March 6, 1951 S. B. MAURER 2,543,979

IMPACT WRENCH TORQUE CONTROL

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Fig. 1.

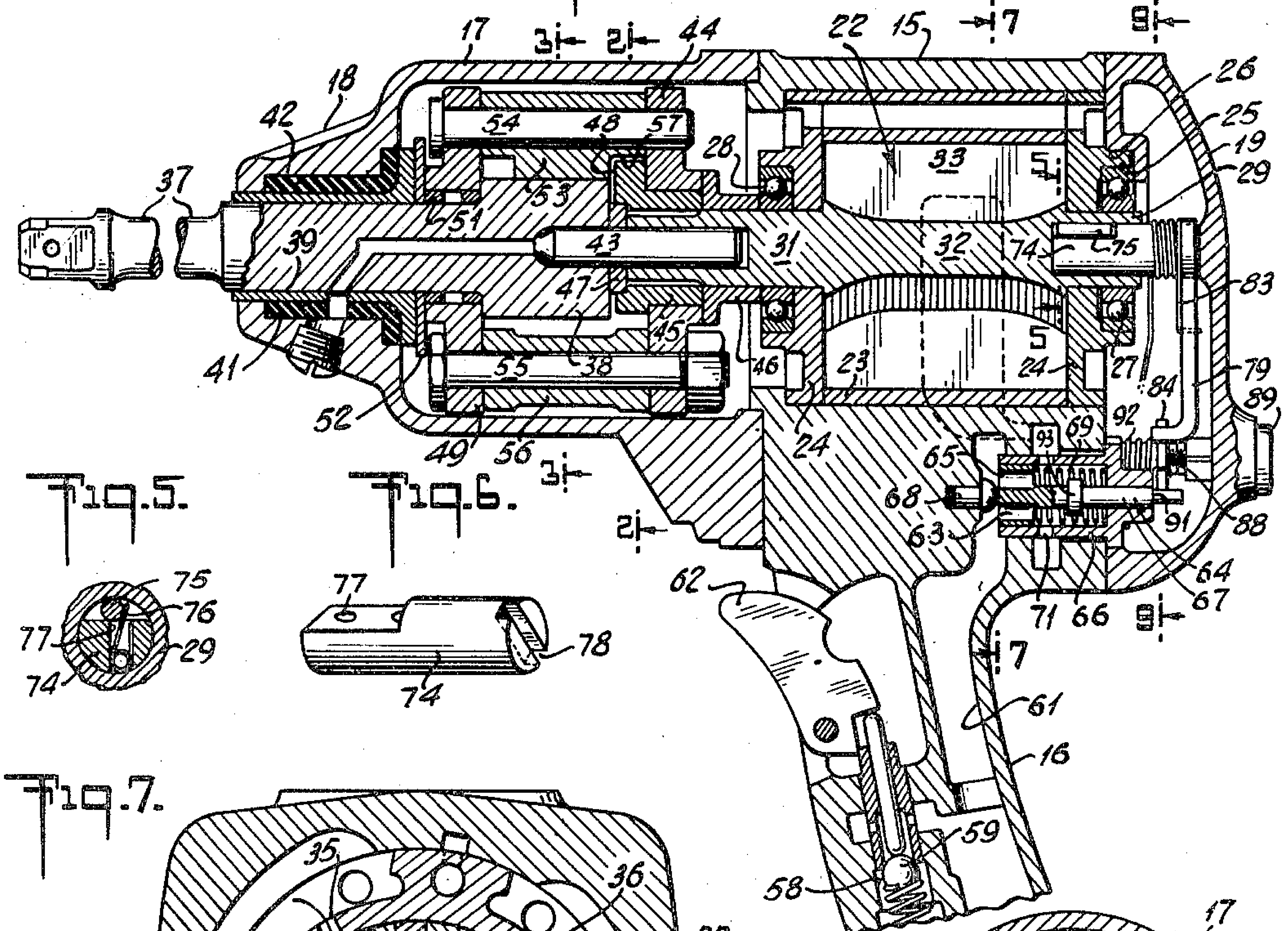


Fig. 5.

Fig. 6.

Fig. 7.

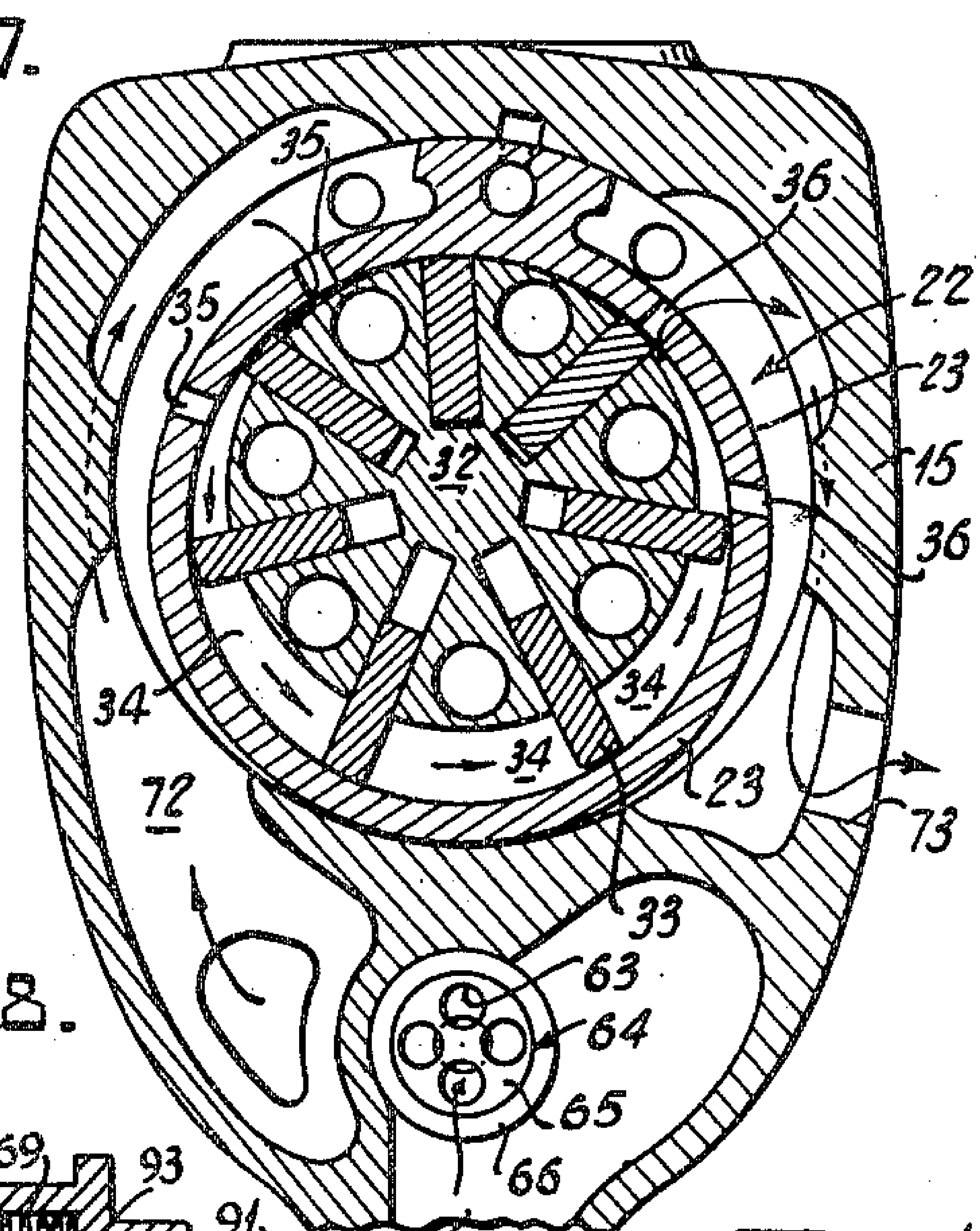


Fig. 2.

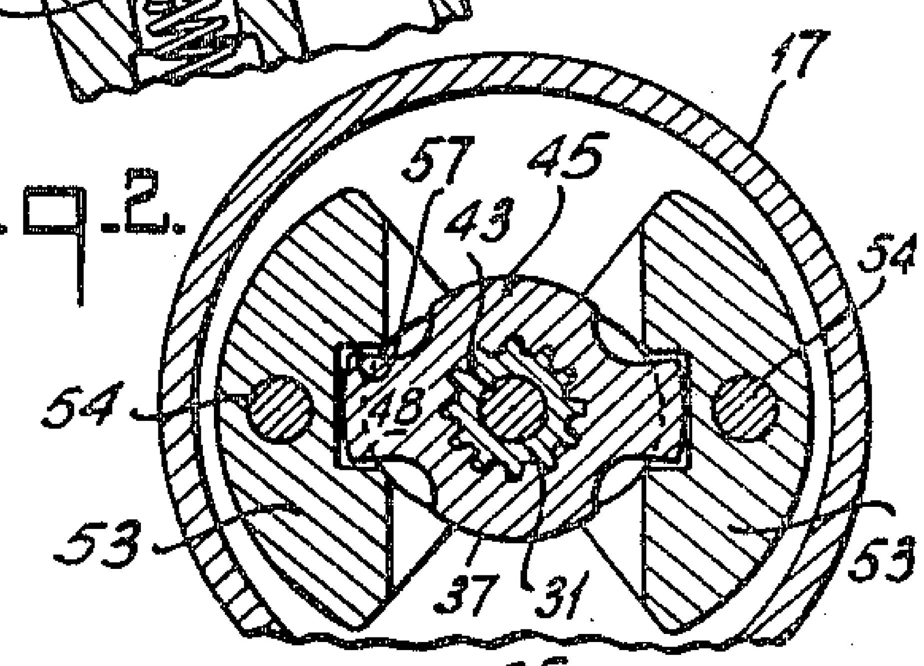


Fig. 3.

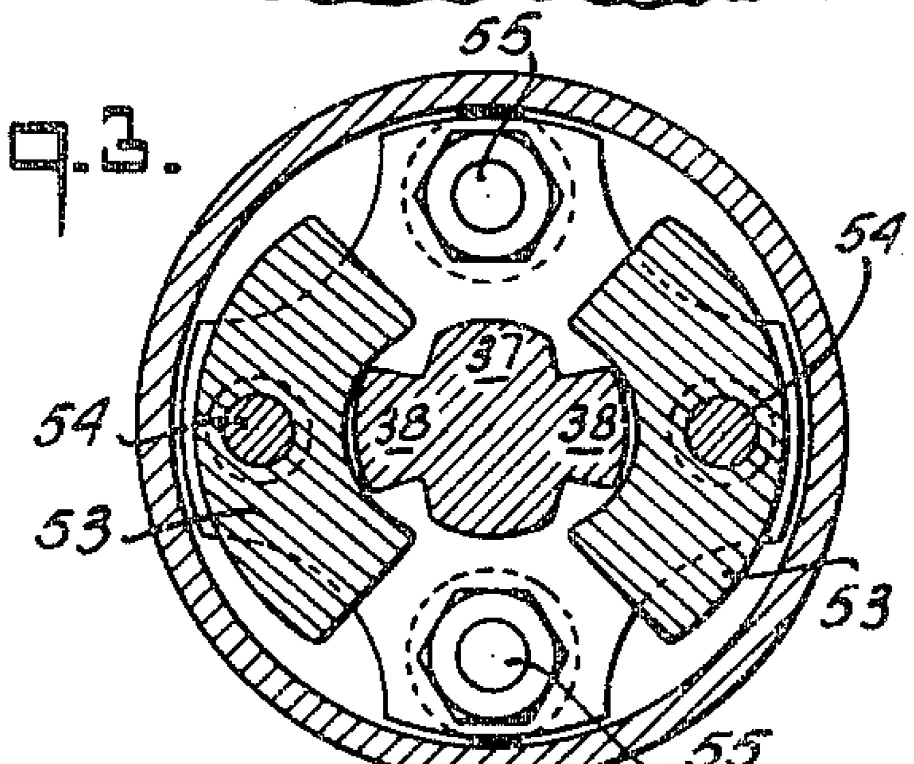


Fig. 8.

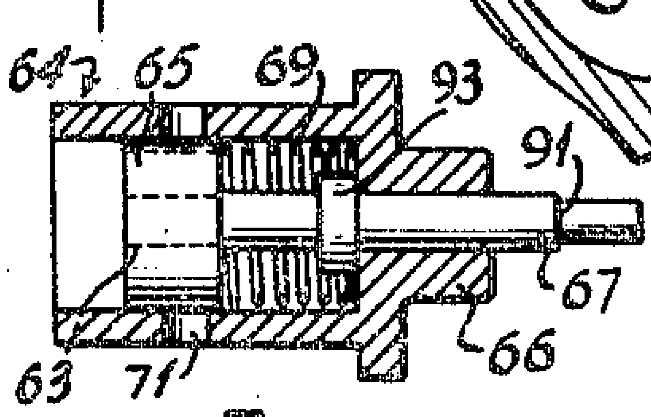
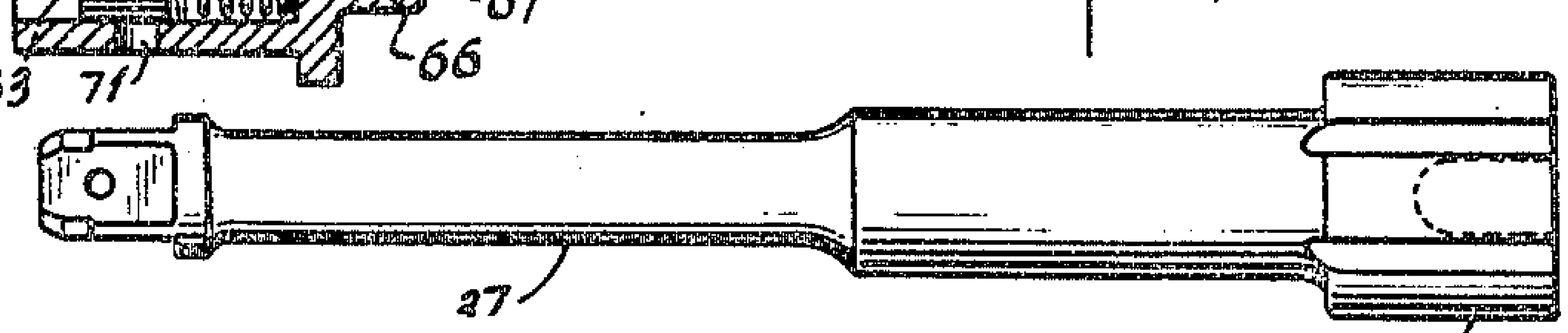


Fig. 4.



INVENTOR
SPENCER B. MAURER.

BY
Raymond G. Mullee
ATTORNEY

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Fig. 9.

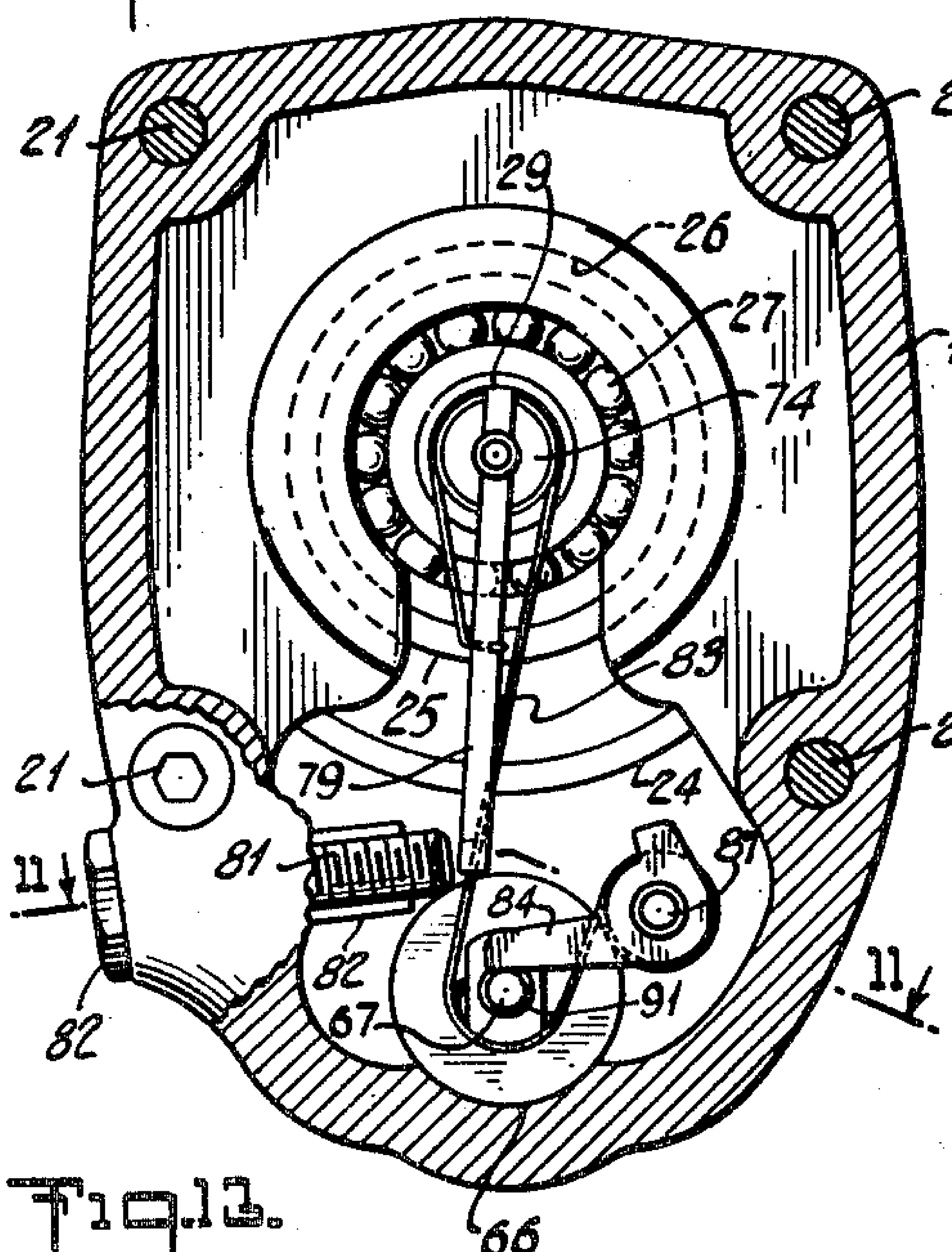


Fig. 11.

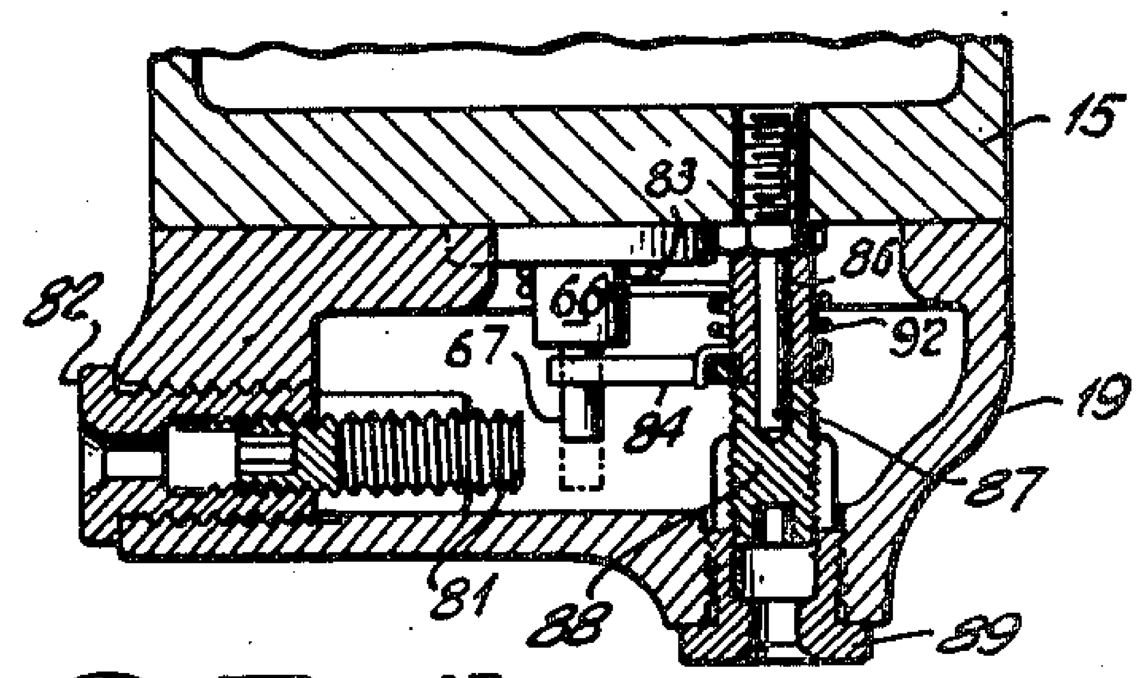


Fig. 10.

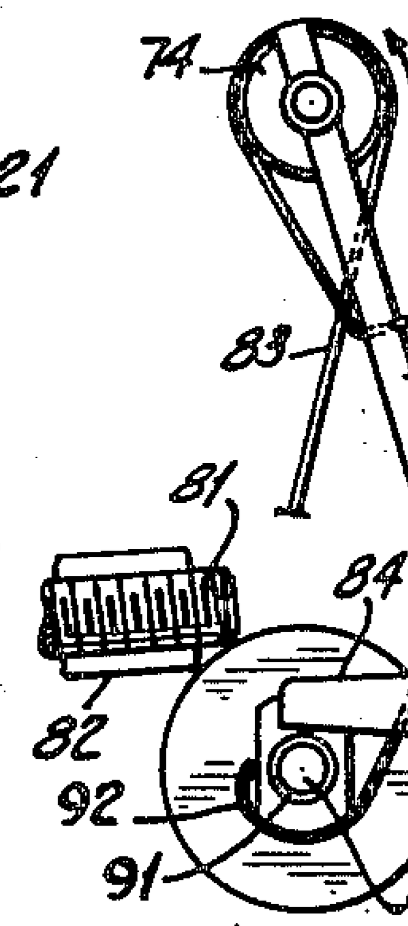


Fig. 12

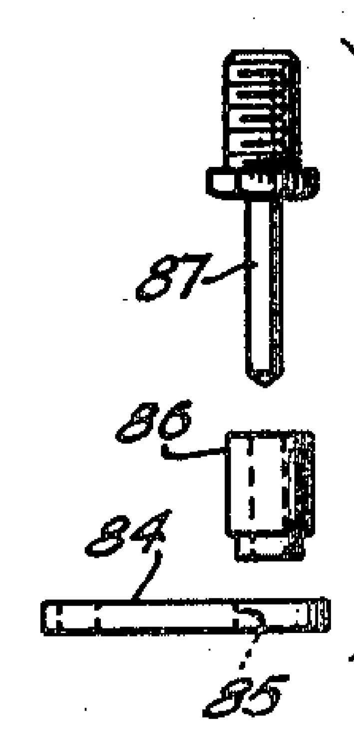
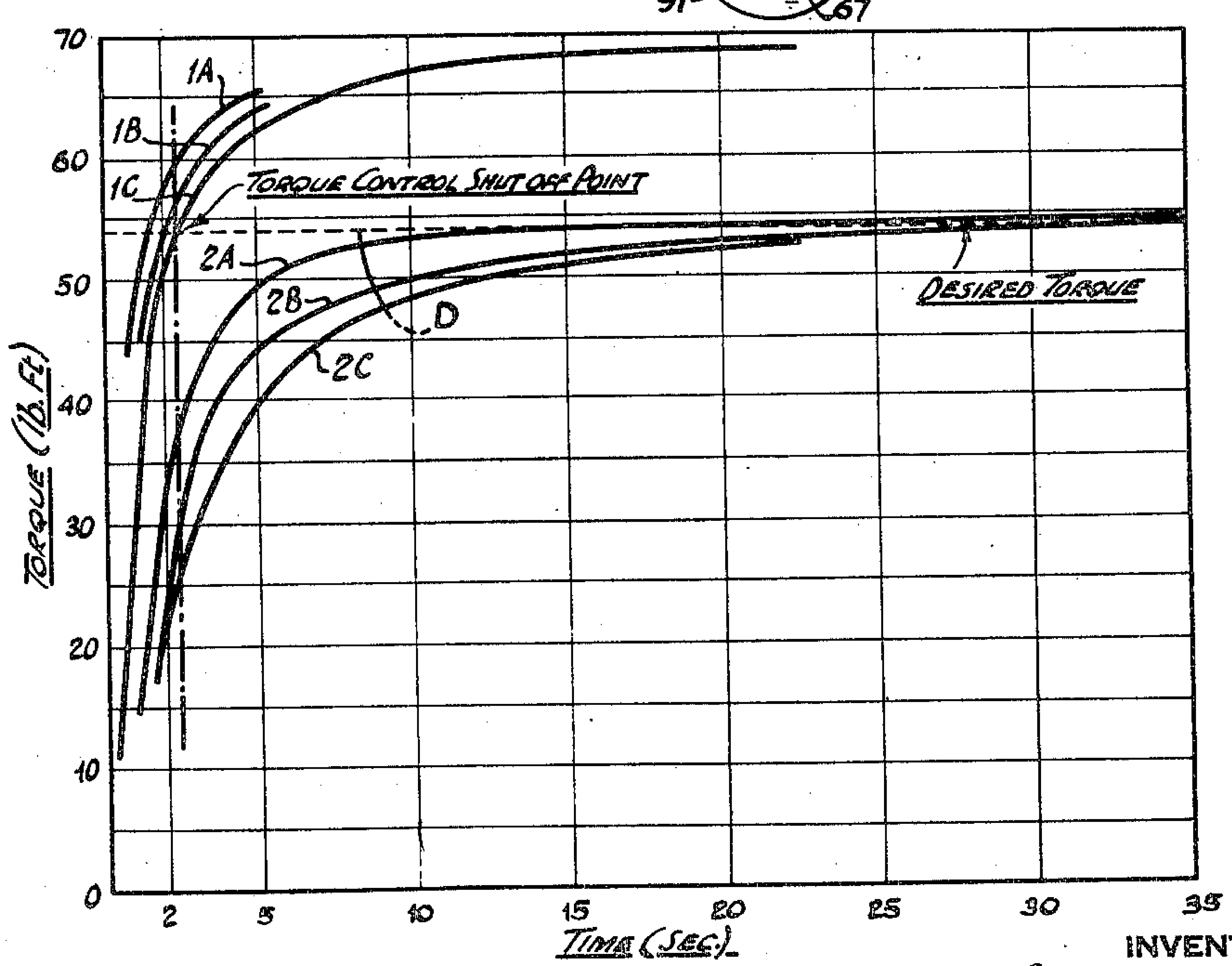


Fig. 13.



INVENTOR
Spencer B. Maurer.
BY
Raymond G. Mullee
ATTORNEY

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2,543,979

IMPACT WRENCH TORQUE CONTROL

Spencer B. Maurer, Cleveland, Ohio, assignor to Chicago Pneumatic Tool Company, New York, N. Y., a corporation of New Jersey

Application January 31, 1946, Serial No. 644,501

18 Claims. (Cl. 192—150)

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This invention relates to impact wrenches and more particularly to an arrangement for measuring and controlling the degree of tightness of the driven bolt or nut. Such control is desirable in many industrial applications, for example in the assembly of parts of internal combustion engines, such as cylinder head bolts, connecting rod bolts, and main bearing bolts.

There are in general two classes of rotary motor operated impact wrenches in common use. One is characterized by a torque responsive resilient accumulator, which stores potential energy in a device such as a spring while the driving shaft rotates ahead of the hammer, releases on overload, and uses the stored energy to accelerate the hammer as it moves into re-engagement with the anvil or driven element of the clutch, the driving shaft being rotated more or less continuously by a motor operating through reduction gearing. In the other class, there is a substantially direct connection between the rotor of the motor and the clutch hammer, whereby the rotor starts, stops, and rebounds in unison with the hammer. With such an arrangement, the declutching and reclutching are usually responsive to centrifugal or cam mechanism which permits satisfactory operation at different speeds.

By varying the air pressure, the maximum speed and maximum torque can be correspondingly altered. This is sometimes done by inserting a pressure regulator in the live air passageway and setting the regulator so that the air pressure and speed will be just enough to hammer the bolt to the desired tightness if the tool is applied for a sufficient length of time, and with forces which never become excessive so as to damage the work. Such a method of regulation is referred to as the "ultimate torque" method.

An impact wrench delivers its power by a series of individual torque impulses transmitted through a spindle or tool head which is integral with the anvil element of the clutch. Assuming the work to be a bolt that is being tightened, or other load whose resistance increases upon displacement, the ultimate torque that can be obtained is limited only by the strength of the work or the flexibility of the tool head transmitting the torque impulses. Provided the strength of the work is sufficient, the ultimate torque obtainable with a given tool at a given air pressure occurs when the entire energy output (not including frictional losses) of each blow is absorbed in elastic deflection or distortion of the tool head or torque transmitting member. This ultimate torque is approached as a limit which is generally reached only after lengthy operation of the tool since the increment of torque impulse per blow decreases as the output torque rises. Therefore, the ultimate torque method of regulation is not considered practical for mass production work.

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An object of this invention is to overcome the disadvantages but retain the advantages of the ultimate torque method by predetermining the bolt tightness with the same degree of accuracy but by enabling the tool to complete a bolt setting operation in one quarter of the time, or less. Another object is to obviate the need for highly skilled operators. A further object is the provision of means for measuring the amount of torque developed upon each blow of the impact wrench. A still further object is to automatically stop the operation of the wrench when the measured torque exceeds a predetermined value. Still another object is to prevent the impact wrench from starting on a new operation until after the operator has first closed, then reopened the throttle valve.

In accordance with the foregoing objects, the invention contemplates the use of an air pressure which is much higher than the pressure which would be employed under the conventional ultimate torque method to attain the same degree of tightness. To limit the output torque of the wrench to some value below the ultimate torque for a given air pressure, and within the normal range of usefulness of the tool, it is necessary to ascertain in some way the maximum instantaneous torque attained during each individual impulse. Since it is not feasible to provide a tool head of sufficient length so that the deflection can be measured directly and used as a measurement of torque, the present invention proposes to measure the spring or potential energy stored in the tool head at the peak torque of each impulse. The amount of energy stored will vary as the square of the maximum torque attained during the impulse and will equal the spring constant of the tool head multiplied by the square of the torque. This peak torque of the impulse is reached when the tool head is at maximum deflection and the clutch hammer has expended all of its kinetic energy and both the work and the clutch are stationary. The work at this point has been tightened up to the peak torque and no further advance motion is possible for the remainder of the impulse. The spring energy stored in the tool head will cause reverse motion and acceleration of the clutch hammer and rotor as the tool head unwinds and converts its potential energy into kinetic energy which it imparts to the hammer and to the rotor of the motor. This reverse motion of the clutch hammer is known as rebound. The energy of rebound is an accurate measurement of the peak torque attained during the impulse.

A feature of this invention resides in the means for measuring the maximum torque of each impulse by measuring the spring energy stored in the tool head and subsequently returned to the clutch hammer as rebound energy of motion. The rebound energy may be measured by 1— rebound velocity attained by the clutch; or 2— distance of rebound motion against a known resisting torque. The latter or second method is utilized in the practice of the present invention. The resisting torque is the nearly constant forward motor torque plus the frictional resistance of the motor being driven backwards. For the purpose of a practical tool this resisting torque has proven to be sufficiently uniform. The distance of rebound then, as measured in degrees of rotation, is an accurate measure of the peak torque of any given impulse. Since there is no relative movement between the rotor and the clutch during the impulse period or during the period of rebound, it is only necessary to provide automatic means for measuring rotor rebound to measure accurately the peak torque of each impulse.

Another feature of this invention resides in the use of an overrunning clutch between the rotor shaft and the rebound shaft of the torque measuring element whereby the rebound shaft remains at rest while the rotor is driving the clutch and tool head and is deflected away from its normal position only at the time of rebound.

Other objects and features of the invention will appear more clearly from the following description taken in connection with the accompanying drawings and appended claims.

In the accompanying drawings which illustrate one embodiment of the invention:

Fig. 1 is a longitudinal section of an impact wrench embodying this invention, a part of the tool head or spindle being broken away, and a part of the grip handle being broken off;

Fig. 2 is a cross section through the impact clutch as indicated by the arrows 2 in Fig. 1;

Fig. 3 is a cross section through the impact clutch as indicated by the arrows 3 in Fig. 1;

Fig. 4 is an elevational view of the tool head;

Fig. 5 is a cross section as indicated by the arrows 5 in Fig. 1 and shows the overrunning clutch between the rotor and the rebound shaft;

Fig. 6 is a perspective view of the rebound shaft;

Fig. 7 is a cross section, as indicated by the arrows 7 in Fig. 1, showing particularly the constant torque air motor;

Fig. 8 is a detail view in longitudinal section showing the automatic valve tripped to closed position;

Fig. 9 is a cross section, as indicated by the arrows 9 in Fig. 1, showing the trigger which releases the automatic valve and the means for actuating the trigger upon the development of a predetermined rebound of the rotor;

Fig. 10 is a sub-assembly view in elevation showing the trigger being actuated to release the automatic valve;

Fig. 11 is a cross sectional view looking downward as indicated generally by the irregular line 11—11 in Fig. 9;

Fig. 12 is a detail view in elevation looking in the same direction as Fig. 11 but showing only the parts of the trigger and the associated supporting pin prior to assembly; and

Fig. 13 is a graph showing the relation between the impact torque delivered by the wrench and the time required to tighten the driven nut or bolt, the upper three curves relating to the torque control wrench of the present invention and the lower three to the conventional system in which the ultimate torque is developed.

Figs. 5 to 12 inclusive are drawn to a larger scale than Figs. 1 to 4.

The invention is applicable generally to impact wrenches of the class in which the motor drives the hammer directly and without any intermediate power accumulator, whereby the rotor of the motor starts, stops and rebounds in unison with the hammer. In the illustrative embodiment, the impact clutch per se conforms substantially to the structure shown in U. S. Patent 2,285,638, granted June 9, 1942 to Lester A. Amtsberg, but is driven only in a clockwise direction (looking forward). The illustrative wrench differs from the device of the patent primarily by the provision of means for measuring the output torque at the peak of an impulse and for automatically cutting off the supply of live air to the motor when such torque attains a predetermined value. To accomplish this aim, the tool head has been modified (as shown in Fig. 4) to make its torsional elasticity much greater than in prior impact wrenches. Also, the design of the rotary motor is such that it imparts to the clutch a more nearly uniform torque throughout a cycle of operations than in the case of the motors commonly used in impact wrenches.

Referring to Fig. 1 of the accompanying drawings, a motor housing 15 is provided with an integral side handle 16 by means of which the tool may be gripped and supported in the hand of an operator. The motor housing is secured in fixed relation to a clutch housing 17 by any suitable means, such as the usual arrangement of bolts and flanges, not shown. The front end of the clutch housing is tapered and fluted at 18 to provide another grip portion. A cover 19 is secured to the rear end of the motor housing 15 by means of bolts 21 (Fig. 9).

A rotary air motor 22 within the motor housing includes a cylinder or cylinder liner 23, the ends of which abut against end plates 24. The rear end plate has a flange 25 fitting part of a recess 26 in the cover, and also has a peripheral portion fitting the motor housing 15. Flange 25 surrounds and supports a ball bearing 27 held between rear end plate 24 and cover 19. A similar ball bearing 28 is mounted in a flange which projects forwardly from the front end plate 24. Ball bearings 27 and 28 support rear and front shafts 29 and 31 respectively, which are integral with and project from a rotor 32. The rotor is of cylindrical shape and is arranged coaxially with its shafts and with the clutch housing 17. The cylinder liner 23, mounted in housing 15, is arranged eccentrically with respect to the rotor 32 to provide a crescent-shaped chamber therebetween. The rotor is provided with a plurality of radial slots in which blades 33 are mounted for movement with their outer edges in scraping contact with the liner 23 to divide the crescent-shaped chamber into a series of pockets 34 intermediate the inlet ports 35 and exhaust ports 35 of the cylinder. As compared with standard rotary motors for impact wrenches, the inlet and exhaust ports are farther apart and there is a greater number of radial blades (seven instead of six) whereby to provide a greater number of active pockets than usually. In this way the motor is arranged to deliver a more nearly constant torque as the position of the blades with respect to the cylinder ports changes because the variation on any one blade is offset and compensated by variations on several other blades, thus producing an average or combined torque which is substantially constant. Hence, the speed developed by the rotor after starting from rest and turning 180° does not depend on the position of the blades at the starting point.

Positioned centrally of the clutch housing 17 is a rotatable tool head or driven spindle 37. At its rear end the spindle has an anvil portion comprising longitudinally extending jaws 38 adapted to receive rotational impacts as hereinafter described. At its front end the tool head is shaped to fit a wrench socket, not shown. As illustrated in Fig. 4, the tool head is longer and more slender than usual and, being made of steel, possesses a considerable degree of torsional elasticity which is utilized in the practice of the present invention. The tool head is supported with a rotating fit in a steel bushing 39. A resilient sleeve 41 made of oil resisting synthetic rubber, such as neoprene, is bonded to the outer surface of the bushing. The resilient sleeve 41 has a press fit within a counterbore 42 near the front end of the clutch housing 17. The rear end of the tool head 37 is supported in axial alignment with the steel bushing 39 and with rotor shaft 31 by means which include a pilot shaft 43 seated in complementary recesses in the rotor shaft 31 and tool head.

The impact clutch comprises a hammer assembly which surrounds the spindle 37 and is supported at the front and rear ends of the anvil jaws 38 for revolution about the axis of the spindle, the arrangement resembling a squirrel cage, as shown in Fig. 3. It extends between two similarly constructed end plates which constitute carriers by which the assembly is revolubly supported. Rear carrier 44 is mounted for oscillatory movement about a bearing surface provided on the rear end of a driving cam 45 splined to the rotor driving shaft 31. A bearing spacer 46 abuts against the rotor bearing 28 at its rear end and the driving cam 45 and hammer carrier 44 at its front end to secure the cam and carrier against rearward axial movement. A thrust plate 47 surrounding the pilot shaft 43 engages the front end of the driving cam and the rear face of the tool head 37 permitting relative rotation therebetween. Forward movement of the carrier 44 is prevented by engagement with cam projections 48 radiating from the front end of the cam 45. The front end plate or front carrier 49 for the hammer assembly is mounted for rotary movement relative to the head 37, a bushing 51 being interposed between the carrier and the tool head. The bushing 51 and front carrier 49 are confined between anvil jaws 38 and a thrust washer 52. The latter seats against a flange on the steel bushing 39 whereby the tool head 37 and carriers 44 and 49 are supported against axial thrusts.

A pair of heavy hammer dogs 53, similarly constructed and arranged, are supported for oscillatory movement about pivot pins 54 which extend through openings in the dogs and carriers, the heads of the pivot pins being retained by the thrust washer 52. A pair of bolts 55 are arranged to pass through openings in the carrier plates and each bolt is surrounded by a spacer sleeve 56 whose ends abut against the carrier plates 44 and 49. The bolts and pivot pins are evenly spaced about the axis of revolution and extend parallel to the axis of revolution and cooperate with each other to hold the carrier rigidly in fixed relation to each other and to the pivot pins. The radial arms 48 on the driving cam 45 project into driving engagement with recesses 57 at the rear end of the hammer dogs 53.

The impact clutch when driven in a clockwise direction (looking forward), operates in the following manner: The driving cam 45, having a direct connection with the rotor shaft 31, delivers force to the hammer dogs 53 in such a direction that the dogs have imparted to them a motion of revolution about the axis of tool head 37 and a component of force which tends to declutch them relative to the anvil jaws 38 on the tool head. The carrier assembly, which includes plates 44 and 49, pivot pins 54, bolts 55 etc., is carried with the dogs as they revolve. The dogs 53 are guided for rocking movement about pivot pins 54 due to camming engagement of the concave inner face of the dog with the anvil jaw 38. Upon completion of the rocking movement the dogs 53 are meshed with the anvil jaws 38. If the resistance to rotation of the driven nut or bolt is relatively slight, the clutch parts may remain for a considerable period in meshed relation, all parts revolving in unison due to friction between the dogs 53 and anvil jaws 38 and between the dogs and their pivot pins 54.

When the tool head 37 encounters substantial resistance to rotation, the forces holding the clutch in mesh are overcome by the declutching force set up by the driving cam 45 on the hammer dog 53 and the dogs are rocked in a releasing direction. As soon as the dogs are declutched, the driving unit is relieved of its load and accelerates to accumulate kinetic energy during a half turn of the motor after which the driving unit is arrested with an impact. The impacts are repeated as long as the operator holds the wrench socket in engagement with the torque resisting nut and continues the supply of air to the motor, unless sooner terminated by the automatic control of the present invention. If the resistance to rotation is moderate the dogs will be declutched before the driving unit comes to a complete rest relative to the clutch housing 17 although the driving unit is arrested, at the time of each impact, relative to the tool head 37.

For further details concerning the structure and operation of the impact clutch, reference is made to Patent 2,285,638, aforesaid. As in the device of that patent, advantage is taken of the torsional elasticity of the tool head 37 to facilitate the release of the clutch dogs from the anvil jaws. Thus, when a rotational hammer blow is delivered to the rear end of the tool head and the front end is held in engagement with a frozen nut or bolt, the tool head is twisted or distorted. Upon termination of the impact, the spindle or tool head 37 unwinds, causing the hammer dogs 53, carrier members 44, 49, 54 and 55, driving cam 45 and rotor 32 to rebound as a unit in a counter-clockwise direction. The rebound acts to momentarily relieve contact pressure between the impact surfaces on the dogs and anvil thereby facilitating declutching and reducing the rubbing action of the impact surfaces over each other. The torsional strain in the illustrative tool head 37 is more than sufficient for declutching purposes, due to the fact that the tool head is extended in length and reduced in cross section (as shown in Fig. 4) in comparison with tool heads commonly employed in standard impact wrenches. The rebound of the hammer assembly may cover an even greater arc of reverse movement than that of the anvil jaws 38, if the heavy dogs 53, and the parts which are reversely driven thereby during the rebound movement, have enough momentum to continue to travel counterclockwise after the tool head has untwisted to its normal position, the hammer assembly finally being stopped by the motor 22, with the hammer dogs in spaced relation to the anvil jaws 38.

Live air is supplied to the tool from an air hose (not shown) through an inlet port 58 in the grip handle 16 and thence through a throttle valve 59 to a handle passage 61, the throttle valve being controlled by the usual manipulative trigger 62. After leaving the handle passage 61, the live air enters a set of longitudinal ports 63 in an automatic control valve 64, which is best shown in Figs. 1, 7 and 8. The control valve has a head 65 in which the ports 63 are drilled, which head slidably fits a counterbore in valve chest 66, the latter being mounted in the motor housing 15 with a press fit. The valve also has a stem or extension 67 fitting a small bore at the rear end of the valve chest and projecting into the space enclosed by the cover 19 in order to connect the valve with other parts of the torque control mechanism. When the throttle valve 59 is off as illustrated in Fig. 1, the automatic control valve 64 is in its foremost position where it rests against a stop pin 68, being held by a compression spring 69 interposed between the valve head 65 and the closed end of the counterbore in the valve chest 66. In this position of the control valve the longitudinal ports 63 establish communication between the handle passage 61 and a set of ports 71 in the valve chest. In the normal operation of the tool and prior to the delivery of a predetermined torque, the automatic control valve is slightly rearward of the position shown in Fig. 1, being locked there by the engagement of the stem 67 with the torque control apparatus, as described hereinafter.

From the ports 71 in the valve chest 66 the live air passes to motor inlet chamber 72, Fig. 7, through inlet ports 35 in cylinder liner 23, through pockets 34 where the pressure fluid expands thereby acting against blades 33 to turn the rotor 32, then out through exhaust ports 36 and 73 in the cylinder liner and motor housing 15 respectively.

When the rotor 32 rebounds, in response to the spring energy stored in tool head 37, it acts as a pump to force air from cylinder exhaust ports 36, through pockets 34 where the air undergoes compression, out through inlet ports 35, and back to the supply line, thus recovering part of the energy used in twisting the tool head.

In accordance with the present invention the rebound motion of the rotor is measured, and is also utilized to cut off the supply of live air when the amplitude of the rebound, which corresponds to the force of the hammer blow, attains a predetermined amount. For this purpose the rear shaft 29 of the rotor is bored to receive a rebound shaft 74, the rear end of which projects beyond the rotor shaft. As shown in Figs. 5 and 6 the front part of the rebound shaft is cut away at its upper side to receive a roller 75. A pair of springs 76 mounted in apertures 77 in the rebound shaft urge the roller in a clockwise direction (looking forward) thereby providing an overrunning clutch which locks the rebound shaft to the rotor shaft whenever the latter turns counterclockwise but permits the rotor to turn clockwise independently of the rebound shaft. The rear end of the rebound shaft is provided with a vertical slot 78 which receives the upper end of a pendulous rod or kicker 79. The rod, which is square in cross section, is retained in the slot by any suitable expedient such as brazing, thereby forming in effect an integral structure with the rebound shaft.

The lower end of the pendulous rod 79 normally rests against an adjusting screw 81 supported in a resilient threaded bushing 82 mounted in the peripheral wall portion of the cover 19 as shown in Figs. 9 and 11. The outer end of the screw has a recess to receive a key (not shown) for turning the screw and thereby adjusting the normal position of the pendulous rod. For urging the latter against the screw a light torsion spring 83 is provided. One end of the spring embraces the rod 79 about half way between the ends of the latter, and from there it extends to the rebound shaft 74, makes several turns around the rebound shaft, then extends downwardly to the left side of a reduced extension on valve chest 66 and then along the bottom of said extension to the right side thereof where the spring 83 terminates.

The lower end of the swinging rod 79 is bent to extend into the plane of a bell-crank lever 84 which has a hole 85 (Figs. 10 and 12) into which the reduced end of sleeve 86 projects, the lever and sleeve being secured together with a press fit. The lever and sleeve assembly, which constitutes a trigger, is mounted for oscillating movement about a pin 87, the inner end of which is screwed into motor housing 15 and the outer end of which is supported in a recessed portion of adjusting screw 88, the latter in turn being mounted in resilient bushing 89, in the same manner that adjusting screw 81 is supported in its associated bushing 82. Normally the bell-crank lever 84 rests on the reduced extremity of the stem 67 on the automatic control valve 64 as shown in Figs. 9 and 11. In that position the lever 84 is adapted to limit the rearward movement of the automatic control valve by the engagement of the lever with a shoulder 91 on the control valve. The amount of such limited movement is regulated by turning the adjusting screw 88. This may be done by inserting a key through the opening in bushing 89 and into the socket in screw 88. The trigger 84, 86, is urged toward the Fig. 9 position by means of a torsion spring 92 one end of which is hooked on the longitudinal arm of the bell-crank lever and the intermediate portion of which coils around the sleeve portion of the sleeve 86 on the trigger. The other end of the spring extends over to and around the lower side of the reduced portion of valve chest 66. Preferably the trigger spring 92 is constructed to act also as a compression spring, thereby holding the bell-crank lever 84 seated against the inner end of screw 88 in any position of adjustment of the latter, as shown in Fig. 11.

The operation of the wrench with specific reference to the automatic control valve 64 is as follows: When the throttle valve 59 is opened live air acts against the front end of the head 65 on the control valve 64 thereby urging the valve rearward. This force is opposed by the spring 69 and by the live air pressure on the rear surface of the head 65 which is not completely balanced because the stem 67 is exposed only to atmospheric pressure at its rear end.

The automatic valve 64 is therefore moved rearward until the shoulder 91 becomes locked in engagement with the trigger or bell-crank lever 84. In the locked position of the valve (not illustrated) the ports 71 in the valve chest 66 are almost entirely uncovered and live air flows freely therethrough to operate the motor 22 as hereinbefore described.

At first the motor drives the impact clutch under a relatively light load and the rotor 32 rotates at high speed independently of the rebound shaft 74 due to the overrunning clutch arrangement which includes the roller 75 interposed between the rotor shaft 29 and the rebound shaft. The hammer dogs 53 deliver rotary impacts to the anvil jaws 38, the intensity of which increases with the resistance of the tool head to rotation. The delivery of such a hammer blow causes the elongated spindle or tool head 37 to twist, within its elastic limit, and then to unwind and cause the hammer assembly and rotor 32 to rebound or revolve in a counterclockwise direction. During such rebound movement the roller 75 of the overrunning clutch catches hold of the rebound shaft 74 and turns it through a corresponding arc. Upon termination of the rebound the rebound shaft turns clockwise, under the influence of the spring 83, until it is restored to its original or normal position, but the rotor 32 continues to turn for an additional 180° or more until the next impact is delivered.

The effect of the rebounds on the kicker rod 79 is that the rod swings away from the Fig. 9 position immediately following the delivery of a rotary hammer blow on the tool head 37, then returns into engagement with the adjusting screw 82, where it awaits the delivery of the next impact, then swings through a greater arc, then returns and remains momentarily at rest, then swings through a still greater distance, and so on. The angle of such swinging movement or deflection is an accurate measurement of the force of the impact because both the force and the deflection correspond to the potential energy stored in the spindle 37 and the angle through which the spindle is twisted.

The amplitude of the swinging movement of the rod 79 increases until it kicks the small arm of the bell-crank lever or trigger 84 as shown in Fig. 10. Thereupon the long arm of the trigger is rocked out of locking engagement with the shoulder 91 on the control valve 64, permitting the live air to force the valve rearward to the Fig. 8 position in which movement of the valve is arrested by the engagement of a flange 93 with the end of the counterbore in the valve chest 66. In this position the head 65 of the automatic valve completely covers the ports 71 in the valve chest, thereby cutting off the supply of air to the motor 22 and of course stopping the operation of the wrench. In order to condition the tool for starting a new cycle of operations, the operator must first release the trigger 62, whereupon the air trapped in handle passage 61 is bled out to exhaust through the slight clearance around valve head 65 and through ports 71, thereby permitting the spring 69 to restore the valve to the Fig. 1 position.

The torque characteristics of the illustrative wrench under various operating conditions are shown in Fig. 13 in which the ordinates represent the torsional force of the blow in pounds feet while the abscissas represent the time that has elapsed between the opening of the throttle valve and the delivery of the respective impacts. For convenience the locus of the points which depict a single operation is represented as a continuous line instead of a series of discrete points, although it will be understood that the torsional forces acting on the driven spindle are discontinuous and recurrent. The curves 1A, 1B and 1C represent the operations under an air pressure of 60 pounds per square inch while the curves 2A, 2B and 2C represent tests under the same conditions respectively but with an air pressure of 25 pounds. In each of these operations or tests the wrench was arranged to drive a ⅜ inch—24 nut or bolt, using a solid slab of steel ½ inch thick as the bolted member. In operations 1A and 2A the threads were dry. In operations 1B and 2B the screw threads were oiled thereby retarding the increase in the resistance to rotation and force of blow as compared with the dry threads used in tests 1A and 2A. In tests 1C and 2C the threads were dry but a lock washer was added, which resulted in shifting the curve still further to the right as compared with the other tests under the same pressure in which no lock washer was used.

Referring to curves 1C, 2A and 2B, which are more nearly complete than the others, it is seen that the impact torque varies from a low value to progressively higher values but with decreasing increments and attains a final ultimate value if the tool is operated for a sufficient length of time. The ultimate torque occurs when the entire usable energy of the blow is absorbed in the torsional elastic deflection of the wrench spindle or tool head. The length of time required to approach the ultimate torque is not always the same for a given air pressure and size of bolt but depends upon the surface finish of the threads, thread fit, degree of lubrication and flexibility of the bolt and bolted members.

The advantage of the present invention over the conventional ultimate torque method of operation will be better understood by comparing curves 1C and 2C which both relate to the same kind of driven element but to the use of different air pressures. Assuming that the operator desires to set the bolt at a torque of 54 pounds feet, with the present invention he uses a relatively high pressure, say 60 pounds, and the wrench starts out under high speed, delivers powerful impacts with a minimum of delay, and as soon as the first blow of the required torque is delivered the wrench automatically stops. The entire operation, depicted by that part of curve 1C which lies below dotted line D, takes place in less than three seconds. If the operator attempted to drive the bolt to the same degree of tightness under the same air pressure and within the same time by skilful manipulation of a conventional impact wrench, he would have to release the throttle valve 59 at precisely the correct instant, as a departure of only a fraction of a second from the critical instant would cause a variation of several pounds feet torque.

Using the ultimate torque method, the operator would first determine by experiment what amount of air pressure if applied indefinitely would be required to drive the bolt to the desired tightness, and then set the air regulator accordingly. Referring to curve 2C the air pressure is selected at only 25 pounds in order to make the curve substantially flat instead of steep when it approaches 54 pounds feet of torque, represented by dotted line D. However, with this method the air may be left on for considerably more than ten seconds before the torque is stabilized and the operator feels safe in stopping the wrench. By turning off the throttle valve prematurely, taking advantage of a tolerance of five per cent below the ultimate torque of 54 pounds feet, but allowing for variations in the resistance of the work and errors in timing, the operator using the ultimate torque method with a conventional wrench would require about eight or ten seconds to set the bolt, which is three or four times as long as required by the torque control system of the present invention.

In the use of the present invention, variations in the torque delivered by the wrench are slight and predictable notwithstanding unknown variations in the air pressure and characteristics of the driven bolt with the same adjustment. The final blow may be slightly greater at some times than at others if the kicker 79 happens to swing to the trigger 84 on one of its strokes without actually unseating the trigger and then, on the final stroke, swing with a little more speed and momentum than is necessary to actuate the trigger. No attempt is made to limit the torque output during any impulse. It is necessary that there be a sufficiently large number of separate impulses so that the increment of torque increase on the last impulse is smaller than the total allowable variation in the final torque. This is easily accomplished in practice by controlling the air inlet so that the total striking energy per blow is of such a value that at least ten blows are required to attain the desired torque. As mentioned before, the increment of torque increase per blow decreases as the work resisting torque increases thus insuring that the torque increase on the last blow or impulse will be much less than one-tenth of the total torque desired. This means that the total error or variation from the desired torque can be no more than the torque increment of the last blow. For greater accuracy a lower air pressure and larger number of blows can be used.

To lower the pressure, the adjusting screw 88 is turned counterclockwise thereby permitting the trigger 84, 86 to move rearward slightly under the compressive force of spring 92. In turn, the automatic valve 64 is permitted a slight additional movement rearwardly under live air pressure before the shoulder 91 is stopped by the trigger in normal operation; and in this new position of adjustment the head 65 of the valve partly covers ports 71 to lower the pressure supplied to the rotary motor 22. Alternatively, a conventional pressure regulator could be provided.

To select the cut off point in the torque curve, the operator turns the adjusting screw 81. Extending the screw closer to the short arm of the trigger or bell-crank lever 84 reduces the size of the arc through which the pendulous arm 79 must swing in order to kick the trigger. This reduces correspondingly the amplitude of rebound movement and the torsional force of the final blow. Conversely, by withdrawing screw 81 further away from the trigger the wrench is conditioned for delivering more powerful impacts.

What is claimed is:

1. A power operated impact wrench comprising a rotary motor, a torsionally resilient tool head driven thereby with a succession of rotational hammer blows, energy measuring means actuated in response to the rebound energy delivered by the tool head for measuring the force of the hammer blow, and automatic means for cutting off the supply of power to the motor said energy measuring means operative upon the development of a predetermined force of blow to initiate actuation of said automatic means whereby the supply of power to the motor is cut off.

2. An impact wrench comprising a motor having a rotor, a hammer assembly rotatable substantially in unison with the rotor, an elongated tool head having at one end an anvil arranged to receive a succession of torsional impulses delivered by the hammer assembly and at the other end being adapted to drive a wrench socket, means movable in response to the force of the individual torsional impulses, said movable means comprising a rebound shaft and an overrunning clutch between the rotor and rebound shaft, the clutch being arranged to permit the rotor to turn independently of the rebound shaft in the driving direction but to drive the shaft during the rebound of the rotor, whereby the deflection of the rebound shaft corresponds to the potential energy stored in the tool head at the time of the impulse and subsequently released during the rebound of the hammer assembly and rotor.

3. An impact wrench comprising a resilient tool head, a rotatable hammer arranged to deliver a series of torsional impulses to the tool head, an air motor having a rotor connected substantially directly with the hammer whereby the rotor starts, stops and rebounds in unison with the hammer as each impulse is delivered, a rebound shaft for moving in response to the rebound movements of the rotor, an overrunning clutch arranged to lock the rebound shaft to the rotor only during the rebound movement of the latter, a passageway for supplying the motor with live air, and automatic valve means for closing said passageway to stop the motor upon the development of a predetermined degree of movement of the rebound shaft.

4. An impact wrench comprising a resilient tool head, a rotatable hammer arranged to deliver a series of torsional impulses to the tool head, a driving motor having a rotor connected substantially directly with the hammer whereby the rotor starts, stops and rebounds in unison with the hammer as each impulse is delivered, a rebound shaft movable correspondingly to the amplitude of the rebound movements of the rotor, an overrunning clutch arranged to lock the rebound shaft to the rotor only during the rebound movement of the latter, a displaceable element driven by the rebound shaft, a limit stop against which the element normally seats, yieldable means constantly urging the element toward the limit stop, whereby the element repeatedly moves away from the limit stop and returns, the extent of movement being a function of the amplitude of rebound motion and hence the force of the impacts delivered by the hammer.

5. An impact wrench comprising a rotatable hammer arranged to deliver a series of torsional impulses to a resilient tool head, an air motor having a rotor connected substantially directly with the hammer whereby the rotor starts, stops and rebounds in unison with the hammer, a rebound shaft for responding to the rebound movements of the rotor, an overrunning clutch arranged to lock the rebound shaft to the rotor only during the rebound movement of the latter, a pendulous rod secured to the rebound sha

a limit stop against which the rod normally seats and from which it is displaced upon rebound, a spring constantly urging the rod toward the limit stop, whereby the rod repeatedly returns to the limit stop after displacement therefrom, the extent of movement being an indication of the amplitude of rebound motion and hence the force of the impacts delivered by the hammer, a trigger interposed in the path of the rod, and means responsive to the impingement of the rod on the trigger for interrupting the supply of live air to the motor.

6. An impact wrench according to claim 5 in which the distance between the limit stop and the trigger is adjustable to permit regulation of the force of torsional impulse required for stopping the motor.

7. An impact wrench according to claim 5 in which the limit stop comprises a screw, the abutting end of which is movable toward and away from the trigger to adjust the amount of swinging movement required for the actuation of the latter.

8. An impact wrench according to claim 5 which includes means on the air supply interrupting means for preventing the motor from starting after the supply of live air is interrupted and until such preventing means is released by manipulative means.

9. A power operated wrench comprising a casing, a rotary motor therein arranged at its front end to drive a tool head, a pendulous rod in back of the motor, an adjusting screw in said casing and having an inner end against which the rod is adapted to rest, a spring constantly urging the rod toward said adjusting screw, automatic means for swinging the rod away from the screw in response to torque developed by the motor, a trigger comprising a bell-crank lever having a short arm interposed in the path of the swinging movement of the rod, a spring urging the long arm of the bell-crank lever against a seat, a plunger movable transversely to the plane of the lever having a shoulder urged into engagement with the long arm of the bell-crank lever when the latter is seated, and means responsive to movement of the shoulder beyond the plane of the bell-crank lever for cutting off the supply of power to the motor whereby the development of a predetermined torque causes the swinging rod to strike the trigger and thereby release the plunger and stop the motor.

10. An impact wrench comprising a rotary air motor, a rotatable hammer driven by the motor, a rotatable tool head arranged to receive torsional impulses delivered by the hammer, said tool head being elongated and having substantial torsional elasticity whereby it delivers rebound energy to the hammer, a passageway for supplying live air to said motor for operating the same, an automatic valve controlling said passageway and movable between open and closed positions to admit or cut off the supply of live air to the motor, and means responsive to the development of a predetermined rebound energy for effecting the movement of the valve to closed position to stop the motor.

11. An impact wrench comprising a rotary air motor, a rotatable hammer driven by the motor, a rotatable tool head arranged to receive torsional impulses delivered by the hammer, said tool head being elongated and having substantial torsional elasticity, a passageway for supplying live air to said motor for operating the same, an automatic valve controlling said passageway and movable between open and closed positions to admit or cut off the supply of live air to the motor, said valve being urged toward closed position, a trigger normally locking the valve in open position, and means responsive to a predetermined torsional strain on the tool head for releasing the trigger and thereby permitting the valve to close.

12. An impact wrench comprising a rotary air motor, a rotatable hammer driven by the motor, a rotatable tool head arranged to receive torsional impulses delivered by the hammer, said tool head being elongated and having substantial torsional elasticity, a passageway for supplying live air to said motor for operating the same, an automatic valve in said passageway and movable between open and closed positions to admit or cut off the supply of live air to the motor, means responsive to pressure in the passageway for urging the valve toward closed position, a trigger normally locking the valve in open position, and means operatively connected to the tool head and responsive to a predetermined torsional strain on the tool head for releasing the trigger and thereby permitting the valve to close, and yieldable means for moving the valve to open position upon reduction in the pressure in said passageway, whereby the operator may restart the wrench subsequent to an automatic stopping operation by first disconnecting and then reconnecting the passageway to the source of live air.

13. A portable pneumatic tool comprising a rotary air motor, a passageway for supplying said motor with live air, an automatic control valve in said passageway and movable between open and closed positions to admit or cut off the supply to the motor, yieldable means maintaining said automatic valve in open position, said automatic valve being arranged to be urged toward closed position, a trigger for locking said automatic valve in open position, and means operatively connected to the motor and responsive to the development of a predetermined torque delivered by the motor for releasing said trigger to permit the automatic valve to close and thereby stop the motor.

14. A power operated wrench comprising a rotary air motor, a passageway for supplying said motor with live air, a manipulative throttle valve in said passageway, an automatic control valve in said passageway between the throttle valve and the motor and movable between open and closed positions to admit or cut off the supply to the motor, yieldable means maintaining said automatic valve in open position and having an unbalanced area exposed to live air for shifting it to closed position, a trigger for locking said automatic valve in open position, and means responsive to the development of a predetermined torque by the motor for releasing said trigger to permit the automatic valve to close and thereby stop the motor, and yieldable means effective upon a reduction in motor torque for urging the trigger toward locking position, said trigger being movable to locking position upon movement of the valve to normal position, said valve being movable to normal position upon closing of the throttle valve, whereby the operator may start the motor by first closing and then reopening the throttle valve.

15. In a pneumatic tool having a rotary motor and a passageway for supplying live air thereto, a pressure regulator interposed in said passageway comprising: a valve chest having a bore and a counterbore in front of the bore, an automatic

valve slidably mounted in the chest and having a stem fitting the bore and a head fitting the counterbore, the chest having one or more ports extending radially from the counterbore and forming part of the passageway supplying live air to the motor, resilient means urging the valve forward, the valve head being positioned to cover the port partly or wholly upon rearward movement away from a normal position against said resilient means, the valve head having an unbalanced area exposed to live air pressure for overcoming said resilient means to move the valve rearward, the stem having a shoulder, a trigger engageable with the shoulder, and automatic means for moving the trigger out of the path of the shoulder upon the development of a predetermined motor torque to permit the valve to completely cover the port and thereby stop the motor.

16. A pneumatic tool according to claim 15 in which the trigger is adjustable forward and rearward relative to the valve chest, whereby to regulate the degree of partial covering of the port prior to the release of the trigger, and thereby control the speed of the motor.

17. In a pneumatic tool, a rotary motor comprising a cylinder, a rotor eccentrically disposed therein and having blades carried thereby, means for supplying live air to drive the rotor in a clockwise direction under a substantially constant torque, a spindle connected to the front end of the rotor and arranged alternately to receive power from the rotor and to impart thereto a series of rebound motions in opposition to the motor torque, a rear shaft on the rotor having a recess, and an overrunning clutch element and a rebound shaft received within said recess, said clutch element being arranged to drive the rebound shaft counterclockwise but permit clockwise movement of the rotor independently of the rebound shaft.

18. An impact wrench comprising a rotatable anvil, a rotatable hammer arranged to deliver a series of rotational impacts to said anvil and to rebound therefrom subsequently to such delivery, means for connecting the anvil to a driven element having gradually increasing resistance to rotation, whereby to cause a step-by-step increase in the peak torque of the impacts and consequently in the distance of rebound of the hammer from the anvil, a movable element operatively connected to the hammer to rebound therewith, a trigger interposed in the path of movement of the movable element and positioned to be operated thereby upon rebound movement of the hammer and element through a predetermined distance, and automatic means responsive to the operation of the trigger for disabling the means for driving the hammer.

SPENCER B. MAURER.

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