Aug. 30, 1955 W. G. MITCHELL 2,716,475

IMPACT TOOLS

Original Filed March 17, 1949 2 Sheets-Sheet 1

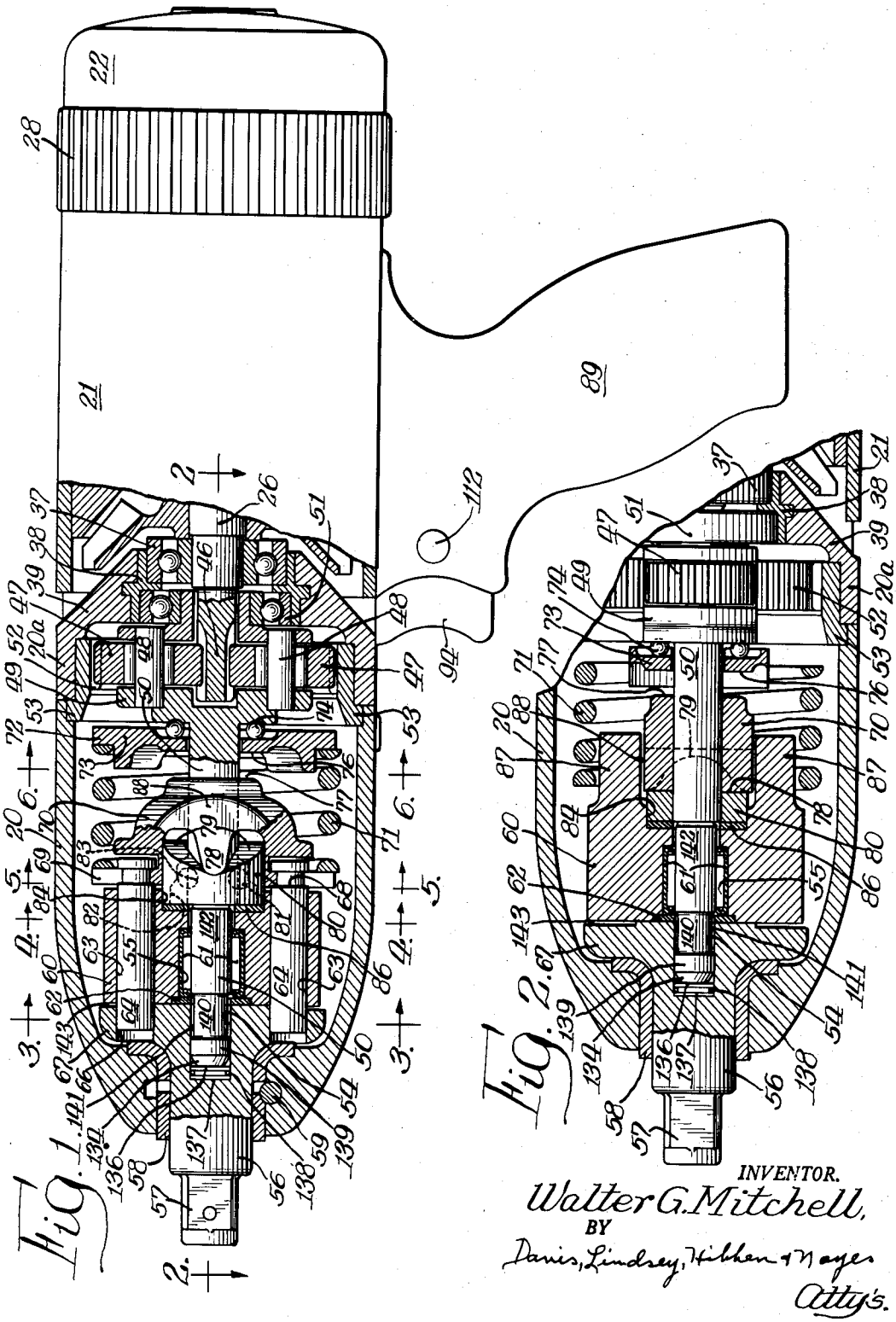


Fig. 1

Fig. 2

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Original Filed March 17, 1949 2 Sheets-Sheet 2

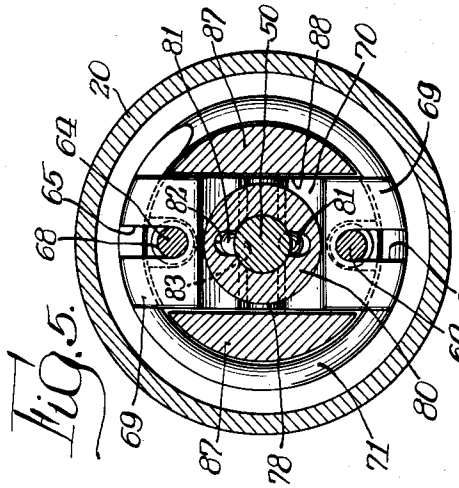


Fig. 5.

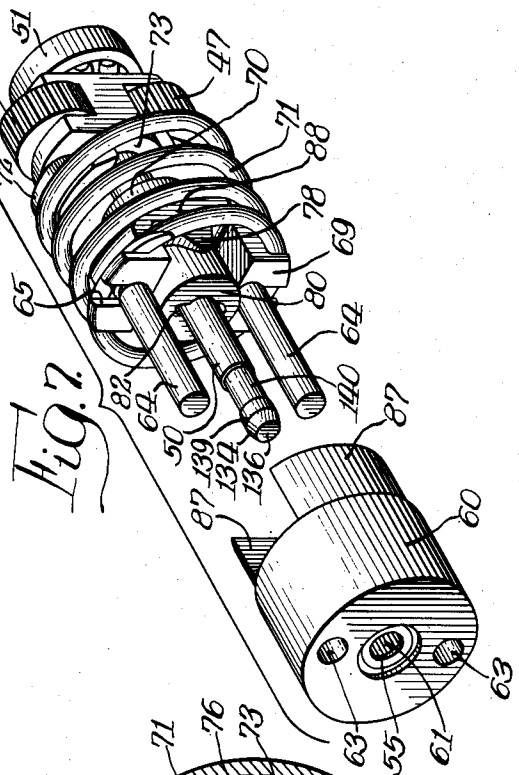


Fig. 7.

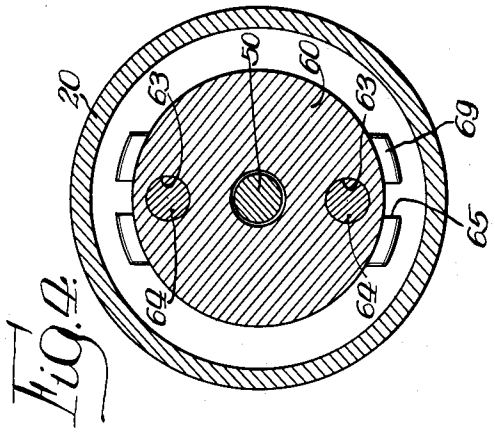


Fig. 4.

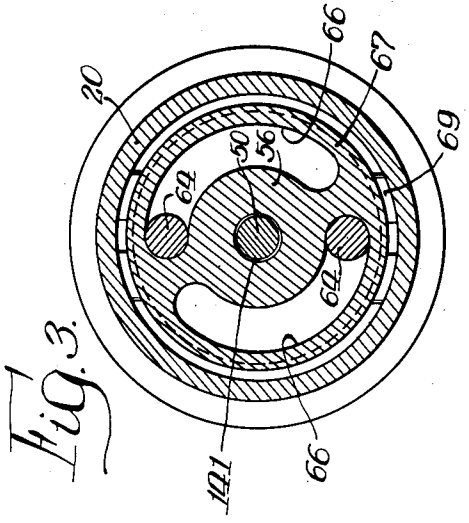


Fig. 3.

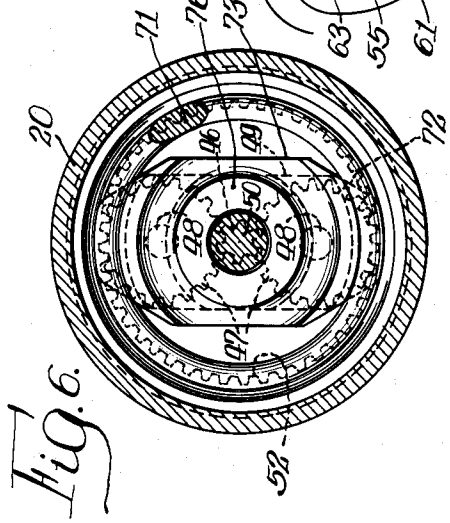


Fig. 6.

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United States Patent Office 2,716,475 Patented Aug. 30, 1955

2,716,475

IMPACT TOOLS

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Original application March 17, 1949, Serial No. 81,966, now Patent No. 2,585,486, dated February 12, 1952. Divided and this application September 7, 1951, Serial No. 245,526

5 Claims. (Cl. 192—30.5)

This application is a division of my copending application Serial No. 81,966, filed March 17, 1949, now Patent 2,585,486, issued February 12, 1952.

This invention relates to improvements in portable power operated impact tools for setting and removing nuts and bolts by power, and more particularly to improvements in an impact clutch mechanism employed in such impact tools for clutching and declutching the hammer and anvil members and for imparting rotational hammer blows on the anvil member to tighten or loosen a nut or bolt.

In my copending parent application Serial No. 81,966, I have described and claimed various novel features of my invention and more particularly an impact clutch mechanism comprising a novel arrangement of a pair of rotatable hammer and anvil members, impact means, and actuating means for the impact means. The present divisional application is directed more specifically to a novel arrangement of a pair of rotatable hammer and anvil members with a supporting shaft whereby to permit limited angular or cocking movement of the anvil member relative to the shaft.

Accordingly, the primary object of my invention is to provide an impact clutch mechanism having novel means for avoiding the inherent tendency of the anvil member of the device toward binding due to misalignment on the drive shaft during operation of the tool.

A further object of the invention is to provide in an impact clutch mechanism a novel combination of axially disposed rotatable anvil and hammer members and a supporting shaft with means for permitting limited angular or cocking movement of the anvil member relative to the shaft during operation of the tool.

Other and further objects and advantages of the present invention will become apparent hereinafter as the description progresses, reference being had to the accompanying drawings in which:

Fig. 1 is a longitudinal sectional view with portions thereof in elevation and showing an impact tool comprising a preferred embodiment of the present invention;

Fig. 2 is a fragmentary longitudinal sectional view of the impact tool taken on the line 2—2 in Fig. 1;

Fig. 3 is a transverse sectional view taken on the line 3—3 in Fig. 1;

Fig. 4 is a transverse sectional view taken on the line 4—4 in Fig. 1;

Fig. 5 is a transverse sectional view taken on the line 5—5 in Fig. 1;

Fig. 6 is a transverse sectional view taken on the line 6—6 in Fig. 1; and

Fig. 7 is a perspective view of the impact unit and gear drive of the tool, showing the hammer in aligned but disassembled position with respect to the spindle and impact rods or jaws.

Referring now to the drawings, the impact tool illustrated therein comprises a four section housing including an impact unit section **20** at the forward portion of the

tool, a gear section **20***a* at the intermediate portion thereof, a field case **21** containing an electric motor, not shown, and a cap **22** at the rear end thereof. As more fully described in my earlier application Serial No. 81,966, the field case **21** is provided at its rearward end with supporting structure for a ball bearing unit having journaled therein the rear end of an armature shaft **26**. A motor reversing ring **28**, which may be composed of plastic or other suitable material, abuts the extreme rearward end of the housing section **21** and is held in position thereagainst for limited rotative movement by the cap **22**. As explained more fully in my earlier application, relative rotative movement of the plastic ring **28** in one direction or the other serves to reverse the operational direction of the motor contained in the housing section **21**. The armature shaft **26** is journaled on its forward side in a ball bearing unit **37** supported within a bushing **38** which is secured within the axial opening of an annular, cup-shaped cross portion **39** extending diagonally inwardly and rearwardly from the forward end of the housing section **20***a*.

Rotary motion of the electric motor contained in the housing section **21** is transmitted to the impact unit within the forward housing section **20** by means of a planetary gear train. The forward end of the armature shaft **26** is shaped to provide a gear **46** in constant mesh with a pair of diametrically opposed, planetary gears **47** mounted for free rotative movement on short shafts **48** secured in a spider member **49**. The spider member **49** is integrally formed as the upper or rearward end portion of a drive shaft **50** and its extreme rearward end is journaled for free rotative movement in a thrust ball bearing unit **51** also received in the bushing **38**. The planetary gears **47** are also in constant mesh with an internal ring gear **52** secured in and projecting somewhat beyond the extreme forward open end of the housing section **20***a*. The ring gear **52** is provided with an outwardly directed annular shoulder **53** which forms a seat against which the rearward end of the impact housing section **20** may be drawn up into tight engagement by means of suitable screws, as shown more fully in my copending application Serial No. 81,966. By this arrangement the assembly and disassembly of the tool is greatly facilitated. The entire front end of the tool including the impact mechanism, its housing section **20**, the drive shaft **50**, the spider **49**, the planetary gears **47** and the ball bearing unit **51** may be withdrawn or separated as a unit from the rear portion of the tool merely by removing the screws and sliding the planetary gears **47** relative to the pinion **46** and the ring gear **52** out of meshing engagement therewith. As will become apparent hereinafter, the impact mechanism may then be removed rearwardly out of the housing section **20**.

The forward end of the drive shaft **50**, comprising the main supporting member or stem of the impact unit, is received within an axial bore **54** of an impact spindle or anvil **56** which terminates in a square section **57** exteriorly of the housing section **20** and is adapted to mount and drive a nut or bolt engaging socket (not shown) in the usual manner. The impact spindle **56** is adapted to rotate within a bushing **58** secured by a set screw **59** within the axial opening at the forward end of the housing section **20**. A rotatable hammer member **60** is mounted on the drive shaft **50** above the impact spindle **56** for free and independent rotative movement relative thereto, the hammer **60** being provided with an axially extending central opening **55** in which is secured a needle bearing **61** facilitating rotative movement between the parts. A thrust washer **62** is also mounted on the shaft **50** between the hammer **60** and the impact spindle **56**.

The hammer **60** is provided with longitudinally ex-

tending openings 63 on diametrically opposite sides of the central axial opening 55, the openings 55 and 63 thus being parallel and in alignment, and the hammer 60 thus being of solid, rugged construction and possessing great mass and inherently great strength. A pair of impacting rods or jaws 64 are slidably mounted within and extend through the hammer openings 63, the lower ends of the rods 64 being adapted to project when in impacting position, as shown in Figs. 1 and 3, beyond the forward end surface of the hammer 60 and into arcuate slotted openings 66 in a radially outwardly directed flanged portion 67 of the anvil 56. As will be explained in detail hereinafter, the impact rods 64 rotate with the hammer 60 and are adapted to strike the anvil 56 at the abutments formed by the ends of the arcuate slots 66 to cause rotation of the anvil.

The upper ends of the impact rods 64 are each provided with an annular groove 68 into which are received the edges of bifurcated portions 69 extending radially from the forward end of an arched cam lifter plate 70. The impact rods 64 may be readily mounted on the cam lifter plate 70 by sliding them inwardly into the slots 65 thereof from the outer edges of the bifurcated portions 69 and they are then free to rotate relative thereof. The cam lifter plate 70 is provided with a central axial opening through which the drive shaft 50 extends, the plate 70 being slidable on the shaft 50 for axial movement relative thereto. A relatively heavy compression spring 71 encircles the cam plate 70 and the forward end thereof seats against the radial edge portions of the bifurcated portions 69 outwardly of the heads of the impact rods 64. By this arrangement the cam plate 70 is constantly urged forwardly so as to project the impact rods 60 into impacting position. The spring 71 also serves the further purpose of retaining the impact rods 60 in assembled position within the slots 65 of the bifurcated portions 69 during assembly of the various parts of the tool and permits the impact mechanism to be assembled and handled as an integral unit.

The upper end of the compression spring 71 bears and seats against radially directed flanges 72 of an upper spring plate 73, this plate 73 also being slidable on the drive shaft 50 and being separated from the forward end surface of the spider member 49 by thrust bearings 74. The upper spring plate 73 is flattened around the shaft 50 on the forward side thereof to provide a surface 76 for engagement with a flat cooperating surface 77 on the rearward end of the cam lifter plate 70, the surface 77 being adapted to be moved upwardly into such engagement upon extreme rearward movement of the lift plate 70.

On the forward or under side of the cam lifter plate 70 are two projecting cam members 78, one on each side of the drive shaft 50, which are forced forwardly into constant bearing engagement with cam surfaces 79 on a drive cam member or bushing 80 fixed on the drive shaft 50 for rotation therewith and restrained against axial movement relative thereto. The cam surfaces 79 slope upwardly in either direction from the low point thereof thus enabling the tool to be operated in a reversible manner in either direction of rotation. The cam bushing 80 is secured on the shaft 50 by means of balls 81 disposed within aligned and cooperating openings 82 and 83 in the bushing 80 and the drive shaft 50, respectively. The forward end of the cam bushing 80 is received within an annular recess 84 in the rearward end of the hammer 60 and bears against a thrust washer 86 at the back of the recess.

As shown clearly in Figs. 2, 5, and 7 of the drawings, the hammer 60 is provided with two diametrically opposed and aligned ears or projections 87 which extend rearwardly in parallel relationship up within the spring 71 and on each of the opposite sides of the cam bushing 80, the cam lifter plate 70 and its cam portions 78. The outer peripheries of the hammer ears 87 are curved so as to conform to the curvature of the spring coils but they are spaced therefrom to permit free relative movement between the spring 71 and the hammer 60. The inner surfaces of the hammer ears 87 are smooth, flat and parallel with each other and parallel with the drive shaft 50, and are in close, slidable engagement with the smooth, flattened surfaces 88 on the opposite sides of the cam lifter plate 70.

The hammer ears 87 thus serve to maintain the parts of the impact unit in perfect alignment and insure perfect alignment of the slots 65 of the bifurcated portions 69 of the cam lifter plate 70 with the axial hammer openings 63. As a consequence, the impact rods 64 are always maintained in perfect and absolute alignment within the hammer openings 63 for free relative slidable axial movement to and from impacting position. Since the parallel inner surfaces of the hammer ears are parallel with the axial openings 63 of the hammer 60 and bear against the opposite flat parallel surfaces of the cam lifter plate 70, absolute parallelism between the parts is maintained at all times. Any forces tending to rotate the hammer 60 in one direction relative to impact rods 64 so as to tend to effect their disalignment and to cause the rods to bear against one side of the hammer slots 63, also, through the hammer ears 87, tend to rotate the rod-carrying cam lifter plate 70 in the same direction and with equal force, thus preventing any such disalignment or binding engagement. It is to be noted that the opposite longitudinal side edges of the cam portions 78 of the cam lifter plate extend slightly beyond the outer periphery of the driving cam bushing 80 so that the cam bushing 80 is out of engagement with inner surfaces of the hammer ears 87. The hammer ears 87 also guide the cam lifter plate 70 in a true axial direction when it is cammed upwardly by the cam bushing 80 and serve to restrain the plate 70 against rotative movement.

On the under side of the housing section 21 is a switch grip handle 89 having a trigger block 94 slidably received within an opening in the front of the handle and adapted to cooperate with a switch device within the handle for closing and opening the electric circuit to the motor, as described in detail in my earlier application Serial No. 81,966. The trigger block 94 may be held in its inward position for continuous operation of the tool by a spring pressed locking plunger 112 extending through the forward portion of the handle and may be quickly unlocked by depressing and releasing the trigger block 94.

When a nut or bolt is to be tightened, the operator first places the socket (not shown), secured on the square 57, over a nut or bolt head and then presses the trigger block 94 inwardly to complete the electric circuit and effect operation of the tool motor. As the armature shaft 26 rotates, the pinion gear 46 on its forward end rotates the planetary gears 47, causing them to move bodily around within the ring gear 52 in a counterclockwise direction as viewed in Fig. 6 and thus causing rotation of the spider 49 and the drive shaft 50. This driving torque is further transmitted to the impact rods 64 and the hammer 60 through the cam bushing 80 and the cam lifter plate 70.

At the outset of the tightening operation, the impact rods 64 are maintained in projected position within the slots 66 of the impact spindle 56, as shown in Figs. 1 and 3, and abut against the respective ends thereof so as to rotate the impact spindle 56 and the socket and tighten the nut or bolt. As long as the nut or bolt is relatively free running, the impact rods 64 remain in constant engagement with the impact spindle 56 and the projecting cam portions 78 of the cam lifter plate 70 are disposed at the low points of the cam faces 79 of the bushing 80. However, when the nut or bolt approaches tightened position and resistance to rotation is sufficient to overcome the driving torque of the tool motor and the expansive force of the spring 71, rotation of the impact spindle 56, impact rods 64, hammer 60, and the cam lifter plate is arrested. But since the spindle 50 and the cam

bushing 80 continue to rotate under the driving force of the motor 29, the cam lifter plate 70 will be cammed rearwardly in a straight longitudinal or axial direction under the guiding influence of the hammer ears 87. Since the cam lifter plate 70 carries and supports the impact rods 64, these rods also are moved rearwardly relative to the hammer 60 and they are thus withdrawn from the slots 66 to a point slightly above the rearward surface of the then stationary impact spindle 56. When the impact rods 64 are fully withdrawn the projecting cam members 78 of the cam lifter plate 70 engage the cam surfaces 79 of the bushing 80 near the top of their high points.

In this withdrawn rearward position, the impact rods 64 are out of driving engagement with the impact spindle 56 and hence are free to rotate under the driving torque of the drive shaft 50. The impact rods 64 and the hammer 60, consequently, resume rotation in a clockwise or tightening direction, pass over the narrow lands between the impact spindle slots 66 and then are projected under the urge of the spring 71 back into the slots 66 so that the impact rods 64 will be impacted against the far end of the slots 66 upon continued rotation. From the time that driving engagement is broken off, as above described, until the rods 64 impact against the ends of the slots 66, the drive shaft 50 accelerates under the driving force of the motor 29 at an ever increasing rate so that the impact rods 64 strike the impact spindle 56 with great force. Because of the solid, massive construction of the hammer 60, this impacting force is of relatively great magnitude.

Because of the fact that the impact rods 64 are circular and are slidably received in the slots 65 of the cam lifter plate 70, they are free to rotate relative thereto. As a result of this rotative connection, the rods 64 are rotated slightly as they are moving into impacting position against the far ends of the slots 66, thus presenting different circumferential portions of the lower or blow-delivering ends of the rods to the slot abutments for each blow. The rods, therefore, present new wearing surfaces and have a long operable life.

It is to be noted further that when the driving engagement is initially broken off between the impact rods 64 and the impact spindle 56, the cam portion 78 of the cam lifter plate 70 engages the cam bushing 80 near the high point on its cam surface. Consequently, as impact rods 64 again accelerate toward impacting engagement with the impacting spindle 56, the heavy spring 71 will force the cam lifter plate 70 axially forwardly, the cam portion 78 moving downwardly and forwardly over the cam surface of the cam bushing 80 toward its low point. The cam lifter plate 70 is thus caused to accelerate ahead of the drive shaft 50 and the impact rods 64 are caused to strike the impacting spindle 56 with an increased impacting force and at a rate of rotation over and above that of the drive shaft 50. After impact, which causes further limited rotative and tightening movement of the impact spindle 56, the impact rods 64 are again withdrawn as above described and are again rotated relative to the impact spindle 56 and thrown again into impacting engagement with the spindle 56 at the far ends of the next succeeding slots 66. The impact rods 64 thus strike the impact spindle 56 two impacting blows for every revolution of the hammer 60 and these blows may be repeated until the work has been tightened to the desired degree.

During the operation of the tool when the impacting blows are being delivered to the work, the impact rods 64 are freely slidable within the hammer slots 63 to and from spindle engaging position since they are maintained in perfect alignment therewith by the hammer ears 87 which engage the flat sides 88 of the cam lifter plate 70. The hammer ears 87 also by their slidable engagement with the flat sides of the cam lifter plate 70 cause that plate to move in a truly longitudinal direction and parallel to the inner sides of the hammer ears. Since the inner sides of the hammer ears 87 are also parallel with the hammer slots 63, the sliding movement of the rods 64 is also parallel therewith. Consequently, it is apparent that by the arrangement of the parts they are maintained in the desired alignment so that their free operation is achieved at all times and no binding of the rods 64 in the hammer slots 63 results.

When it is desired to utilize the tool for the purpose of loosening a nut or bolt, the operator need only rotate the plastic reversing ring 28 to its other position as above described, and the electric motor will thus operate in the reverse direction and the drive shaft 50 will also be rotated in the reverse direction. Since the work is in a tightened state, the impact rods 64 will immediately commence imparting blows against the spindle 56 until the resistance to rotation offered by the work is less than the driving torque of the cam lifter plate 70, at which point the impacting blows cease and the impact rods 64 remain in constant free running engagement with the impact spindle 56. The tool may then be allowed to remain in engagement with the work until the work is entirely disengaged from a threaded opening.

Having thus described the general mode of operation of a rotary power tool utilizing an impact clutch mechanism of the type illustrated, I will now describe the specific improvement to which the present application is particularly directed. It will be understood that the specific features hereinafter described are broadly applicable to various impact clutch arrangements other than those involving the specific impacting and actuating means described above.

As shown more particularly in Figs. 1 and 2 of the drawing, the lower or forward end portion of the drive shaft 50 is constructed so as to permit limited tilting or cocking movement of the impacting spindle or anvil 56 relative to the drive shaft 50. When the tool is in operative engagement with the work and is in the hands of an operator, the impacting forces and the handling of the tool by the operator tend to force the impacting spindle 56 out of alignment with the drive shaft 50. Because of this inherent tendency toward misalignment which would cause binding of the impact spindle 56 if there was a tight fit between the parts and hence would reduce the effective output of the driven shaft 50 and the spindle 56, the extreme forward end of the driven shaft 50 is rounded or tapered as at 134 leaving a reduced flat end face 136 for abutting engagement with a thrust washer 137 of hardened metal retained within the forward end of the bore 54 of the impact spindle 56 by a spring ring 138. Immediately rearwardly of the tapered portions 134, the drive shaft 50 is provided with a relatively short, annular bearing surface 139 which is adapted to engage the walls of the spindle bore 54 over a limited area. This limited bearing surface 139 serves to prevent too loose a mounting between the spindle 56 and the shaft 50 but at the same time permits limited tilting movement therebetween. The forward and inward edges of the bearing surface 139 wear away or round off after a relatively short period of tool operation so that this bearing surface, as the tool wears in and the parts adjust themselves to each other, also becomes slightly rounded to the desired degree.

The portion of the drive shaft 50 next above or rearwardly of the bearing surface 139 is of reduced diameter, as indicated at 140, and is slightly spaced from the sides of the spindle bore 54 and the thrust washer 62 to provide a clearance 141 for the limited tilting movement. The diameter of the drive shaft portion indicated at 142 within the central bore 55 of the hammer 60 is also of lesser extent than the diameter of that bore so as to provide clearance therebetween, and the needle bearing 61 serves to maintain the proper bearing relationship between the hammer 60 and the drive shaft 50. Also for the purpose of permitting the relative tilting movement, the marginal or radially outer portion of the annular flanges 67 of the impact spindle 56 are cut away so as not to extend rear-

wardly as far as the central portion thereof, thus providing a slight clearance **143** between the opposed end faces of the hammer **60** and the spindle **56** at their outer portions and for engagement therebetween and with the thrust washer **62** only at their central portions around the drive shaft **50**.

Thus, by reason of the special construction of the shaft and the pair of anvil and hammer members thereon, the clearances indicated at **141** and **143** are provided so that the anvil or spindle **56** can undergo limited cocking movement or axial misalignment without binding and thereby facilitating easy operation of the tool.

It is to be further understood that while the preferred embodiment of the present invention has been described above as an impact wrench, the very same tool by means of special attachments secured on the square **57** may be adapted for other uses such as a screw driver, tapper, reamer, drill, sander, polisher or wire brush driver. When utilized for these latter purposes, the tool will be primarily free running without operation as an impacting wrench. However, such impacting features are permitted to become effective when used as a screw driver or when used as a drill and the drills or bits become frictionally held against rotation before the drill hole has been completed. The impacting blows of the hammer rods **64** then are applied to the drills and bits to overcome such frictional resistance and any overload on the motor is eliminated.

Although there has been described above and illustrated in the drawings a preferred embodiment of an impact tool comprising the present invention, it is apparent that modifications and changes may be made in the details of structure and mode of operation without departing from the spirit and scope of the appended claims.

I claim:

1. An impact clutch comprising relatively rotatable axially aligned hammer and anvil members having aligned axial bores therein and an aligning shaft in said hammer bore supporting said hammer for rotatable movement thereon and extending into the bore of said anvil for rotative movement relative thereto, the portion of the shaft within said anvil bore being of lesser diameter and spaced from the walls of said bore along a major portion of its extent and having a generally cylindrical enlargement of relatively short axial extent in bearing engagement with the walls of said bore whereby said anvil and hammer members may move angularly out of axial alignment to a limited extent without interference with free relative rotation.

2. An impact clutch comprising relatively rotatable axially aligned hammer and anvil members having aligned axial bores therein and an aligning shaft in said hammer bore supporting said hammer for rotatable movement thereon and having a free end extending into the bore of said anvil for rotative movement relative thereto, the portion of the shaft within said anvil bore being of lesser diameter and spaced from the walls of said bore along a major portion of its extent and having adjacent the free end thereof a generally cylindrical enlargement of relatively short axial extent in bearing engagement with the walls of said bore whereby said anvil and hammer members may move angularly out of axial alignment to a limited extent without interference with free relative rotation.

3. An impact clutch comprising relatively rotatable axially aligned hammer and anvil members having aligned axial bores therein and an aligning shaft in said hammer bore supporting said hammer for rotatable movement thereon and having a free end extending into the bore of said anvil for rotative movement relative thereto, the portion of the shaft within said anvil bore being of lesser diameter and spaced from the walls of said bore along a major portion of its extent and having its free end rounded and having a generally cylindrical enlargement of relatively short axial extent adjacent said rounded end in bearing engagement with the walls of said bore whereby said anvil and hammer members may move angularly out of axial alignment to a limited extent without interference with free relative rotation.

4. An impact clutch comprising relatively rotatable axially aligned hammer and anvil members having aligned axial bores therein and having axially opposed wall portions with abutments on the axially opposed wall portion of the anvil member, impact means rotatably movable with said hammer and axially movable independently of said hammer into and out of the path of the abutments for clutching and declutching said members and for imparting rotational hammer blows on the anvil member at said abutments, and a driving shaft in said hammer bore supporting said hammer for rotatable movement therein and having a free end extending into the bore of said anvil for rotative movement relative thereto, the portion of the shaft within said anvil bore being of lesser diameter and spaced from the walls of said bore along a major portion of its extent and having a generally cylindrical enlargement of relatively short axial extent in bearing engagement with the walls of said bore, and said hammer and anvil members being in bearing engagement with each other at their central axial portions and being axially spaced from each other at their outer portions whereby said anvil and hammer members may move angularly out of axial alignment to a limited extent during operation thereof without interference with free relative rotation.

5. In an impact clutch, a rotatable hammer member having an axial bore therethrough, a rotatable anvil member having an axial recess in alignment with said bore, and a shaft extending through said bore and having an end portion extending into said recess, said end portion being provided adjacent its outer extremity with a generally cylindrical enlarged bearing portion of relatively short axial extent substantially less than the axial depth of said recess and disposed in bearing engagement with the wall of said recess for normally retaining the hammer and anvil members in axial alignment, the remainder of said end portion inwardly of said bearing portion being of reduced diameter relative to said recess whereby to provide an annular clearance therebetween, and said hammer and anvil members having their opposed ends axially spaced from each other adjacent the outer peripheries thereof, whereby to permit limited angular displacement of said anvil member relative to said shaft and said hammer member without interference with relative rotary movement of the parts.

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