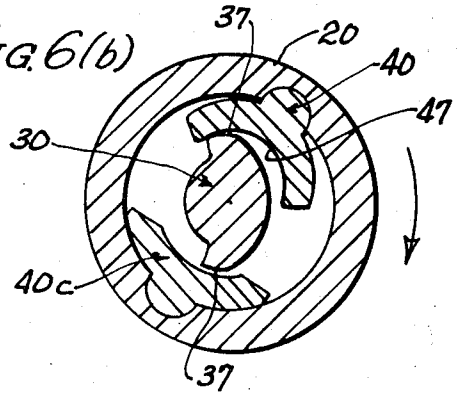


FIG. 6(c)

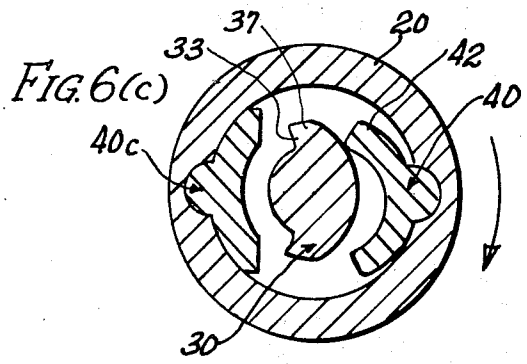


FIG. 6(d)

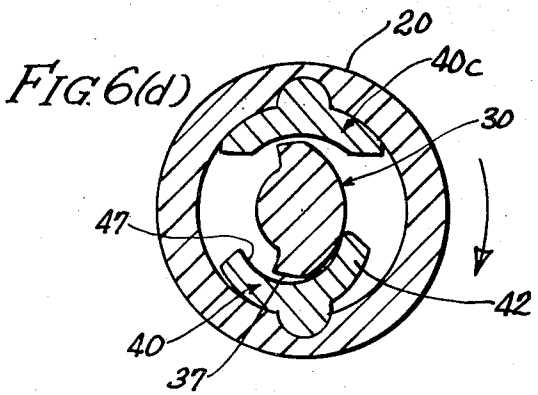


FIG. 7(a)

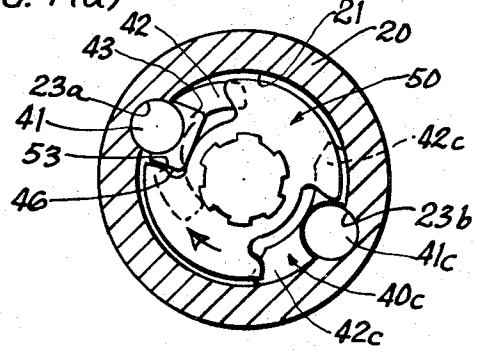


FIG. 7(b)

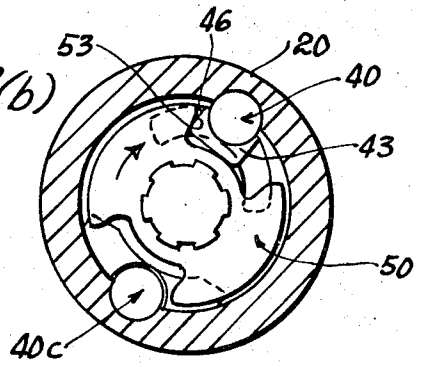


FIG. 7(c)

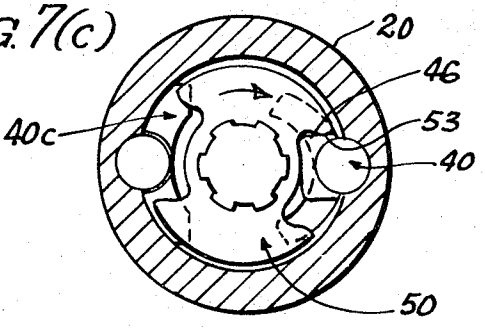


FIG. 7(d)

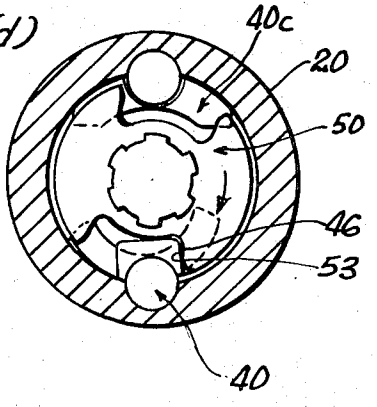


FIG. 6(e)

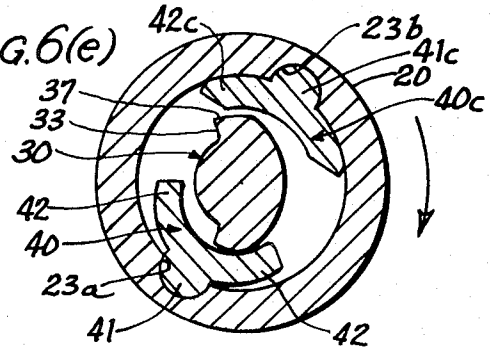


FIG. 7(e)

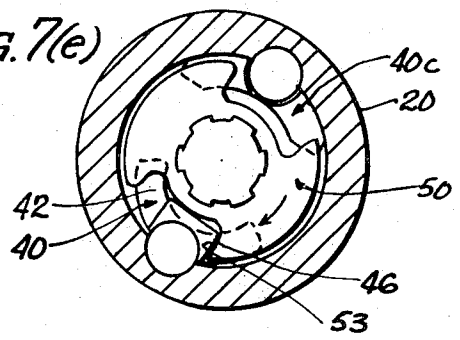


FIG. 6(f)

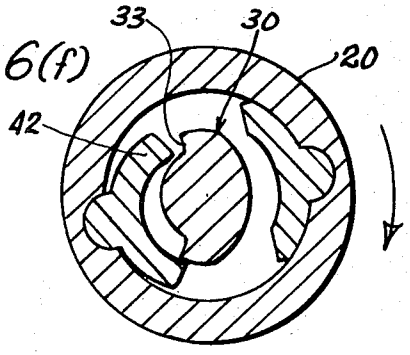


FIG. 7(f)

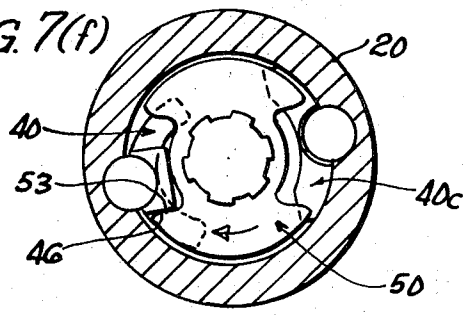


FIG. 6(g)

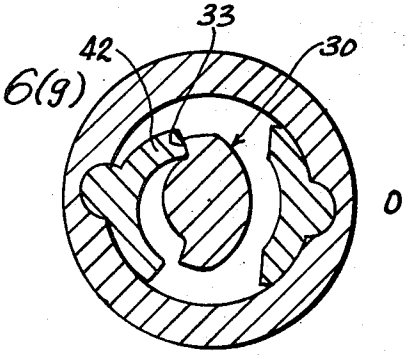


FIG. 7(g)

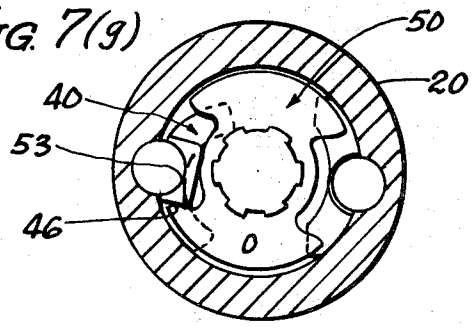


FIG. 6(h)

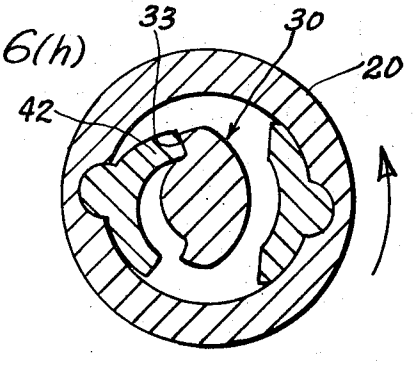


FIG. 7(h)

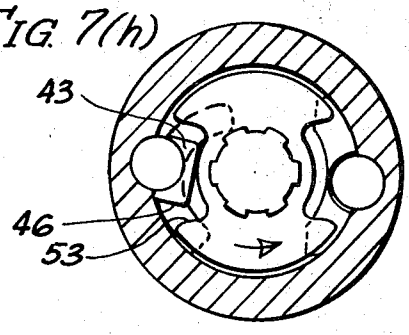


FIG. 6(i)

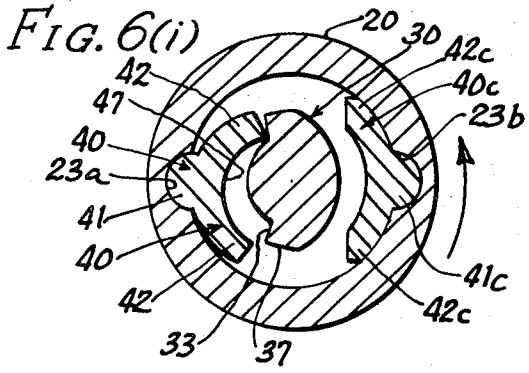


FIG. 7(i)

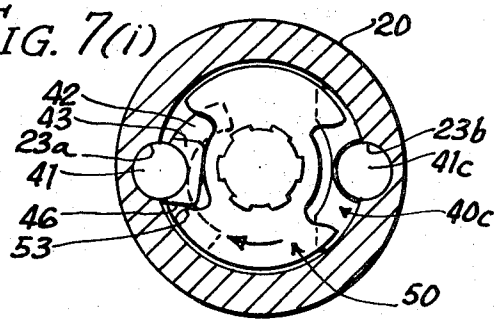


FIG. 6(j)

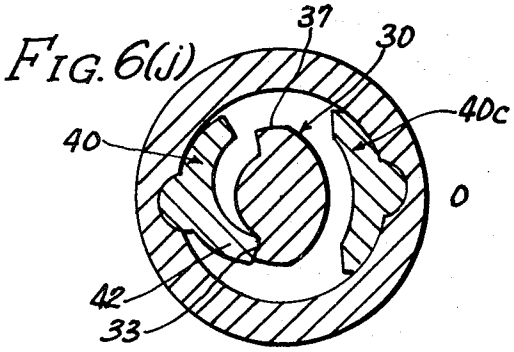


FIG. 7(j)

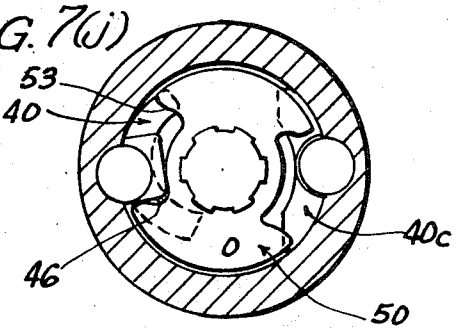


FIG. 6(k)

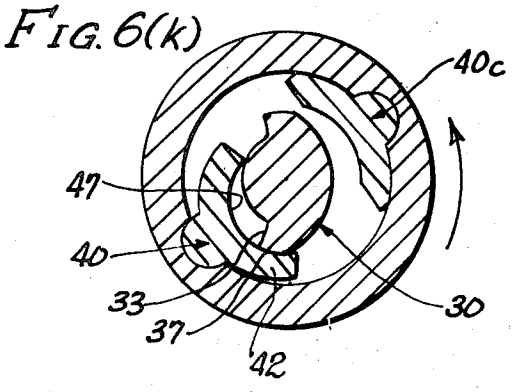


FIG. 7(k)

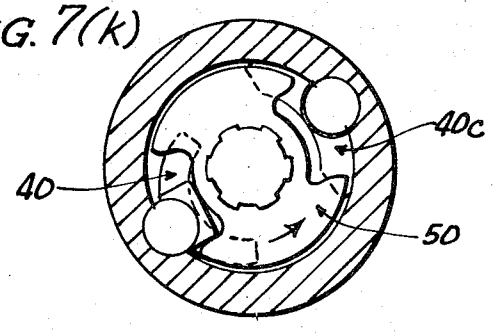


FIG. 6(l)

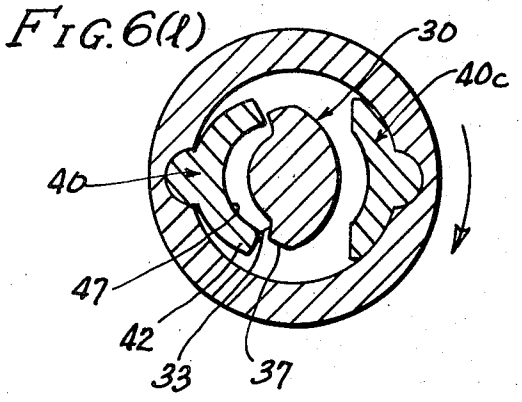


FIG. 7(l)

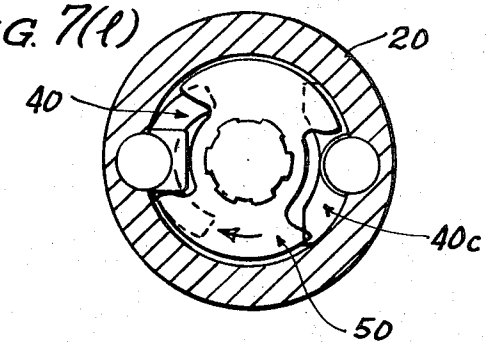
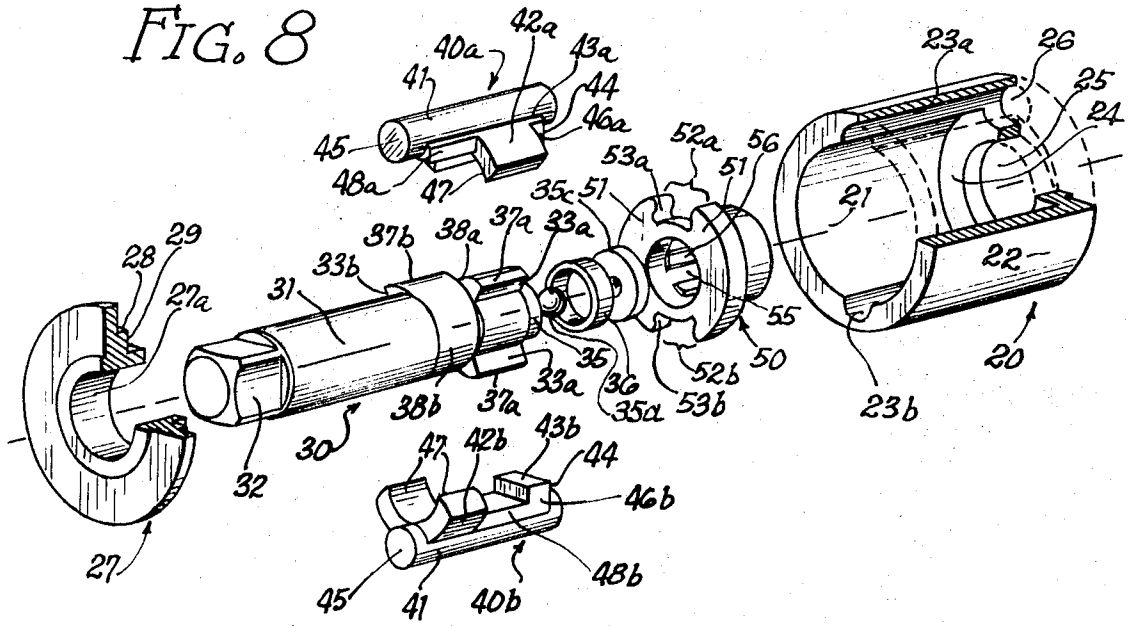


FIG. 8


42a
40a
43a
41
44
46a
45
48a
47
23a
26
25
24
52a
56
53a
51
35c
51
21
37b
38a
37a
33a
28
29
33b
31
27a
22
55
35
36
50
52b
20
38b
37a
33a
35a
53b
23b
30
32
47
42b
43b
44
46b
27
45
41
40b
48b

IMPACT MECHANISM WITH IMPROVED HAMMER AND HAMMER FRAME ASSEMBLY THEREFOR

BACKGROUND OF THE INVENTION

This invention relates generally to impact mechanisms, and more particularly to impact mechanisms in which a driven rotary hammer assembly incorporates a hammer dog which is rocked about an axis spaced radially from the axis of rotation of the assembly and wherein the rocking movements of the hammer dog are controlled to deliver a succession of impact blows to an anvil surface.

Of the various types of rotary motor operated impact devices, the type having a hammer dog which rocks radially within a rotating frame mass to effect engagement and disengagement with a rotary anvil, has been found to have high efficiency. That is, a hammer dog of small mass, journaled in a rotary frame of relatively larger mass, is pivoted, or rocked for engagement and declutching about an axis which is parallel to the anvil axis. High efficiency is attributed to the development of momentum by the combined frame and hammer dog mass rotating in a single direction, with only the smaller mass of the hammer dog changing directions by rocking in and out for impact engagement with the anvil and relative to an axis with respect to which it is symmetrical and balanced. An impact clutch of this type was first disclosed in U.S. Pat. No. 2,285,683 issued June 9, 1942, to L. A. Amtsberg.

In an impact mechanism of this type (the Amtsberg device being typical), the hammer frame was bored for reception of a pivot rod having ends journaled in end plates or flanges in the hammer assembly. This type of structure has a large amount of stress presented at the hammer dog journals during impact. Since only relatively short end portions of the pivot rod were journaled in apertures of short length, the stress and fatigue factors, naturally, were concentrated. Consequently, a high percentage of failures were attributable to breakage and the wear which occurred at the positions of stress concentration.

SUMMARY OF THE INVENTION

Therefore, to overcome the foregoing and other difficulties of the prior art, the general object of this invention is to provide an improved impact mechanism having longer service free life. To this end, the present invention features an elongated and extended journal bearing surface between the hammer frame and hammer dog in order to spread stress for minimal wear and fatigue thereat. This has been accomplished by providing a socket of arcuate section running the axial length of a portion of the hammer frame which presents an extended bearing slot through an inside wall of the hammer frame. The hammer dog has an axially extending and radially projecting pivot bearing rod portion of substantial length journaled in the elongated bearing slot. An impact portion of the hammer dog extends radially inward from one side only of the rod portion and inwardly of the hammer frame for impact delivering engagement with an anvil member.

Thus, one of the objects of this invention is to minimize wear and breakage by spreading stress in relatively movable bearing elements of the hammer structure of an impact mechanism.

Another object of this invention is to improve the journal between the hammer frame and hammer dog of an impact mechanism by providing extended bearing surfaces thereat.

As another object this invention provides a durable and efficient impact mechanism which is easily assembled and economical to produce by utilizing a minimum of components and utilizing structure which lends itself to standard mass production manufacturing techniques.

It is another object to provide an improved hammer frame with a hammer dog mounted for relative rocking movement therein and an elongated wear and fatigue resistant journal bearing structure therebetween.

This invention further has within its purview the provision of a hammer assembly adapted to the use of either one or two hammer dogs, and which, in either instance, is balanced for rotational movement.

Further, the invention comprehends and impact mechanism in which the interchange of only a portion of the operating parts of the impact mechanism is required to effect the change from a one to a two hammer dog assembly which has rotational balance in both instances.

Further and other objects, and a more complete understanding of the invention may be had by referring to the following description and claims, taken in conjunction with the accompanying drawing.

DESCRIPTION OF THE DRAWING

For the purpose of illustrating the invention, there is shown in the drawing the form which is presently preferred, it being understood, however, that this invention is not necessarily limited to the precise arrangements and instrumentalities there shown.

FIG. 1 is a cross-sectional side elevation of a preferred embodiment of the improved impact mechanism;

FIG. 2 is an exploded perspective view with portions cut away, showing components of the impact mechanism of FIG. 1;

FIG. 3 is an end view of an anvil member taken in the direction of arrows 3–3 in FIG. 2;

FIG. 4 is an end view of a hammer dog taken in the direction of arrows 4–4 of FIG. 2;

FIG. 5 is an end view of a cam member taken in the direction of arrows 5–5 in FIG. 2;

FIGS. 6*a*–1 show a sequence of component positions taken along a schematic cross section at lines 6–6 of FIG. 1;

FIGS. 7*a*–1 show a sequence of component positions taken along a schematic cross section at lines 7–7 of FIG. 1; and

FIG. 8 is an exploded perspective view similar to FIG. 2 with portions cut away showing a modified version of the improved impact mechanism.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Having detailed reference to the drawing, wherein like numerals indicate corresponding elements, there is shown in FIG. 1, an illustration of an improved impact mechanism embodying the present invention which is designated generally at 10. As illustrated, the impact mechanism 10 is utilized within a rotary power driven tool 15, shown in phantom. A shaft 17 of a rotary driving motor (not shown) is supported in a bearing 16 carried by an end plate 14 of the motor housing, for driving the impact mechanism 10 about axis A–A, as shown in FIG. 1. A second bearing 18, also carried by the end plate 14, is coaxial to and axially spaced from the bearing 16 to provide support for one end of the impact mechanism.

A hammer frame 20 of desired mass is made of a relatively heavy and durable material, such as steel, which, in the structure shown, is a hollow cylinder having concentric inside and outside cylindrical walls 21, 22, respectively. Elongated sockets 23*a* and 23*b* of arcuate section extend the axial length of the hammer frame interiorly of the cylindrical wall 22 and open inwardly through the inside wall 21 at diametrically opposed positions in the hammer frame, as illustrated by reference to FIG. 2. It may be observed that the elongated sockets 23*a* and 23*b* each form an elongated axial bearing slot along the inside cylindrical wall 21. An integral end flange 24 extends radially inward at one end of the hammer frame, as shown in FIGS. 2 and 8, and has a central opening 25 therein. Continuations of sockets 23*a* and 23*b* through the integral end flange 24 forms bearing apertures such as 26 at diametrically opposed positions therein which are coaxial to the arcuate surfaces of the respective sockets therein. The remaining end of the hammer frame 20 is capped by a removable end cap 27 which has a central bearing opening 27*a* therein. End cap 27 includes a shoulder portion 28 which fits snugly within the inside wall 21 of the hammer frame, as shown in FIG. 1. A radially disposed abutment surface 29 is also provided at the inside of the removable end cap 27, as shown.

A protruding lip 27*b* may be provided around the openings 27*a*, at the inside surface of the end cap 27, in order to enlarge

the bearing surface area of the opening 27*a*. Also, a lip 24*a* projects axially of the outer surface of the end flange 24 on the hammer frame, to serve as a spacer for the hammer frame in the assembly of the mechanism relative to the bearing 18.

As shown in FIG. 1, an anvil 30 is coaxially received within the hammer frame 20. The anvil 30 includes an arbor portion 31 upon which one end of the hammer frame is journaled in the opening 27*a* of the end cap 27 to provide for relative rotational movement between the hammer frame and anvil. An appropriately shaped end portion 32 on the anvil 30 extends outwardly from the tool 15 for reception by, and driving engagement with various tools, as for example, socket wrenches (not shown). Portions of the arbor 31 are shaped to provide radially disposed impact areas 33 shown in FIGS. 2 and 3. An inside end portion 35 of the anvil 30 serves as a reduced spindle for reception of a spacer ring 36 through which the inner end of the anvil is supported. Also, for the transmission of end thrust from the anvil, without friction between the ring 36 and the end face of the anvil, a thrust ball 35*a* is disposed in a central socket 35*b* in the end of the anvil, and engages a thrust disc 35*c* which is seated in a bore 55 and against splines 56 of a driven cam member 50, the support and function of which will be more fully described. With reference to the end view of the anvil in FIG. 3, it may be seen that projecting cam surfaces 37 are symmetrically provided adjacent to each of the impact surfaces 33. Portions of the anvil 30 opposed to the impact surfaces 33 are also shaped to form an arcuately curved back surface 38 which extends between and connects the cam surfaces 37, as shown.

A hammer dog 40 is shown in FIGS. 2 and 4. This hammer dog 40 is an integral structure which includes a longitudinally extending pivot bearing portion 41, impact portions 42 extending laterally from opposed midregional sides of the pivot bearing portion, and a boss portion 43. With reference to FIG. 4, it may be observed at the impact portions 42 and boss portion 43 extend from one side of bearing portion 41, as defined by a diametric plane B–B of the bearing portion. The bearing portion 41 is of a size to fit snugly and movably in socket 23 and aperture 26 of the hammer frame 20. Endwise retention of bearing portion 41 within socket 23 is accomplished by abutment of boss end 44 and bearing end 45 against the side of integral flange 24 and surface 29 of removable cap 27, respectively. Also, the bearing end 45 is held radially in the socket 23 of the hammer frame by the protruding lip 27*b* of the end cap 27. The boss portion 43 also has symmetrical side surfaces 46 which are perpendicular to plane B–B. The two impact portions 42 are curved and symmetrical laterally with respect to the bearing portion 41, each having an inside cam surface at 47. The impact portions 42 and boss portion 43 extend from the bearing portion 41 away from the socket 23 and from the inside cylindrical wall 21 when assembled with the hammer frame. With this construction, the entire length of bearing portion 41 is journaled within the entire length of socket 23 to spread stress for minimal wear and metal fatigue.

In order to provide rotational balance to the hammer assembly when only one hammer dog is used for driving the anvil, as in the form depicted in FIGS. 1 and 2, a counterweight 40*c*, having the same mass as the hammer dog 40, is utilized and made a part of the hammer assembly. The hammer frame 20, its flange 24 with diametrically separated openings 26 and the end cap 27 are adapted to the support and operation of two hammer dogs, when desired. Thus, for convenience in mounting and ease of accomplishing weight balance in the hammer assembly when only one hammer dog is used, the counterweight 40*c* has a bearing portion 41*c* like that of the hammer dog which mounts in the socket 23*b* opposite the hammer dog and which is supported at its opposite ends in a flange aperture 26 and by the end cap as in the instance of the hammer dog. Since the counterweight does not have any engagement with the anvil, and need not move like the hammer dog does, it has weight segments 42*c* which are integral with the bearing portion 41*c* and of a size to balance the weight of the hammer dog, the mass of the composite counterweight being distributed in a structure which clears the anvil during rotation of the hammer assembly. In the form disclosed, the weight segments 42*c* have outer surfaces which conform substantially to the curvature of the inner wall 21 of the hammer frame and normally rest thereagainst when the bearing portion 41*c* is engaged in the socket 23*b*. The inner surface of the counterweight is arcuately curved to afford clearance for the anvil.

A driven cam member 50 is separately illustrated at FIGS. 2 and 5. This driven cam member 50 extends through the opening 25 in the hammer frame 20 and has a radial rim 51 fitting within the inside cylindrical wall 21 and against the inside of integral flange 24 of the hammer frame as shown in FIG. 1. The radial rim 51 has like recesses 52*a* and 52*b* at diametrically opposed positions in the rim which are each wider than the width of the boss 44 between its side surfaces 46 and which, respectively, have opposed cam faces 53*a* and 53*b* at the opposite sides thereof, as shown at FIG. 5. A central bore 55 at one end of the driven cam member 50 receives the disc 35*c* and spacer ring 36 with the ball 35*a* disposed between the anvil and disc, which ring, in turn receives the spindle portion 35 of the anvil 30 and is rotatably supported therein. A splined end of the drive shaft 17 (partially visible in FIG. 1) fits into the end of the bore 55 opposite the spacer ring 36 between splines 56 on the cam member, so that the cam member 50 is driven thereby. It is also notable that the cam member 50 is supported by the bearing 18 which is concentric with respect to the shaft bearing 16. In the assembly and in operation, the boss 44 is disposed in the recess 52*a* with cam faces 53*a* disposed to engage side surfaces 46 of the boss for driving the hammer assembly and effecting movements of the hammer dog relative to the hammer frame, as will be described.

In addition to the recesses 52*a* and 52*b* being wider than the boss 44, between the generally planar surfaces 46 thereof, so that the boss surfaces 46 are separately engageable with the cam faces 53*a* in the two possible directions of rotation of the mechanism, there is room for some lost motion or rebound in certain phases of operation, as will be described. Also, it may be observed that the cam faces 53*a* and 53*b* are similar and opposed with oppositely disposed convexly curved portions 53*c* and 53*d* adjacent the periphery of the rim 51 which have adjoined oppositely disposed concavely curved portions 53*e* and 53*f* radially inward of the rim from the convexly curved portions, as shown in FIG. 5. The convexly curved portions of the cam faces provide contact with the boss surfaces 46 in operation.

An alternate or double hammer version of the disclosed structure is shown in FIG. 8 of the drawings. Identical numbers designate like elements. This modification utilizes both of the two diametrically opposed sockets 23*a* and 23*b* in the hammer frame 20 for movably supporting two hammer dogs 40*a* and 40*b* in operative relationship to the hammer frame and anvil. These hammer dogs 40*a*,*b* are modified in that the impact portions 42*a*,*b* of each have notched out portions 48*a* and 48*b*, which are displaced axially of the hammer dogs, as shown so that the respective impact portions are aligned for engagement with axially spaces segments of the anvil. Modification of the anvil 30 includes two symmetrical, axially separated, oppositely disposed and shaped portions for engagement with respective hammer dogs 40*a*,*b*, and which have thereon radially disposed impact surfaces 33*a*,*b*, cam surfaces 37*a*,*b*, and back surfaces 38*a*,*b* for cooperation with associated portions and surfaces of the two hammer dogs 40*a*,*b*. In this modified form both recesses 52*a* and 52*b* of the driven cam member 50 are utilized for reception of and coaction with the bosses 43*a*,*b* on a respective hammer dogs and engagement with boss faces 46*a*, *b*, respectively.

Operation of the tool impact mechanism will be described with reference to FIGS. 6 and 7. In the single hammer dog form illustrated in the sequential steps of part movements, it is to be understood that the counterweight 40*c* does not move relative to the hammer frame or engage the anvil. It is for weight balance. It may be understood that the double hammer

modification shown in FIG. 8 has similar principles of operation, but that with two hammer dogs 40*a*,*b* operating simultaneously, impact blows are delivered to diametrically opposed surfaces of the anvil for effecting balanced impact engagement. The operation described hereafter can be applied to each hammer dog of the structure of FIG. 8.

FIG. 6*a* may be taken as the beginning of an operational cycle, and is arbitrarily chosen as a position after an impact of the hammer dog against the anvil and after the impact portion 42 of the hammer dog has moved radially outward from the impact area 33 of the anvil and after the hammer frame and hammer dog have moved somewhat in the direction of the arrow. The hammer frame 20 is shown as rotating clockwise in the illustrated sequence, although it is to be understood that the symmetry of the structure disclosed is provided in the various portions and surfaces thereof to enable identical operation in either direction. The part positions shown in FIG. 6*a* and FIG. 7*a* are concurrent in time, and, as shown in FIG. 7*a* face 53 of driven cam 50 is drivingly engaged with surface 46 of the boss 43 for driving the hammer assembly and effecting counterclockwise rotation of the hammer dog relative to the hammer frame. That is, the clockwise rotation of the driven cam 50 effects engagement of the cam face 53 with the boss surface 46, thereby transmitting rotational force and movement to the hammer frame 20 through the hammer dog 40, while at the same time, cam engagement between the boss surface 46 and face 53 causes the hammer dog 40 to rock or pivot about the bearing 41 in a counterclockwise direction until the hammer dog portion 42 engages the inner surface 21 of the hammer frame.

The hammer frame 20 and hammer dog 40 continue to rotate clockwise with respect to the anvil 30, until, as shown in FIG. 6*b*, the trailing cam surface 37 engages the inside cam portion 47 of the trailing impact portion 42. This engagement causes the hammer dog 40 to start moving clockwise relative to the hammer frame. The resulting relative movement of the hammer dog 40 with respect to the hammer frame retards the cam in relation to the hammer frame, and moves the position of contact between the surface 53 and boss surface 46, as shown in FIG. 7*b*. However, once the trailing portion 42 of the hammer dog has passed cam surface 37 of the anvil, as shown in FIG. 6*c* clockwise rotation of the driven cam member 50 will again, through force applied by cam face 53 to boss surface 46, turn the hammer dog in a counterclockwise direction relative to the hammer frame. Further rotation of the hammer frame 20 and hammer dog 40 brings the components to the positions shown in 6*d* and 7*d*, wherein the leading cam surface 37 of anvil 30 is about to engage the cam portion 47 of trailing impact portion 42. Upon engagement therebetween the hammer dog 40 will again rock and pivot clockwise relative to the hammer frame, bringing the leading impact portion 42 inward, as shown in FIGS. 6*e*, and *f*, and 7*e*, and *f*. With the leading impact portion 42 rocked to its inwardmost position, it will engage the impact area 33 of the anvil 30, to deliver an impact blow thereto and cause impact rotation thereof. The instant of impact engagement is shown at FIG. 6*g* and the corresponding relationship between the driven cam member 50 and boss portion 43 is shown in FIG. 7*g*.

At the instant of impact, the delivery of force from the hammer assembly to the anvil through engagement of the leading impact portion 42 of the hammer dog with the opposed impact surface 33 of the anvil either retards or stops the movement of the hammer assembly, depending upon the resistance which is offered to movement of the anvil against the work being done thereby and the resultant rate of absorption of the energy of the hammer blow and driving force. When the resistance to movement of the anvil 30 is small enough to permit substantial resultant driven movement of the anvil as a result of each blow, the blow delivered by the hammer is sometimes referred to as a "soft" blow. As the resistance to anvil movement increases, the blows become "harder", until a point is reached at which each blow of the hammer assembly against the anvil produces very little or no perceptible movement of the anvil. The "harder" blows produce more reaction against the hammer assembly and also against the cam and prime mover rotor which generally have less mass than the hammer assembly which results in a tendency toward rebound of the hammer parts and the cam and prime mover rotor. The normal and expected variations of the reactive forces against the action of the hammer mechanism and the prime mover, and the desirability of relatively uniform and consistent operation of the impact mechanism under such varying conditions in operation give desirability to providing for lost motion between the prime mover and the hammer assembly. In the disclosed structure, as has been pointed out, the recess 52, in which the boss 44 extends, is wider than the distance between the side surfaces 46 of the boss, so that relative movement can occur between the driving cam 50 and the hammer assembly.

As the resistance to movement of the anvil by the hammer increases, a reaction force develops upon impact of the hammer with the anvil which can be, and is sufficient that the prime mover rotor and cam 50 are reversed in rotational movement against the normal driving force of the prime mover. In this instance a separation occurs between the boss surface 46 on hammer dog and the surface 53 on the cam, as shown in FIG. 7*h*, which separation is provided for by the relative widths of the boss 43 and cam recess 52. The reaction of the impact of the hammer dog against the anvil also retards or stops the hammer assembly, but since the hammer assembly has considerably more mass than the cam and prime mover rotor, the reaction of the hammer assembly is much slower and of smaller magnitude, even though the forces involved are greater. As a practical matter, the reaction force which is effective on the hammer assembly causes a relief of the force between the leading edge of the portion 42 of the hammer dog and the impact surface 33 of the anvil, as indicated in FIG. 6*i*.

Following the effects of these reaction forces, and within an extremely small time interval, the prime mover stops its reactive reverse rotation and again starts moving in its driven clockwise direction of rotation. This can happen quickly, because of the usual small mass of the prime mover rotor and the cam. The acceleration is rapid until the space between the boss surface 46 and the cam surface 53 is closed, as indicated in FIG. 7*i*, at which time, force is transmitted from the cam surface 53 to the hammer boss surface 46 to rotate the hammer dog 40 in a counterclockwise direction relative to the hammer frame, thereby to cause the leading portion of the hammer dog to clear the anvil impact surface for continued movement and operation.

When the impact blow of the hammer dog against the impact surface 33 of the anvil is sufficiently "hard" and the reaction forces are of a high magnitude, the resultant counterclockwise rotation of the hammer assembly may continue until stopped by engagement of the normally trailing portion 42 of the hammer dog with the opposed anvil surface 33, as shown in FIG. 6*j*. When thus disposed, the hammer dog is in a position relative to the hammer frame such that the leading portion of the hammer dog will clear the cam surface 37 of the anvil upon being driven forward by the force of the prime mover, and operation will continue as illustrated from the positions of parts shown in FIGS. 6*a* and 7*a* through succeeding cycles.

When resistance to anvil movement is insufficient for a "hard" impact blow of the type described, a "soft" impact blow may not produce sufficient reaction to the hammer assembly that the surface of the normally trailing portion of the hammer dog will engage the opposed surface 33 of the anvil, as shown in FIG. 6*j*, for arresting reactive movement. However, even in this instance, action of the cam surface 53 against the boss surface 46 during rotation motivated by the prime mover will still position the hammer dog relative to the hammer frame for passing over the cam surface 37 of the anvil, whereupon action may continue, as described.

It is further worthy of note that the operating actions of parts of the disclosed mechanism can be changed in some respects by variations of the angle between the two impact

surfaces **33** of the anvil. As an example, and while the sequences of operation will be the same as illustrated in FIGS. **6***a* and **7***a* through FIGS. **6***i* and **7***i*, a reduction of the angle between the two impact surfaces of the anvil can prevent the hammer dog portion **42** from reactive engagement with the anvil surface **33**, as shown in FIG. **6***j*, whereupon the hammer dog portion **42** will pass over the cam surface **37** of the anvil, as shown in FIG. **6***k*. In this instance, the counterclockwise rotation of the hammer assembly will be stopped by the prime mover acting through the cam **50**. After stopping such reactive movement of the hammer assembly, the prime mover will again effect clockwise rotation thereof. After the trailing portion **42** of the hammer dog pass over the cam surface **37** of the anvil, the action of the cam against the hammer dog boss will effect counterclockwise rotation of the hammer dog relative to the hammer frame in the manner depicted in FIGS. **6***l* and **7***l*, whereupon the cycle will return to that starting with FIGS. **6***a* and **7***a*.

The operation is the same for left-hand rotation of the hammer assembly, except that symmetrically opposed surfaces of the hammer dog, anvil and cams become effective.

The operation of the modified embodiment shown in FIG. **8** is similar, except that there is a simultaneous impact engagement by both hammers **40***a*,*b* during each cycle, at diametrically opposite impact areas **33***a*,*b* of the anvil to provide balanced impact for producing rotation.

We claim:

1. In an impact mechanism for translating rotational force of a prime mover into impact energy for application to a tool, the combination comprising a hammer assembly supported for rotation about a central axis, said hammer assembly including a hammer frame having inner and outer walls concentric with said axis and having an arcuate socket type recess extending axially along and opening through the inner wall of said hammer frame, and anvil supported within the said hammer frame for rotation about said axis and relative to the hammer frame, a hammer dog supported for rocking movement relative to said hammer frame by an elongated pivot bearing portion journaled in said socket type recess along a substantial length of the hammer frame to distribute stress and minimize wear therebetween, said hammer dog extending radially into the hammer frame from said recess and being movable relative to the axis of said arcuate socket to and from engagement with said anvil, means for transmitting rotational movement to said hammer frame, and cam means for effecting periodic engagement and declutching movement of said hammer dog relative to said hammer frame for transmitting a succession of impact blows from the hammer dog to the anvil, said cam means including a driven cam member having thereon a radially projecting rim with an outwardly opening recess in the periphery thereof, a boss on said hammer dog which extends, laterally from said pivot bearing portion and has surface engagement with said rim at one side of said recess to transmit rotational movement to the hammer frame through a part of the hammer dog, and said one side of said recess and the adjacent side of said boss being engaged at positions providing leverage against the hammer dog for normally effecting declutching movement of the hammer dog relative to the anvil after impact engagement therebetween.

2. In an impact mechanism in accordance with claim **1**, the combination wherein said boss has a relatively flat side surface disposed for engagement with a convexly curved side surface of said outwardly opening recess in the rim of said cam means.

3. In an impact mechanism in accordance with claim **1**, the combination wherein said boss is narrower than the circumferential width of said outwardly opening recess in the rim of said cam means to provide space for reactionary movement between said boss and said one side of the recess as a result of an impact blow delivered to the anvil by the hammer dog.

4. In an impact mechanism for translating rotational force of a prime mover into impact energy for application to a tool, the combination comprising a hammer assembly supported for rotation about a central axis, said hammer assembly including a hammer frame having inner and outer walls concentric with said axis and having an arcuate socket type recess extending axially along and opening through the inner wall of said hammer frame, an anvil supported within the said hammer frame for rotation about said axis and relative to the hammer frame, a hammer dog supported for rocking movement relative to said hammer frame by an elongated pivot bearing portion journaled in said socket type recess along a substantial length of the hammer frame to distribute stress and minimize wear therebetween, said hammer dog extending radially into the hammer frame from said recess and being movable relative to the axis of said arcuate socket to and from engagement with said anvil, means for transmitting rotational movement to said hammer frame, and cam means for effecting periodic engagement and declutching movement of said hammer dog relative to said hammer frame for transmitting a succession of impact blows from the hammer dog to the anvil, said hammer frame comprising a hollow cylinder having an end flange at one end and an end cap at the other end, said bearing portion of the hammer dog extending from said end cap through an opening in said end flange for supporting the hammer dog movably within the hammer frame.

5. In an impact mechanism in accordance with claim **4**, the combination wherein said cam means includes a boss at one end of said hammer dog which has an end surface disposed laterally of the axis of said bearing portion of the hammer dog and engageable with said end flange of the hammer frame for limiting axial movement of the hammer dog away from said end cap.

6. In an impact mechanism in accordance with claim **4**, the combination wherein said end flange is an integral part of the hammer frame and has a bearing aperture therethrough in axial alignment with the arcuate socket type recess, said bearing surface of the hammer dog having an end portion projecting therefrom and extending into said aperture in the hammer frame flange at one end of the hammer frame and having a portion at the opposite end which abuts said end cap for retention of the hammer dog therebetween.

7. In an impact mechanism in accordance with claim **5**, the combination wherein said hammer dog, including said bearing portion and said boss, and said anvil have relative positions in which both are symmetrical in sectional shape with respect to a plane passing through said axes, so that the mechanism is operable in both directions of rotation of the hammer frame.

8. In an impact mechanism for translating rotational force of a prime mover into impact energy for application to a tool, the combination comprising a hammer frame which constitutes the major element of mass of a hammer assembly and which is in the form of a hollow cylinder having outer and inner surfaces radially separated with respect to a central axis, said hammer frame having therein a bearing recess of arcuate section with a span of less than 180° and opening inwardly of the inner wall in a direction radial to said central axis and extending axially along a major portion of the length of said inner wall, and a hammer dog supported for rocking movement relative to the hammer frame by a bearing portion of convex arcuate section with a sectional span of more than 180° and which fits into and extends along said bearing recess.

9. In an impact mechanism in accordance with claim **8**, the combination wherein the span of the arcuate section of said bearing recess, the sectional span of said bearing portion and the contours of the hammer dog adjacent said bearing portion are related to one another and to the interior of said hollow cylinder to effect limitation of the rocking movement of the hammer dog relative to the hammer frame.

10. In an impact mechanism in accordance with claim **8**, the combination wherein said hammer frame has an integral flange at one end with a bearing opening therein concentric with the axis of said arcuate bearing recess, and said hammer dog has a bearing stud thereon fitting into said bearing opening in the hammer frame flange.

11. In an impact mechanism in accordance with claim **8**, the combination wherein an end cap fits onto one end of said

hammer frame and has a radial surface and a centering shoulder thereon, said hammer dog has an end projection of circular section concentric with the axis of said bearing portion thereof, and means on the end cap for providing radial support for the end projection of the hammer dog and means for limiting endwise movement of the hammer dog in one direction relative to the hammer frame.

12. In an impact mechanism in accordance with claim **8**, the combination wherein said hammer frame has a second bearing recess like the aforementioned bearing recess at a position directly opposite thereto in the hammer frame, and a counterweight of a mass equal to that of the hammer dog immovably mounted in said second bearing recess.

13. In an impact mechanism for translating rotational force of a prime mover into impact energy for application to a tool, the combination comprising a hammer frame which constitutes the major element of mass of a hammer assembly and which is in the form of a hollow cylinder having outer and inner surfaces radially separated with respect to a central axis, said hammer frame having therein a bearing recess of arcuate section with a span of less that 180° and opening inwardly of the inner wall in a direction radial to said central axis and extending axially along a major portion of the length of said inner wall.

14. In an impact mechanism for translating rotational force of a prime mover into impact energy for application to a tool, the combination comprising a hammer dog which is symmetrical in sectional shape with respect to a central plane with an axially extending bearing portion of convex arcuate section and a span of over 180° equally divided by said central plane, and said hammer dog having laterally projecting impact portions on opposite sides of the bearing portion and extending axially of the midregion thereof.

15. In an impact mechanism in accordance with claim **14**, the combination wherein said hammer dog has bearing studs at its opposite ends which are concentric with respect to said arcuate bearing portion.

16. In an impact mechanism in accordance with claim **14**, the combination wherein said hammer dog has a boss at one end of the impact portions thereof which has cam surfaces symmetrically and oppositely disposed with respect of said central plane.