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PRESSURIZED FLUID TORQUE DRIVER  
CONTROL AND METHOD Patent  
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## AWARDS ABSTRACT

MSC-22385-1

This invention is a fluid driven, torque limiting rotary drive mechanism operable to transmit a precisely controllable degree of torque to a driven member such as a nut, screw, or the like. It relates to torque control mechanisms such as nut runners, automatic screw drivers, and other rotary drive mechanisms. Commercial applications include uses as a torque drive mechanism wherein it is necessary to apply torque simultaneously to multiple driven members such as nuts, screws, or the like. Representative industrial applications include vehicle assembly operations in which multiple bolts or screws are to be driven, as in the installation of head bolts in an engine block, the driving of multiple bolts or screws during the assembly of a consumer product, and in auto service functions, such as those in which wheel lugs are to be threadingly driven, spark plugs installed, or motor heads torqued down.

In one embodiment, the apparatus includes an input shaft (shaft 58, Fig. 2) connectable to a pneumatic or electrical drive motor or the like for receiving an input torque which may be greater than the desired output torque. A pinion or drive gear (54) is mounted coaxially on the input shaft (58) in lateral alignment with a driven gear (52) which is mounted upon a laterally translatable gear shaft supported on a flexible coupling (66). Coupling (66) is drivingly connected at its opposite end to an axially rotatable but non-translatable output shaft (68). A biasing means, comprising a controllable, self-vented pneumatic piston/cylinder mechanism (100, Fig. 4), is positioned alongside the driven gear for urging it into engagement with the drive gear. The teeth of the drive gear and driven gear are of a particular wedge-shaped configuration, whereby they remain in mutual engagement until a predetermined torque limit is exceeded, whereupon radially outward force components produced by the confronting gear teeth faces are sufficient to overcome the oppositely directioned biasing forces produced by the pneumatic biasing mechanism and disengage the gears momentarily, until the next set of teeth are engaged. The slope of the confronting surfaces of the teeth, and the amount of biasing force, are selected to provide a desired level of torque.

Novel and technically advantageous features of the invention include the fluid powered biasing system, by which the torque limit may be adjusted conveniently and precisely by modifying the fluid pressure or by altering the pneumatic piston diameter, or by varying the diameter of one of the gears. Another innovative feature is the configuration of the flexible drive shaft upon which the driven gear is mounted. For certain applications, the invention has commercial advantages over prior mechanisms in that a precise degree of torque can be controllably applied to each of a plurality of driven members. A further technical advantage of the invention over some prior torque-applying devices is its use of mating, displaceable gears configured to minimize frictional wear during operation. Some prior devices tended to have limited useful life because the frictional drive mechanisms used tended to overheat due to friction when subject to heavy loads.

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Title: "Pressurized Fluid Torque Driver Control and Method"

PRESSURIZED FLUID TORQUE DRIVER CONTROL AND METHODORIGIN OF THE INVENTION

The invention described herein was made by an employee of the United States Government and may be manufactured and used  
5 by or for the Government of the United States of America for governmental purposes without the payment of any royalties thereon or therefor.

Technical Field

The present invention generally relates to torque control  
10 mechanisms and methods in threaded fastener drivers, such as nut runners and some types of screw drivers. More specifically, the present invention relates to a driver having a torque driver configuration for precisely limiting torque applied to the threaded fastener.

15 Background Art

It is often desirable to apply a highly accurate and repeatable torque to one or more fasteners, such as nuts, screws, or the like. Prior art methods, both manual and



automatic, do not provide an uncomplicated mechanism that limits torque applied to a fastener with sufficient accuracy that the fastener is repeatably tightened with a torquing force having a tight, consistent tolerance range. This is particularly true for drivers used to simultaneously install multiple fasteners. The torque applied to each fastener by a torque driver ideally is adjustable, although the torque applied to one or more fasteners in a particular fastener configuration may be required to be different than the torque applied to other fasteners in the same configuration. Numerous prior art drivers have attempted to solve these problems.

U.S. Patent No. 3,845,673 to Karden et al. discloses a two-speed nut runner that has a low torque clutch designed for disengagement at a predetermined, relatively low torque. A second high torque clutch automatically takes over the transmission of torque at a lower speed and higher torque while holding the low torque clutch in a fully released position by means of a piston that is placed inside the output shaft of the nut runner.

U.S. Patent No. 2,069,882 to W. Hall discloses a wrench for tightening a plurality of securing members that includes a plurality of rotatable spindles operable to transmit tightening forces to the securing members, and a transmission

operable by a single drive to rotate the several spindles independently to tighten the members to the same degree of tightness. The transmission includes pinions associated with the spindles, and an element, rotated by the drive, operable  
5 to successively cooperate with pairs of substantially opposite pinions to partially rotate the same.

U.S. Patent No. 2,781,682 to W.B. Herndon discloses a torque wrench with multiple spindles that provides a steady torque for setting screw-threaded fastenings while allowing  
10 the drive to slip when the fastening is tight.

U.S. Patent No. 4,909,105 to Namiki et al. discloses an automated nut driving apparatus having a plurality of motors for rotating respective ones of a plurality of drive shafts to respective sockets holding nuts. The nut driver includes a  
15 plurality of universal joints connecting the sockets and drive shafts while allowing the sockets to be tilted with respect to the drive shafts. Universal joints are movable axially with respect to the drive shafts to allow the sockets to be retracted under reactive forces produced when the nuts engage  
20 the wheel attachment bolts.

U.S. Patent No. 5,092,410 to Wallace et al. discloses a hydraulic torque impulse generator using a dual piston arrangement to provide impacts to a rotatable anvil. Automatic shut-off and control apparatus is provided for

limiting the pressure without reversing the direction of the driving clutch cage. A pressure venting arrangement permits one impact per revolution.

U.S. Patent No. 4,533,337 to K.C. Schoeps discloses a  
5 hydraulic torque impulse tool having a power inertia drive member, a hydraulic fluid chamber, and a cam driven piston in the fluid chamber for reciprocating movement.

U.S. Patent No. 5,125,298 to C.O. Smith discloses an automatic wheel assembly line in which an array of fastener  
10 members is prepared, corresponding in number and geometric pattern to the array of co-acting fastener members on the vehicle hub and to the array of bolt holes on the wheel assembly. The assembly includes a source of fasteners, a feeder mechanism having an outboard face, means defining a  
15 plurality of fastener receptacles, means operative to transport fasteners and means to move the loaded fasteners.

U.S. Patent No. 4,989,478 to Trivedi et al. discloses an apparatus for tightening or loosening a plurality of bolts or other rotatable elements in which a drive socket and a pair of  
20 reaction sockets are supported by an elongated beam member.

U.S. Patent No. 3,905,254 to Palatnick et al. discloses a tool for loosening and removing the lug nuts of an automobile and truck wheels with selectively positioned non-rotating stabilized sockets.



As can be understood from a review of the background as discussed above, there remains the need for an improved torque control mechanism that offers variable control over torque applied to one or more fasteners, a simplified construction  
5 for higher reliability, flexibility as to configuration of the fasteners, and a driver which may be produced at reduced levels of capital investment. Those skilled in the art will appreciate that the present invention provides solutions to these and other problems.

10

STATEMENT OF THE INVENTION

The present invention provides a method and apparatus for an improved torque controlled driver for applying torque to a fastener. The torque driver includes a driver housing, and a first gear carried by the driver housing for relative  
15 rotation with respect thereto. The first gear has first gear teeth mounted around a circumferential portion thereof. A second gear is provided to be displaceable with respect to the first gear between an engaged and a disengaged position. The second gear also has second gear teeth mounted around a  
20 circumferential portion thereof. The second gear teeth and the first gear teeth mesh in the engaged position, and at least one of the first and second gear teeth define an angled

tooth profile to produce a separating force in response to torque transfer between the first and second gears for moving the second gear to the disengaged position to thereby limit torque transferred between the first and second gears and to the fastener. A pressurized-chamber assembly is provided to bias the second gear toward the engaged position. The pressurized-chamber assembly includes a chamber for pressurization and a constant pressure fluid supply to maintain a constant pressure within the chamber to thereby produce a substantially constant bias on the second gear as the second gear is displaced from the engaged to the disengaged position.

In operation torque is transferred from the first gear to the second gear and to the fastener. The angled tooth profile on at least one of the first and second gears produces a separating force between the first and second gears during the transfer of torque between the first and second gears. The second gear is biased toward the first gear with a constant bias that remains substantially constant as the second gear moves away from the first gear in response to the separating force and the first gear is rotated to apply torque to the second gear and to the fastener until the separating force overcomes the constant bias and separates the first and second



gears to limit torque transfer between the first and second gears.

An object of the present invention is to provide an improved torque driver.

5 Another object is to provide a torque driver for selectively controlling torque to provide either a constant or varying torque to each of a plurality of fasteners.

Another object of the present invention is to provide a torque driver that will tighten one or more fasteners to a  
10 more precise torque.

Yet another object of the present is to provide a torque driver that is adaptable to a variety of fastener configurations.

A feature of the present invention is a pressurized  
15 cylinder assembly.

Another feature of the present invention includes a constant pressure pneumatic supply system.

Yet another feature of the present invention is a calibrated gage used to select a torque limit.

20 An advantage of the present invention is an uncomplicated, economical construction.

Another advantage of the present invention is a construction that allows the driver to be adapted to a wide variety of fastener configurations.

Another advantage is the ability to provide different torque to different fasteners within the same fastener configuration.

Other objects, features and intended advantages of the present invention will be readily apparent by the references to the following detailed description in connection with the accompanying drawings and claims.

#### BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic representation showing reaction forces and principal features with a pinion (driver) and driven gear;

FIG. 2 is an elevational view, in section, of a torque driver with gears in engaged position;

FIG. 3 is an elevational view, in section, of the driver of FIG. 2 with gears displaced from each other;

FIG. 4 is a top view, in section, of a driver according to FIG. 2 but with a pressurized cylinder biasing assembly;

FIG. 4A is a side view, in section, of the driver of FIG. 4 along the line 4A-4A;

FIG. 5 is a schematic representation of a self-venting pressure bias control system;

FIG. 6 is an elevational view, in section, of a bevel gear driver;

FIG. 6A is a sectional view of the driver of FIG. 6 along the lines 6A-6A;

5        FIG. 6B is a sectional view of the driver of FIG. 6 along the lines 6B-6B;

FIG. 6C is a sectional view of the driver of FIG. 6 along the lines 6C-6C;

10       FIG. 7 is an elevational view, in section, of the driver of FIG. 6 with gears displaced;

FIG. 7A is an elevational view, in section, of the axial force clutch of FIG. 6 in the disengaged position;

FIG. 8 is an elevational view, in section, of a bevel gear driver with pneumatic bias;

15       FIG. 8A is an elevational view, in section, of a bevel gear driver without an intermediate bevel gear assembly;

FIG. 9 is an elevational view, in section, of a spur gear driver with adjustable fastener driver configuration;

20       FIG. 9A is a sectional view of the driver of FIG. 9 along line 9A-9A; and

FIG. 10 is a top view, in section, of a driver for a non-circular fastener configuration.

While the present application includes claims directed to a driver having a pneumatic biasing system as explained

herein, it will be understood that figures which show spring biasing may be modified for pneumatic biasing and are necessary to provide a complete understanding of the present invention. It will also be understood that claims directed to  
5 a pneumatically biased driver will also include drivers that may, in addition to pneumatic biasing, include a spring or other biasing means.

Thus, while the invention will be described in connection with the presently preferred embodiments, it will be  
10 understood that it is not intended to limit the invention to these embodiments. On the contrary, it is intended to cover all alternatives, modifications, and equivalents as may be included in the spirit of the invention.

#### DESCRIPTION OF THE PREFERRED EMBODIMENTS

15 The present invention relates to an improved torque driver mechanism operable to transmit a precisely controllable degree of torque to a driven member such as a nut, screw, or the like. The torque driver may receive an input torque which is greater than a desired output torque and is operable to  
20 limit the input torque to the desired output torque level.

In very general terms, the application of gears for torque transmission between shafts falls into three categories

of (1) parallel shafts, (2) shafts with intersecting axes, and (3) shafts neither parallel nor intersecting but skew.

While the present application includes claims directed to a driver having a pneumatic biasing system as explained  
5 herein, it will be understood that figures which show spring biasing may be modified for pneumatic biasing and are necessary to provide a complete understanding of the present invention. It will also be understood that claims directed to a pneumatically biased driver will also include drivers that  
10 may, in addition to pneumatic biasing, include a spring or other biasing means.

The basic gear forces involved in operation of a torque driver in accord with the present invention are best described in connection with FIG. 1 wherein gear teeth 12 and 14, on  
15 respective gears 13 and 15, are shown in some detail. Respective gear teeth 12 and 14 provide a positive drive, maintaining exact velocity ratios between pinion or driver gear 13 and driven gear 15 to positively rotate one fastener, or more than one fasteners simultaneously, up to the desired  
20 torque.

Pitch circles, 16 and 18, are generally defined as the circles whose periphery is the pitch surface, or surface of an imaginary cylinder that would transmit by rolling contact, the same motion as toothed gears 13 and 15. Motion transmitted

between gears 13 and 15 is similar to that of rolling surfaces identical with the gear pitch surfaces, but the action of gear teeth 12 on gear teeth 14 is generally a combination of rolling and sliding motion.

5       The radial distances between bottom lands 24, 26 and respective pitch circles 16, 18 is referred to as the respective dedendums. Circles 28, 30 are the base circles of pinion 13 and driven gear 15 that are generally defined as the circle from which an involute tooth curve is generated. The  
10 addendum circle (not shown) is the circle connecting the top lands 32, 33 of teeth 12, 14 and the corresponding radial distance to pitch circles 16, 18 is the addendum. Faces 35, 37 are the surfaces of teeth 12, 14 radially outwardly of pitch circles 16, 18. Flanks 39, 41 are the surfaces of teeth  
15 12, 14 radially inward of pitch circles 16, 18.

Pitch point 38 is the point at which pitch circles 16 and 18 are tangent and this is typically the only point where contact between the teeth is pure rolling contact depending on the teeth profiles which may be of numerous types or modified  
20 types including involute, involute stub tooth, cycloid, American Standard spur gear tooth forms, and the like, made by different methods including hobbing, shaping, milling, pre-shaving, and the like.



Each pair of teeth has its points of contact co-linear with pressure line 36. The angle  $\phi$  between pressure line 36 and tangent line 40 is the pressure angle. Force P is the reaction force on bearings of pinion 13 and force D is the  
5 reaction force on bearings of driven gear 15. Force N is the force normal to tooth profiles at the pitch line. Force T is the tangential or driving force. Force R is the radial or separating force.

While tangential force T is used to produce output  
10 torque, radial force R is often considered a wasteful byproduct. However, force R is used to control output torque by automatically disengaging the gear at a specified torque.

When the fastener is tightened to the specified torque, the radial force R at the engaged teeth equals a bias force,  
15 discussed hereinafter that urges pinion 13 and driven gear 15 into engagement. Any higher torque causes gears 13 and 15 to move laterally away from each other in this embodiment of the invention.

As gears 13 and 15 move laterally away from each other,  
20 pressure angle  $\phi$  typically increases depending on the gear teeth profile (such as the frequently used involute or modified involute gear teeth profiles) causing a decrease in tangential force T (and output torque) and an increase in radial force R to thereby facilitate additional displacement.

Thus, the separation process typically accelerates to completion rapidly after it first begins.

After gears 13 and 15 are displaced by the working depth (sum of the addendums of the teeth 12 and 14), pinion 13 is  
5 disengaged. Pinion 13 continues to turn and biasing will cause pinion 13 to re-engage with the next set of teeth. The process of displacement and disengagement continues until input torque is removed. In this manner, the torque applied to the threaded fastener is limited to the specified torque.

10 FIG. 2 and 3 show how this principle is applied for use with spur gear driver 10. Spur gears are generally considered to be those gears that transmit torque between parallel shafts and may have straight teeth parallel to the gear axis or helical or herringbone teeth patterns and may include  
15 external, internal, and rack and pinion type configurations. Gears 50 and 52 are external spur gears with teeth 54 and 56 which point outwardly from shafts 58 and 60, respectively. Gears 50 and 52 and other components discussed hereinafter are preferably mounted in driver housing 61 with antifriction  
20 bearing assemblies such as ball-bearing assemblies 64 or other types of antifriction bearing assemblies as desired. Various bearing assemblies may be used with the rotating components as desired. Thus, while various sliding surfaces may be shown or discussed herein without specifying types of antifriction



bearings, or indicating the presence of bearing surfaces in the associated drawings, it will be understood that those skilled in the art will be aware of the need to supply the same where desired.

5       As well, various types of lubrication may be used for any sliding surfaces discussed or shown herein. For instance, self-lubricated teflon coated or impregnated bronze surfaces may be used for relatively sliding components. Other suitable coating materials could be used for self-lubricated surfaces  
10 on the various components shown and discussed herein as are known to those skilled in the art. Those skilled in the art will recognize that many methods for lubricating any relatively sliding surfaces that are discussed herein. For instance, the various chambers and suitable elements may be  
15 flooded with oil. Pneumatic biasing systems, as discussed hereinafter, may be circulated through oil baths for air lubrication. It is also to be understood that antifriction means include lubricants and self-lubricating materials that may be used on the various relatively sliding components  
20 discussed and shown herein.

Input shaft 58 is preferably connected to an input torque source (not shown) that supplies torque at a higher level than the preferred specified output torque limit from shaft 60. The input torque source may include a manual ratchet wrench,

electric drill, pneumatic motor or the like. Output shaft 60 is connected through flexible linkage or coupling 66 to output shaft attachment 68 which is then coupled to a fastener (not shown).

5       Flexible linkage 66 preferably has a coil configuration that is flexible in all directions and may be obtained commercially with specified coil configurations, windup, spring rates, special end connections, axial offsets, bending angle features, and other specified features, from companies  
10 such as Helical Products Company, Inc. Flexible linkage 66 could also be provided with U-joints and constant velocity splines (not shown), or other linkage components including combinations of components including a coiled linkage as desired. However, coiled linkage 66 is presently preferred  
15 for simplicity, lower cost, and ease of construction and maintenance. As well, flexible linkage 66 may be disposed in an angled, rounded, or other configuration as desired to tighten fasteners located in difficult to reach positions.

Driven gear 52 and shaft 60 are mounted for rotation in  
20 displaceable support member 70. Displaceable support member 70 is laterally displaceable in a direction transverse to input shaft axis 72 which, in this embodiment, is parallel to output shaft axis 74. Displaceable support member 70 moves laterally with respect to pinion 50 within chamber 71 of

driver housing 61 between an engaged position and a disengaged position shown, respectively, in FIG. 2 and FIG. 3. Displaceable support member 70 may have various cross-sectional shapes such as, but not limited to, circular, 5 elliptical, or square shapes.

Displaceable support member 70 and its related bearings 64 for supporting shaft 60 are loaded by compressed spring 76 to produce a biasing force that tends to move displaceable support member 70 towards pinion 50. To control the specified 10 level of torque limit, the level of the bias, or other factors discussed hereinafter, may be adjusted.

The required bias can be obtained from geometrical relationships using the gearing configuration that includes e.g. effective gear radius and pressure angle. Various sizes 15 and combinations of gears may be selected based on the particular requirements, speed, and so forth of the application. Load adjustment shims 78 or other means such as a moveable adjustment plug (not shown) may be used to adjust the bias to the desired level. Thus, spring bias, or other 20 types of bias, may be either preset or adjustable.

A preset version can be used, for instance, with critical assembly operations where the operator needs only one torque setting. An adjustable version, which could be manually or

automatically adjustable, could be used for a wider assortment of operations when multiple torque settings are needed.

The input torque source (not shown) rotates input shaft 58 and also rotates pinion 50, which is secured to input shaft 58 for rotation therewith. When the fastener is tightened to the specified torque limit, the separating force at engaged teeth 54 and 56 equals the biasing force. Higher torque causes driven gear 52 to move laterally away from pinion 50. As driven gear 52 moves laterally away from pinion 50, the pressure angle increases as discussed hereinbefore, causing a decrease in tangential force  $T$  (and related output torque) and an increase in radial or separating force  $R$  to facilitate lateral movement of displaceable support member 70 toward a disengagement position shown in FIG. 3.

After driven gear 52 is displaced by its working depth to disengage from pinion 50 as shown in FIG. 2, pinion 50 continues to rotate. Spring bias 76 then biases displaceable support member 70 to move driven gear 52, which is substantially no longer rotating, into engagement with the next set of teeth 54 on pinion 50. The process of displacement and disengagement repeats until input torque is removed.

Flexible linkage 66 remains connected to the fastener throughout the torque sequence and adapts for movement of



displaceable support member 70. Adapter 68 is preferably rotatably secured, with a bearing assembly 80, to driver housing 61 so that adapter 68 rotates but is axially fixed with respect to housing 61, at least in the embodiment of FIG.

5 2 and FIG. 3. Bearing assembly 80 may be of several types and shapes but is preferably used to prevent radial forces that could interfere with loading of displaceable support member 70 and thereby provide more accurate torque limiting action. See discussion of bearings and lubrication hereinbefore.

10 The design selection of pinion 50 and driven gear 52 requires consideration of such factors as pressure angle, contact ratio (ratio of length of path two gear teeth are in contact to base pitch), teeth depth, radii, tooth profile, and the like chosen for the particular application.

15 FIG. 4 and FIG. 4A show a top view of alternative embodiment driver 10A that is substantially the same as driver 10 except for an alternative pneumatic biasing assembly 100. Components with substantially identical counterparts in driver 10 are labeled with an "A" suffix and reference may be made to  
20 the earlier discussion for those components so labeled.

As with driver 10, displaceable support member 70A moves within cavity 71A between an engaged and a disengaged position. Cavity 71A has a first section 73 that has a substantially consistent rectangular cross-section along its



length in which a rectangular portion, as shown in FIG. 4A, of displaceable support member 70A moves. It will be understood that this cross-section could also be cylindrically, elliptically, or otherwise configured as desired. As with  
5 driver 10, cavity 71A also has an expanded portion 75 which allows lateral movement of driven gear 52A within cavity 71A.

Sealed cavity 102 is preferably cylindrically shaped and is preferably sealed with sealing elements such as O-ring 104 and lubricated to reduce friction. As displaceable support  
10 member 70A moves between its engaged and disengaged position, the volume 105 of sealed cavity 102 and also cavity 106 varies accordingly. Piston 108 is threadably secured to driver housing 61A but is moveable relative to displaceable support member 70A.

15 Passageway 110 through piston 108 interconnects cavity 102 to a preferred constant pressure pneumatic source, discussed hereinafter with respect to FIG. 5, to maintain a constant pressure within cavity 102 and thereby produce a constant biasing force on wall 112 of cavity 102.

20 A constant pressure pneumatic biasing source of this type is inherently more accurate than a spring bias for several reasons. The spring bias tends to have a relatively larger tolerance variation in spring pressure over relatively short distances than a pneumatic bias assembly.

As well, the spring bias increases and decreases with compression and expansion to adversely affect torque limiting accuracy. Even though the pressure angle between the gear teeth typically increases to thereby increase the radial or  
5 separating force after the desired torque is reached, the spring bias force also increases with respect to the bias force of the engaged position to inhibit the desired slippage operation between pinion 50 and driven gear 52. With a constant pressure bias, the movement towards the disengaged  
10 position occurs more rapidly.

Thus, the constant pressure biasing assembly 100 produces a constant bias even as displaceable support member 70A moves laterally with respect from pinion 50A between an engaged position and a disengaged position. As the pressure angle  
15 changes and the radial force increases, the load on displaceable member 70A remains constant to correspond to the specified output torque. Preferably, a calibration record is obtained to measure cavity pressure versus output torque so as to provide an accurate calibration of each machine. Thus,  
20 torque control level may be set with a sufficiently accurate pressure gage having sufficient resolution to be calibrated in terms of the desired torque limit and tolerance required thereof.

Various configurations of pressurized cylinder-piston assemblies with related seals and shafts may be used with that shown in FIG. 4 being a presently preferred configuration. The bias force produced to act against displaceable support member 70A is a function of piston/cylinder geometry (e.g. internal diameter and pressure). Maintaining a constant pressure in the cylinder ensures that during lateral displacement of gear 52A towards or away from pinion 50A, the bias will always remain at the preset level which corresponds with the specified torque output.

FIG. 5 discloses a schematic of a self-venting pressure regulator system 130 that relieves downstream pressure at two different desired setpoints for applying different, but readily adjustable, torque limits to different threaded fasteners in a configuration of fasteners. The multiple fastener torque drivers shown in FIG. 9 and FIG. 10 are examples of drivers that may require such adaptability. System 130 could also be used to supply constant pressure to a single driver such as that of FIG. 4 or FIG. 8 or could be modified to provide for more than two different torque limits if desired. Safety devices such as relief valves (not shown) may be used therein as desired e.g. in the supply line.

Pressurized air enters inlet 132 and is regulated by first check valve 134. Pressure indicated at first pressure

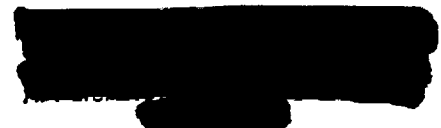




gage 136 will preferably be higher than subsequent output pressures at outlets 1 through N. Adjustable self-venting pressure regulator 138 may be used in conjunction with gage 136 to set a desired pressure corresponding to a desired torque level at outputs 1 through I one of which could be connected, for instance, to passage 110 of FIG. 4. The letter I indicates that as many connections as desired up to the limits of the system may be used.

To obtain a second selective torque limit, the pneumatic line branches at 142 to second check valve 144. The output of second check valve 144 goes to second adjustable self-venting pressure regulator 146 that may be adjusted in accord with gage 148 to obtain a second pressure output at outputs I + 1 to N. While system 130 is a particular pressure supply system, other systems may also be used. The outputs of system 130 may be ramped, sequenced, or otherwise varied to control torquing of fasteners in a desired manner. Even while pressure is relatively slowly ramped, it will be understood that the pressure within a piston/cylinder assembly remains substantially constant during the short time in which a laterally or axially displaceable gear moves from an engaged to a disengaged position.

FIG. 6 and FIG. 7 discloses a conical or bevel gear driver 160 having an adjustable bias configuration. Driver



160 operates on a similar principle as that illustrated in FIG. 1 of loading a displaceable gear to control output torque. However, the bevel gear configuration of driver 160 has an axial load scheme such that the displacement occurs axially, with respect to a gear shaft axis, rather than radially with respect to a gear axis, as with the spur gear arrangement. In the embodiment illustrated, the input gear or drive gear is displaceable rather than the output or driven gear.

10 Bevel or conical gears are normally those which may be used to change the shaft axis direction or to change shaft speed as well as direction. Bevel gears include many variations such as external bevel gears, internal bevel gears, crown gears, straight bevel gears, spiral bevel gears, zero  
15 bevel gears, and the concept of operation of this embodiment of the invention, as discussed hereinafter, could conceivably be developed with hypoid gears and worm gears which are therefore also considered, for the purposes of this application, to come under the general term of bevel gears.

20 Bevel driver 160 includes three basic gear assemblies including displaceable gear 162, intermediate gears 164, and output gear 166. Displaceable gear 162 is axially displaceable in a direction parallel to axis 168 of input shaft 170 and axis 172 of output shaft 174. Displaceable gear

162 is biased into engagement with intermediate gears 164 by biasing spring 176. Key 163, shown also in FIG. 6B, rotationally secures displaceable gear 162 with respect to input shaft 170 but permits axial movement therebetween.

- 5 Thrust bearing 167 is preferably connected to the lower end of shaft 170 for support, to reduce play, and to reduce friction between shaft 170 and disc 180 during rotation.

While gear 162 is shown as a monolithic component, it could be comprised of an assembly of components including  
10 bearings, discs, and so forth. The gear train of the bevel gear driver 160 could also be arranged so as to be angled if desired as may be necessary to apply torque to fasteners located in difficult to reach positions.

The components of driver 160 are mounted within  
15 cylindrical driver housing 178. Various types of bearings may be used between driver housing 178 and any movable components such as in the gear train as well as between the shafts and the supporting members (see general discussion of bearings and lubrication above).

- 20 Shafts 165, shown also in FIG. 6C for intermediate bevel gears, are preferably held in their respective positions with respect to driver housing 178 i.e. their associated shaft axes are fixed with respect to driver housing 178. They may be pinned to driver housing 178, held by internal slots within



driver housing 178, or otherwise affixed thereto. As well, disc 180, which supports intermediate gears 164 and their related shafts 165, may be fixed by pins or other means to secure disc 180 in position with respect to driver housing 5 178. Because the axes of shafts 165 are fixed with respect to housing 178, bevel gears 164 may be used to rotate output gear 164 via output gear teeth 169 (see FIG. 6C) in response to rotation of displaceable input gear 162. Thrust bearing 167 may be provided to reduce friction and play between disc 180 10 and shaft 170.

Cap 182 holds the internal components of driver 160 within driver housing 178 and may be threadably secured thereto, as shown, with fine adjustment micrometer type threads 184. Fine adjustment threads 184 allow bias, and 15 therefore torque limit, to be adjusted by rotation of cap 182. Driver 182 may be calibrated so that torque indication marks 186 provide a direct reading of the selected torque limit for driver 160. Lock nut 188, or other locking means, may be used to fix cap 182 with respect to driver housing 178 and maintain 20 a selected torque limit.

Driver 160 also includes axial load control or clutch assembly 200 that comprises input attachment 202, attachment housing 204, axial load control spring 206, and splined drive disc 208. As shown in FIG. 6A, spline-groove interfaces 210

between drive disc 208 and attachment housing 204 are engaged to transmit torque to input shaft 170. The spline-groove interfaces disengage in the manner shown in FIG. 7A to prevent torque transmission if axial force applied, as by an operator, 5 is greater than a selected amount. As discussed hereinbefore, spline-groove interfaces 210 may be self-lubricated or otherwise lubricated as desired.

Input shaft 170 is fixably secured to drive disc 208 by means not shown such as welds, pins, threads, snap-rings, or 10 other means. Axial load control or clutch assembly 200 therefore acts to prevent excessive axial load being applied to the fastener through driver 160. Excessive axial load could alter bias applied with spring 176 to thereby change the selected torque limit.

15 If the axial load is greater than a selected amount, attachment housing 204 moves towards cap 182 to thereby disengage drive disc 208 with respect to attachment housing 204 as shown in FIG. 7A. This action prevents torque transmission to shaft 170. As well, it is noted that clutch 20 200 could be positioned elsewhere along the shaft and gear assembly prior to connection with fastener 212 to prevent torque transmission to fastener 212.

In operation, torque applied through input shaft 170 is transmitted through output shaft 174 to fastener 212. If an



operator presses too hard on the fastener 212 with driver 160 then clutch 200 disengages torque to prevent any torque transmission to fastener 212. Otherwise, torquing of fastener 212 continues until a desired torque limit is reached as indicated by calibration marks 186. At that point, displaceable gear 162 moves upwardly with respect to intermediate bevel gears 164, as shown in FIG. 7, to a disengaged position. The next set of teeth on displaceable gear 162 will then re-engage bevel gears 164 due to bias spring 176, or other bias mechanism, and the process repeats until torque is removed from input attachment 202.

FIG. 6C shows four bevel gears with a fixed and equidistant circumferential spacing with respect to driver housing 178 and driven gear 166. Other numbers of bevel gears could be selected based largely upon the anticipated torque-related stress to be carried by each of bevel gears 164. Thus, in some applications three bevel gears in an intermediate bevel gear assembly may be sufficient to handle the anticipated torque requirements. The multiple number of bevel gears 164 provide additional durability because wear between the multiple gears carrying the torque reduces wear which would otherwise occur between a single driver-driven gear arrangement.

In addition to increasing torque adjustment accuracy, the bevel gear assembly is relatively inexpensive, is easy to manufacture, and provides opportunity for considerable flexibility of tooth profile design for the intended slippage  
5 type operation and/or for angled operation of the driver.

FIG. 8 provides an alternative bevel gear driver configuration 160A with pneumatic biasing assembly 230. Pressurized cylinder sleeve 232 contains pressurized fluid, preferably gas held at a constant pressure in variable volume  
10 cavity 233, as discussed hereinbefore, to produce a biasing force against piston 234. Piston 234, in turn, acts against displaceable gear 162A in the manner discussed hereinbefore.

Seal 236, an O-ring seal, or other types of seals may be used to seal cavity 233 from air leakage. For instance,  
15 grease from grease inserts, grease wicks, or other sources may act to form suitable seals and simultaneously lubricate relatively close fitting bearing surfaces around shafts 170A and 174A.

Pressurized gas preferably passes through passageway 238  
20 of connector 240 to cavity 233. For best accuracy, pressurized gas will be from a constant pressure regulated air supply, such as system 130 shown in FIG. 5, or other constant pressure regulated fluid supply systems. A sufficiently accurate pressure gage may be calibrated and operated with a

vented pressure regulator to set driver 160A at a desired torque limit.

As explained hereinbefore with respect to driver 10A shown in FIG. 4 and FIG. 4A, a constant pressure used with a pressurized cylinder, as shown in FIG. 8, creates a constant bias that is more accurate than a spring bias. The spring bias increases with distance and may also have significant bias force tolerances that limit accuracy and thereby affect the desired torque level.

FIG. 8A discloses another alternative bevel gear driver arrangement 160B wherein the intermediate bevel gear assembly is absent but which includes a torque limit adjustment mechanism as may be conveniently hand-held. This embodiment may relate to consumer oriented specifications which may not require the long lasting features for only occasional or hobby usage. Upper gear 162B directly drives output gear 166B to thereby rotate output shaft 174B up to the torque limit as discussed above. Thrust bearing 167B may be included to reduce friction and for the same purposes discussed hereinbefore in relation to thrust bearing 167. While it is desirable to include the intermediate bevel gear assembly for the reasons discussed above, it is also possible to provide a selectable torque nut driver for removing or tightening fasteners without that assembly. As with the other



configurations, the bevel gear driver of FIG. 8A may be used with a battery driven electric motor as may be conveniently hand held. This configuration could also be used with a pneumatic motor and include the pneumatic biasing assembly 230  
5 discussed above as may be more desirable for shop use. Other similar configurations and arrangements in accord with the teachings herein as will be apparent to those skilled in the art.

FIG. 9 and 9A show a multiple fastener driver 250 that  
10 may be used to drive multiple fasteners 252 for simultaneous tightening or loosening. Socket 255 may be adapted to include fastener storage and control means (not shown) for storing and controlling multiple nuts 252 and 253 as might be removed from four tires for subsequent reinstallation. As will be  
15 appreciated, simultaneous removal of fasteners could save significant time, for instance, in a business that changes out large numbers of tires each day. If desired, socket 255 could be a standard socket that receives only a single nut 252.

Driver 250 may be used to drive fasteners 252 in the same  
20 configuration to different selectable torque levels by coupling plugs, such as coupling plugs 254 and 256, to pressurized gas or fluid at different pressures as may be obtained from system 130 of FIG. 5. The pressurized gas supply obtained from system 130 may also be ramped during the



nut driving process to limit torque applied to any particular nut in a fastener configuration to thereby verify that all fasteners are limited to the same torque at all times - a feature that overcomes a problem with prior art devices. It will be noted as discussed hereinbefore, that a ramped pressure remains substantially constant during the short time period in which a moveable gear, such as output gear 262, moves from an engaged position to a disengaged position. A sequencer (not shown) could be used to specially sequence the order of fastener installation by controlling, for instance, the bias from system 130.

The basis of operation of driver 250 is similar to that of driver 10 and 10A shown in FIG.2 - FIG. 4. Input torque is applied to shaft 258 and pinion 260. Pinion 260 drives each output gear 262 at the same rate to thereby torque multiple fasteners 252 simultaneously as may be required for some applications. Each output gear 262 is carried by an associated displaceable gear support member 264 moveable within a respective support member cavity 266.

Coiled flexible drive shafts 268 each individually adjust to lateral movement of the respective output gears 262. Shaft connector members 270 are preferably rotatably secured to outer leg housing 272 and inner leg housing 271 by intermediate bearing members 273 which may be of various

types. Intermediate bearing members 273 isolate radial forces on flexible coupling 268 and 292 to prevent interference with displaceable gear support 264 control loading. Inner and outer leg members 271 and 272 are substantially cylindrical  
5 (see FIG. 9A) in the present embodiment but could be separate leg members for each fastener shaft 268. Hole 281 may be included to allow torquing operation around a centrally located shaft.

Driver 250 also includes a gauged selector mechanism 280  
10 shown more clearly in FIG. 9A which is a cross-section of FIG. 9 along the line 9A-9A. Gauged selector mechanism 280 allows driver 250 to be used with more than one fastener configuration by movement of fastener adapters 290.

For this purpose, ring 282 is rotatable via control  
15 handle 286 to move translatable blocks 284 laterally inwardly or outwardly. Output adapters 290 are then also movable within slots 288 in end member 294 as indicated in FIG. 9A. Antifriction bearings 285 are preferably used to rotatable secure output adapters 290 in translatable blocks 284. As  
20 discussed herein before, various lubrication means are preferably provided between moving parts such as blocks and other sliding or rotating surfaces. Flexible coiled fastener drive shafts 292 flex as required to adjust shaft length and orientation to compensate for movement of output adapters 290.

Other types of shaft linkage such as universal joints could be used in place of or in conjunction with flexible coiled shafts 268 and 292 as discussed hereinbefore.

Ring 282 is supported by end member 294 that is secured  
5 to outer leg housing 272 with connectors 298. Bias springs 296 bias translatable blocks 284, having curved guide surfaces 300, to move along inner ring cam surfaces 302 that determine the lateral orientation of fastener adapters 290. Guide and cam surfaces 302 and 300 may have various shapes as desired.  
10 Preferably there is a single point interaction between the surfaces so that translatable blocks 284 follow cam surface 302 more readily. Control handle 286 moves in slot 304 of end member 294 which may be calibrated or marked as desired for various lateral orientations. While end 303 is shown as a  
15 point, it could also be square shaped, round or otherwise shaped. Other adjustment mechanisms, such as individually adjustable gauge selector members (not shown) could also be used. Other adjustment paths for fastener adapters 290 defined herein by slots, such as slots 288, could also be  
20 used.

FIG. 10 provides another driver 350 embodiment suitable for torquing fasteners in non-circular fastener configurations such as might be suitable for manifold fastener configurations. While various types of gear drives could be

use, this particular embodiment uses inner pinion 352 may be used for torque input to drive output gears 353 and to transfer torque via idler gears 354 to outer pinions 356 which, in turn, drive output gears 358. Other drive means  
5 could be used. For instance, a centrally disposed elongate worm gear could be used as the principal torque input and transfer means. Such a worm gear could extend across driver 350 from each gear 356 in place of the present gear chain and would tend, at least in this embodiment, to reduce or  
10 eliminate gear chatter noise.

As discussed hereinbefore, each output gear 358 is laterally displaceable within a chamber 362 and is supported with displaceable gear support member 364. Fasteners may be simultaneously rotated up to a desired torque limit. The  
15 torque limit may vary, it may be ramped, or otherwise controlled as desired between fasteners in the same configuration as discussed hereinbefore. Each air connector 360 is preferably connected to a constant pressure pneumatic supply system such as system 130 shown in FIG. 5 for biasing  
20 each respective gear support member 364.

In general, in the various torque drivers discussed, the gear teeth on one or both gears may be modified so as to be short, stubby, and/or rounded to further facilitate this operation as desired. As well, for high speed operation,



means could be used to completely disengage the gears after the specified torque is reached and displacement begins if this was desired. However, design of short, stubby, teeth may be more economical than a kick-out mechanism or other such  
5 means.

The foregoing disclosure and description of the invention is illustrative and explanatory thereof, and it will be appreciated by those skilled in the art, that various changes in the size, shape and materials as well as in the details of  
10 the illustrated construction, reliability configurations, or combinations of features of the various torque driver elements of the present invention may be made without departing from the spirit of the invention.

ABSTRACT OF THE INVENTION

Methods and apparatus are provided for a torque driver including a displaceable gear to limit torque transfer to a fastener at a precisely controlled torque limit. A biasing  
5 assembly biases a first gear into engagement with a second gear for torque transfer between the first and second gear. The biasing assembly includes a pressurized cylinder controlled at a constant pressure that corresponds to a torque limit. A calibrated gage and valve is used to set the desired  
10 torque limit. One or more coiled output linkages connect the first gear with the fastener adaptor which may be a socket for a nut. A gear tooth profile provides a separation force that overcomes the bias to limit torque at the desired torque limit. Multiple fasteners may be rotated simultaneously to a  
15 desired torque limit if additional output spur gears are provided. The torque limit is adjustable and may be different for fasteners within the same fastener configuration.



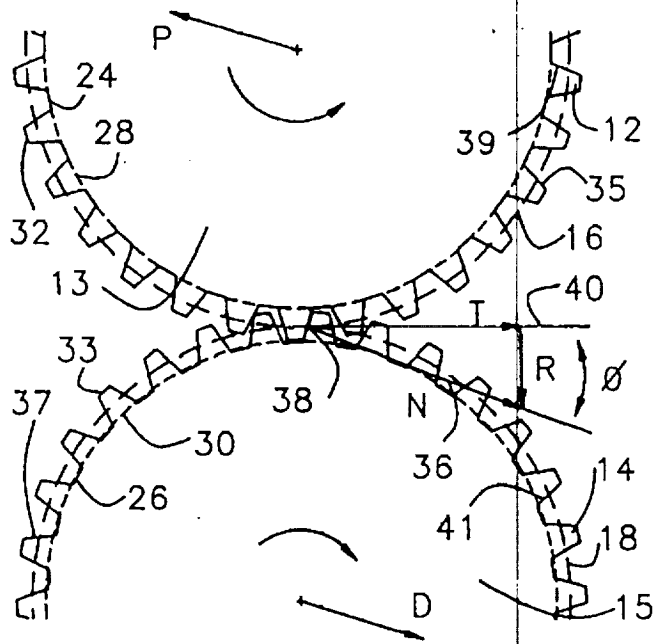


FIG. 1

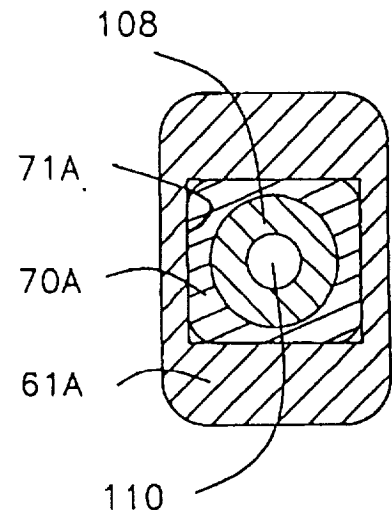


FIG. 4A

10A

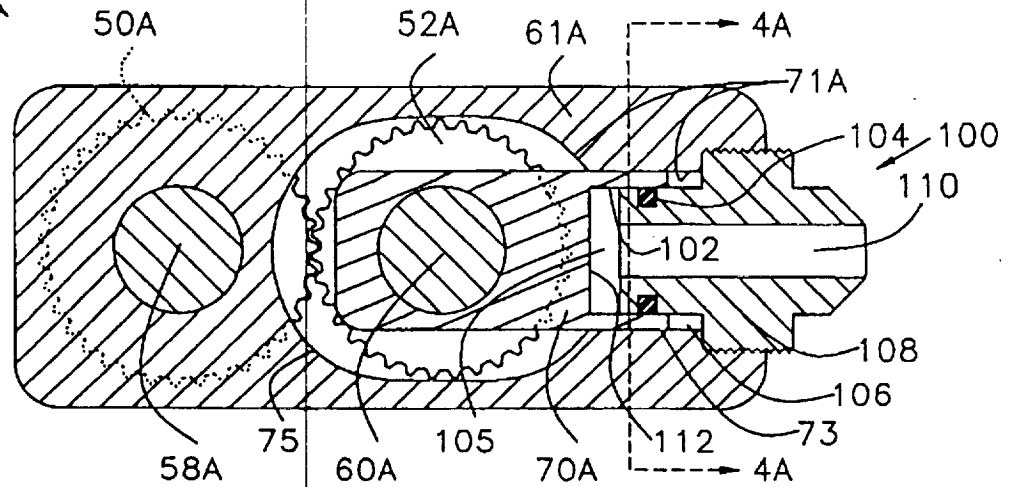


FIG. 4



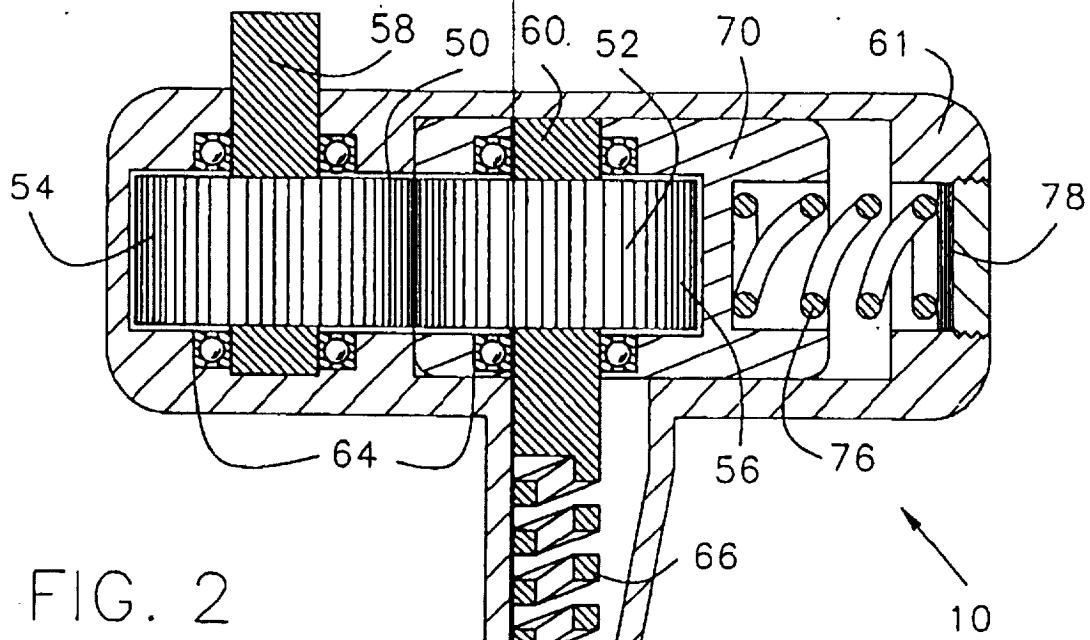


FIG. 2

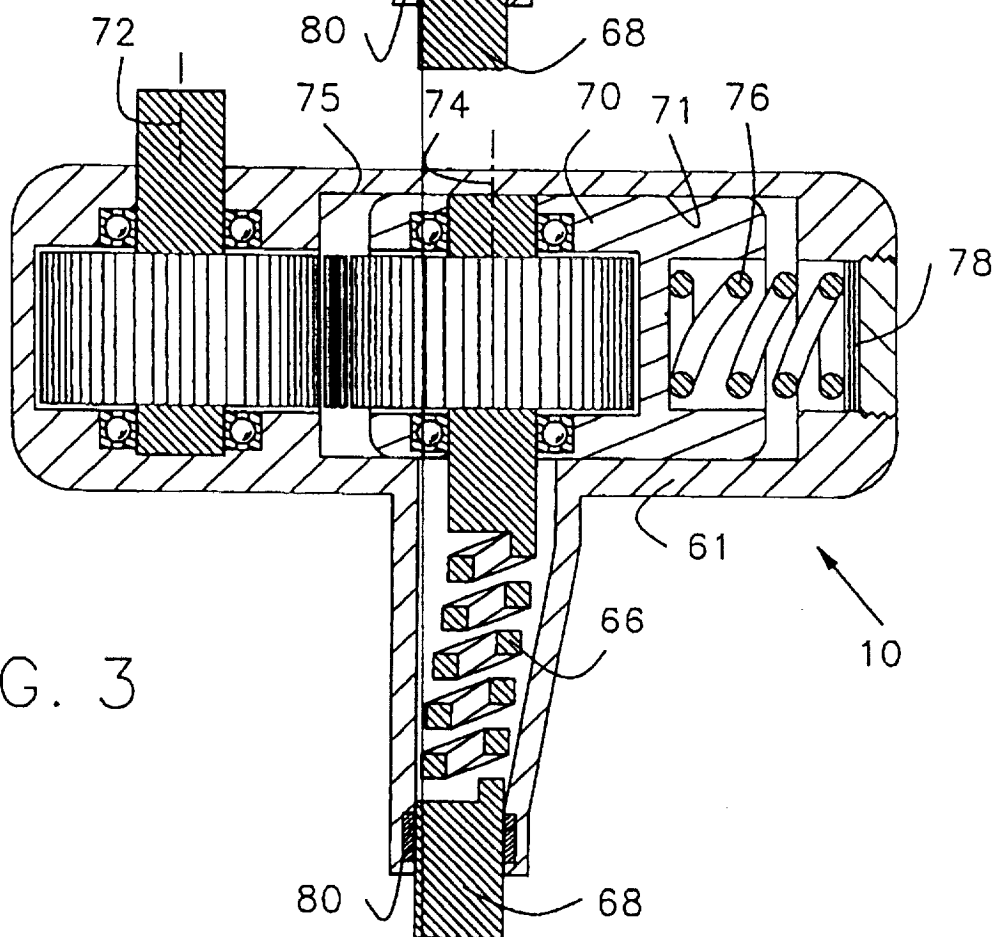
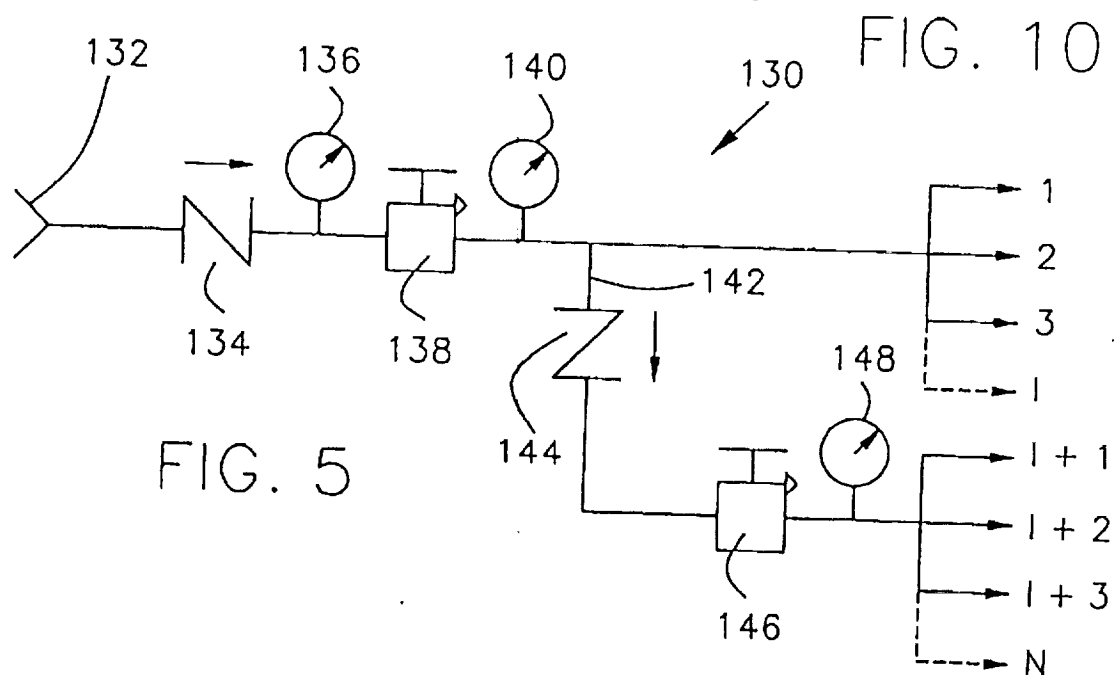
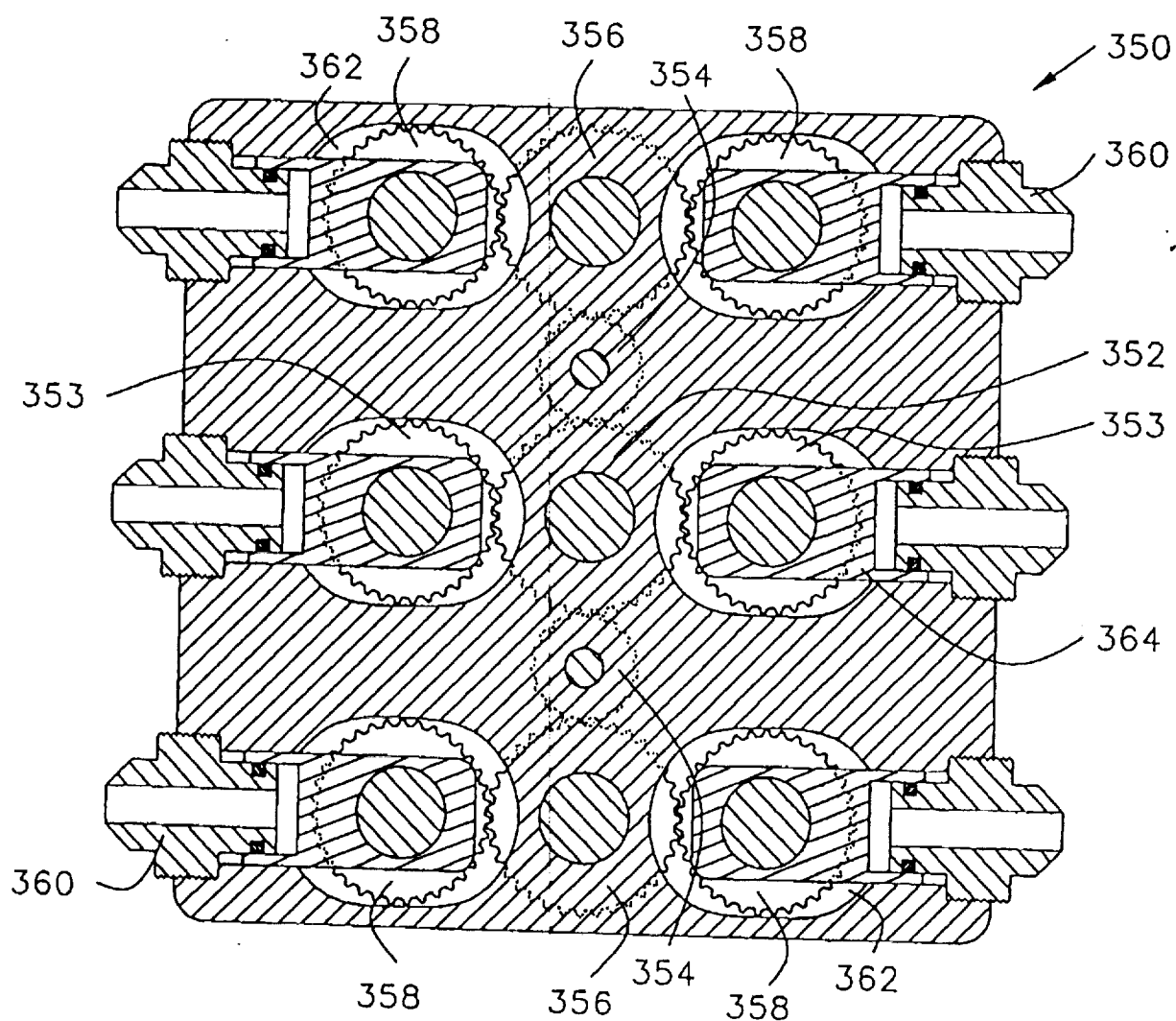


FIG. 3



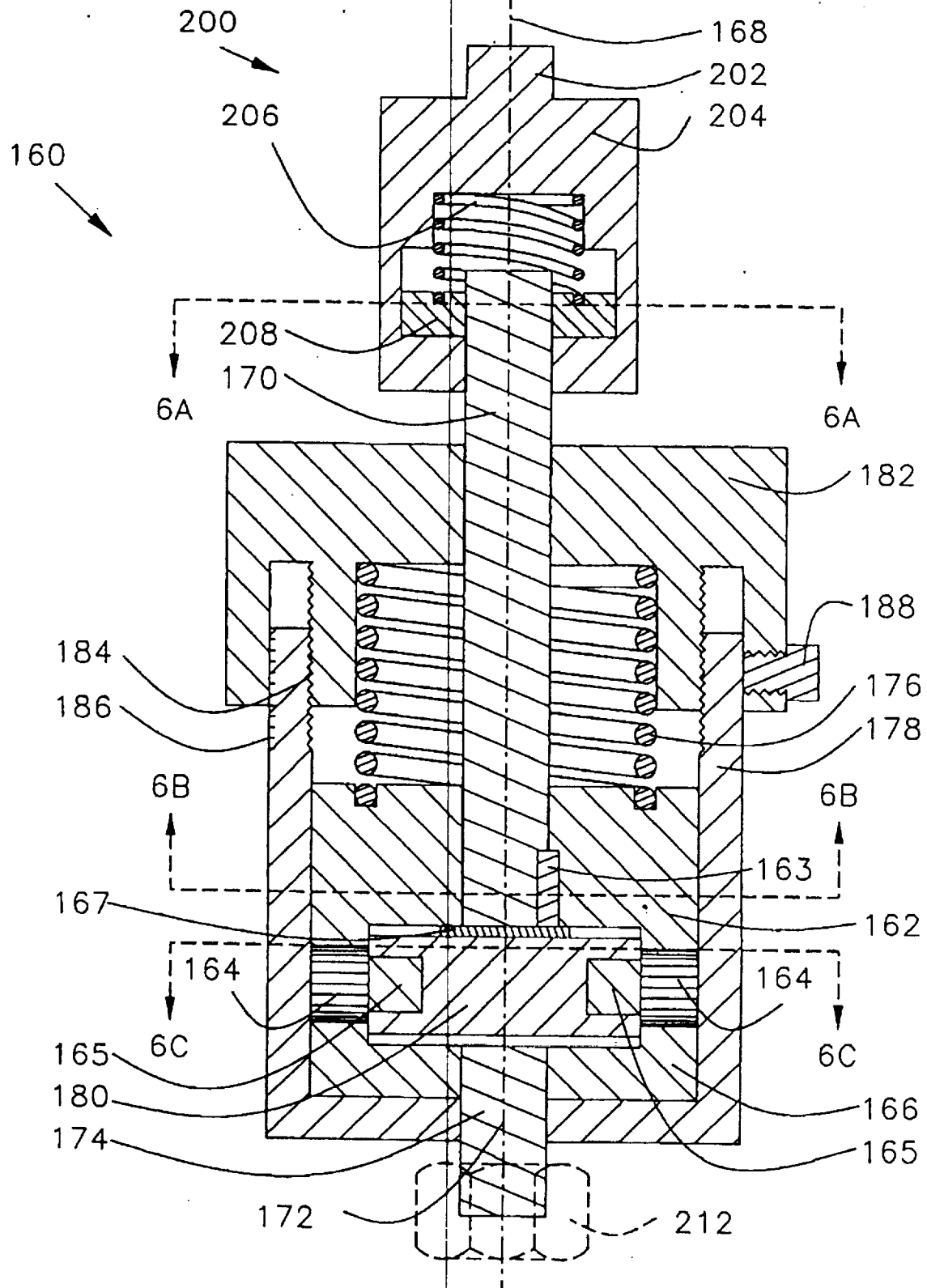


FIG. 6

FIG. 7A

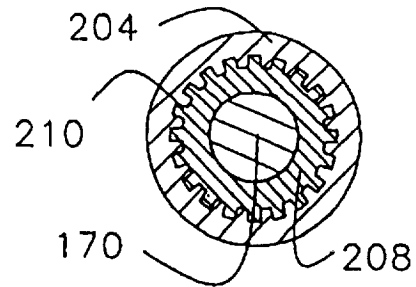
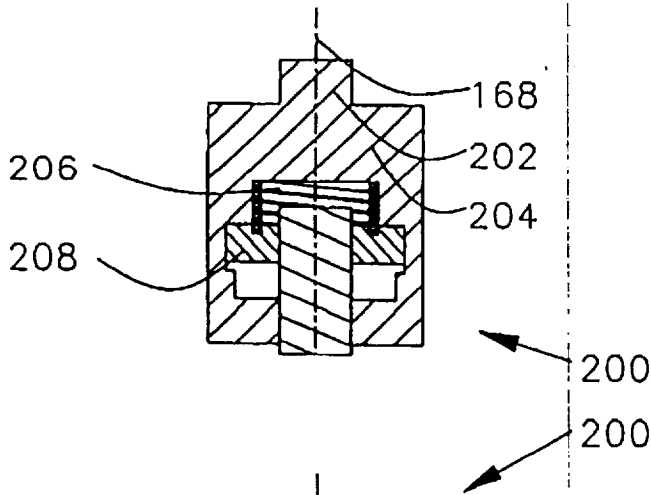


FIG. 6A

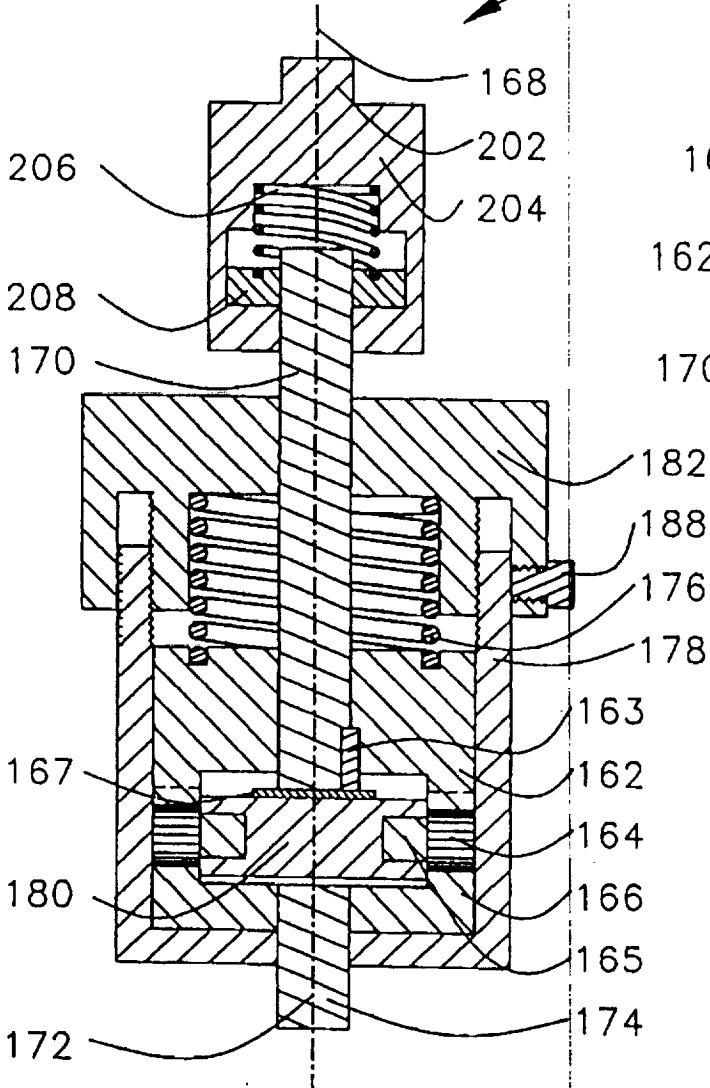


FIG. 7

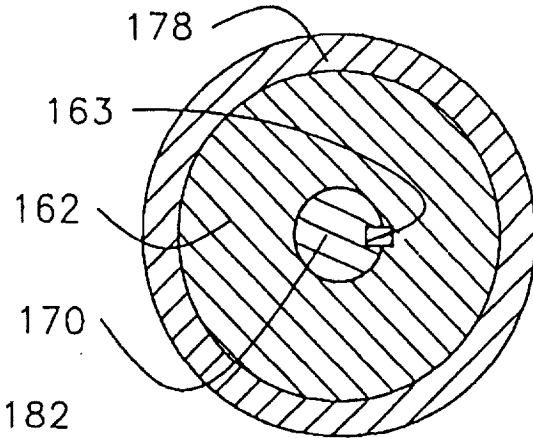


FIG. 6B

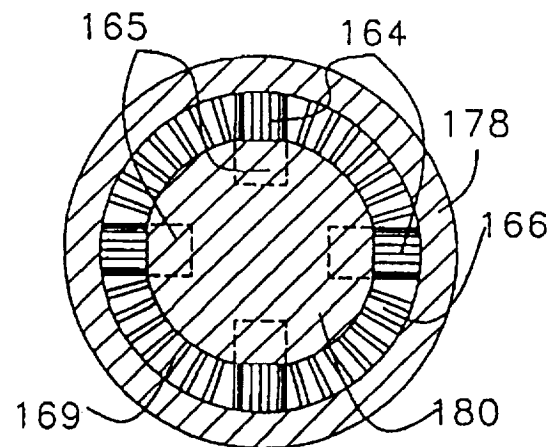
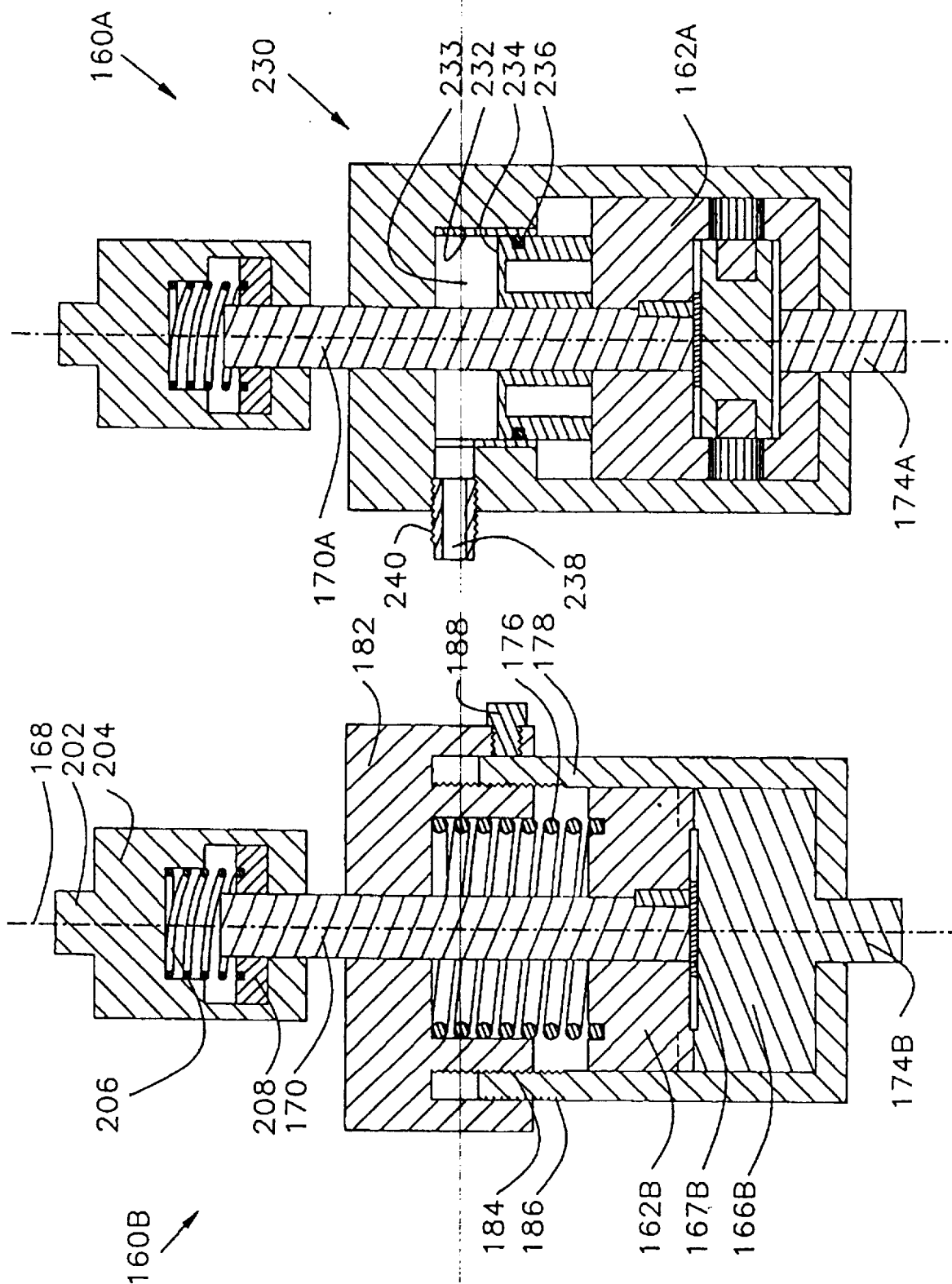


FIG. 6C



56

FIG. 8A

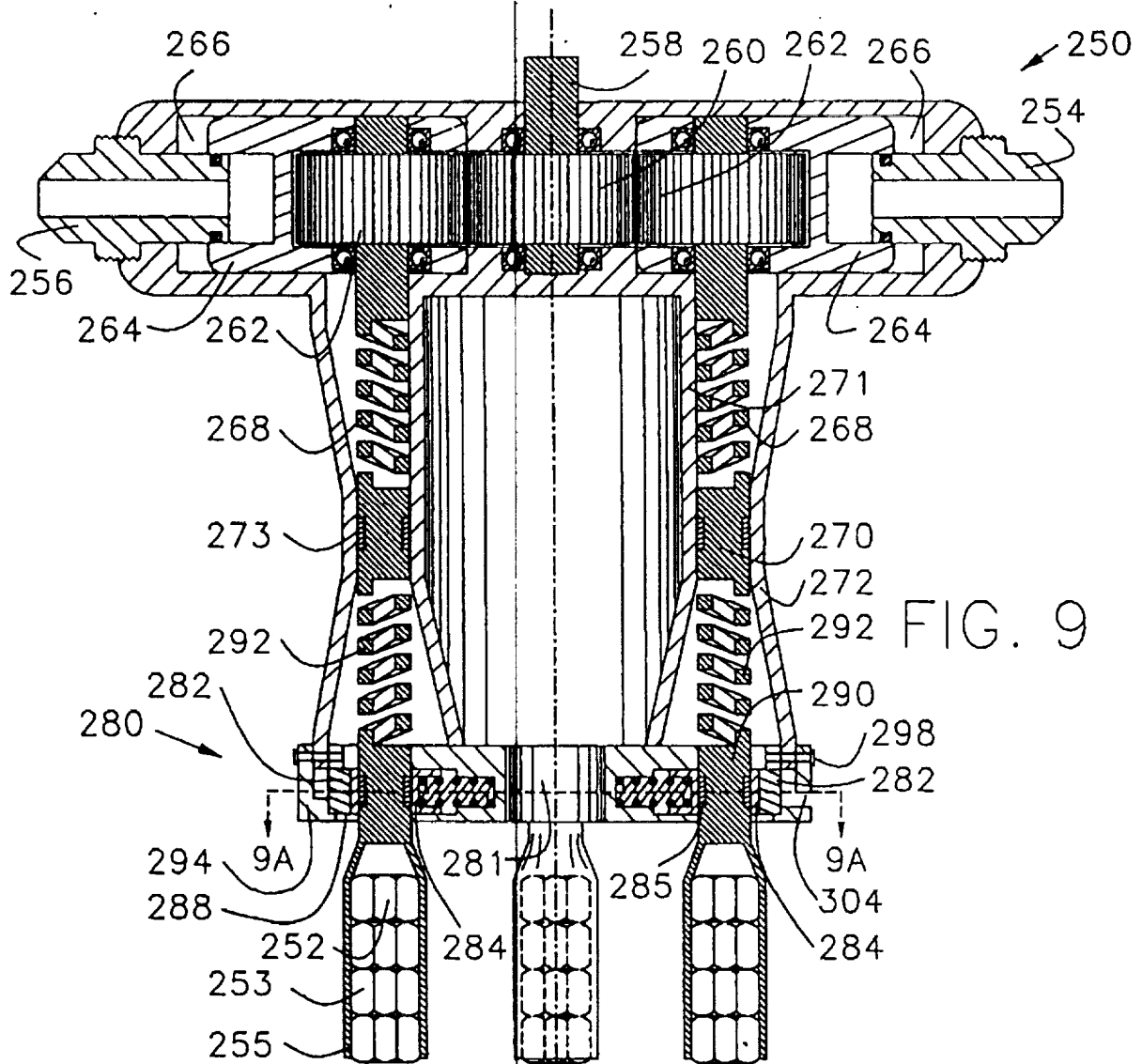


FIG. 9

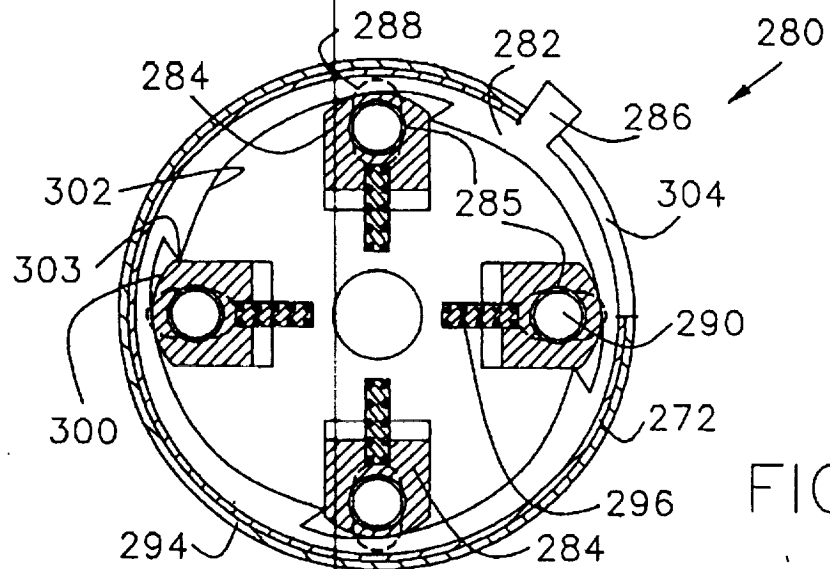


FIG. 9A