United States Patent

Schoeps

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[54] **IMPACT WRENCH WITH TORQUE CONTROL MEANS**

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[73] Assignee: **Atlas Copco Aktiebolag,** Nacka, Sweden

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April 24, 1970 Sweden.....5745

[52] **U.S. Cl.**.....**173/12,** 173/93.5

[51] **Int. Cl.**.....**B25b 21/02**

[58] **Field of Search**.....173/12, 93, 93.5; 81/52.4

[56] **References Cited**

UNITED STATES PATENTS

3,318,390	5/1967	Hoza et al.	173/12
3,643,749	2/1972	Pauley	173/12

FOREIGN PATENTS OR APPLICATIONS

460,669	9/1968	Switzerland	173/12

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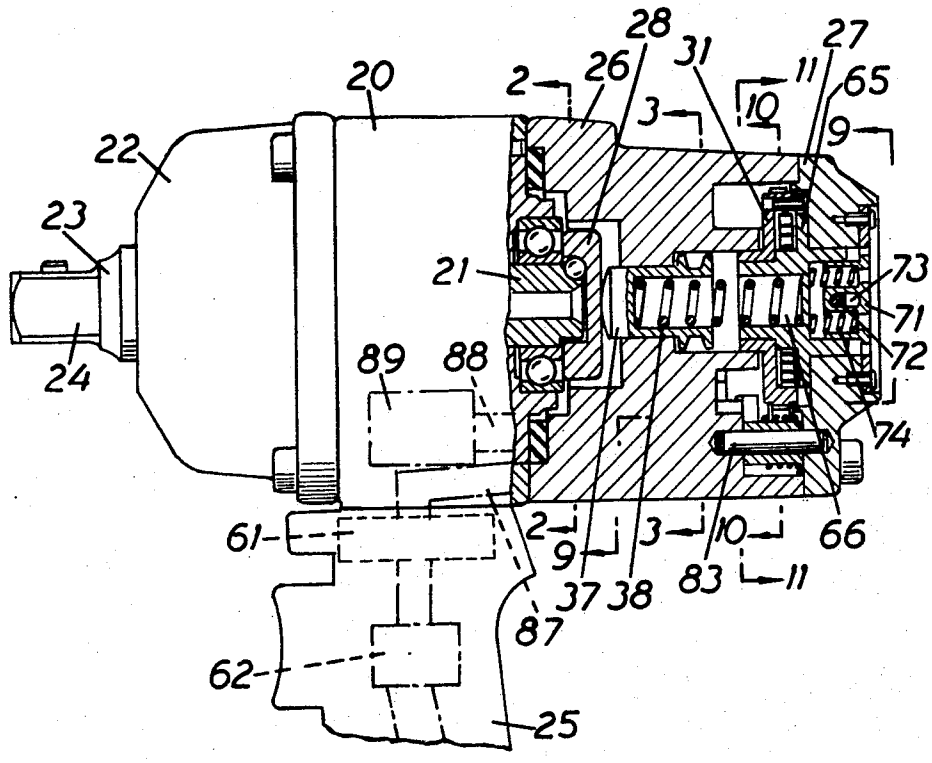
Attorney—Bauer & Goodman

[57] **ABSTRACT**

In an impact wrench provided with torque control means, a cut-off for the wrench drive motor is activated at attainment in an adding mechanism of a predetermined sum total of the lengths of displacement of a body which in response to rotational resistance at each impact is displaced against the action of a return force.

9 Claims, 13 Drawing Figures



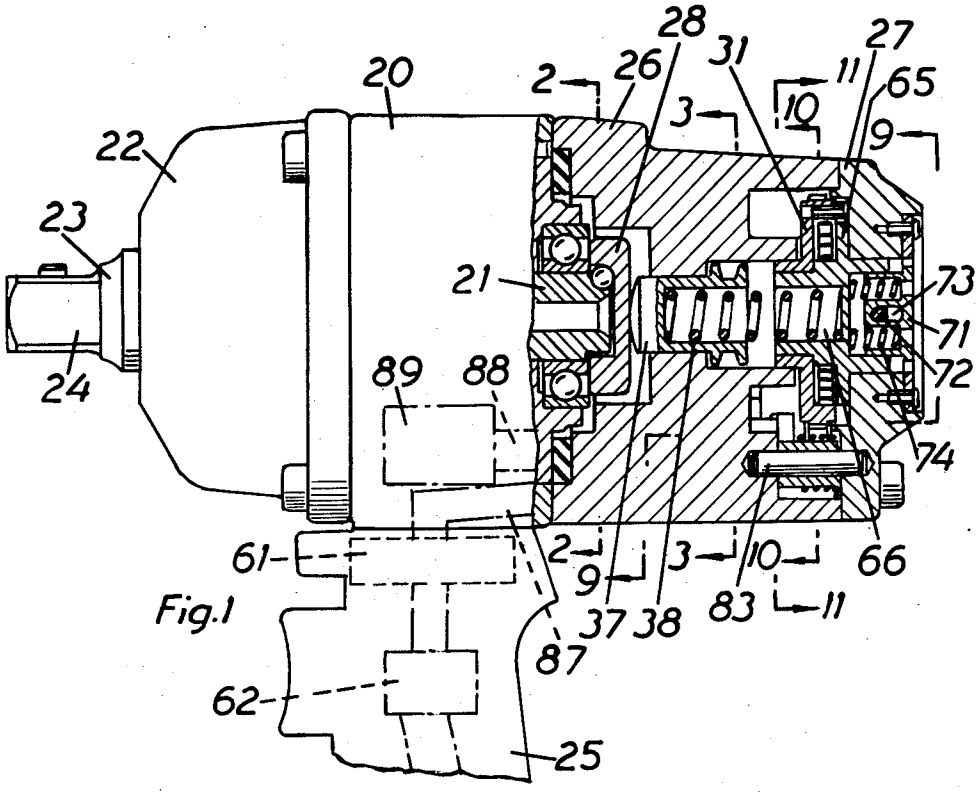


Fig. 1

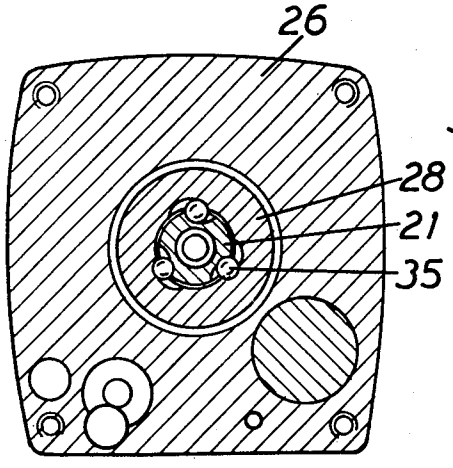


Fig. 2

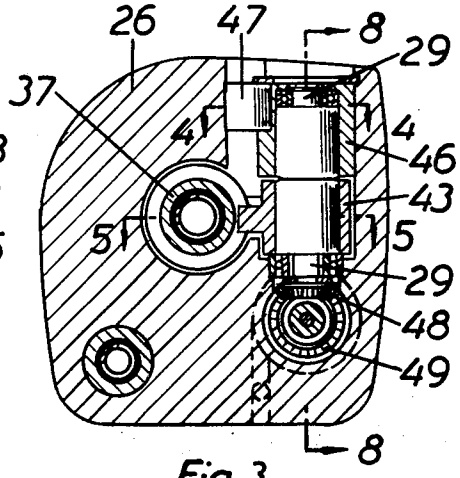


Fig. 3

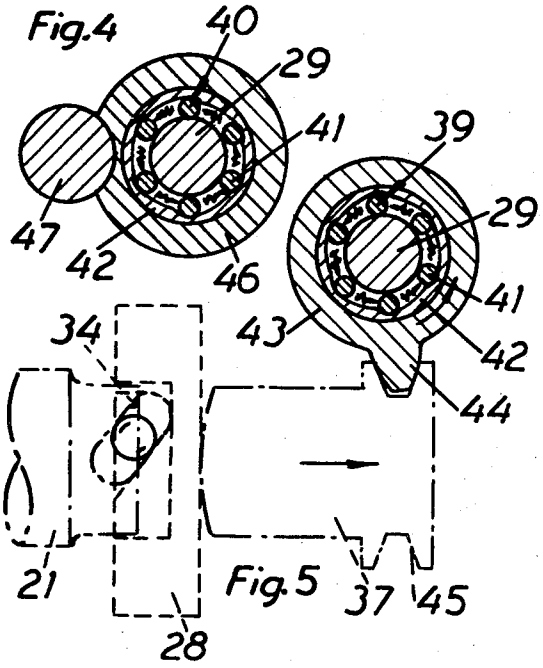


Fig.4
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Fig.5
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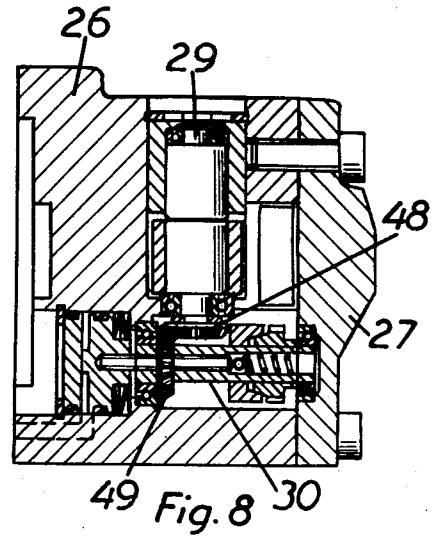
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Fig. 8
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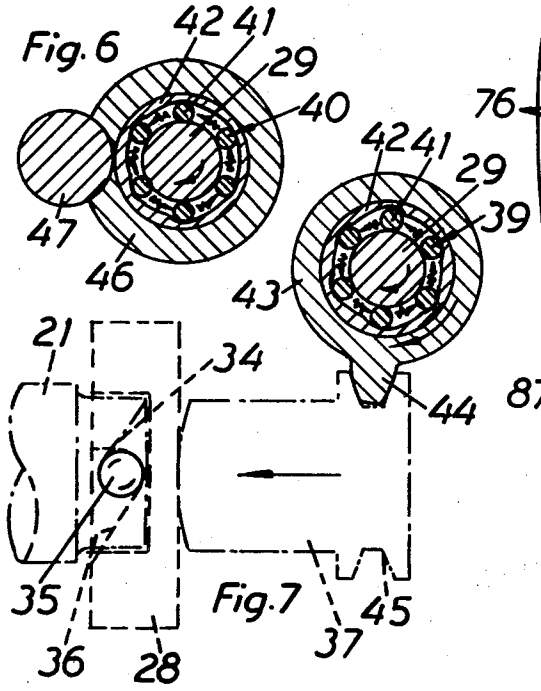
Fig. 6
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Fig.7
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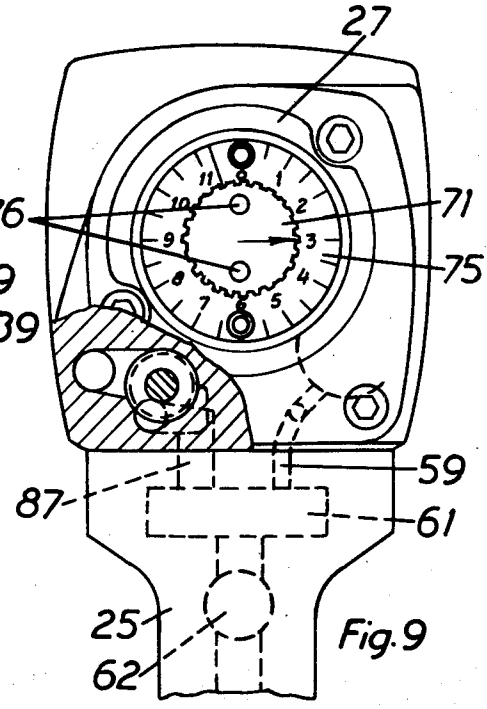
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Fig.9

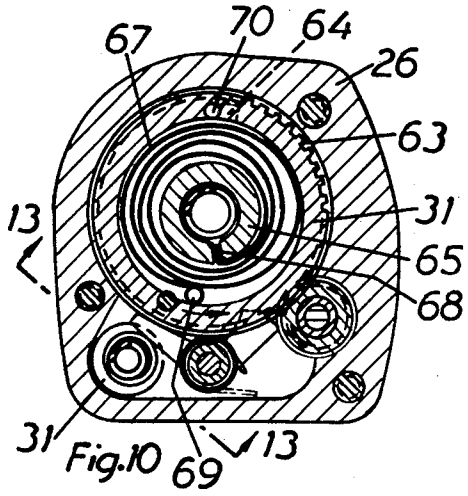

Fig.10

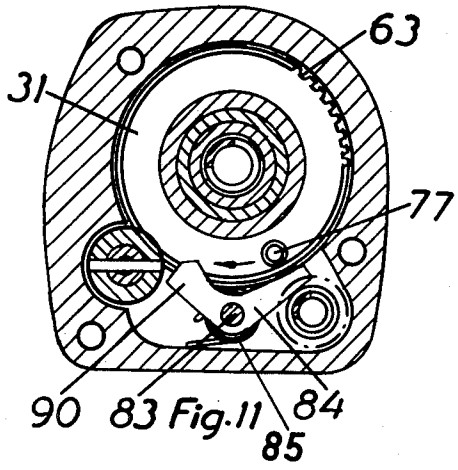

Fig.11

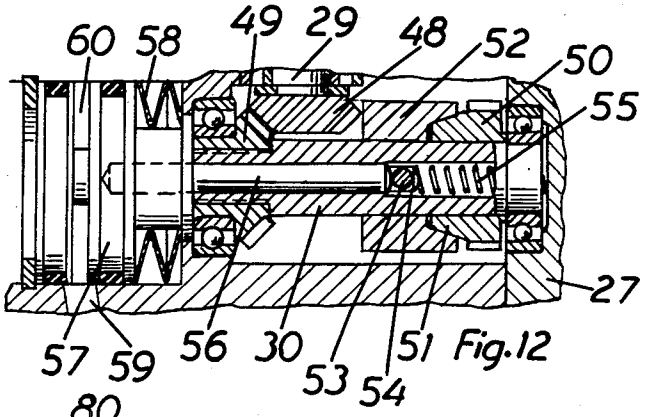

Fig.12

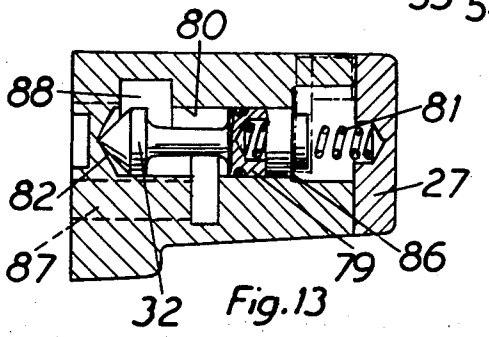

Fig.13

IMPACT WRENCH WITH TORQUE CONTROL MEANS

The invention relates to impact wrenches incorporating torque control means for controlling the tightening of threaded fasteners, for example screws and nuts, by means of such power driven tools. More particularly the invention relates to impact wrenches with torque control means of the type incorporating a drive motor, a cut-off means for the motor, a drive shaft in the motor, and a body operatively associated with the drive shaft which body in response to rotational resistance at each impact is displaced against the action of a return force. In such devices the body is subjected to inertial forces or rebound forces when impacting occurs. These forces increase with increasing rotational resistance and are conventionally employed for controlling the torque by releasing the motor cut-off when the individual successively increasing length of displacement of the body finally reaches a predetermined selective value. A difficulty in this connection is that the first impact often tends to be stronger than the following producing an excessively long first displacement which tends to activate the motor cut-off too early before the desired final torque has been reached in the fastener to be tightened. It is an object of the invention to provide an impact wrench with torque control means of the aforementioned type in which the risk for such too early a release of the motor cut-off is eliminated.

For the above and other purposes there is according to the invention provided an impact wrench with torque control means comprising a housing, a drive motor in said housing, cut-off means for said motor in said housing, a drive shaft in said motor, an anvil rotatably journalled in said housing and adapted to be connected to a workpiece, a rotary impact motor in said housing operatively interconnected between said drive shaft and said anvil for applying a succession of rotational impacts against said anvil and thereby to said workpiece, a body in said housing operatively associated with said drive shaft for being displaced at each impact relative to said drive shaft in response to rotational resistance of said anvil, means for producing a return force active against said body to cancel the displacements thereof, an adding mechanism in said housing for adding the lengths of displacement of said body, and means in said housing associated with said adding mechanism for activating said cut-off means at attainment in said adding mechanism of a predetermined sum total of said lengths of displacement.

The above and other purposes of the invention will become obvious from the following description and from the accompanying drawings in which an embodiment of the invention is illustrated by way of example. It should be understood that this embodiment is only illustrative of the invention and that various modifications thereof may be made within the scope of the claims following hereinafter.

In the drawings FIG. 1 shows a fragmentary side view drawn partly in section of a wrench embodying the invention. FIG. 2 is a cross section on the line 2—2 in FIG. 1. FIG. 3 is a cross section on the line 3—3 in FIG. 1 FIG. 4 is a fragmentary section on the line 4—4 in FIG. 3. FIG. 5 shows a fragmentary section on the line 5—5 in FIG. 3 and illustrates by broken lines an operating position of the elements covered by the section. FIGS. 6 and 7 correspond to FIGS. 4 and 5 but show an alternative position of the elements involved. FIG. 8 is a section on line 8—8 in FIG. 3. FIG. 9 is a rear end view drawn partly in section and seen on the line 9—9 in FIG. 1. FIG. 10 is a section on the line 10—10 in FIG. 1. FIG. 11 is a section on the line 11—11 in FIG. 1. FIG. 12 shows an enlarged partial view of FIG. 8. FIG. 13 is a section on the line 13—13 in FIG. 10. rotor

As an example for the description has been chosen an application of the invention to a pneumatically driven impact wrench but the invention may also be applied with advantage in connection with electrically or hydraulically driven impact wrenches. The pneumatic impact wrench in FIG. 1 includes a housing 20 in which is journalled the rotor 21 of a motor preferably made as a vane motor. Within a front piece 22 the rotor 21 drives a conventional rotary impact motor, not shown, the anvil 23 of which projects from the front portion with a square end 24 intended for taking up a socket wrench. The socket wrench, not shown, is brought into engagement with a workpiece such as a nut to be tightened and the motor housing 20 is directed and held by way of a handle 25.

The motor housing 20 carries a backhead 26 having a cover 27, these parts enclosing the torque control means. The main parts of the latter are formed by an inertial or flywheel body 28, FIGS. 1,2,5, and 7, an adding shaft 29, FIGS. 3–8, a clutch shaft 30, FIGS. 8 and 12, a setting drum 31, 65. FIGS. 10, 11, and a motor cut-off means 32, FIG. 13.

The inertial or flywheel body 28 encloses in cap-like manner the rear end of the rotor 21 which is provided with cam means in the form of oblique grooves 34 into which are inserted cam balls 35. The outer halves of the cam balls 35 project into internal oblique grooves in the inner portion of the inertial body 28 radially around the rear end of the rotor 21. A piston 37 is axially slidably supported in the backhead 26 and abuts under the load of a helical spring 38 in axial direction against the outer side of the inertial body 28, producing a return force active to but the inertial body 28 against the end face of the rotor 21 whereby the cam balls 35 are caused to take inner positions in the respective oblique grooves 34, 36, FIG. 7. Normally the inertial body rotates conjointly with the rotor 21 occupying the position illustrated in FIG. 7. However, as soon as the impact motor disposed within the front piece 22 delivers an impact against the anvil 23, the rotor 21 is momentarily retarded by the rotational resistance while the inertial body 28 in response to such resistance persists in its rotational movement and forces the cam balls 35 to roll outwardly in the oblique grooves 34 and 36, respectively, to the position illustrated in FIG. 5. As a result thereof the inertial body 28 is displaced axially relative to the rotor 21 a distance increasing with the increased rotational resistance at impact and under compression of the helical spring 38 of the piston 37 and axial retraction of the latter. During the acceleration phase of the impact motor prior to the next impact thereof, the piston 37 is returned by the spring 38 towards the rotor 21 and the inertial body 28 and the cam balls 35 are simultaneously returned to the position in FIG. 7.

The adding shaft 29 is journalled in the backhead 26 on suitable ball bearings in a plane perpendicular to the rotational axis of the rotor 21. The adding shaft 29 is

associated with means for transforming the axial movements of the inertial body **28** to unidirectional angular movement of the adding shaft **29**, which means are provided by a pair of free-wheel-mechanisms **39,40** each comprising rollers or needle bodies **41** which at rotation in one direction are pressed by an outer ring **42** provided with cam surfaces into non-rotary engagement with the adding shaft and at rotation in the opposite direction through the intermediary of a needle holder provided with resilient elements allow free rotation of the outer ring **42** relative to the adding shaft **29**. The outer ring **42** of the free-wheel-mechanism **39** is carried non-rotatably by a ring shaped hub or driving member **43** which by means of a tooth **44** is in engagement with a ring groove **45** on the piston **37**. The other free-wheel-mechanism **40** is received at the outer ring **42** thereof non-rotatably in a ring shaped hub **46** which by a locking plug **47** is non-rotatably locked in the backhead **26**. The locking direction of the two free-wheel-mechanisms **39, 40** is chosen such, FIGS. **4,5** and **6,7**, respectively, that during movement of the inertial body **28** axially away from the rotor **21**, FIG. **5**, the ring hub **43** will rotate freely in counter-clockwise direction with respect to the adding shaft which by the free-wheel-mechanism **40** is locked against rotation in counter-clockwise direction. At return of the inertial body **28** under the action of the return force of spring **38** and the piston **37**, the groove **45** entrains the tooth **44** and the ring hub **43** in clockwise direction. The free-wheel-mechanism **39** now locks against the adding shaft **29** and entrains the latter in its movement, FIG. **7**, simultaneously with the free-wheel-mechanism **40** allowing such movement by free-wheeling relative to the adding shaft **29**. As a result of this function the adding shaft **29** at each return movement of the piston **37** will be turned uni-directionally an angle increasing step by step.

The adding shaft **29** is by means of a bevel gear **48** in engagement with a bevel gear **49** on a clutch shaft **30** which is supported on suitable ball bearings in the backhead **26** and the cover **27** thereof. On the clutch shaft **30** is freely rotatably journalled a gear **50** the hub of which is formed as one half **51** of a conical friction clutch. The cooperating other half **52** is carried slidably in axial direction on the clutch shaft **30** and is non-rotatably fixed thereto by a cross pin **53** traversing the clutch half **52** and the shaft **30**, and being slidable axially in an axial groove **54** formed in the clutch shaft **30**. Furthermore the clutch shaft is made hollow and carries a helical spring **55** bearing against the cross pin **53** and the cover **27** and striving to move the cross pin **53** and thus the clutch half **52** to such a position in the groove **54** as to release the clutch half **52** from the clutch half **51** of the gear **50**. In the clutch shaft **30** is furthermore slidably supported an axial pin **56** for actuating the cross pin **53** which axial pin for purposes of engaging the clutch halves **52, 51** is actuatable by actuating means in response to actuation of the main control means of the impact wrench for the drive motor thereof. In electrically driven wrenches such actuation will be attained by application of a suitable solenoid. In pressure fluid actuated wrenches such as in the present example a servo piston motor including a servo piston **57** affixed to the axial pin **56** is applied , the servo piston **57** striving under the force bias of annular springs **58** to move the axial pin **56** to inactivated position in which the cross pin **53** is unaffected. When pressure fluid is supplied via a passage **59** to the cylinder **60** of the servo piston **57**, the clutch halves **52, 51** are caused to engage by the axial pin **56** applying a thrust against the cross pin **53**. The passage **59** is pressurized or can be exhausted, respectively, by means of the reversing valve **61** of the wrench , diagrammatically illustrated in FIGS. **1,9**, in the two alternative positions of said valve **61**, the reversing valve **61** being arranged in usual manner downstream of the throttle valve **62**. The gear **50** driven via the clutch halves **52, 51** by the coupling shaft **30** is in engagement with a peripheral gear rim **63** on the setting drum **31**, FIG. **10**.

The setting drum **31, 65** includes a drum portion **31** which is rotatably journalled on a base portion **65**, FIG. **1**, in its turn rotatably supported in a bore in the cover **27** coaxially with the piston **37**. A bore **66** in the base portion **65** facing the piston **37** receives the helical spring **38** of the piston **37**. In an annular cavity between the base portion **65** and the drum portion **31** a clockwork spring **67** is tensioned between a groove **68** in the base portion **65** and a pin **69** in the drum portion **31**, FIG. **10**. The clockwork spring **67** is pre-stressed striving to but a pin **70** on the drum portion **31** against a tooth **64**, FIG. **10**, arranged on the base portion **65** on the rim of a flange thereon. The base portion **65** is adjustably affixed to the cover **27** in a manner to prevent rotation between these parts . To this end a cogwheel-shaped locking plate **71** is axially slidably but non-rotatably mounted in the base portion **65** on a cross pin **72** engaging an axial groove **73** in the locking plate **71** and is depressable against the action of helical springs **74** which are inserted between the base portion **65** and the interior of the locking plate **71** to urge the latter outwardly into locking position, FIG. **1**. In the locking position the locking plate **71** mates with a toothed ring **75** which by means of screws is affixed to the cover **27**, FIG. **9**. When the locking plate is depressed axially towards the base portion **65**, it will be released from the toothed ring **75** so that the locking plate **71** by a suitable tool insertable into axial bores **76** in the plate **71** may be turned together with the base portion **65** and the drum portion **31** to the desired angular position in the cover **27**, whereupon the plate **71** is permitted to spring out into locking engagement with the toothed ring **75**, the base portion **65** thus becoming locked in the desired angular position. The drum portion **31** carries an abutment embodied by a release pin **77** for activating the motor cut-off means **32**.

In the exemplary wrench intended to be driven by compressed air, the motor cut-off **32** is formed by a cut-off valve provided with a closing head which is slidable in a bore **80** in the backhead **26** through the intermediary of a base shaped as a piston **79** of equal size with the head **32**. A helical spring **81** is inserted between the cover **27** and the piston **79** and strives to move the cut-off **32** in a direction away from the cover **27** to but against the bottom **82** of the bore **80** in which position both faces of the closing head of the cut-off **32** are pressurized. A bell crank lever **84** is pivotally journalled on a pivot **83** adjacent the piston **79** between the cover **27** and the backhead **26** and is actuated by a turning spring **85** to latch in the bottom butting position of the cut-off **32** under the action of the turning spring

85 behind an annular shoulder 86 on the piston 79, thus locking the cut-off 32 in inactivated position between the lever 84 and the shoulder 82, FIG. 13. The bore 80 is supplied with pressure fluid from the reversing valve 61 via a passage 87 and the pressure fluid flows from the bore 80 past the cut-off 32 via a passage 88 to the motor inlet 89 in the motor housing 20, FIG. 1. By pressure in the bore 80 acting against the piston 79, the cut-off 32 is biased in the direction of the cover 27. The release pin 77 of the setting drum 31 cooperates with the lever 84 and is capable of applying a thrust against the arm 90 thereof remote from the shoulder 86 whereby the lever 84 is turned against the action of the turning spring 85 away from the ring shoulder 86. At such instant the pressure in the bore 80 will displace the piston 79 towards the cover 27 so that the head of the cut-off 32 engages the central part of the bore 80 thus closing off the supply of pressure fluid from the passage 87 to the passage 88 whereby the drive motor of the wrench is stopped.

In electrically driven wrenches the cut-off preferably will be embodied by a conventional solenoid-actuated switch for the main current with the solenoid striving to displace the switch member past the lever 84 when the latter is released by the release pin 77 acting against the arm 90.

By axial depression and turning of the locking plate 71 the base portion 65 of the setting drum 31, 65 is first adjusted to the desired angular starting position in the backhead 26. This means that one selects by way of adjustment of the base portion 65 the desired angular starting position of the release pin 77, FIG. 11, and of the tooth 64 of the base portion 65, against which the pin 70 is butted by the clockwork spring 67 in the starting position. Thereupon the locking plate 21 is allowed to spring out into locking engagement with the toothed ring 75 so that the position of the base portion 65 is fixed. The wrench is connected to a suitable pressure fluid supply, for example to a source of compressed air. Upon the socket wrench having been placed on a nut to be tightened, the throttle valve 62 of the handle 25 is depressed so that the reversing valve 61, set to cause clockwise rotation of the motor 21 when the tool is viewed from the rear in FIG. 1, is pressurized as is the case with the passages 87,59.

From the passage 59 the pressure fluid enters the cylinder 60 actuating the servo piston 57 to bring the clutch halves 52, 51 of the clutch shaft 30 into frictional engagement. The pressure fluid passes through the passage 87 and the bore 80 past the motor cut-off 32 and thence to the passage 88 and to the motor inlet 89 to rotate the rotor 21 in clockwise direction. The screw rotates until it has been driven down against the workpiece whereupon the anvil 23 stops and the rotary impact motor starts impacting. Prior to each impact the inertial or flywheel body 28 is in the state of rapid rotation and when the anvil 29 at impact causes the rotor 21 to stop, the inertial body 28 will continue its rotation performing, by reason of the oblique grooves 34, 36 and the cam balls 35, an axial rearwardly directed stroke for each impact delivered. Thus the piston 37 for each impact delivered is firstly displaced freely rearwardly, FIG. 5, and thereafter returned in forward direction under the bias of the spring 38 entraining the adding shaft 29. The uni-directional angular movements of the adding shaft 29 are transmitted directly to the clutch shaft 30 and via the clutch halves 52, 51 to the gear 50. As a result of each reciprocation of the inertial body 28 the adding shaft 29 sums up the length of displacement by uni-directional rotation and turning of the gear 50, the rotation of the latter being transmitted via the gear rim 63 to the drum portion 31. The turning motion forces the pin 70 of the setting drum 31 to leave the tooth 64 and as the increment lengths of the displacements generate turning movement of the adding shaft 29 and the setting drum 31, the pin 77 will be successively displaced towards the arm 90. After a sufficient number of impacts the release pin 77 hits the arm 90 , the lever 84 is unlatched and the motor cut-off is released so that it can fall into the middle portion of the bore 80 thus closing the air supply to the motor inlet 89. Upon cut-off the operator has to release the throttle valve 62, the latter being preferably made such as to exhaust in the released position thereof the passages 87 and 59 and the reversing valve 61 via the throttle valve 62. When the pressure is relieved in the passage 87, the closing bias acting against the piston 79 disappears and the spring 81 of the cut-off 32 is thus in the position to return the cut-off to open position supported against the bottom of the hole 82. Relief of the passage 59 relieves the servo piston 57 and the annular springs 58 thereof displace the axial pin 56 away from the cross pin 53. This permits the helical spring 55 to move the cross pin 53 and the clutch half 52 away from the clutch half 51, the gear 50 thus becoming released for free rotation on the clutch shaft 30. This allows the tensioned clockwork spring 67 to return the setting drum back to the starting position, the pin 70 thus being returned against the tooth 64 and the release pin 77 to its starting position.

When a series of screws are to be tightened under similar conditions, substantially the same tightening torque will be reached in the screws provided they have received the same number of impacts from the wrench up to cut-off of the latter, such condition in the above described device meaning that the angular distance between the starting position of the release pin 77 and the arm 90 during each tightening operation must remain unchanged. During pressure variations in the pressure fluid line a variable number of impacts will be generated up to cut-off. Within pressure variation limits occuring in practice, however, the final torque will tend to be constant since a lower pressure gives an increased number of somewhat smaller angular increments to be totaled while a higher pressure gives fewer but somewhat larger angular increments. If it is desired to change the final torque in the screw, another angular starting position will be chosen for the release pin 77 by adjustment of the locking plate 71 to a different angular position relative to the toothed ring 75. For more easy adjustment the toothed ring 75 is provided with suitable marking with numerals.

When the wrench is set for reverse rotation, the passage 59 is exhausted directly via the reversing valve 61 so that the clutch halves 52, 51 do not engage. Simultaneously, during impacting in counter-clockwise direction the inertial body 28 is locked by the cam balls 35 against continued rotation at the moment of impact and thus the inertial body 28 will not perform any axial displacements and leaves the adding shaft 29 unaffected.

As obvious from the foregoing, the first impact in the exemplary embodiment only represents a fraction of the total turning movement of the adding shaft up to activation of the cut-off, and an excessively long first displacement of the inertial or flywheel body, therefore, will be unable to have any adverse effects on the resulting final torque.

What I claim is:

1. An impact wrench with torque control means comprising a housing, a drive motor in said housing, cut-off means for said motor in said housing, a drive shaft in said motor, an anvil rotatably journalled in said housing and adapted to be connected to a workpiece, a rotary impact motor in said housing operatively interconnected between said drive shaft and said anvil for applying a succession of rotational impacts against said anvil and thereby to said workpiece, a body in said housing operatively associated with said drive shaft for being displaced at each impact relative to said drive shaft in response to rotational resistance of said anvil, means for producing a return force active against said body to cancel the displacements thereof, an adding mechanism in said housing for adding the lengths of displacement of said body, and means in said housing associated with said adding mechanism for activating said cut-off means at attainment in said adding mechanism of a predetermined sum total of said lengths of displacement.

2. An impact wrench according to claim 1 in which said body is an inertial body axially displaceably mounted on said drive shaft, axial cam means between said drive shaft and said body for rotating the latter conjointly with said drive shaft and for displacing said body at retardation of said shaft by way of inertia axially of said shaft, said adding mechanism including an adding shaft, a driving member between said body and said adding shaft, and means for transforming the axial displacements of said body and driving member into uni-directional turning movement of said adding shaft.

3. An impact wrench according to claim 2 in which said transforming means include a pair of free-wheel-mechanisms, one coupled between said driving member and said adding shaft and the other coupled between said adding shaft and said housing.

4. An impact wrench according to claim 2 in which said activating means comprises a setting drum rotatably mounted in said housing, a base portion on said setting drum, means for adjusting and affixing said base portion angularly with respect to said housing, a drum portion on said setting drum connectable to said adding shaft for being turned thereby relative to said base portion, spring means interposed between said base and drum portions for counteracting such turning movement of said drum portion, and a member on said drum portion for activating said cut-off means upon appropriate turning movement of said drum portion.

5. An impact wrench according to claim 4 in which a clutch is provided for connecting said drum portion to said adding shaft, and means for disengaging said clutch at activation of said cut-off means to permit resetting of said drum portion relative to said base portion under the action of said spring means.

6. An impact wrench according to claim 5 in which there are provided manually actuatable starting means in said housing for said motor, and means in said housing for engaging said clutch in response to actuation of said starting means.

7. An impact wrench according to claim 4 in which said motor is actuated pneumatically, a passage in said housing for supplying compressed air to said motor, said cut-off means being a normally open cut-off valve in said passage, means for biasing said valve to the cut-off portion thereof, a locking arm pivotally mounted in said housing and cooperating with said cut-off valve to releasably arrest said cut-off valve in the open position thereof, and said member on said drum portion cooperating with said locking arm for releasing movement of said cut-off valve to the cut-off position thereof.

8. An impact wrench according to claim 6 in which said motor is actuated pneumatically, a passage in said housing for supplying compressed air to said motor, a throttle valve in said passage providing said starting means, and said clutch engaging means being a pneumatically actuated servo piston means controlled by said throttle valve and operatively connected to said clutch.

9. An impact wrench according to claim 2 in which said turning movement of said adding shaft is generated by said return force.

* * * * *