Dec. 12, 1950 G. C. WILHIDE ET AL 2,533,703

IMPACT NUT RUNNER

Filed June 21, 1947 5 Sheets-Sheet 1

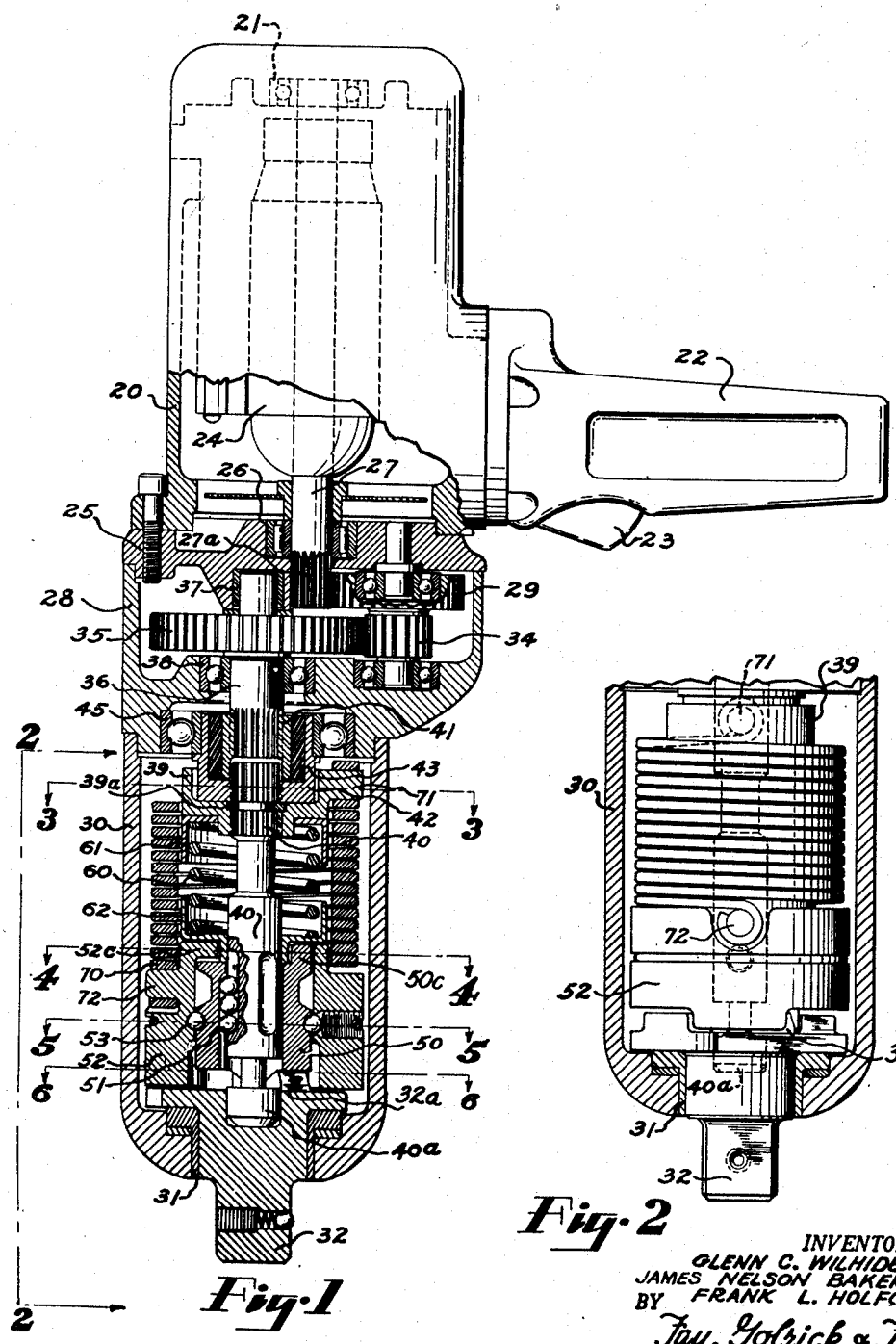


Fig. 1

Fig. 2

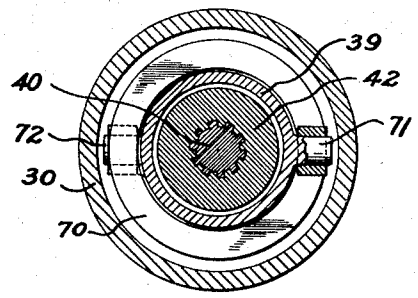
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Fig. 3



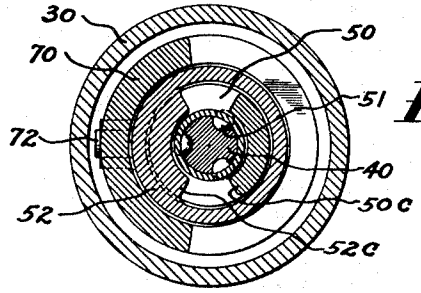
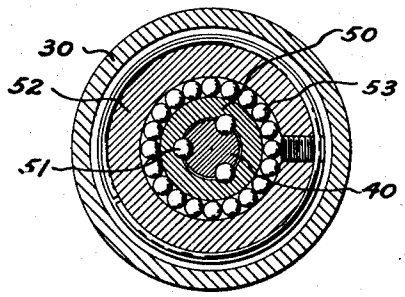


Fig. 4

Fig. 5



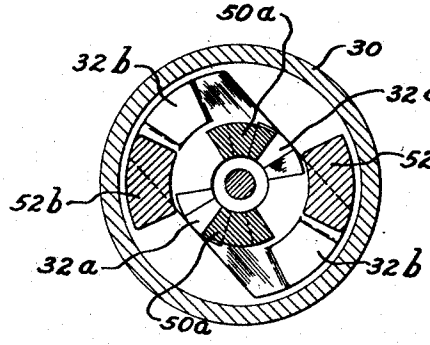


Fig. 6

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IMPACT NUT RUNNER

Filed June 21, 1947 5 Sheets-Sheet 3

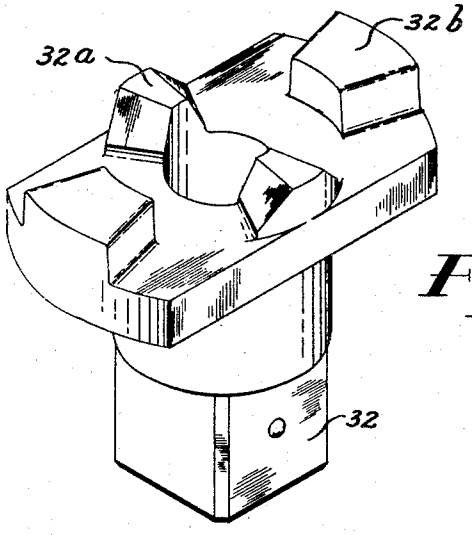


Fig. 7

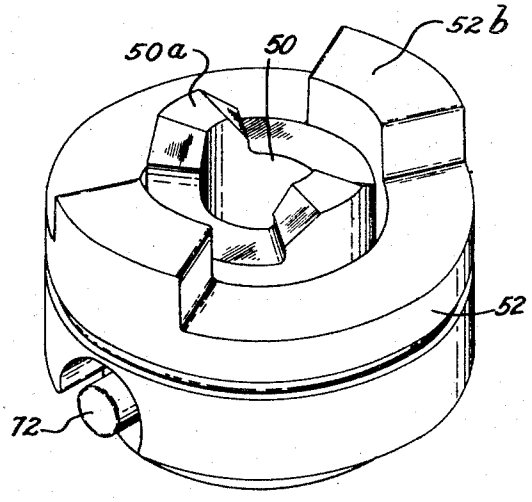


Fig. 8

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IMPACT NUT RUNNER

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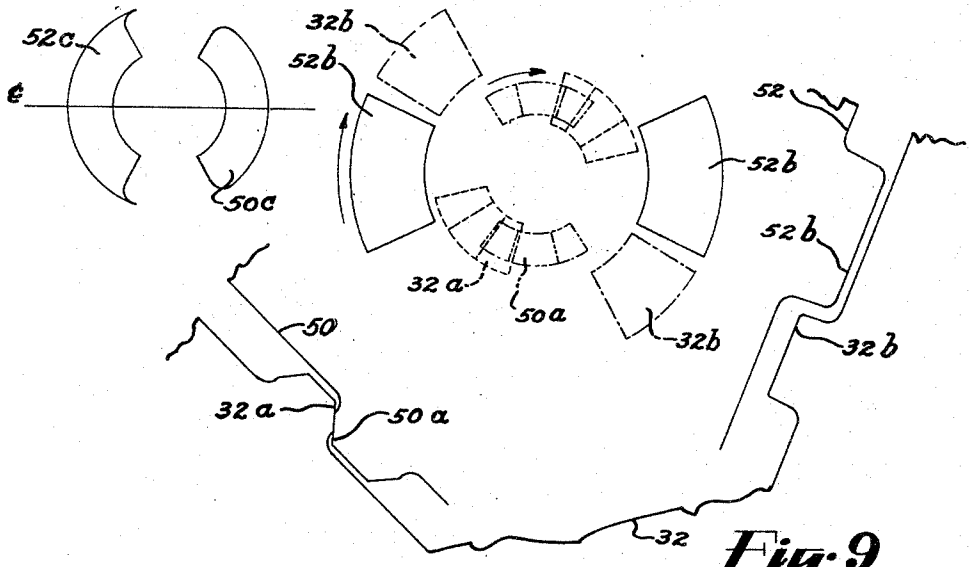


Fig. 9

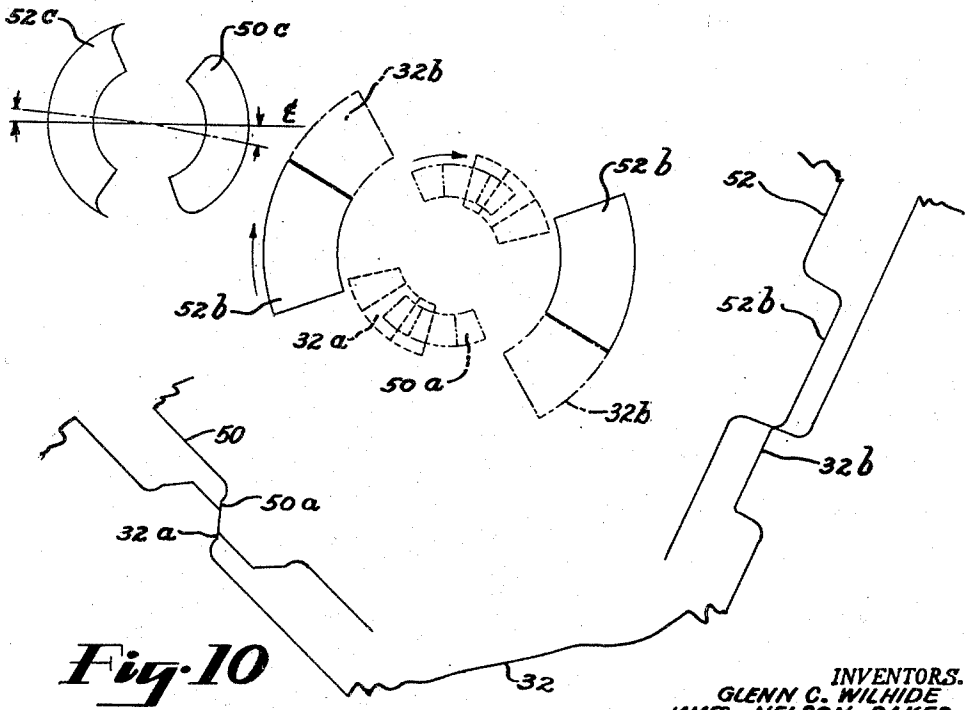


Fig. 10

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IMPACT NUT RUNNER

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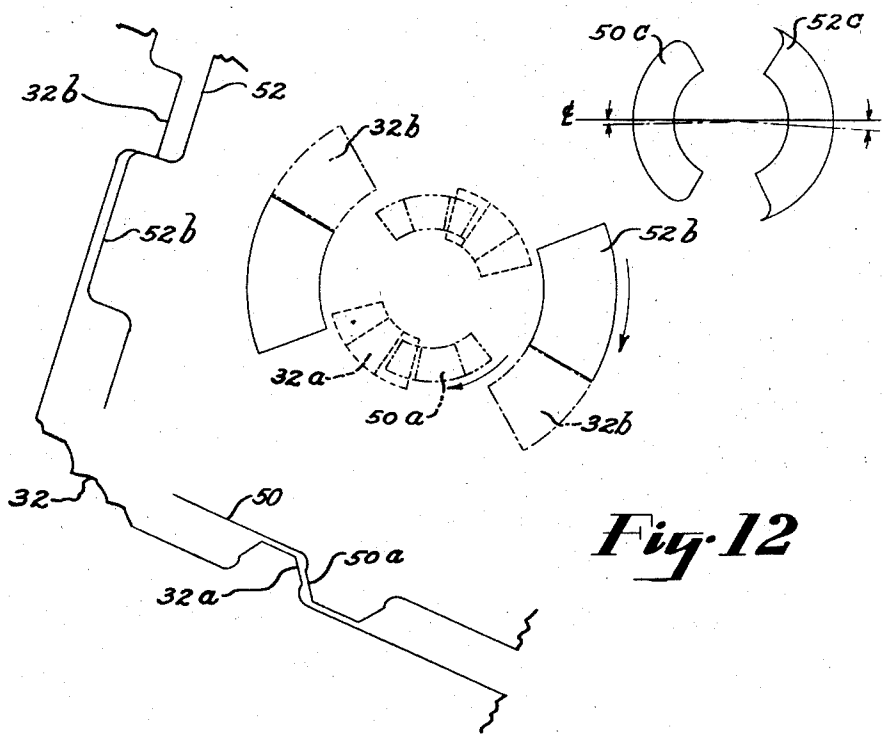


Fig. 12

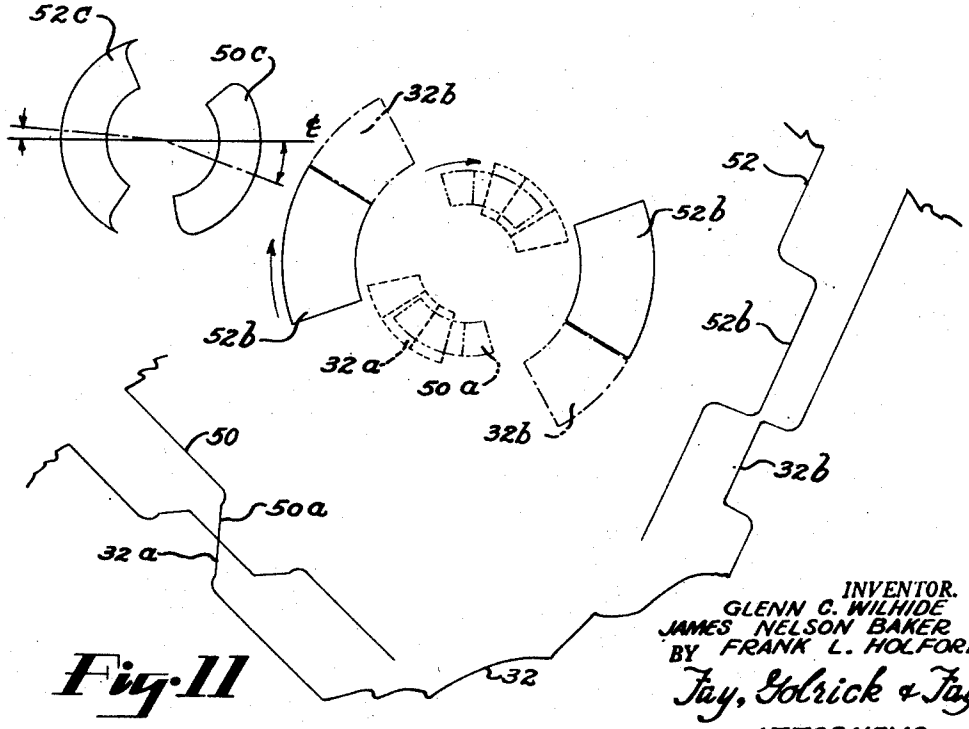


Fig. 11

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Patented Dec. 12, 1950

2,533,703

UNITED STATES PATENT OFFICE

2,533,703

IMPACT NUT RUNNER

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Application June 21, 1947, Serial No. 756,114

13 Claims. (Cl. 192—30.5)

1

This invention relates to power operated portable tools of the general type commonly in use for running nuts and bolts into or out of assembly and to hammer mechanisms for cinching or loosening bolts, nuts, etc.

The general object is to provide a clutch and hammer mechanism which will be durable and accurate in torque effort throughout long periods of use and yet be sufficiently low in overall weight as to not cause undue fatigue when used throughout a work day.

More specifically, our invention is concerned with the provision of a novel hammer mechanism which can be associated with the usual yieldable tooth clutch mechanisms characteristic of portable nut runner tools and in such manner as not to be involved mechanically in that part of the transmission mechanism of the tool which exerts the normal torque effort upon the wrench or socket.

A further object is the provision of a power driven portable nut runner incorporating a hammer mechanism in such manner that a rebound build up of the impact forces can be accomplished without introducing additional problems other than the ultimate efficient functioning of the parts comprising the hammer mechanism.

Another object is the provision of a nut runner mechanism of the inertia hammer type wherein the force of the hammer blows may be transmitted directly to the socket or wrench mechanism thereby to avoid the transmitting of the hammer blows through the tooth and cam portions of the clutch mechanism.

Other objects and advantages of the invention will be apparent from the following detailed description of preferred forms of embodiment of the invention, reference being made to the accompanying drawings wherein:

Fig. 1 is an elevational view of an electrically driven portable nut runner incorporating the features of our invention;

Fig. 2 is an assembled elevation of the hammer, clutch and socket mechanism;

Fig. 3 is a cross-sectional view taken along the line 3—3 of Fig. 1;

Fig. 4 is a cross-sectional view taken along the line 4—4 of Fig. 1;

Fig. 5 is a cross-sectional view taken along the line 5—5 of Fig. 1;

Fig. 6 is a cross-sectional view taken along the line 6—6 of Fig. 1;

Fig. 7 is a perspective view of the socket driving means compromising a driven clutch means and impact receiving or anvil means;

Fig. 8 is an upside down perspective of the driving clutch and hammer means;

Fig. 9 shows a diagrammatic linear and an angular representation of the clutch teeth and

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hammer impact shoulders during normal driving of the socket wrench member;

Fig. 10 is a diagrammatic linear and angular representation of the clutch teeth and hammer impact shoulders during the camming period of the clutch teeth and while hammer spring is being tensioned;

Fig. 11 is a diagrammatic linear and an angular representation of the clutch teeth and hammer impact shoulders, but showing the impact shoulders shifted to escape position upon complete driving disengagement of the clutch teeth but before the hammer has been accelerated by the torsion spring; and

Fig. 12 is a diagrammatic linear and an angular representation of the clutch teeth and impact shoulders at the time of impact.

Portable nut runners have been produced heretofore provided with some form or other of hammer means for delivering a series of blows to the wrench socket member after the nut, bolt, etc. has become seated. In some, this action has been obtained by having the clutch teeth on the driving clutch member rotatively pound upon the clutch teeth of the driven clutch member, the latter comprising an integral part of, or being directly connected to, the socket member. This rapid engaging and disengaging of the clutch teeth under such impact conditions caused deterioration of the clutch members to such an extent as to require frequent replacement. Other types of tools incorporated energy storing means in the form of distortable rubber torsion members or springs which comprised part of the torque transmitting means from motor to socket wrench or reactive auxiliary cam and clutch means, all of which resulted in mechanical compromises with resulting lack of durability and efficiency.

In our present invention the well known tooth cam type of clutch mechanism is used for normal transmission and delivery of torque effort to the wrench or socket member up to a predetermined amount and the clutch maintaining spring has no other purpose or function than to maintain the clutch teeth engaged up to such predetermined torque effort after the nut or bolt becomes seated and partly cinched, but gives way before the nut or bolt is cinched home. We propose to utilize a rotating hammer arrangement which will be actuated by energy stored in a second spring arrangement and this storing of energy is accomplished during interruption of the rotative movement of the hammer by a continuously rotating drive shaft. A camming period immediately follows the seating of the nut or bolt with consequent arresting of the rotating movement of the socket or wrench member and the timed action of the hammer is such that under dynamic

conditions a torsion pendulum build up is obtained.

The hammer spring is in the form of a torsion member connected at one end to the drive shaft of the tool and to the hammer at the other end, thus forming a flexible connection between the hammer and the shaft. The hammer is carried by the driving clutch member in an independently revolvable manner but is fixed axially on this clutch member thereby to be shiftable axially with this clutch member. The torsion spring comprises no part of the means transmitting torque effort through the clutch member to the socket member, thus prolonging the life of the clutch members. A further advantage is that the clutch maintaining spring is burdened with no other function than to maintain the clutch and camming teeth in socket driving engagement up to a predetermined torque effort upon the socket member and accordingly can be designed accurately to perform this single function.

Referring to the drawings, in Figs. 1 and 2 we show a portable, electrically-driven nut runner tool embodying the features of our invention.

The tool casing comprises a housing structure forming a motor housing member 20, a handle 22 mounted thereon having a switch mechanism 23 associated therewith for convenient starting and stopping of the motor 24. The upper end of the motor housing is formed to be closed as shown in Fig. 1, to support a bearing 21 attached to the top end of the motor shaft.

A reduction gear mechanism is provided and gear supporting frame structures 25 and 28 are secured respectively to the lower end of the motor housing and to each other. The frame member 25 supports a motor shaft bearing 26 which supports the lower end 27 of the motor shaft. The second frame structure 28 affords bearing mountings for reduction gearing.

A fourth housing member 30, in the form of a cylinder having a converging bottom is secured to the frame member 28 and supports a bearing member 31. The bearing member 31 serves to support an anvil member or wrench turning means 32 against the thrust of a clutch closing spring 60, as will be described.

The motor shaft 27 has a pinion formation 27a on the lower end thereof and meshing with an intermediate gear 29, preferably integrally formed with a pinion 34. The pinion 34 drives a large reduction gear 35 carried on a stud shaft 36. The stud shaft 36 is supported by bearings 37 and 38, these bearing members being supported by the frame members 25 and 28, respectively.

The mechanism arrangement so far described is to be found in nut runner tools heretofore manufactured and sold and generally is designed for the delivery of rated torque effort to be exerted upon a socket, nut or bolt and the like, through a yieldable cam tooth type clutch mechanism maintained normally closed by the spring 60. A clutch driving shaft 40 is supported independently of the gear reduction mechanism, as will be described.

The gear shaft 36 is arranged in alignment with the clutch shaft 40. The shafts 36 and 40 are connected by a flexible coupling mechanism comprising an inner sleeve 41 connected to shaft 36 by a fluted or spline connection. An outer sleeve member 42 is spline connected to the upper end of shaft 40. A rubber medium 43 is in pressed relation to the outer cylindrical surface of member 41. Member 42 is supported by a bearing mechanism 45, and the bearing mechanism 45 is supported by the frame member 28. It will be seen that shaft 40 thus is power connected to the motor 24 and shaft 40 will be constantly driven so long as the motor is energized. The lower end 40a of shaft 40 extends into a bore formed in the socket driving member 32 to provide a bearing for supporting the lower end of shaft 40. The upper portion of shaft 40 has secured thereto an annularly flanged spring seating and spring anchoring member 39 which thrusts against small collar 39a seated in a groove formed in the splined part of shaft 40.

Spline connected to the intermediate part of shaft 40 is a clutch member 50, there being three sets of bearing balls 51 riding in keyways spaced 120° apart and formed in the shaft 40 and clutch member 50 so that the clutch member 50 is shiftable along shaft 40.

Loosely mounted upon the driving clutch member 50 is a cylindrical inertia or hammer member 52, held rotatively upon clutch member 50 by bearing balls 53 which roll in external and internal grooves formed respectively in the outer face of the clutch member 50 and the bore face of hammer member 52 whereby revolvable displacement between clutch member and hammer member can take place, but the two members will remain relatively axially fixed.

The lower end surface of clutch member 50 has two clutch teeth 50a of the clutch and cam type spaced 180° apart which engage two complementary clutch teeth 32a formed on the upper face of the socket wrench driving means or anvil member 32. These pairs of clutch teeth have the respective clutching surfaces thereof sloped or inclined to provide clutch opening or camming surfaces. The compressed spring 60 is of predetermined compression to maintain the clutch members normally closed against the camming action of the clutch teeth up to a predetermined torque effort factor. The spring is disposed between upper and lower flange collar members 61 and 62, respectively engaging the anchoring member 39 and the top of clutch member 50. The spring member serves to hold the camming faces of the clutch teeth 50a and clutch teeth 32a in clutch closed position until the nut, bolt, etc. become seated and partly cinched.

A second spring in the form of part of an energy storing means comprises a flat coiled structure 70 wound to extend outside of the compression spring 60. The upper end of the torsion spring terminates in a loop secured to a lug 71 formed on the flanged thrust member 39. The lower end of the spring terminates in a loop attached to a lug formation 72 formed on the hammer member 52. It has been found in practice that having the spring ends loosely connected to the hammer and to the flanged thrust member 39 will minimize rupturing of spring ends. A total play of the two spring anchorages of about four to seven degrees assures free movement of the hammer. The arrangement of the mechanism is such that this spring member is in a normally unloaded condition other than the effort required to rotate the hammer mass, but is designed to have a sufficient torsional strength to store predetermined amounts of energy when stressed by either a spring winding or unwinding exertion upon the ends of the spring. Hence, whenever relative, angular displacement of the ends 71—72 is brought about, energy is stored

and this displacement will take place during a cam escapement action of the clutch teeth brought about by an arresting of rotation of the anvil. This can happen only when rotation of member **32** is arrested by the seating action of the bolt or nut and rotation of the hammer member **52** in synchronism with clutch member **50** is interrupted. For the purpose of impact delivery a pair of arresting and impact shoulders **52b** on the hammer lower end extend into the normal path of shoulders **32b** on the clutch and socket driving member **32**. The cooperating impact surfaces of these shoulders are disposed in a plane displaced about five degrees from a vertical plane, and the shoulders of each pair are spaced 180° apart.

To prevent the hammer shoulders from rebounding or escaping from properly poised relation to the impact shoulders on the wrench driving member **32** and to the clutch teeth of clutch member **50**, a radially extending lug **52c** is formed on the top end of the hammer **52** to span about 120°. The center line of this lug is approximately coincident with the center line of the hammer shoulders **52b** and the hammer shoulders extend over an arc of about 50°. A cooperating lug **50c** is formed on the top of clutch member **50** to extend into the path of the hammer lug. The lug **50c** spans an arc of about 120° and normally is disposed approximately radially opposite lug **52c**. Lug **50c** has its center line displaced about 90° from the center line of clutch teeth **50a**. Thus relative revolving movement between the hammer and the clutch member **50** is limited to a distance of not more than 60° in either direction from a normal nut running position.

To a better understanding of the sequential actions which take place between the camming and clutch teeth and between the impact shoulders of the socket or driven clutch member **32** and the shoulders **52b** on the hammer, several diagrams have been included in the drawings. They show the positions during normal nut runing; the positions at the time the nut has been seated and the clutch teeth are camming the clutch member **50** and hammer **52** upwardly, i. e., axially relative to shaft **40**; and the positions at the instant of escapement of the clutch teeth and hammer shoulders and the positions at the time of impact. In these diagrams the direction of rotation of shaft **40** is assumed to be clockwise and the diagrams have been arranged by way of both linear and annular demonstrations. It is important there to understand that the effective operating heights of both sets of clutch teeth and the impact shoulders are shown in the diagrams as being equal, although in practice the effective height of the clutch teeth is one thirty-second of an inch higher, for reasons stated later.

In Fig. 9 (normal nut running, that is, with the rotation of the wrench member **32** unarrested) the hammer shoulders are shown in rotative alignment but out of contact with impact shoulders **32b** of socket driving member **32**, but the camming faces of the clutch teeth **50a** and **32a** are being maintained in normal driving contact by spring **60**. Under this condition torsion spring **70** is unloaded.

In Fig. 10 the nut has been seated and rotation of the socket driving member has been arrested by the seating of the nut. The cam teeth of clutch member **50** have been rotated relative to the cam teeth on member **32** a distance greater than the gap normally existing between the impact shoulders of the two clutch members with the result that clutch member **50** has been revolved relative to the arrested hammer, thus slightly stressing the torsion spring **70**. The cam action of the clutch teeth in axially separating the impact shoulders and the clutch member has progressed beyond the relationship shown in Fig. 9.

In Fig. 11 the shoulders and clutch teeth have been shifted apart by the continued action of the camming faces of the clutch teeth to the point of instant escapement of the impact shoulders and the clutch teeth, and some energy has been stored in the torsion spring **70**. The spring **70** is now free to actuate the hammer. The design of the spring **70** is such as to greatly accelerate the revolving movement of the hammer in the direction of rotation of the shaft **40** and its clutch member **50**.

In Fig. 12 the position of the clutch teeth and shoulders is shown at the time of the impact or blow delivery to the socket driving member **32**. Spring **60** has snapped the clutch member **50** and hammer **52** downwardly as viewed in Fig. 1, but the impact shoulders **52b** of the hammer have struck the shoulders **32b** of the socket driving member before the camming faces of the clutch teeth of the clutch member **50** have again been revolved into driving engagement with the camming faces of the socket driving member **32**. Then follows a rebound of the hammer shoulders relative to the anvil shoulders as shown in Fig. 9.

It will be seen that at no time can the hammer be caused by spring **70** to overrun the clutch member **50**, by reason of the interacting stop shoulders **50c** and **52c** on the clutch member **50** and hammer member **52**, respectively. Also it will be understood that the design of the torsion spring **70** and the weight of the hammer will be such as to deliver the desired amount of blow to the socket driving member within the time interval indicated by the foregoing diagrammatic explanation of actions, it being understood that spring design and hammer weight may vary in accordance with the size and capacity of the portable tool contemplated. In the instant design, for example, the contemplated speed of the clutch shaft **40** is between 1200 to 1500 R. P. M. and the torsion spring **70** is wound left-handed. Thus in a clockwise drive of shaft **40**, the energy storing action on the spring **70** comprises a tendency to "wind up" the spring, although it has been found that a right-hand wound spring will give the desired action even though there would be present an "unwinding" tendency.

The actions set forth in the diagrams are on the basis of a slow motion analysis and there is approximately a 15° spacing between anvil and hammer shoulders and a clutch tooth displacement of about 23° before they are cammed apart resulting in a take-up of play and a slight distortion of the torsion spring totaling about 8°.

At speeds of about 1490 R. P. M. of the drive shaft **40**, the inertia and acceleration factors result in a torsional pendulum action producing a rebound build up. This build up may be understood by considering the upper end of the torsional spring to be stationary. The torsion spring **70** and hammer **50** then become a torsional pendulum with respect to the rotating spindle as a base, and follow the harmonic laws of a torsional pendulum with respect to the spindle. The hammer shoulders **52b**, during extremely high resistance to rotation, become practically stationary but with respect to the spindle as a base possess

a high speed of rotation relative to the spindle. Thus the slight winding of the torsion spring takes place at a high rotative speed equal to the relative speed of the hammer shoulders to the spindle. Therefore, relative to the spindle the harmonic cycle of the torsional pendulum starts with a velocity displacement from normal, and the inertia of the hammer will swing the torsional pendulum through a maximum displacement much higher than 8° with respect to the spindle. It follows, if no frictional losses were present, that the torsional pendulum would swing back through the normal to displacement with a speed equal to the velocity of displacement from zero, or with a speed equal to the actual spindle speed superposed upon the velocity of the spindle which serves as the reference base. Since the velocity of displacement is relative to the spindle as a base and is equal to the spindle speed itself, the absolute velocity of the return of the hammer through neutral is equal to twice the fixed spindle speed.

Just after returning through the neutral position with respect to the spindle and at a speed equal to the original velocity of displacement, the hammer shoulders **52** strike the anvil shoulders **32**. This dynamic blow results in a rather high coefficient of restitution from the driven system manifest in a hammer rebound at considerable velocity relative to the stationary shoulders **32**. During this recoil of the hammer shoulders, the displacement of the pendulum with respect to the spindle at impact is seen to be made at a much higher relative velocity, which is the sum of the rebound velocity relative to shoulder or anvil **32**, plus the relative velocity of the shoulders **32** to the spindle. Consequently, if torsion spring surges are disregarded the hammer returns from larger displacements at greater velocity and tends to acquire a still higher rebound velocity.

Such a rebound build up will become uniform when the velocity of the hammer at impact is equal to the sum of the relative spindle speed plus the product of the impacting velocity of the hammer and its coefficient of rebound velocity.

Should it be assumed that Fig. 9 portrays a dynamic condition with the lift cams beginning to open instead of the slow motion explanation hereinbefore given, the hammer shoulders are rebound rotating, i. e., counter-clockwise at a high velocity of rebound, probably at a higher rotational speed than that of the cams rotating in the opposite direction. The energy of this velocity will be quickly absorbed by the torsion spring. It has been determined that the hammer lugs rebound through a substantial angle very rapidly and the hammer then dwells in a suspended condition while the rotation of the drive shaft continues to wind up the spring coils until the inertia of the suspended hammer is overcome whereupon hammer rotation again takes place. While reference herein has been made to the torsion spring as an energy storing means it will be apparent to those skilled in the art that the energy of the mass spring system under dynamic condition of operation is composed of the kinetic and potential energy in the spring and the kinetic energy of the rotated hammer. The importance of the presence of the stop shoulders or lugs **52c** and **50c** on the hammer and driving clutch member respectively during a condition of extreme dynamic cyclical build up become apparent for the oscillatory motion of the hammer otherwise could increase to the point where the action of the lift cams could be completed before the hammer should normally have recovered from rebound.

We have determined also that it is advisable to have the effective height of the clutch teeth exceed the effective heights of the slightly sloping impact faces of the hammer shoulders so that the thrusting action of the clutch spring **60** will be taken by the wrench or anvil member **32** and thus prevent pounding on the spline balls **53** for the spline grooves should be of sufficient length to prevent impact on the balls **53**. Thus the clutch closing action of spring **60** terminates by having the clutch teeth lands or top faces in contact with the valley faces thereof. Wrench member **32** transmits this thrust to the radial flange of bearing member **31**. Friction between the surfaces of the hammer and anvil is reduced to a minimum. The present tool is adaptable for both screwing and unscrewing bolts, nuts, etc., by the use of a reversing motor and appropriate switch mechanism, for both the impact shoulders and the clutch teeth are functionally symmetrical.

It should be noted that the impact load, other than the usual ratchet effect, is not transmitted to the socket driving means through the clutch mechanism which drives that means. Accordingly, the clutch mechanism and spring **60** can be designed in a practical and accurate way to perform the clutch and cam functions intended for delivery of a predetermined low torque effort at the socket.

Accordingly the hammer mechanism can be designed without the distractions of considerations involved in the designing and manufacturing of clutch mechanisms which would have to withstand the shock of hammer impacts at the rate of several thousand per minute.

We claim:

1. In a tool of the character described, the combination of a casing a driving and a driven clutch member having cam declutching teeth, a spring reacting on the casing for yieldingly maintaining the members engaged against the camming action of the teeth during a normal nut running operation, a hammer revolvably mounted upon the driving clutch member and having an impact shoulder adapted to strike an impact shoulder on the driven clutch member, a drive shaft surrounded by the spring and spline connected to the hammer carrying clutch member, the last mentioned clutch member being movable on the drive shaft to move the hammer axially of said shaft, means for limiting relative revolving movement between the hammer and its clutch member, and a torsion member connecting the shaft and hammer with the said impact shoulders normally circumferentially spaced apart, the declutching camming action being such that relative rotation between the shaft and the hammer effects accumulation of rebound energy in the torsion member during the separating actions of the clutch teeth and while the hammer is being shifted by the camming action axially relative to the shaft.

2. In a portable nut runner tool, two separable tooth clutch members one of which comprises a wrench turning means, a spring yieldingly maintaining the two clutch members in driving-driven relation during a normal nut running operation, the teeth of the clutch members also comprising a camming means for effecting separation of the two clutch members against the influence of said spring when the spring rating is exceeded by the camming forces, a cylindrical hammer mem-

ber revolvably mounted upon one of the clutch members to have a revolving movement relative to that member, stop means operatively disposed between the hammer member and the last mentioned clutch member for limiting said relative revolving movement to less than 180°, a drive shaft drivingly splined to said last mentioned clutch member while permitting axial movement of that clutch member, an energy storing spring connected to the hammer member and to the shaft, and cooperating impact shoulders arranged on one end of the hammer member and on the wrench turning means clutch member and serving to effect rebound storing of energy in the energy storing member during camming apart actions of the clutch members and as an impact means upon closing of the clutch members.

3. In a portable electrically driven nut runner tool, the combination of a power driven shaft, a wrench turning means having impact shoulders thereon, a pair of clutch members having clutch opening camming teeth for connecting the shaft to the wrench turning means, one of the clutch members comprising cams integral with the wrench turning means, the other of the clutch members being spline connected to and shiftable axially of the shaft, a spring acting on the shiftable clutch member to maintain the clutch members in driving relation until the shaft exerts a predetermined maximum torque effort on the wrench turning means, a hammer member revolvably mounted on the shiftable clutch member and being in the form of a cylindrical inertia member having impact shoulders engageable with said shoulders on the wrench turning means, and a normally unloaded torsion spring connected to the hammer member and the shaft and adapted to store energy by reason of relative rebound movement between shaft and hammer subsequent to an escapement movement between the clutch members.

4. In a portable electrically driven nut runner tool, the combination of a power driven shaft, a wrench turning means into which the shaft extends, a pair of clutch members of the camming tooth type for connecting the shaft to the wrench turning means, one of the clutch members comprising part of the wrench turning means and the other clutch member being spline connected to and shiftable axially of the shaft, said clutch members comprising camming clutch teeth, a precompressed spring surrounding the shaft and acting on the shiftable clutch member to maintain the clutch mechanism closed against the camming action of the clutch teeth while the shaft exerts a predetermined torque effort on the wrench turning means through the clutch mechanism, an impact member revolvably mounted on the shiftable clutch member and being in the form of a cylindrical inertia means having impact relation intermittently with the wrench turning means upon escapement of the clutch members, and a torsion spring connected to the cylinder and to the shaft and adapted to store energy by reason of relative rebound movement between shaft and cylinder subsequent to an escapement movement between the clutch members.

5. In a nut runner, a drive shaft, a wrench turning means, a revolvable hammer, said hammer and wrench turning means having cooperating impact shoulders, a torsion spring interconnecting the shaft and hammer for actuating the hammer, cam means for causing relative movement between the hammer and wrench turning means to move said shoulders into and out of alignment with one another during rotation of said shaft, said hammer being revolvable to oscillate independently of said cam means, and cooperating stops on said hammer and cam means for limiting the degree of oscillation of the hammer relative to the cam means.

6. In a nut runner, a drive shaft, a wrench turning means, a revolvable hammer, said hammer and wrench turning means having cooperating impact shoulders, torsion storing means interconnecting the shaft and hammer for actuating the hammer, a cam member revolvably supporting the hammer for causing movement of the hammer relative to the wrench turning means to move said shoulders into and out of alignment with one another during rotation of said shaft, said hammer being revolvable to oscillate independently of said cam member, and stops on said hammer and cam member for limiting the degree of oscillation of the hammer relative to the cam member.

7. In a nut runner, a drive shaft, a wrench turning means, a clutch member driven by the shaft and a clutch member drivingly connected with the wrench turning means, said members having cooperating cam faces and resilient means biasing the cam faces in engagement with one another for effecting transmission of torque through the members, one of said members being intermittently cammed from and toward the other axially of the shaft when the members rotate relative to one another, a hammer revolvably mounted on said axially cammed clutch member and moved axially thereby, said hammer and wrench turning means having alignable impact shoulders, and a torsion spring interconnecting the shaft and hammer for actuating the hammer, said impact shoulders being aligned but separated while said clutch members are in torque driving relation and being intermittently moved into and out of alignment during said intermittent moving of said axially shiftable clutch member.

8. In a tool of the character described, the combination of a casing, a shaft within the casing, a driving and a driven clutch member having clutch teeth of the cam-separating type, one of the clutch members being in the form of an anvil member supported by the casing and the other clutch member being loosely splined to the shaft for axial movement along the shaft, a compression spring on the shaft yieldingly maintaining the members engaged against the camming action of the clutch teeth, a hammer revolvably mounted upon the splined clutch member and having an impact shoulder adapted to strike an impact shoulder on the anvil clutch member and a torsion spring connecting the shaft and hammer with the said impact shoulders angularly spaced out of contact, the declutching camming action being such that relative rotation between one of the clutch members and the hammer to accumulate energy in the torsion member by a rebound build up in successive cycles is effected subsequent to the separating actions of the teeth of the clutch members.

9. In a nut runner, a casing, a wrench turning means rotatably supported by the casing, a drive shaft rotatably supported within the casing, said wrench turning means having anvil lugs and clutch teeth formed thereon, a clutch member loosely spline connected to the shaft and having clutch teeth to engage the clutch teeth on the wrench turning means and the respective clutch

teeth having cam faces for opening the clutch mechanism, a cylindrical hammer element revolvably fixed on the clutch member and having lugs on one end circumferentially disposed to impact against the anvil lugs on the wrench turning means, a compression spring on the shaft for normally maintaining the clutch member with its clutch teeth in engagement with the wrench teeth thereby to rotate the wrench turning means, a torsion spring surrounding the compression spring and having one end connected to the shaft and the other end connected to the hammer with the respective anvil and hammer lugs normally out of engagement thereby to form with the hammer a torsion pendulum on the shaft and the angular spacing of the respective lugs and clutch teeth being such as to constitute an escapement mechanism to permit rebound pendulum action of the hammer relative to the camming action of clutch teeth.

10. In a nut runner, a casing, a wrench turning means rotatably supported at an open end of the casing to extend therethrough, a drive shaft rotatably supported at one end by the wrench turning means and at its other end by a flexibly mounted power coupling means disposed at the opposite end of the casing, said wrench turning means having anvil lugs and clutch teeth formed thereon, a clutch member loosely spline connected to the shaft and having clutch teeth to engage the clutch teeth on the wrench turning means and the respective clutch teeth having cam faces for opening said clutch mechanism, a cylindrical hammer element revolvably fixed on the clutch member and having lugs on one end circumferentially disposed to impact against the anvil lugs on the wrench turning means, a compression spring on the shaft for normally maintaining the clutch member with its clutch teeth in engagement with the wrench teeth thereby to rotate the wrench turning means, a torsion spring surrounding the compression spring and having one end connected to the shaft in the region of the power coupling and the other end connected to the hammer with the respective anvil and hammer lugs normally out of engagement thereby to form a torsion pendulum and the angular spacing of the respective lugs and clutch teeth being such as to constitute an escapement mechanism to permit rebound pendulum action of the hammer relative to the camming action of the clutch teeth.

11. In a tool of the character described, the combination of a casing, a shaft within the casing, a driving and a driven clutch member having camming teeth, one of the clutch members being in the form of an anvil member supported by the casing and the other clutch member being loosely splined to the shaft for axial movement along the shaft while being driven by the shaft, a spring on the shaft yieldingly maintaining the clutch members engaged against the camming action of the clutch teeth, a hammer revolvably mounted upon the splined clutch member and having an impact shoulder adapted to strike an impact shoulder on the anvil clutch member and a normally unstressed pendulum torsion spring connecting the shaft and hammer with the impact lugs of the anvil and hammer angularly and out of impact contact, the declutching camming action being such that relative rotation between one of the clutch members and the hammer to accumulate energy in the torsion member by a rebound build up is effected subsequent to the separating actions of the teeth of the clutch members.

12. In a nut runner of the reversible type, the combination of a casing, a drive shaft within the casing, an anvil member rotatably supported by the casing and having impact lugs and clutch teeth annularly arranged within the lug locations, a driving clutch member loosely splined on the shaft to be axially shiftable of the shaft while being driven by the shaft and having clutch teeth to drivingly engage the clutch teeth on the anvil member and the shapes of the anvil member and the driving clutch teeth being in the form of cams for effecting separation of the respective clutch teeth when the driving clutch member is rotated in either direction, a compression spring on the shaft for maintaining the respective clutch teeth in engagement to effect rotation of the anvil member, an oscillatable hammer revolvably mounted upon and axially shiftable by the driving clutch member, said hammer having impact lugs disposed to impact upon the anvil lugs, an unstressed pendulum torsion spring connected to the drive shaft and to the hammer in such manner that the impact lugs of the hammer and anvil member are out of contact during unarrested rotation of the anvil member and the spacing of the camming surfaces of the clutch teeth relative to the disposition of the hammer lugs is such that successive cammed escapements of the driving clutch member upon arresting of rotation of the anvil member causes the rotative impacts of the hammer lugs upon the anvil lugs to build up a successively increasing reverse motion of the hammer and a successively increasing magnitude in the impact force delivered through the hammer lugs to the anvil lugs by the pendulum spring.

13. In a nut runner of the portable type, a power driven shaft, wrench turning means, a clutch mechanism connecting the shaft and wrench turning means, a spring on the shaft yieldably maintaining the clutch mechanism closed during a preliminary nut running operation, a cam means disposed between the wrench turning means and an element of the clutch mechanism for overcoming the action of the spring and for opening the clutch mechanism, a torsion pendulum oscillatable relative to the clutch mechanism and having its center of oscillation on the shaft and arranged to deliver a series of tangential blows to the wrench turning means consequent to an intermittent yielding of the clutch mechanism, and stop means rotatable with the shaft and acting on the pendulum to limit the magnitude of angular displacement of the pendulum relative to the shaft.

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