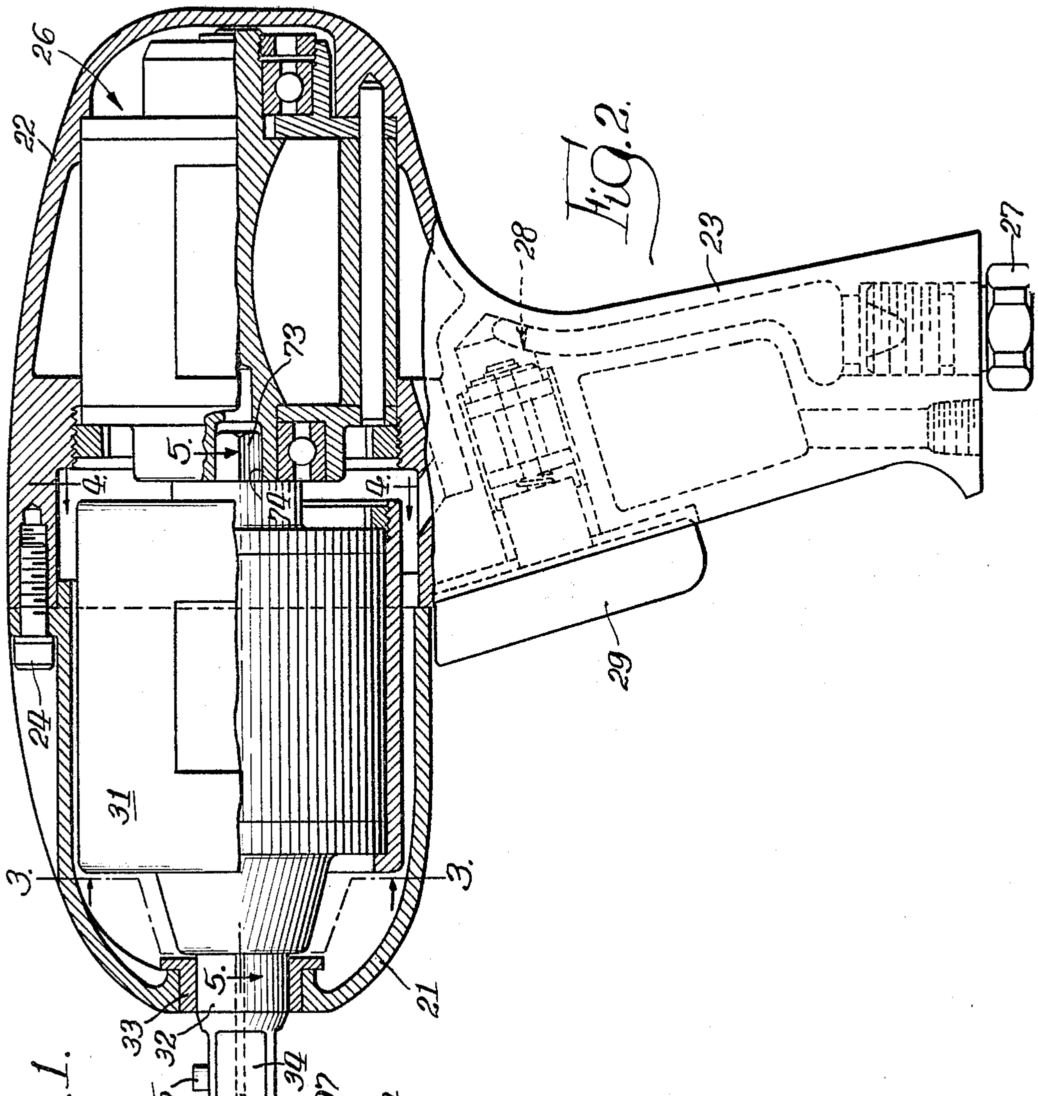
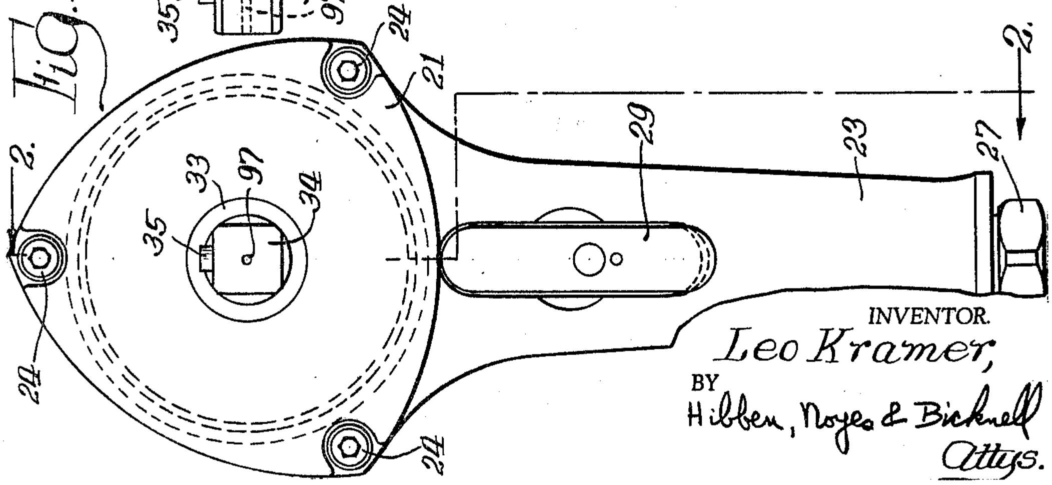
IMPULSE TOOL

Filed Nov. 22, 1963 5 Sheets-Sheet 1





INVENTOR.
Leo Kramer,
BY
Hibben, Noyes & Bicknell
Attys.

Fig. 3.

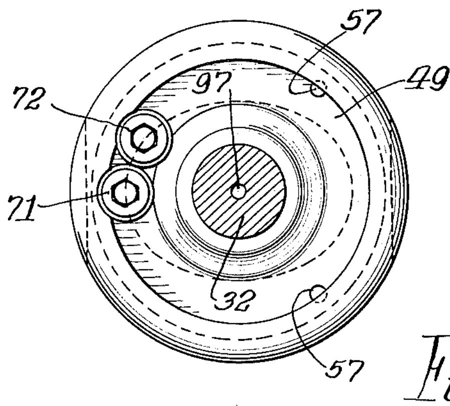


Fig. 4.

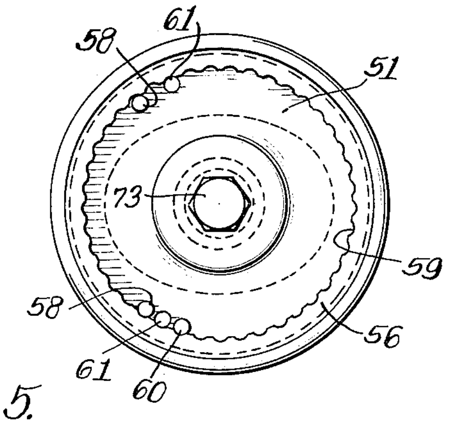
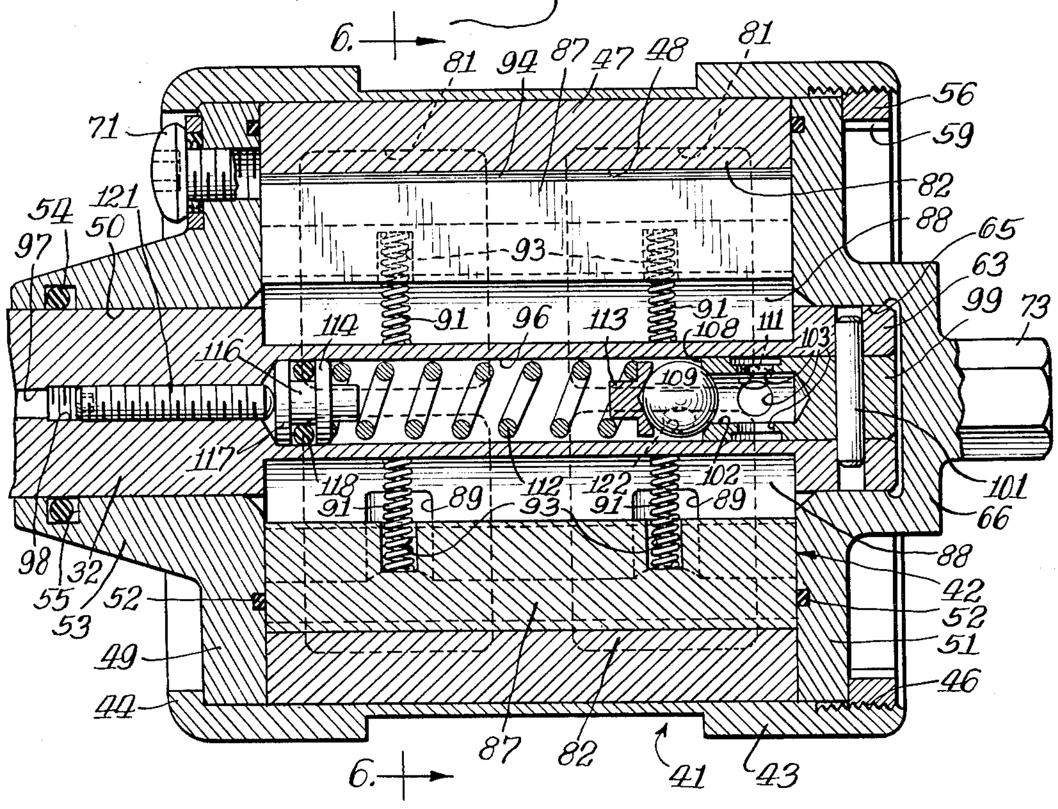


Fig. 5.



INVENTOR.
Leo Kramer,
BY
Hibben, Noyes & Bicknell
Attys.

Aug. 2, 1966 L. KRAMER 3,263,449

IMPULSE TOOL

Filed Nov. 22, 1963 5 Sheets-Sheet 3

Fig. 6.

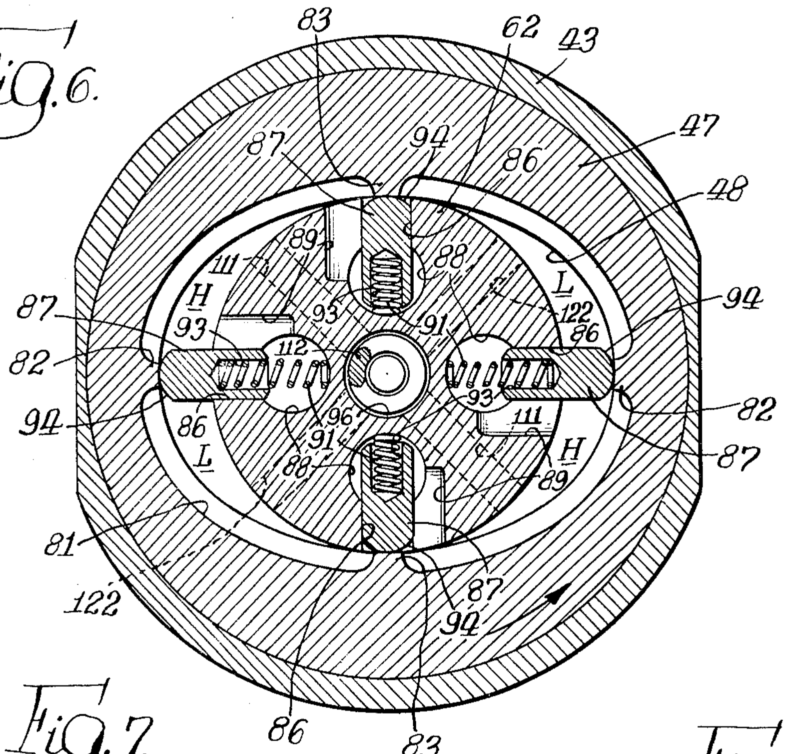


Fig. 7.

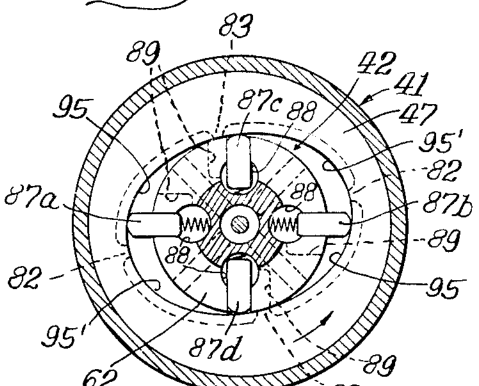


Fig. 9.

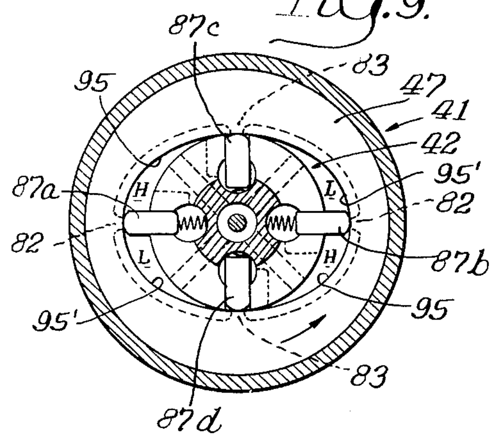
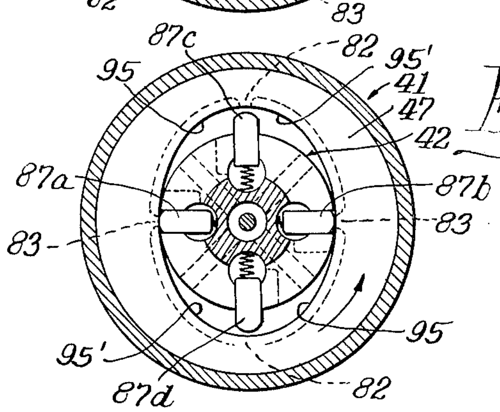


Fig. 8.



INVENTOR.

Leo Kramer,

BY

Hibben, Noyes & Bicknell

Attys

IMPULSE TOOL

Filed Nov. 22, 1963 5 Sheets-Sheet 4

Fig. 10.

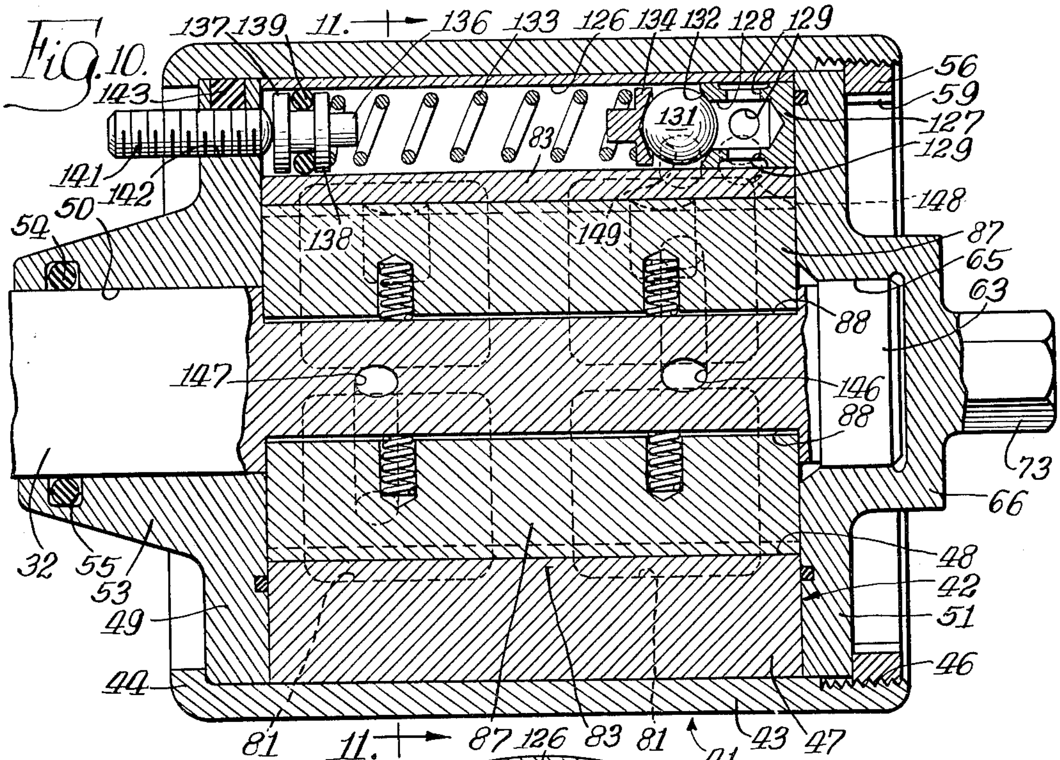
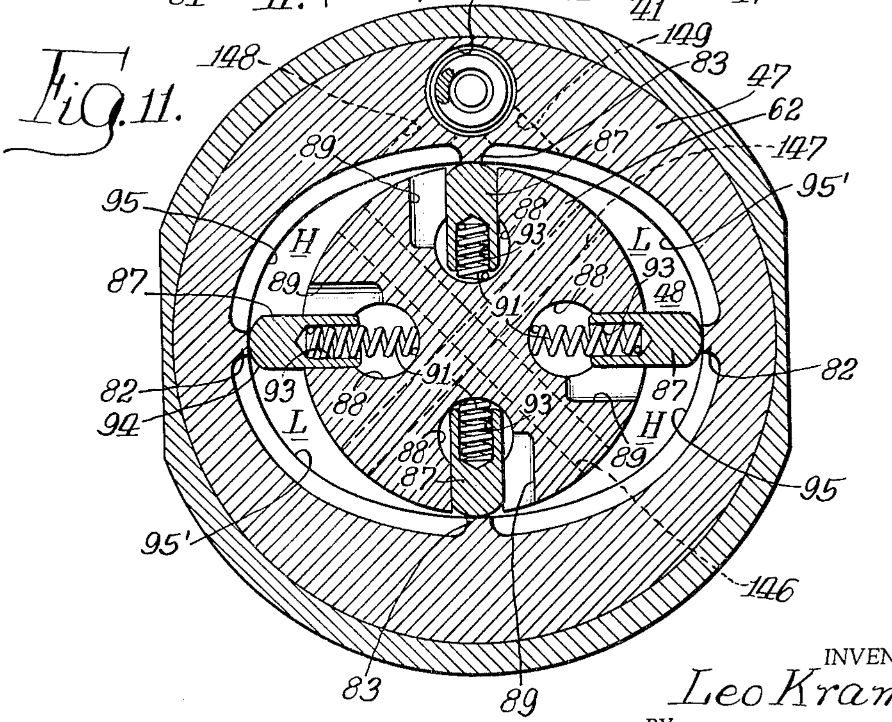


Fig. 11.



INVENTOR.
Leo Kramer,
BY
Hibben, Noyes & Bicknell
Attys.

IMPULSE TOOL

Filed Nov. 22, 1963 5 Sheets-Sheet 5

Fig. 12.

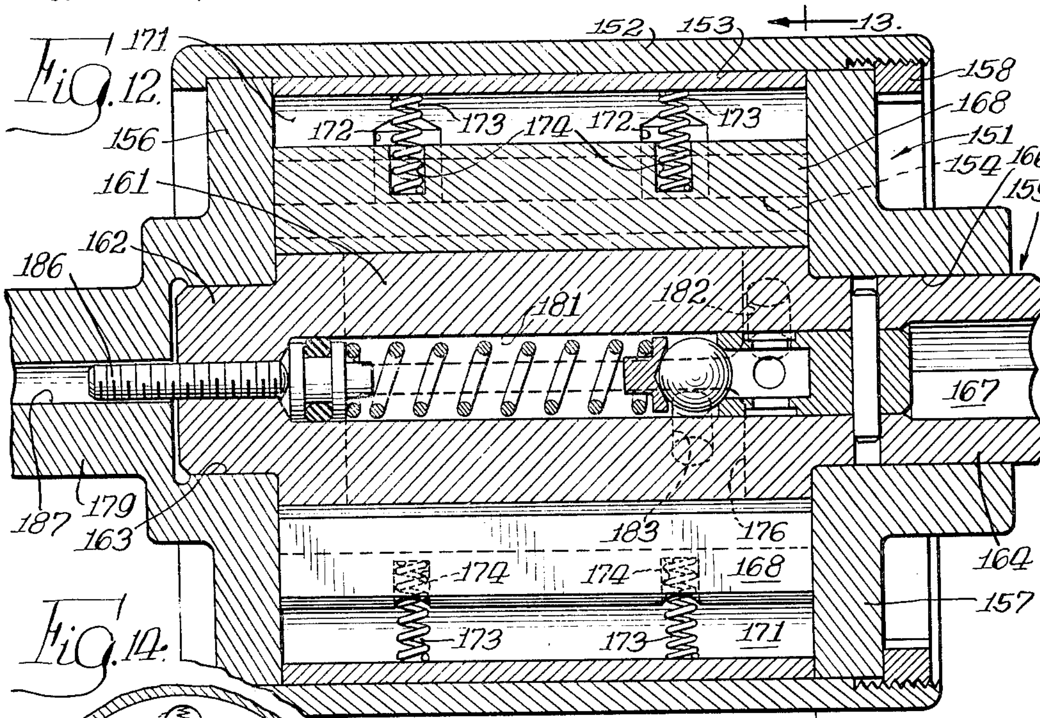


Fig. 14.

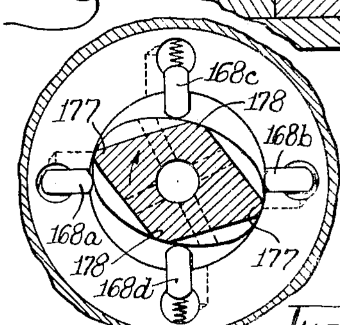


Fig. 13.

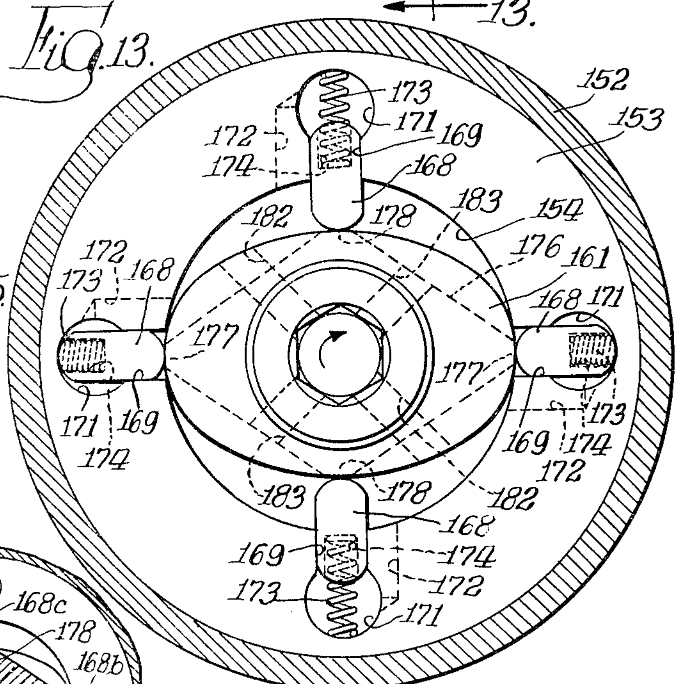


Fig. 15.

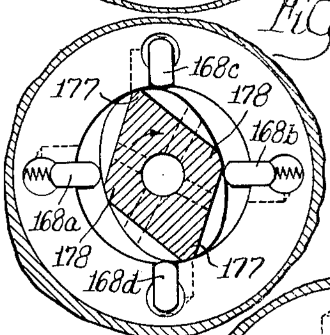
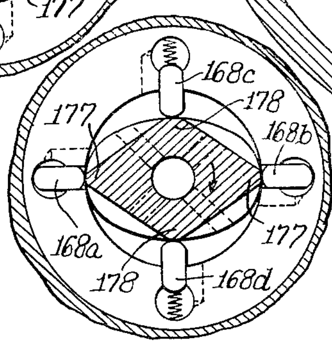


Fig. 16.



INVENTOR.
Leo Kramer,
BY
Hibben, Noyes & Bicknell
Attys

United States Patent Office

3,263,449
Patented August 2, 1966

3,263,449

IMPULSE TOOL

Leo Kramer, Chester, N.J., assignor to Thor Power Tool Company, Aurora, Ill., a corporation of Delaware

Filed Nov. 22, 1963, Ser. No. 325,629
11 Claims. (Cl. 64—26)

This invention relates to improvements in rotary, power driven, torque applying tools, such as a wrench or the like for driving nuts, bolts, screws or other fasteners.

In a conventional impact wrench, the kinetic energy of a massive rotating hammer member is transferred to a spindle member by repeated collision or striking between cooperating impact jaws on the hammer and spindle. Such tools have well-known disadvantages and limitations, particularly the relatively limited life of the impact jaws and the difficulty in the control of output torque.

In U.S. Patent No. 3,116,617 by Donald K. Skoog, a different type of tool known as an impulse tool is described and claimed in which the torque generating mechanism includes a pair of non-impacting or non-striking, relatively rotatable, eccentrically spaced members, one of which is driven by a motor and the other of which is adapted to mount a tool such as a nut or bolt engaging socket or the like. The space between the two members is filled with oil or other pressure transmitting liquid, and the internal structure is such that as one member rotates relative to the other, the pressure of the oil rapidly increases and decreases in a cyclic manner so as to create a series of impulses. These impulses are transmitted from the driving member to the driven member through the medium of the oil but without any direct physical impact or collision between jaws or other metal parts.

In my copending application Serial No. 250,160, filed January 8, 1963, now U.S. Patent No. 3,214,940, certain improvements in an impulse tool of the foregoing type are described and claimed, said improvements relating primarily to the internal mechanism responsible for the generation of the impulses. However, in each instance the tool mechanism involves an eccentric rotating member which may tend to impart undesirable effects under certain conditions. The present invention relates to additional improvements where by I am able to eliminate the use of an eccentric rotating mass so as to provide a tool having a symmetrical structure and a balanced operation.

Accordingly, a primary object of the invention is to provide a novel and improved impulse tool of the general character described above.

A further object of the invention is to provide, in an impulse tool of the general character described, novel and improved means for obtaining impulses which does not involve the use of an eccentric rotating mass.

A more specific object of the invention is to provide, in an impulse tool of the aforementioned general character, an improved impulse generating means having a symmetrical structure and a balanced operation.

Other objects and advantages of the invention will become evident from the subsequent detailed description taken in conjunction with the accompanying drawings, wherein:

FIG. 1 is a front end elevational view of a tool comprising one specific embodiment of the invention;

FIG. 2 is a side elevational view, partially in section, taken along the line 2—2 of FIG. 1;

FIG. 3 is a reduced scale end elevational view of a portion of the internal mechanism of the tool as seen along the line 3—3 of FIG. 2;

FIG. 4 is a view similar to FIG. 3 but taken at the opposite end of the internal mechanism along the line 4—4 of FIG. 2;

FIG. 5 is an enlarged scale horizontal sectional view of the internal mechanism taken along the line 5—5 of FIG. 2;

FIG. 6 is a transverse sectional view taken along the line 6—6 of FIG. 5;

FIGS. 7–9 are reduced scale sectional views, generally similar to FIG. 6, showing successive operating positions of the internal mechanism of the tool;

FIG. 10 is a sectional view generally similar to FIG. 5 but showing a modification of the invention;

FIG. 11 is a sectional view taken along the line 11—11 of FIG. 10;

FIG. 12 is a sectional view generally similar to FIG. 5 but showing a second embodiment of the invention;

FIG. 13 is a sectional view taken along the line 13—13 of FIG. 12; and

FIGS. 14–16 are reduced scale sectional views showing successive operating positions of the second embodiment.

For the sake of convenience, the present invention is illustrated in connection with a rotary power driven wrench or nut setter, but it will be understood that the principles of the invention which pertain to the torque generating or impulse unit may be utilized to advantage in a wide variety of power tools.

Referring first to FIGS. 1 and 2, the tool has a front casing section **21** and a rear casing section **22** with a depending handle portion **23**, the front and rear casing sections being detachably connected, as by a plurality of cap screws **24**. The rear casing section **22** contains a rotary air-actuated motor of a conventional type designated generally at **26**. An air hose (not shown) may be connected to an air inlet connection **27** at the lower end of the handle portion **23**, and the air supply to the motor **26** is controlled by means of a throttle valve assembly **28** movably mounted in the handle portion **23** and operable by a depressible trigger **29** for regulating the operation of the tool. An impulse unit **31** is contained within the front casing section **21**, as hereinafter described in greater detail. The impulse unit **31** has a rotatable spindle with an elongated forward shaft portion **32** journaled in a bearing or bushing **33** which is mounted in an opening in the forward end of the casing section **21**. The outer terminus of the spindle portion **32** is provided with a square end **34** having a depressible detent **35** for detachably mounting a nut-or bolt-engaging socket or other tool member.

Referring particularly to FIGS. 3–6, the impulse unit **31** comprises a housing assembly, indicated generally at **41**, which is adapted to be driven by the air motor **26**, and an internal spindle **42** having the forwardly projecting shaft portion **32** heretofore mentioned.

The housing assembly **41** includes an outer casing **43** having a flanged forward end **44** and an internally threaded rear end portion **46**. A cylinder bushing **47** (FIGS. 5 and 6) having a generally elliptical or oval shaped internal cylinder cavity **48** is tightly fitted within the outer casing **43** and is disposed between a pair of forward and rear cap members **49** and **51** (FIG. 5), respectively, with O-ring seals **52** mounted in suitable grooves in the inner faces of the respective cap members opposite the axial ends of the bushing **47**. The forward cap member **49** abuts the flange **44** and has a forwardly projecting tapered bearing portion **53** with a bore **50** which receives the spindle shaft portion **32**. A rotary O-ring seal **54** is mounted in an annular groove **55** in the bearing portion **53**. An annular locking ring or nut **56** threadedly engages the casing opening **46** and abuts the rear cap member **51** for retaining the impulse unit **31** in unitary assembled relation. The front and rear cap members **49** and **51** are retained in predetermined orientation with respect to the elliptical cavity **48** of the bushing **47** by means of a plurality of roll pins

57 and 58 (FIGS. 3 and 4) which are fitted in suitable openings in the bushing and the respective cap members. As seen in FIG. 4, the inner periphery of the locking ring or nut **56** is provided with a plurality of recesses or serrations **59** so that after the nut **56** has been tightened to the desired extent, a set screw **60** can be inserted through one of the recesses **59** and threaded into a selected one of a plurality of threaded sockets **61** in the rear cap member **51** for locking the nut **56** in its tightened position.

The spindle **42** has an enlarged central portion **62** with a circular cross-sectional configuration disposed coaxially within the elliptical cylinder cavity **48** of the bushing **47**, as best seen in FIG. 6. A short integral stub shaft portion **63** (FIG. 5) projects rearwardly from the spindle portion **62** and is journaled in a socket **65** provided in a projecting boss portion **66** on the rear cap member **51**. The opposite end of the spindle **42** is journaled in the bearing portion **53** of the front cap member **49** by means of the elongated integral shaft portion **32** (FIG. 5) heretofore described. The spindle shaft portions **32** and **63** are coaxial with the axis of rotation of the housing assembly **41**.

The elliptical cylinder cavity **48** of the bushing **47** is filled with a suitable oil or other pressure transmitting liquid, and a detachable fill plug **71** (FIGS. 3 and 5) is provided in the front cap member **49** for filling the cavity **48** with oil. A removable vent plug **72** (FIG. 3) is also provided in the cap member **49** adjacent the plug **71** for venting air from the cavity **48** during the introduction of oil. The rear cap member **51** has a hexagonal drive connection **73** (FIG. 5) projecting from the boss portion **66** and engaging a driving socket connection **74** (FIG. 2) on the air motor **26**. Thus, when the air motor **26** is in operation, the housing assembly **41** (comprising **43**, **47**, **49**, **51**, and **56**) of the impulse unit **31** is rotated and, as described in detail below, fluid pressure impulses are generated in and are transmitted through the oil medium to the spindle **42**.

The elliptical cavity **48** of the bushing **47** is provided with a pair of circumferentially extending recesses or undercuts, as designated at **81** (FIGS. 5 and 6), intermediate the opposite ends of the bushing **47**. These circumferential recesses **81** are interrupted by a pair of axially extending lands or seal points **82** diametrically disposed at the longer or major axis of the elliptical cavity **48** and also by a similar pair of lands or seal points **83** disposed at the shorter or minor axis of the cavity. Thus, the distance between the lands **82** is appreciably greater than the distance between the lands **83**. Moreover, as shown in FIG. 6, the diameter of the circular spindle portion **62** is such that there is only a slight clearance between the spindle portion **62** and the lands **83** at the minor axis of the cavity **48**.

The circular spindle portion **62** is provided with four radially extending slots **86** disposed at equal circumferential spacings around the spindle, and each slot **86** has an elongated blade **87** slidably mounted in close-fitting relation therein with the blades **87** extending the full axial length of the spindle portion **62** and spanning the recesses **81**, as best seen in FIG. 5. The base or innermost end of each slot **86** is enlarged in the form of a circular axially extending bore **88** (FIG. 6) so that the slot **86** and its connecting bore **88** have a keyhole shape in cross-section. Each bore **88** communicates with the outer periphery of the spindle portion **62** through a pair of fluid pressure loading passages **89** machined in the spindle at one side of the corresponding slot **86**. As shown in FIG. 6, the passages **89** are arranged in opposed relation at adjacent sides of a pair of blades **87** so that the corresponding bores **88** of each pair of blades **87** are placed in fluid communication with the same region of the oil filled cavity **48**, for the purpose hereinafter described in greater detail. A pair of compression springs **91** bear against the inner wall portion of each bore **88** and extend into a pair of bores **93** extending from the inner edge of each blade **87** for normally urging the blades **87** outwardly into contact with the wall of the bushing cavity **48**. As best seen in FIG. 6, the outer edge of each blade portion **87** is rounded or curved, as at **94**, to provide intermittent sealing engagement with the lands **82** and **83** during relative rotation of the housing assembly and the spindle.

Referring now to FIGS. 7–9, a typical operating cycle of the tool will be described. It will be assumed that the housing assembly **41**, including the bushing **47**, is being driven in a counter-clockwise direction by the air motor **26**, as indicated by the arrow. The square end **34** on the forward spindle shaft **32** is assumed to be in engagement through a suitable socket (not shown) with a fastener element (not shown) such as a nut or bolt which is to be tightened. At the outset of the tightening operation, very little torque resistance is encountered by the spindle and, consequently, the frictional engagement of the spring-pressed blades **87** with the inner wall of the bushing cavity **48** is sufficient to cause the spindle **42** to rotate in a counter-clockwise direction in unison with the rotating housing assembly **41**.

As tightening of the fastener element proceeds and substantial torque resistance is encountered by the rotating spindle **42**, the housing assembly **41** begins to rotate relative to and ahead of the spindle **42**. In the position of the mechanism shown in FIG. 7 the housing assembly **41** has rotated in a counter-clockwise direction relative to the spindle **42** so that the lands or seal points **82** and **83** have just left the outer ends **94** of the blades **87**, which are separately identified as **87*a***, **87*b***, **87*c***, and **87*d***. The zones designated at **95** in the elliptical cavity **48** between blades **87*a*** and **87*c*** and between blades **87*b*** and **87*d*** are gradually decreasing in volume, whereas the zones designated at **95′** in the cavity **48** between blades **87*a*** and **87*d*** and between blades **87*b*** and **87*c*** are gradually increasing in volume. However, since the blades **87** are out of engagement with the seal points **82** and **83**, the oil or other pressure transmitting fluid in the cavity **48** is free to flow between the zones **95** and **95′** through the recesses or undercuts **81** so that the pressure throughout the cavity **48** is substantially uniform. Thus, the motor **26** continues to accelerate the housing **41** about the spindle **42**.

In FIG. 8 the housing **41** has rotated not quite 90° from its FIG. 7 position so that the lands **82** and **83** are aligned with the blades **87**. However, in this position the more closely spaced lands **83** have caused the oppositely disposed pair of blades **87*a*** and **87*b*** to be cammed or depressed substantially fully into their slots **86** and bores **88** and thereby causing the corresponding springs **91** to be substantially fully collapsed. At this point, zones **95** of the cavity **48** are now increasing in volume and zones **95′** are decreasing in volume. However, since the inner edges of the other pair of blades **87*c*** and **87*d*** are in fluid communication, through passages **89** and bores **88**, with the relatively lower pressure expanding zones **95**, it will be evident that the relatively greater fluid pressure in the contracting zones **95′** will tend to depress the blades **87*c*** and **87*d*** inwardly in their slots against the spring action, as shown in slightly exaggerated form in FIG. 8. Thus, fluid in the cavity **48** is free to flow between the zones **95′** and **95** at opposite sides of the blades **87*c*** and **87*d*** so that the fluid pressure throughout the cavity **48** is again essentially uniform and the housing **41** continues to accelerate ahead of the spindle **42**. As the housnig **41** rotates beyond its FIG. 8 position, the lands **82** and **83** will once again pass beyond the blades **87** so that fluid pressure is again equalized throughout the cavity **48** by reason of the recesses **81**.

In FIG. 9 (which also corresponds to FIG. 6) the housing **41** has rotated through approximately 180° or one-half revolution from its FIG. 7 position, and just before reaching the FIG. 9 position the zones **95** are again decreasing in volume while the zone 95' are increasing in volume. When the FIG. 9 position is reached wherein the lands 82 and 83 are again aligned with the blades 87, it will be seen that the inner edge of each blade 87 is subjected, through the passages 89 and the bores 88, to the relatively high fluid pressure of the contracting zones 95. Consequently, all four blades 87 are simultaneously subjected to pressure loading by high pressure fluid and are, therefore, urged outwardly by the combined action of fluid pressure and the springs 91 so that the outer blade edges 94 are pressed into dynamic sealing relation with respect to the moving lands 82 and 83 and fluid is no longer free to pass between the zones 95 and 95' through the recesses 81. As a result of this sudden sealing-off of the fluid, a momentary substantial increase in fluid pressure is obtained in the zones 95 of the cavity 48, as indicated by the letter H in FIG. 9 to designate the high pressure zones. In the other sealed-off zones 95' of the cavity 48, the fluid pressure is substantially lower, as designated by the letter L to indicate the low pressure zones. In a typical example, the pressure in the zones 95 may be as high as 3,000 lbs. per square inch.

By the time the housing 41 reaches the dynamic sealing position of FIG. 9 it has acquired considerable momentum since it has been accelerating through substantially 180°. Consequently, the pressure in zones 95 will continue to rise until the housing has lost its momentum or until the lands 82 and 83 have moved beyond the blades 87. This momentary pressure increases in the zones 95 imposes an eccentric force on the blade 87 and thereby causes the spindle 42 to rotate in a counterclockwise direction to tighten the fastener element.

As the housing assembly 41 rotates beyond its FIG. 9 position ahead of the spindle 42, the fluid pressure in the cavity 48 is again substantially equalized by reason of the free fluid communication provided by the recesses or undercuts 81 once the moving parts have passed beyond the dynamic seal position. This condition prevails until the housing has rotated 180° to another dynamic sealing position which is just prior to the FIG. 7 position of the mechanism. Continued rotation of the housing assembly 41 causes the foregoing cycle to be repeated with the result that successive pressure increases are generated in the high pressure zones 95 of the cavity 48 each half revolution of the housing, i.e. each time the lands 82 are aligned with the blades 87*a* and 87*b*, and these pressure increases result in impulses which are imparted to the spindle 42 through the medium of the fluid and the blades 87 until the desired tightening operation has been completed.

Because of the rapid rotation of the housing assembly 41, it will be understood that the dynamic sealing relationship described above is repeated over and over again. However, to insure proper operation of the tool, it may be desirable to provide a small static leakage between the relatively rotatable housing assembly 41 and spindle 42. For example, in assembling the impulse unit 31, the casing nut 56 is tightened sufficiently to retain the inner faces of the end cap members 49 and 51 in dynamic sealing relationship with the opposite axial ends of the spindle portion 62, but the operating clearance therebetween is sufficient to allow a predetermined static leakage between the high and low pressure zones of the cavity 48. This relationship is desirable in order to insure that excessive driving torque from the air motor 26 will not be required in order to rotate the housing assembly 41 beyond one of its dynamic sealing positions such as illustrated in FIGS. 6 and 9.

As a further aid in this respect, the inner face of one, and preferably both, of the housing cap members, such as the cap 49, may be provided with appropriately located grooves of predetermined dimensions which are capable of transmitting fluid pressure between the high and low pressure sides of the blades 87*a* and 87*b* when the operating parts are in dynamic sealed relation. The use of a metering groove for this purpose is described in more detail in my aforementioned copending application Serial No. 250,160.

To reverse the operation of the tool, the direction of rotation of the air motor 26 is reversed, thereby rotating the housing assembly 41 in the opposite direction and consequently interchanging the high and low pressure zones 95 and 95' illustrated in FIGS. 7–9, but the operation is otherwise the same as hereinabove described. Thus, the mechanism is readily reversible and proper pressure loading of the blades 87 is not dependent upon the operation of a check valve or the like in an internal fluid pressure passageway.

Although in the illustrated embodiment the housing assembly 41 is driven by the air motor 26 and the spindle 42 drives the tool member, it will be appreciated that these functions may be interchanged by minor and obvious modifications of the structure. Thus, the rear spindle shaft portion 63 can be connected to the air motor 26 and the forward cap member 49 of the housing assembly 41 can be modified to mount the tool member.

One of the principal advantages of an impulse tool, in addition to the elimination of impacts or collisions between metal parts, is the facility with which such an impulse mechanism lends itself to the control of output torque of the tool. Certain minor provisions for controlling the pressure differential between the high and low pressure sides of the impulse unit have been described above for the purpose of minimizing the torque requirements of the air motor 26. However, in order to provide a positive upper limit on the torque output of the impulse unit, an adjustable pressure relief valve is preferably provided which will prevent the fluid pressure in the high pressure zones from exceeding a predetermined maximum.

One such pressure relief arrangement is shown in FIGS. 5 and 6 and comprises a centrally located axial bore 96 in the spindle portion 62. At the forward end of the spindle portion 62 the bore 96 communicates with an elongated smaller diameter bore 97 having a threaded portion 98. At the opposite or rearward end of the spindle portion 62 the bore 96 is closed by means of a plug or cap 99 which is keyed by a pin 101. The inner end portion of the plug 99 has an axial recess 102 with a plurality of lateral or radial ports 103. The inner axial end of the plug 99 provides an annular shoulder 108 constituting a valve seat for a ball element 109. High pressure fluid from the zones 95 or H communicate with the recess 102 through a pair of radial angular drilled passages 111 in the spindle portion 62. The inner ends of the passages 111 are aligned with the enlarged ports 103 in the plug 99. The ball element 109 is normally retained in seated relation on the shoulder 108 by means of a spring 112 having its inner end operatively engaging the ball 109 through a cap member 113. The opposite end of the spring 112 seats against a flange 114 of a piston member 116 disposed adjacent the forward end of the bore 96. The piston 116 has another flange 117 spaced from the flange 114 with an O-ring seal 118 mounted therebetween. An adjusting screw 121 is mounted in the threaded portion 98 of the small bore 97 and engages the outer end of the piston 116 for regulating the position of the piston 116 within the bore 96 and the degree of compression of the spring 112. The bore 97 extends forwardly the length of the spindle and is accessible at the front end of the square 34 (FIGS. 1 and 2) so that a suitable elongated tool, such as an Allen wrench can be inserted in the bore 97 for adjustment of the screw 121.

When high pressure fluid in the recess 102 exceeds a predetermined pressure, which is dependent upon the pressure exerted by the spring 112 as determined by the adjusting screw 121, the ball 109 is disengaged from its seat 108 and the high pressure fluid bleeds into the bore 96 and thence through another pair of drilled radial angular passages 122 in the spindle portion 62 into the low pressure zones L or **95′** of the cavity **48**. Thus, when the spindle **42** encounters a predetermined torque resistance during a tightening operation, the pressure in the high pressure zones H or **95** of the cavity **48** will tend to build up because of the limited rotatability of the spindle at which point the relief ball valve **109** will open so as to prevent the imposition of excessive torque on the fastening element.

To recapitulate, it will be understood from the foregoing description that the present invention affords an improved impulse generating mechanism in which the relatively rotating parts are mounted coaxially and are in symmetrical or non-eccentric arrangement relative to the common axis of rotation. Thus, broadly speaking, the mechanism includes (1) a pair of relatively rotatable, coaxial, non-eccentric members having a fluid filled cavity therebetween, (2) a plurality of symmetrically disposed spring-pressed blades or vanes slidably mounted on one of the members and extending into the fluid-filled cavity and (3) a plurality of symmetrically disposed seal points on the other member arranged for periodic engagement with the blades at non-uniform distances from the common axis of rotation. Upon relative rotation between the members, the fluid-filled cavity is intermittently divided into zones which are alternately increasing in volume (low pressure) and decreasing in volume (high pressure). A fluid pressure loading arrangement is also provided for the blades such that, at certain of the periodic conditions of alignment between the blades and the seal points, each blade is subjected to the high fluid pressure of a contracting zone of the fluid-filled cavity whereby to provide a momentary dynamic sealing relation between the relatively rotating members. Consequently, the momentary large pressure increases necessary for impulse generation are obtained. As a result of the symmetrical non-eccentric relationship of the rotating parts, no eccentric forces are developed and the impulse mechanism of the tool operates under a balanced load. Such uniform balanced operation avoids unbalanced forces on the bearings and insures a balanced pressure condition at the housing end plates or caps of the impulse unit. In many cases, the tool operating under balanced load will also be more comfortable and less tiring for the operator.

FIGS. 10 and 11 illustrate a modification of the impulse mechanism **31** previously described in that the pressure relief valve for torque control is mounted in the housing means of the impulse unit rather than in the spindle. Otherwise identical parts in FIGS. 10 and 11 are identified by the same reference numerals used in the comparable FIGS. 5 and 6. Thus, in FIGS. 10 and 11 one of the thick wall portions of the bushing **47** is provided with an axial bore **126** having a plug **127** at its rearward end adjacent the end cap **51**. The plug **127** has an axial recess **128** and a plurality of radial apertures or ports **129**. A ball element **131** is held in normally seated engagement against an end seat **132** on the plug **127** by means of a spring **133** and a cap **134**. The forward end of the spring **133** is engaged by an adjustably positioned piston **136** having spaced flanges **137** and **138** with an O-ring seal **139** therebetween. An adjusting screw **141** is mounted in a threaded opening **142** in the end cap **49** and engages the piston **136**. A tension plug **143**, of nylon or the like, is provided in the cap member **49** and engages the threads of the screw **141** for retaining the latter in adjusted position. Of course, in this modification the spindle shaft portions **32** and **63** are solid and integral with the main spindle portion **62**.

In this case the spindle portion **62** has a pair of diametric bores **146** and **147** arranged at right angles so as to interconnect the two high pressure zones **95** or H and the two low pressure zones **95′** or L, respectively, when the mechanism is in dynamic sealing relation as shown in FIG. 11. The bushing **47** also has a pair of angular passages **148** and **149** extending from the bore **126** at opposite sides of the valve seat **132** to opposite sides of one of the lands **83** and thereby connecting the bore **126** with the high and low pressure zones **95** and **95′**, respectively, of the elliptical cavity **48** when the unit is in dynamic sealing position. Thus, at impulse position when fluid pressure in the zones **95** or H exceeds the maximum determined by the spring adjustment, high pressure fluid can pass from the interconnected zones **95** or H through passage **148** and ports **129** into the recess **128** and, upon displacement of the ball **131** from its seat **132**, the high pressure fluid bleeds through the passage **149** and thence into the interconnected low pressure zones **95′** or L. If desired, an empty axial bore (not shown) may be provided in the opposite thick wall portion of the bushing **47** diametrically across from the bore **126** in order to balance the weight of the unit and insure uniform balanced operation. Reference is made to my previously mentioned copending application Ser. No. 250,160 for an illustration of this feature.

FIGS. 12 to 16 show another embodiment of the invention which differs from the previously described structure in that: (1) the housing cavity of the impulse unit is circular in cross-section and the spindle body is substantially oval or elliptical in cross section instead of the reverse relationship shown in FIGS. 1 to 11, (2) the slidable blades or vanes are mounted in slots in the housing rather than in the spindle, and (3) the spindle is driven by the air motor of the tool and the housing is utilized as the output portion of the tool instead of the opposite arrangement heretofore illustrated and described.

The housing assembly, designated at **151**, includes an outer casing **152**, a rigidly fitted cylinder bushing **153** having a circular cavity **154**, a forward end cap **156**, a rear end cap **157**, and a locking ring **158** threadedly engaging the casing **152** for retaining the impulse unit in assembled relation. A spindle, designated generally at **159**, has a central body portion **161** of substantially oval or elliptical cross-sectional configuration which is disposed coaxially within the cavity **154**. A shaft portion **162** at the forward end of the spindle **159** is journalled in a recess **163** in the end cap **156**, and a shaft portion **164** at the rearward end of the spindle **159** is journalled in a bore **166** in the end cap **157**. The rear shaft portion **164** of the spindle is provided with a non-circular socket **167** for driving engagement with the air motor of the tool.

A plurality of blades or vanes **168** are slidably mounted in slots **169** in the bushing **153** which extend outwardly from the wall of the cavity **154** and terminate in enlarged axially extending bores **171** having connecting pressure loading passages **172** for communicating fluid pressure from the cavity **154** to the outer edges of the blades **168**. Each blade **168** has associated therewith a pair of compression springs **173** extending between the outer wall of the corresponding bore **171** and a pair of spaced bores **174** in the outer edge of the blade whereby the blade is normally urged inwardly into contact with the elliptical spindle portion **161**. The peripheral surface of the elliptical spindle portion **161** is provided with an encircling recess or undercut **176** which is disposed centrally between the axial ends of the spindle portion **161** and is interrupted by a pair of axially extending lands or seal points **177** disposed at the ends of the major axis of the ellipse and also by a pair of lands or seal points **178** disposed at the minor axis of the ellipse. The major axis dimension of the elliptical spindle portion **161** is such that there is only a slight clearance between the spindle portion **161** and the circular cavity **154**.

The operation of the impulse generation unit is in all respects analogous to the operation heretofore described in detail in connection with the previous embodiments. Thus, referring particularly to FIGS. 14–16, an impulse position is obtained every half revolution when the lands **177** and **178** are in dynamic sealing relation with the blades **168**. Assuming rotation of the spindle **159** by the air motor in a clockwise direction, the spring pressure of the blades **168** against the periphery of the spindle portion 161 is sufficient at the outset of operation to effect clockwise rotation of the housing 151 and also the wrench socket or other tool member which is secured to the forward end (not shown) of an axial extension 179 on the end cap 156. As torque resistance on the extension 179 increases, the spindle 159 begins to accelerate ahead of the housing 151. At the position shown in FIG. 14 the volumes of the upper left quadrant (between blades 168*a* and 168*c*) and the lower right quadrant (between blades 168*b* and 168*d*) of the cavity 154 are decreasing and the volumes of the remaining quadrants are increasing. However, fluid pressure is equalized between adjacent contracting and expanding zones by reason of the recess 176. In FIG. 15 the lands 177 and 178 are approaching alignment with the respective blades 168, and blades 168*c* and 168*d* are substantially fully depressed by the camming action of the lands 177. However, because of the orientation of the pressure loading passages 172, only low pressure fluid from the now expanding zones of the upper left and lower right quadrants of cavity 154 has access to the outer edges of the blades. Consequently, there is no pressure seal and high pressure fluid in the remaining contracting quadrants can force the blades 168 outwardly to equalize the pressure. As the spindle continues to rotate ahead of the housing, the zones in the upper left and lower right quadrants start to decrease in volume and finally the position of FIG. 16 is reached in which the lands 177 are aligned with blades 168*a* and 168*b* and the lands 178 are aligned with blades 168*c* and 168*d*. At this position the outer edge of each blade is exposed to high pressure fluid from a contracting zone and a dynamic seal is achieved between the blades 168 and the lands 177–178. The cycle is repeated in the same manner so that an impulse is imparted to the housing 151 through the blades 168 every half revolution of the spindle relative to the housing.

A pressure relief valve arrangement for torque control is provided in a bore 181 extending axially of the spindle 159 in essentially the same manner as shown in FIGS. 5 and 6 so that a detailed description is unnecessary. High pressure fluid from the high pressure zones of the cavity 154 communicates with the upstream side of the relief valve bore 181 by means of a pair of drilled passages 182 in the elliptical spindle portion 161, and such fluid is vented from the downstream side of the relief valve through another pair of spindle passages 183 into the low pressure zones of the cavity 154. The spring setting for the relief valve is adjusted by means of a screw 186 threaded into the spindle shaft 162 and accessible from the front end of the tool through an elongated axial passage 187.

Although the invention has been described with particular reference to certain specific structural embodiments thereof, it should be understood that other modifications and equivalent structures may be resorted to without departing from the scope of the invention as defined in the appended claims.

I claim:

1. In an impulse tool, an improved impulse generating means comprising a rotatable housing member having a symmetrical, non-eccentric, coaxial, internal cavity containing a pressure transmitting fluid, a spindle member rotatably disposed in coaxial non-eccentric relation in said cavity for relative rotation between said housing member and said spindle member, means on one of said members defining a plurality of fluid pressure sealing portions disposed in symmetrically spaced relation around said cavity, a plurality of movable elements carried by the other of said members in symmetrically spaced relation around said cavity, the configurations of said cavity and said spindle member being such that, during relative rotation of said members, said fluid pressure sealing portions and said movable elements are engageable at different distances from the axis of rotation of said members whereby to divide said cavity into a plurality of zones of alternately increasing and decreasing volume, and fluid passage means in said other member for simultaneously communicating the resultant increasing fluid pressure from the zones of decreasing volume to all of said movable elements at certain predetermined relative rotational positions of said members so as to urge said movable elements into momentary sealing relation with said fluid pressure sealing portions, whereby to generate intermittent substantial fluid pressure increases in sealed-off portions of fluid in said zones of decreasing volume and thereby transmitting rotational torque impulses through said fluid between said members.

2. The structure of claim 1 further characterized in that said other member is provided with a plurality of symmetrically spaced slots and said movable elements comprise blades extending axially of said cavity and radially slidable in said slots, and spring means is provided in said slots for normally urging said blades outwardly of said slots and in engagement with said one member whereby said blades are adapted to span said cavity between said members for defining said zones.

3. The structure of claim 2 further characterized in that said fluid passage means comprises a plurality of fluid passages in said other member communicating between said cavity and the base portions of said slots behind said blades, and said fluid passages being disposed relative to said members and said blades so as to register said fluid passages with said zones of decreasing volume during said certain predetermined positions of said members.

4. The structure of claim 1 further characterized in that one of said housing or spindle members is provided with adjustable pressure relief valve means and is also provided with a plurality of fluid passages communicating said valve means with said zones of increasing volume and with said zones of decreasing volume when said movable elements and said fluid pressure sealing portions are in sealing relation, whereby to prevent fluid pressure in said zones of decreasing volume from exceeding a predetermined maximum.

5. The structure of claim 1 further characterized in that said cavity is generally elliptical in cross-section, and said spindle member has a circular cross-section.

6. The structure of claim 5 further characterized in that four fluid pressure sealing portions are provided on said housing member in opposed relation at the major and minor axes of the elliptical cavity, and said movable elements comprise four blades slidably mounted in uniformly spaced relation around the periphery of said spindle member.

7. The structure of claim 6 further characterized in that said housing member has means for drivingly connecting the same to a rotary motor and said spindle member has means for connecting the same to a tool element.

8. The structure of claim 1 further characterized in that said cavity is circular in cross-section, and said spindle member has a generally elliptical cross-section.

9. The structure of claim 8 further characterized in that four fluid pressure sealing portions are provided around the periphery of said spindle member at the ends of the major and minor axes thereof, and said movable elements comprise four blades slidably mounted in said housing member in uniformly spaced relation around said cavity.

10. The structure of claim 9 further characterized in that said spindle member has means for drivingly connecting the same to a rotary motor and said housing member has means for connecting the same to a tool element.

11. An impulse generating mechanism for use in a rotary tool or the like, comprising a pair of relatively rotatable members having a common axis of rotation, said members being symmetrical relative to said axis and defining a cavity therebetween containing a pressure

transmitting fluid, and means on one of said members providing a plurality of points of sealing engagement with the other of said members during relative rotation therebetween, said points of engagement being disposed in symmetrically spaced relation around said axis and the configurations of said members and said cavity being such that, during relative rotation of said members, said points of sealing engagement are disposed at different distances from said axis, whereby to divide said cavity into a plurality of zones of alternately increasing and decreasing volume and thereby generating intermittent substantial fluid pressure increases in sealed-off portions of fluid in said zones of decreasing volume.

References Cited by the Examiner

UNITED STATES PATENTS

887,781	5/1908	Fischer	192—58
1,156,819	10/1915	Rich	192—58
1,544,721	7/1925	Bredlow et al.	192—58
2,169,747	8/1939	Treer	192—58
2,564,212	8/1951	Ramsey	192—58
3,116,617	1/1964	Skoog	64—26

MILTON KAUFMAN, *Primary Examiner.*

BROUGHTON G. DURHAM, *Examiner.*

H. C. COE, *Assistant Examiner.*