June 1, 1954

C. LEAVELL

2,679,826

PNEUMATIC TOOL

Filed Aug. 9, 1948

3 Sheets-Sheet 1

Fig. 1

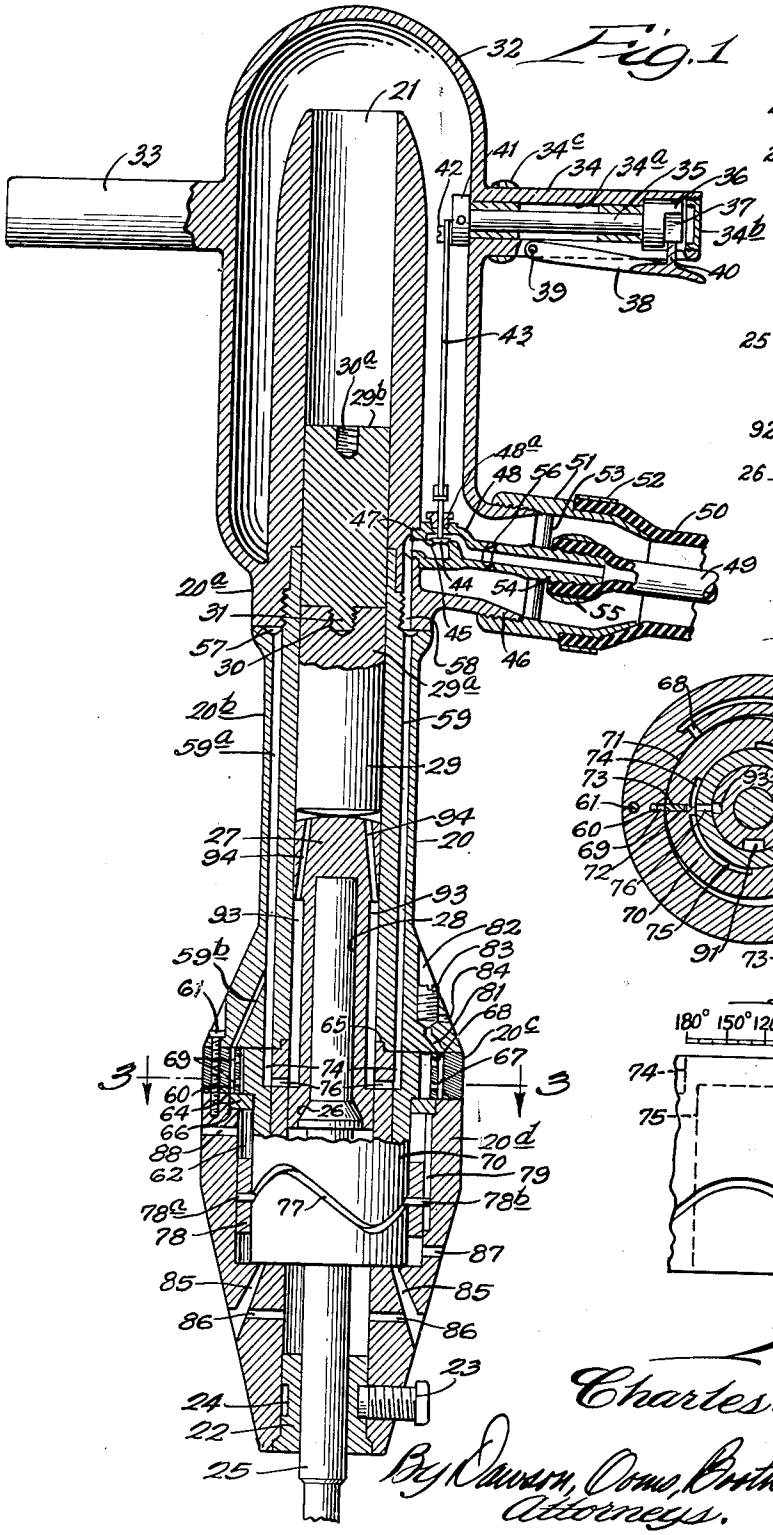


Fig. 2

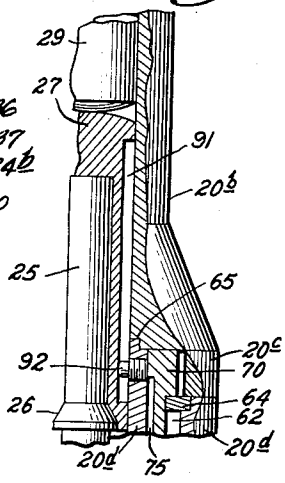


Fig. 3

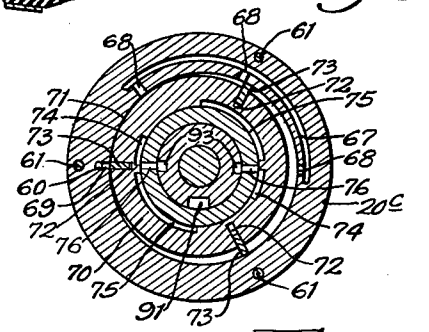
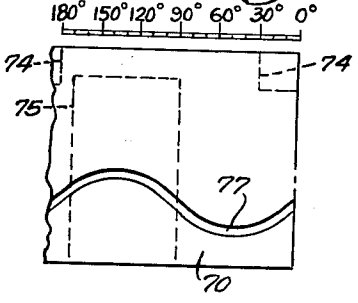


Fig. 4



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PNEUMATIC TOOL

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Fig. 11

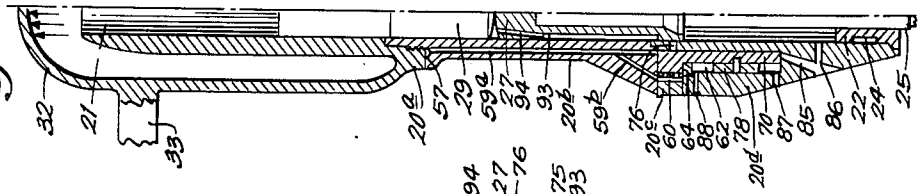


Fig. 10

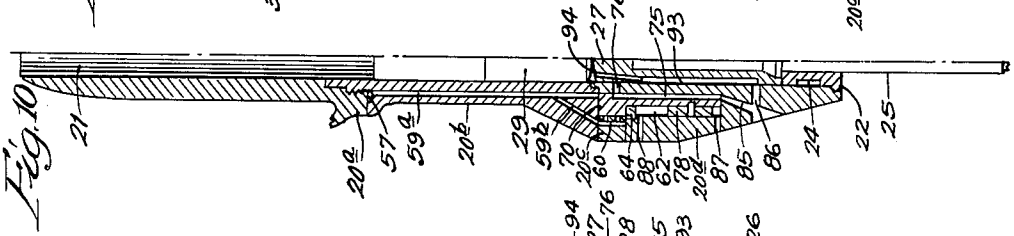


Fig. 9

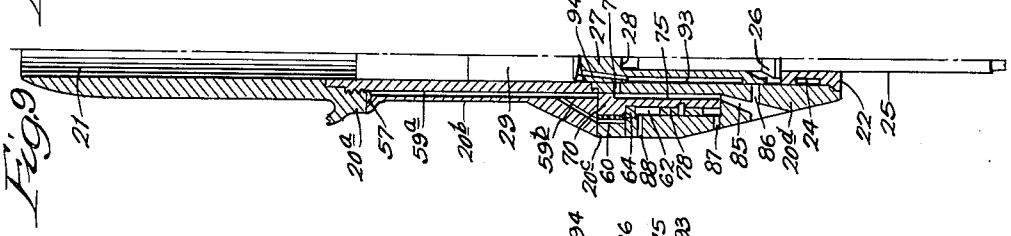


Fig. 8

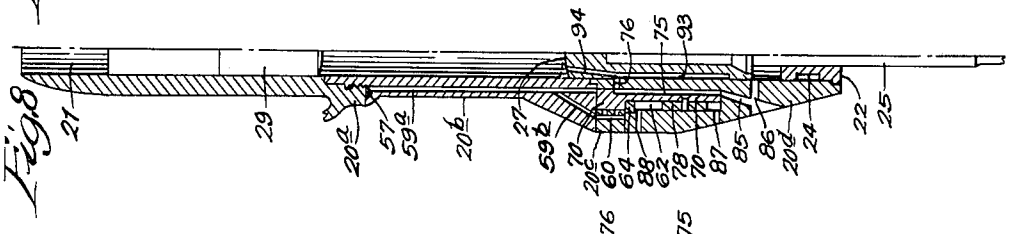


Fig. 7

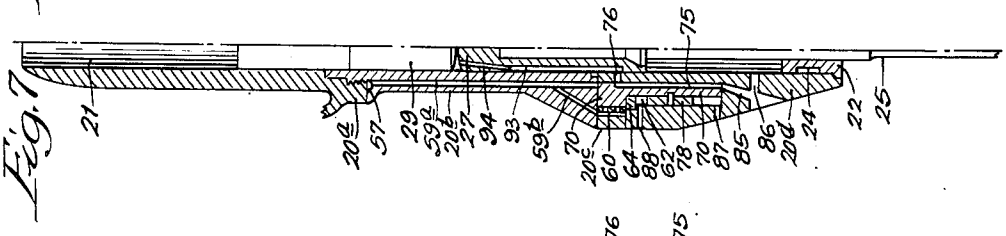


Fig. 6

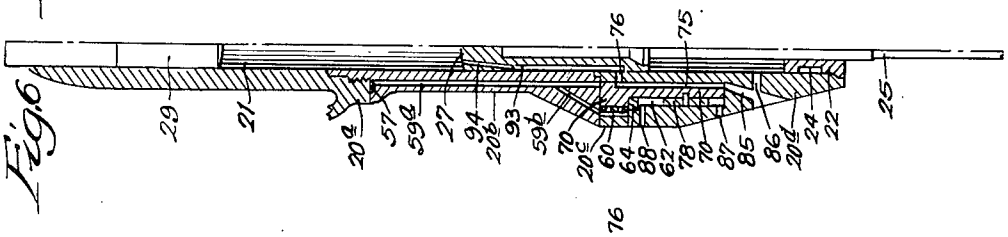
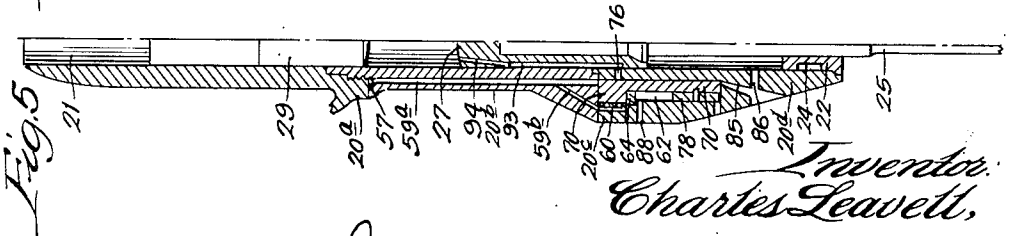


Fig. 5



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PNEUMATIC TOOL

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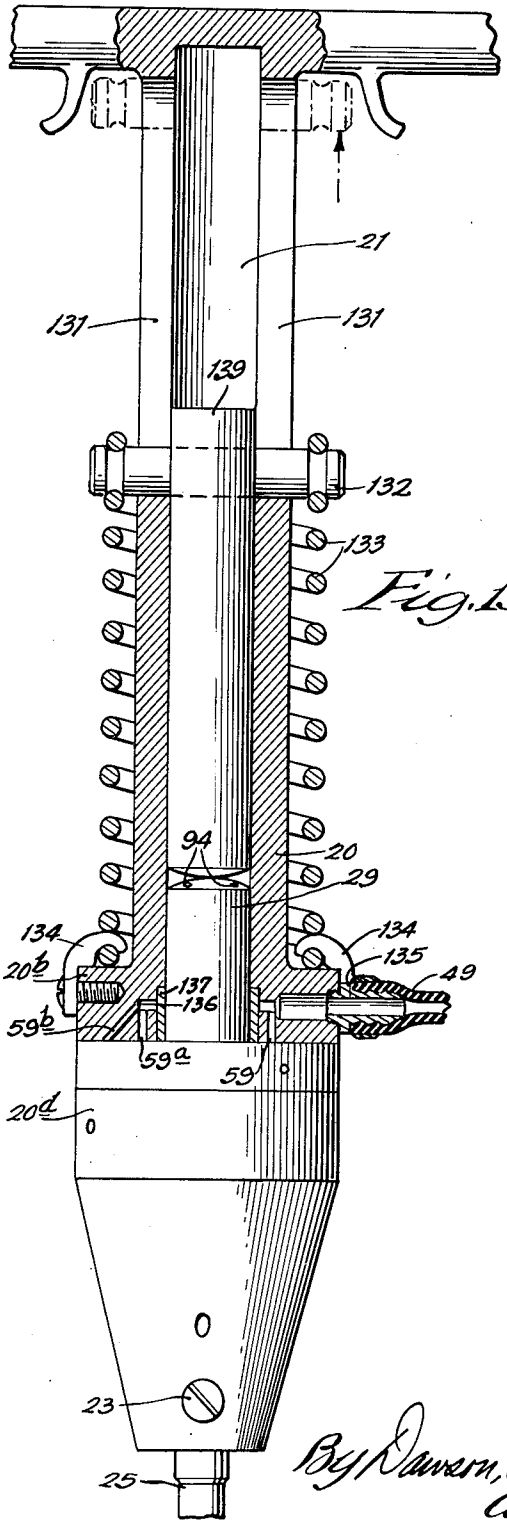

Fig. 13

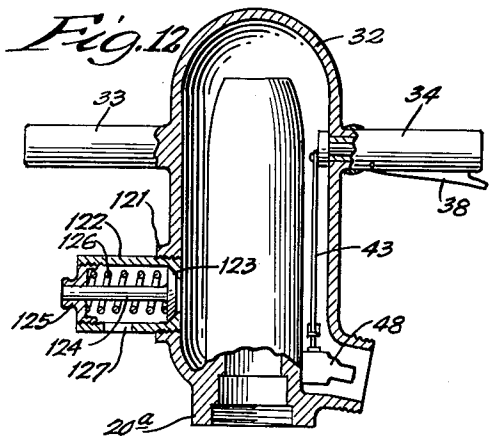

Fig. 12

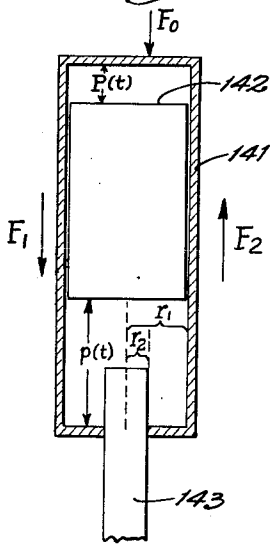

Fig. 14

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UNITED STATES PATENT OFFICE

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PNEUMATIC TOOL

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Application August 9, 1948, Serial No. 43,300

28 Claims. (Cl. 121—13)

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This invention relates to percussive tools such as paving breakers, air hammers, pneumatic riveters, and the like.

As will be hereinafter set forth in more detail, this invention is directed to the elimination of objectionable features heretofore inherent in such tools—particularly vibration and noise—and to their improvement with respect to working speed, efficiency, versatility, and safety. The invention will be illustratively described with reference to a pneumatic paving breaker.

With reference to the problem of improving working speed, it should be understood that there are very wide disparities between the types of blows which are respectively ideal for the various working situations to which a paving breaker may be applied. It has been found, for example, on extensive jobs of wall and floor demolition, especially if the slab is thick, of high quality, and strongly reinforced, that demolition can be accomplished with much greater speed and economy by swinging and dropping a 100 to 300 lb. steel ball suspended by a cable from a mobile crane than by any other presently available means. Such a procedure will demolish concrete far more rapidly than will the use of hand-held pneumatic paving breakers. Although the number of blows delivered by the paving breaker per minute may exceed the number delivered by the heavy ball by a factor of 300, as against energy and momentum excesses of the ball blow over the paving breaker's blow by factors of only 10 to 20, the destructiveness of the individual blow increases so rapidly with its energy and momentum magnitudes that the working speed of the heavy ball is far greater than that of the paving breaker, in spite of the great superiority of the breaker with respect to blow frequency.

It is desirable, then, for the purpose of demolishing strong slab, that a paving breaker be able to deliver heavy blows with a massive piston approximating the blows delivered by the steel ball, which in energy terms may be of the order of 1000 foot-pounds, in spite of the decrease in blow frequency which would attend the substitution in a paving breaker of, say, a 40 lb. piston delivering an 800 foot-pound blow for a 5 lb. piston delivering a 75 foot-pound blow. The latter case may be taken as typical of current paving breaker design. On the other hand, when a paving breaker is to be used with a spade point for digging in clay, it is desirable that it be able to produce a light, high-frequency impact—for example, a 60 foot-pound blow delivered with a 5 lb. piston.

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From the foregoing it will be clearly seen that a paving breaker should be designed to enable the operator to adjust the type of blow delivered by it, with respect to the working application, from a light, high-frequency impact to a heavy, low-frequency impact, and it is obvious, in view of the many varieties of work intermediate in blow type requirement between the illustrative examples given, that a graduated adjustability of blow type is necessary to secure maximum working speed in all applications.

Vibration is the property of heretofore existing hand-held percussive tools which has limited the momentum and energy magnitudes of the blows they could be designed to deliver, because the amplitude of the vibration of any such tool when held free of constraint, a condition often well approximated in use, is proportional for any given casing weight to the weight and length of stroke of its piston. For example, an 80 lb. paving breaker of conventional type oscillating a 40 lb. piston over a 1-foot stroke length, measured with respect to a fixed point, would exhibit handle vibration amplitudes up to 1 foot, a prohibitive value for hand-held use.

Elimination of vibration from percussive tools would clearly, therefore, in the light of the foregoing, constitute a great forward step in improving the effectiveness of such tools, and is a required condition to the development of a variable-blow paving breaker capable of adjustment for the delivery of the very powerful blows necessary for maximum working speed in heavy demolition applications. To provide a percussive tool substantially free from vibration is one of the principal objects of this invention.

A related object of my invention is to provide a hand-held percussive tool capable of operation by one man and nonetheless having a piston which may have a weight of 50 pounds or more, an increase by a factor of about 10 over present piston weights.

Another object of my invention, the pertinence of which to the problem of maximizing working speed in each of various types of application will be understood from the explanation just given, is to provide means within a percussive tool whereby the weight of its piston, its length of stroke, and the frequency of its reciprocations can be varied at will within wide limits. To illustrate, in one design of tool, the range by piston weights might be from 4 to 50 pounds, the range of stroke length might be from 5 inches to 2 feet, and the range of frequency might be from 100 to 3000 blows per minute.

Since the operator should not be burdened with the weight of an excessive amount of unused cylinder length when the tool is adjusted for short-stroke use, it is a related object of my invention to provide a percussive tool whose length of casing can be adjusted by the removal or insertion of sections, as required.

Pneumatic percussive tools constructed according to the techniques of the prior art have a cylinder with a piston which is reciprocated in the cylinder by the application of the full line pressure alternately to the top and bottom of the piston. The compressed air in the cylinder is vented after both the down-stroke and the up-stroke of the piston.

Accordingly, in prior art tools, the compressed gas which actuates the tool is vented at substantially the same pressure at which it is taken from the compressor. This mode of operation is inefficient and unsatisfactory for two reasons—first, it wastes by far the greater part of the potential energy stored in the compressed gas, and second, it causes a loud and most unpleasant burst of noise each time the cylinder is vented, since the sudden expansion of the compressed air on being vented to atmosphere necessarily produces a loud report similar to a thunder clap. When the tool is operating at normal speed, these thunder claps follow one another in rapid succession and cause the nerve-wracking noise which makes the use of pneumatic tools so objectionable to persons within hearing.

In the pneumatic tool provided by this invention, means are provided whereby the compressed gas which supplies energy to the tool is allowed to expand to substantially atmospheric pressure within the cylinder of the tool before being vented. As a result, a large proportion of the energy of compression is reclaimed in the form of useful work, rather than being dissipated in generating noise, and the tool constructed according to this invention thus is more efficient and quieter than tools designed in accordance with the prior art. It is thus another principal object of this invention to provide a pneumatic tool wherein the energy of compression stored in the compressed air is in large part utilized to actuate the tool, the gas being at substantially atmospheric pressure when vented from the tool. Still another object of this invention is to provide a pneumatic tool substantially quieter in operation than tools designed according to the prior art, the reduction in noise being achieved by venting air from the tool at atmospheric pressure rather than at the high pressure at which the air is fed from the compressor.

In pneumatic tools constructed according to the prior art, the upper and lower limits of the piston stroke have been defined within narrow limits by the valve provided for directing flow of compressed air alternately to the upper and lower ends of the cylinder. Thus, it has been necessary for the operator of the tool to "follow" the tool into the work-object as the cutting spike or other contact member advanced under the impacts of the reciprocating piston. It is still a further object of this invention to overcome this limitation and to provide a pneumatic tool wherein the contact member and the piston are free to "drift" down the cylinder as the tool causes the contact member to cut into the work-object. This advantage of my present invention enables the operator to adjust his position to that best adapted to give a firm, steady push on his tool and to hold that position until the tool has advanced into the work-object to the full extent of the drift-path, which in a typical embodiment of my invention might be eight inches or more.

Another object of my invention is to provide a means by which the tool is automatically cut off from its source of compressed air when the spike or other contact member is at or near the bottom of its "drift-path"—that is, when the spike or other contact member is at or near its point of greatest outward movement. This mechanism thus informs the operator automatically, by turning off the compressed air, when the tool has completed its full cut and is ready for re-positioning. Perhaps the most important virtue of this feature of my invention, however, is that of safety, since, as will be seen, it prevents the air from being accidentally applied to the piston while the tool is being carried from one place to another. This cut-off mechanism provides another advantage, as will be shown, in that it insures that the operator will at all times impose a substantial push on the tool during operation and thereby impart to it adequate momentum for its operation against the work-object, since failure on his part to deliver the proper push to the tool results in the air being shut off automatically.

An additional object of this invention is to provide a pneumatic tool having a new and greatly improved means whereby the compressed air source assists the operator in removing the tool from a work-object, such as a block of concrete, in which it has become embedded.

Still another object of this invention is to provide a pneumatic tool wherein the valve mechanism and other massive parts of the tool have been removed to a point below the lower limit of the piston stroke, thereby lowering the center of gravity of the tool and raising the limit of the free stroke length for a cylinder and piston of given size.

The foregoing and other specific objects and advantages of my invention will appear as the specification proceeds.

An illustrative embodiment of a pneumatic tool according to my invention, which for example has been shown as a paving breaker, is portrayed in the accompanying drawings, of which Figure 1 is a view in side elevation, partly in section, of the paving breaker, the section being taken through the axis of the cylinder; Fig. 2 is a fragmentary view, partly in section, of the paving breaker of Fig. 1, the section being taken through the axis of the cylinder in a plane at right angles to that of the section in Fig. 1; Fig. 3 is a horizontal sectional view of the tool of Fig. 1, the section being taken along the line 3—3 of Fig. 1; Fig. 4 is a flat projection of approximately half of the curved outer surface of a cylindrical rotary valve which is an important component of the pneumatic tool shown in Fig. 1; Figs. 5 to 11 inclusive are fragmentary sectional views showing the position of various components of the Fig. 1 embodiment of the invention during the course of its operating cycle; Fig. 12 is a fragmentary sectional view, partly in front elevation, showing a modification of the pneumatic tool of Fig. 1; Fig. 13 is a view, partly in section and partly in front elevation, showing another modification of the pneumatic tool of Fig. 1; and Fig. 14 is a diagrammatic showing of the essential elements of a pneumatic percussive tool of the reciprocating piston type, such showing being herein employed in connection with the discus-

sion of the basic principle of operation of this invention.

Before describing in detail the structure of the pneumatic tool shown in the drawings, with its various modifications, I shall, with particular reference to Fig. 14, develop briefly the mathematical basis for the underlying principle of my invention. Fig. 14 shows, diagrammatically, a pneumatic tool comprising a cylindrical casing 141, a piston 142 reciprocable therein, and an impact-transmitting element 143 which is slidably mounted in the bottom of cylinder 141. Fig. 14 being diagrammatic, it will be understood that piston 142 is to be considered as making a gas-tight fit with the walls of cylinder 141, and impact-transmitting member 143 is to be taken similarly as making a gas-tight fit with the aperture in cylinder 141 within which it is slidable. The mechanism illustrated in Fig. 14 may be considered to be a generalized pneumatic tool reduced to its simplest terms; thus, in a practical embodiment, element 143 would normally comprise an anvil and a spike. For purposes of Fig. 14, the single element 143 is taken as the functional equivalent of the anvil and spike of an actual tool.

It will be apparent, by reference to Fig. 14, that if element 143 is to be prevented from imparting vibration to the casing 141, element 143 must be capable of free and substantially frictionless reciprocation, in response to impact from the piston, over a range substantially greater—preferably several times greater—than the maximum expected advance of the force-transmitting element under a single impact. The following argument assumes, and the practical embodiments of my invention hereinafter described provide, such freedom of movement by the impact-transmitting element with respect to the casing. Hereinafter in this specification this range within which the impact-transmitting element is free to move with respect to the casing without mechanical, pneumatic, or other force-transmitting coupling will be denoted the "force-free drift range."

The provision in my invention of such a force-free drift range for the impact-transmitting element within the cylinder represents a novel departure from prior-art pneumatic tools, since in such tools the range of motion provided for the anvil-block is extremely short, and an air-trap is usually provided as a "cushion" between the impact-transmitting element and the bottom of the cylinder. Such a cushion, of course, functions as a pneumatic coupling between the two elements, and in such tools a considerable part of the casing vibration is attributable to accelerations imparted to the casing by the impact-transmitting element.

The pneumatic pressure operative between the upper end of piston 142 and the top of cylinder 141 is denoted $P(t)$, representing a variable function of time. The pneumatic pressure operative within the lower portion of cylinder 141, pressing against the under side of piston 142, is denoted $p(t)$, also representing a variable function of time. The radius of the piston 142 and the inner radius of cylinder 141 are equal and are denoted r_1. The radius of the impact-transmitting element 143 is denoted r_2. The force exerted on the top of the cylinder 141, urging the tool toward the work-object, is denoted F_o. The quantity F_o thus symbolizes the operator's "push." The total force tending to accelerate the piston toward the member 143 is denoted F_1. The total force tending to accelerate the casing toward the operator is denoted F_2.

From an examination of the mechanism of Fig. 14, it will be seen that the total force tending to accelerate the piston 142 toward the element 143 is equal to the total force exerted on the top of piston 142 by the pressure $P(t)$ less the total force exerted on the bottom of piston 142 by the pressure $p(t)$. In equation form, this may be written:

$$F_1=\pi r_1^2P(t)-\pi r_1^2p(t) \tag{1}$$

The total force tending to accelerate the casing toward the operator equals the total upward force exerted on the top of cylinder 141 by the pressure $P(t)$ minus the sum of the operator's push and the total force exerted on the bottom of cylinder 141 by the pressure $p(t)$. Expressed as an equation, this may be written:

$$F_2=\pi r_1^2P(t)-[\pi(r_1^2-r_2^2)p(t)+F_o] \tag{2}$$

In order to make the pneumatic tool represented in Fig. 14 perform useful work and at the same time be free from casing vibration, two conditions must be met: (I) F_1 must alternate in sign as time increases; and (II) F_2 must remain equal to zero at all times.

The first of the conditions just stated is clearly necessary to achieve reciprocation of the piston within the cylinder, and the second condition is the condition of "no vibration." It may be seen by reference to Equation 1 that in order for F_1 to alternate in sign as time increases, $p(t)$ must be alternately greater and less than $P(t)$.

Expressing the second of the aforementioned conditions as an equation we have:

$$\pi r_1^2P(t)-[\pi(r_1^2-r_2^2)p(t)+F_o]=0 \tag{3}$$

By an obvious transformation, Equation 3 becomes:

$$P(t)-\frac{r_1^2-r_2^2}{r_1^2}p(t)-\frac{F_o}{\pi r_1^2}=0 \tag{4}$$

Differentiating Equation 4 with respect to time we obtain:

$$P'(t)-\frac{r_1^2-r_2^2}{r_1^2}p'(t)=0 \tag{5}$$

Dividing through by the derivative of $p(t)$, Equation 5 becomes:

$$\frac{P'(t)}{p'(t)}=\frac{r_1^2-r_2^2}{r_1^2} \tag{6}$$

The physical nature of the mechanism under consideration necessarily implies that no instantaneous change in the values of $P(t)$ and $p(t)$ can occur. Therefore, $P(t)$ and $p(t)$ are entirely continuous functions, and their derivatives are always finite. Moreover, as is well known in the theory of continuous functions, if two continuous functions of the same variable are so related that one of the functions becomes alternately greater and less than the other as the variable increases, then the derivative of the one will alternately be greater and less than the derivative of the other as the variable increases. Since, in the present example, from condition I, $p(t)$ will be alternately greater and less than $P(t)$, it follows that the derivative of $p(t)$ will be alternately greater and less than the derivative of $P(t)$.

It should be noted that the right-hand member of Equation 6 is a constant. Therefore, the value of the left-hand member of Equation 6 must likewise be a constant. Since the derivative of $p(t)$ is alternately greater and less than the derivative of $P(t)$, zero is the only constant value which

could be assigned to the left and right members of Equation 6 consistently with the conditions heretofore stated. It must follow, therefore, that the derivative of $P(t)$ is at all times equal to zero, and accordingly $P(t)$ is constant. Also, to make the right-hand member of Equation 6 equal to zero, r_1 must equal r_2.

The mathematical deductions just stated lead to the conclusion that to eliminate from a practical tool casing vibration due to piston actuation, the pressure on the top of the piston must be constant, and the area of the cross section of the impact-transmitting element which provides seal for the alternately greater and lesser pressure on the bottom of the piston must be equal to the cross-sectional area of the cylinder.

The foregoing analysis summarizes very briefly my basic discovery which is embodied in my invention. I have set forth the conditions which must be met to prevent vibration communication between the impact-transmitting element and the casing, and between the piston and the casing, and I have shown in the drawings a practical vibrationless tool which meets those conditions and which attains the objects hereinbefore stated.

In the drawings, a tool casing 20 is shown, casing 20 being, for convenience in construction, formed of the separable pieces 20*a*, 20*b*, 20*c*, and 20*d*. Casing 20 is constructed to provide a cylinder 21 extending therethrough. The bottom end of cylinder 21 carries a closure member 22, held in position by pin 23, which is threaded into a suitable aperture in casing piece 20*d* and engages annular recess 24 in closure member 22.

A spike 25 is slidably carried in a central aperture in closure 22; near its upper end within cylinder 21 spike 25 has an annular shoulder 26. Anvil 27 is machined to have a snug sliding fit within cylinder 21; a central recess 28 in anvil 27 is formed to receive the upper end of spike 25, and the spike and anvil normally occupy a nested position as shown in Fig. 1, although they may slide apart. Anvil 27 has a slightly convex upper surface as shown.

Piston 29 is carried within cylinder 21; it may be formed from a single piece, although I have shown it in the drawing as comprising two parts, 29*a* and 29*b*. The lower end of piston piece 29*a* is slightly convex as shown. I have shown the upper end of piston piece 29*a* as having an internally threaded central recess 30. Piston piece 29*b* has at its lower end an axial projection 31 externally threaded and adapted to be screwed into the recess 30. As shown in the drawing, piston piece 29*b* has at its upper end a recess 30*a* similar to recess 30 in the lower piston piece; thus provision is made for the addition to the piston of a third section should such be desired. The preferable piston construction shown in the drawing is intended to facilitate the use of a piston having mass appropriate to the work to be performed; this flexibility with respect to piston size and mass is one of the distinctive features of my invention, as has been heretofore stated. It will be understood that piston 29 is machined to provide a snug sliding fit with cylinder 21.

Casing piece 20*a*, forming the upper part of casing 20, is formed with an inner portion comprising the upper part of cylinder 21 and an outer portion which envelops the upper end of cylinder 21 to form a bell-like hollow housing 32 therearound. Handles 33 and 34 extend outwardly from housing 32; they may be integrally formed with casing piece 20*a* as shown.

Near the lower end of housing 32, and in an axial line extending vertically downward from handle 34, a cylindrical coupling member 46 extends outwardly. Coupling member 46 is externally threaded to receive a hose fitting. In the side wall of cylinder 21 a recess 47 is located at a point approximately in line with the axis of coupling member 46 and is internally threaded to receive a second, smaller coupling member 48. Coupling member 48 is hollow, and its forward end contains a smoothly machined seat adapted to receive a resilient washer. The rear end of member 48 is open, providing communication between the hollow interior of member 48 and the recess 47.

Handle 34 is hollow, being provided with a cylindrical bore 34*a* extending from the inner wall of housing 32 to the outer end of handle 34. The portion of bore 34*a* immediately adjacent the outer end of handle 34 is enlarged to a diameter somewhat greater than that of the remainder of the bore. Within bore 34*a* a cylindrical member 35 is carried, and at the outer end of member 35 a cam-like end piece 36, cut away to provide a flat surface 37, is carried by member 35 within the enlarged outer portion of bore 34*a*. A cap 34*b* is threaded into the end of bore 34*a* to hold member 35 in position. A lever 38, secured to handle 34 by pin 39, has a projection 40 extending through a suitable aperture in the wall of handle 34 and adapted to engage flat surface 37 so as to rotate member 35 slightly when lever 38 is pressed against the flat surface 37. The inner end of member 35 carries a crank element 41 having an eccentric pin 42. Pivoted on pin 42 is a valve rod 43 which extends downward and passes through stuffing box 48*a* into the interior of coupling member 48. At its lower end lever arm 43 carries a valve member 44 which is adapted to cooperate with valve seat 45, so that when member 44 engages valve seat 45 communication through the interior of coupling member 48 is interrupted. A ring 34*c* is provided on handle 34; if desired, it may be slid outward along handle 34 to hold lever 38 in its upper position, in order that, during continuous operation, the manual valve can be dispensed with and reliance placed entirely on the automatic valve action provided by elements 93 and 76, as hereinafter described.

To the coupling members 46 and 48 the ends of a pair of hoses running to a pair of compressed air sources are respectively attached. The hoses are arranged coaxially one within the other; the inner hose is denoted 49 and the outer hose is denoted 50. The hoses terminate in a hose coupler 51 which includes a cylindrical outer casing having at one end a gradually reduced portion, over which hose 50 is stretched and secured in place by band 52, and having at the other end a threaded portion adapted to screw onto coupling member 46. A spider 53 carried internally by coupler 51 supports in the center thereof an inner hollow coupling member 54. One end of member 54 has a tapering portion over which the end of hose 49 is stretched and secured by band 55; the other end of member 54 terminates in a resilient washer 56 which may be cemented or otherwise securely affixed to the end of member 54. Washer 56 is adapted to seat tightly in the machined orifice at the forward end of coupling member 48 when coupler 51 is screwed onto outer coupling member 46. Hose 50 should be connected to a source of pneumatic pressure which is held at as nearly

constant value of pressure as possible. Hose 49 is connected to a source of pneumatic pressure which develops a pressure substantially greater than that of the constant-pressure source connected to hose 50.

As shown in the drawing, casing pieces 20a and 20b are screwed securely together, a portion of the upper end of piece 20b being threaded to engage a threaded recess within the lower end of casing piece 20a. Around the upper surface of casing piece 20b is an annular recess 57; a vertical bore 58 in casing piece 20a provides communication between recess 47 and the annular recess 57. Descending vertically from recess 57 within casing piece 20b are a pair of bores 59 and 59a, extending respectively entirely through casing piece 20b. A branch bore 59b descends diagonally outward and downward from bore 59a to a point on the bottom of casing piece 20b where it registers with a vertical bore 60 in the casing piece 20c.

Casing piece 20b can be, it will be noted, removed from the body of the tool and replaced by another piece 20b of different length or different mass or both. This feature of my invention is provided in view of the great versatility of my invention with respect to piston mass and stroke length. When the tool is adjusted for short stroke length and high impact frequency, a casing piece 20b of short length can be used; when the work at hand calls for a long stroke, a piece up to several feet in length might be used as casing piece 20b.

The lower portion of casing 20 comprises pieces 20c and 20d. Casing piece 20d has a central aperture therethrough which has the same inner diameter as the remainder of cylinder 21. On its upper surface, surrounding the central aperture, piece 20d carries a raised shoulder 65 which cooperates with a corresponding recess in the lower face of casing piece 20b to facilitate centering of the piece on assembly.

Casing piece 20d contains, concentric with the central aperture, a deep annular recess 62. The wall of recess 62 is set back near its upper end to provide a ledge or shoulder 66, and a short distance above ledge 66 the outer wall of casing piece 20d is cut away to provide accommodation for casing piece 20c. A ring member 64 rests on ledge 66 and provides support for the rotary valve member 70, as will be hereinafter more fully described.

Casing piece member 20c is a ring, adapted to fit between the outermost portions of casing pieces 20b and 20d. As shown in Figs. 1 and 3, casing pieces 20b, 20c, and 20d are securely joined together by a plurality of screws 61. Casing piece 20c contains a carefully machined axial aperture of circular cross section. This aperture in casing piece 20c is, however, slightly eccentric with respect to cylinder 21, as is best shown in Fig. 3. As previously stated, casing piece 20c contains a vertical bore 60, from which horizontal ports 69 communicate with the eccentric aperture. Spaced around said eccentric aperture in a clockwise direction from ports 69 are a plurality of ports 68, providing communication between the aperture and an arcuate channel 67 in casing piece 20c. Channel 67 is adapted to register with a bore 81 which connects the lower surface of casing piece 20b with a cylindrical recess 82 cut into the side of casing piece 20b. Recess 82 is, near its lower end, internally threaded to receive a screw 83, the lower end of which is tapered and machined to cooperate with the bottom of recess 82 to form an adjustable valve. A port 84 is drilled from the outside surface of casing piece 20b to a point on the inner wall of recess 82 near the bottom thereof.

A pair of ports 85 run from the outside of casing piece 20d to the bottom of recess 62, joining recess 62 at points immediately adjacent the inner wall thereof. Another pair of ports 86 connect the outside of casing piece 20d with the central cylinder 21 slightly above the top of closure member 22.

A port 87 in the outer wall of casing piece 20d provides an air vent at the side wall of recess 62 near the bottom thereof; a similar port 88 provides an air vent for recess 62 at a point slightly below the ledge 66.

A rotary valve member 70, circular in cross section and machined to fit snugly the inner wall of recess 62, occupies recess 62. The upper portion of member 70 has an annular shoulder which rests on ring 64. Vertical radial slots 72 are cut into the wall of the annular shoulder at the top of valve member 70 at intervals of 120° around the circumference, as best shown in Fig. 3. Carried within slot 72 are vanes 73. The width of each of the vanes 73 is slightly less than the depth of one of the slots 72, and each vane is equipped with a leaf spring (not shown) pressing against the bottom of the slot in which the vane rides, urging the vane outward against the wall of the aperture in casing piece 20c.

As shown in Fig. 3, the diameter of the aperture in casing piece 20c is slightly greater than the diameter of the annular shoulder on valve member 70, and the aperture in piece 20c is eccentric with respect to the annular shoulder on member 70 so that member 70 is tangent to the wall of the eccentric aperture at point 71, while at other points around its circumference a gap exists between member 70 and the wall of the eccentric aperture in casing piece 20c.

Around the inner surface of rotary valve member 70 are cuts 74 and 75. The pair of cuts 74 form recesses which run from the upper edge of valve member 70 downward along its inner surface for a distance about one-fifth of the length of valve member 70. In the embodiment of the invention shown in the drawings, the width of the cuts 74 is equal to an arc of 30° measured along the inner circumference of valve member 70. The pair of cuts 75 form recesses in the inner wall of valve member 70 running from the bottom thereof to a point slightly above the lowest portion of cuts 74. In the embodiment shown, cuts 75 have an angular width of 80° and are oriented with respect to cuts 74 such that their adjacent walls are separated by an arc of ten degrees. Each set of cuts 74 and 75 are symmetrically arranged around the inner circumference of valve member 70, as shown in Fig. 3. A pair of ports 76 are cut through the inner wall of casing piece 20d in a position to register successively with cuts 74 and 75 as valve member 70 is rotated. Bores 59 and 59a, heretofore described, register with the upper ends of cuts 74 when the rotary valve member is in the appropriate angular position, as shown in Fig. 1.

The choice of the angular width of the respective cuts 74 and 75 is a design factor of great importance in this invention; the means by which the appropriate widths may be determined in designing any given tool will be hereinafter described.

Near the lower end of rotary valve member 70, in the portion shown in elevation in Fig. 1, a curved slot 77 is cut on the outer surface entirely around the circumference of the valve member 70. The exact shape of this slot 77 is likewise a design factor which in particular cases might be somewhat different from the shape shown in Figs. 1 and 4; in the normal case, however, the slot will have a conformation roughly approximating that of a sine function of period equal to half the circumference of the valve member 70, and its position relative to the cuts 74 and 75 will normally be substantially that illustrated in Fig. 4.

A ring member or vibration compensating member 78 is slidably carried in recess 62 between the outer wall of that recess and the outer wall of valve member 70; ring member 78 carries on one side, in a suitable aperture, a pin 78*a* which engages slot 77, and in a corresponding position on the other side ring member 78 carries a pin 78*b* which passes entirely through ring member 78, engaging at its inner end the slot 77 and engaging at its outer end the vertical slot 79 in the casing piece 20*d*. It will be seen that as a result of the construction described, ring member 78 will oscillate in the axial direction as rotary valve member 70 rotates, the pin 78*b* and slot 79 preventing its rotation.

Cuts 75 register at their bottom ends with the ports 85 heretofore described.

The anvil 27 carries, in the plane at right angles to the section shown in Fig. 1, an axial slot 91, as shown in Figs. 2 and 3; and a pin 92, carried by the casing piece 20*d*, cooperates with slot 91 to prevent anvil 27 from rotating, while permitting its free movement in the axial direction to the full extend permitted by slot 91. The upper end of slot 91 is chosen to limit downward movement of anvil 27 at the point where the lower end of anvil 27 comes into contact with closure member 22. The lower end of slot 91 is placed at a position such that the anvil can move upward to a point somewhat above the position in which it is illustrated in Fig. 1, as shown in Fig. 11.

In the plane shown in Fig. 1, anvil 27 carries in its outer wall a pair of axial slots 93, extending from a point near the bottom of anvil 27 to a point near the top thereof. Each of the slots 93 is connected with the upper surface of anvil 27 by a bore 94 extending diagonally upward from the upper end of slot 93 and providing communication between the slots 93 and the space above anvil 27.

The upper and lower limits of slots 93, as will be hereinafter described in more detail, define the limits of the useful force-free drift range of anvil 27 and spike 25. The slots 93 register, when the anvil 27 is appropriately positioned, with the ports 76. The upper ends of slots 93 are placed at a point on anvil 27 such that communication between ports 76 and slots 93 ceases when the lower edge of the anvil drops below ports 86. The anvil is shown in that position in Fig. 9.

Operation of the Figure 1 embodiment

When the two sources of pneumatic pressure connected to hoses 49 and 50 are in operation, the constant pressure is transmitted through hose 50, coupling member 46, and bell chamber 32 to the upper end of cylinder 21 and is accordingly impressed on the upper end of piston 29. If the spike is not being pressed against a work-object, this pressure on the top of piston 29 will cause the piston, anvil, and spike to descend in cylinder 21 to their lowest position, as shown in Fig. 10.

To place the tool in readiness for operation, the spike is placed against the work-object and sufficient force imposed on the handles 33 and 34 to push casing 20 down, relatively raising against the aforesaid constant pneumatic pressure, the spike, anvil, and piston to a point near their maximum upward position—for example, to the position shown in Fig. 1. Thereupon the operator closes his hand over lever 38, causing valve rod 43 to rise and permitting high-pressure air to pass from hose 49 into the vertical bores 59 and 59*a*. The high-pressure air passes from bore 59*a* through the branch bore 59*b*, bore 60, and ports 69, and valve member 70 is thereupon placed in rotation, since the structure at the upper end of valve member 70 constitutes a rotary-vane motor. The path of the air within the eccentric chamber contained in casing piece 20c is from ports 69 around the chamber in a counter-clockwise direction as viewed in Fig. 3 and out through ports 68, bore 67, bore 81, and past valve member 83 to outlet port 84. The rate at which air can traverse this path can be adjusted by appropriate adjustment of valve member 83, so that the speed of rotation of the rotary valve 70 is at all times within the control of the operator.

When the rotary valve reaches the point in its counter-clockwise rotation at which cuts 74 register with ports 76—such position being hereinafter called "the 0° position"—high-pressure air flows from bores 59 and 59*a* into slots 93 and bores 94 to the space above anvil 27. Air under high pressure continues to flow in this path so long as cuts 74 register with ports 76—that is, throughout 30° of rotation of valve member 70. The piston 29 is accelerated upward under a substantially constant force during this interval of time. The position of the piston and rotary valve at the 30° rotation point is illustrated in Fig. 5. As the rotary valve 70 passes the 30° position, the air from bores 59 and 59*a* is cut off, and for the next 60° of rotation no air can either enter or escape from the space between anvil 27 and piston 29.

During the 30° to 90° portion of the rotation of valve member 70, the air trapped between anvil 27 and piston 29 expands and thus continues to force piston 29 upward, although its rate of acceleration begins to decrease with the decreasing pressure of the expanding air as soon as the 30° point is passed.

Design factors are determined so that by the time the rotary valve reaches its 90° position, the air trapped between piston 29 and anvil 27 will have expanded to substantially atmospheric pressure and the piston wil have slowed to a stop. It will be understood that the piston 29 will begin to slow down as the pressure on the under side of the piston approaches and drops below the constant value of pressure which is imposed on its upper side, but the piston will continue to rise, with negative acceleration, until its accumulated upward momentum has been overcome by the force on the top of the piston. If the rotary valve 70 has been properly designed with respect to the relative pressures of the two pressure sources, the piston will come to a stop at its point of maximum upward movement at a time when the air below the piston is at substantially atmospheric pressure. The position of the piston and the rotary valve at the 90° position is shown in Fig. 6.

When the rotary valve passes the 90° position, the cuts 75 register with ports 76, and the constant force imposed on the top of piston 29 there-

upon causes the piston to start downward under constant acceleration, the air before it being vented as it drops by passage through bores 94, slots 93, ports 76, cuts 75, and ports 85. This condition continues for about 70° of rotation, at the end of which time piston 29 will strike the upper surface of anvil 27 with a heavy impact. A period of 10° to allow for late impact is followed by a "settling" period of 10° to permit piston 29 to come to rest after fully transmitting its momentum to anvil 27 and spike 25, which in turn transmit the impact to the work-object. Fig. 7 shows the positions of the piston, anvil, and rotary valve as the valve approaches the 160° point of rotation, just as the piston strikes the anvil and 10° before the exhaust path is closed off by the movement of cuts 75 beyond the point of registration with ports 76.

As the valve 70 passes the 180° point, a new cycle of operation similar to that just described is commenced. As a result of the impact from the piston, however, anvil 27 and spike 25 will normally have been driven downward slightly into the work-object, and as a result the points of maximum upward and downward movement of the piston on the second stroke will be slightly lower in each case than on the first stroke. This shift, however, it will be seen, does not affect in any way the operation of the tool. The stroke is the same length as before, and the valve operation is identical to that described for the first cycle.

This course of operation can continue, with the anvil and spike drifting downward, by penetrating the work-object, with each piston impact until the anvil has "drifted" down to that point where further downward movement will cause the upper ends of slots 93 to pass below the ports 76. Fig. 8 shows the tool at the 90° position just as the piston is about to start downward, with the anvil at that point where the upper ends of slots 93 are barely in registration with ports 76.

When the piston drops from its Fig. 8 position to that shown in Fig. 9, it drives the anvil downward by a sufficient amount to place the upper ends of slots 93 below the ports 76, and the operation of the tool will thereupon cease, since there is no longer any means for the high-pressure air from hose 49 to reach the space above anvil 27.

In the examination of Fig. 9, it will be noted that the last impact of the piston has driven the spike 25 downward to its point of maximum downward movement, where shoulder 26 is resting on closure member 22. The more massive anvil 27, however, cannot strike the closure member 22 suddenly, since, with its downward movement, it has cut off the vent ports 85, thereby creating an air cushion between anvil 27 and closure member 22.

Fig. 10 shows the tool a few seconds after the position illustrated in Fig. 9; the air trapped between anvil 27 and closure member 22 has now had time to leak out around closure member 22 and anvil 27 has now moved to its position of maximum downward movement wherein it also rests on closure member 22. It will be noted that the lower ends of slots 93 now register with ports 85, permitting the venting of any residual air which may remain in the space between piston 29 and anvil 27.

Fig. 11 illustrates one of the important features of this invention which makes for convenience and ease in operation—namely the feature by which the constant pneumatic pressure in the upper part of the cylinder can be employed to assist the operator in extracting the impacted spike from the work-object, e. g. reinforced concrete, which in some instances will impose strong frictional resistance to the removal of the spike. In such a case, when the spike has penetrated the work-object to the full extent desired, the operator releases valve handle 38, thereby cutting off the air from the high-pressure source, pushes downward on the handles 33 and 34 until the tool casing 20 has been forced through its full possible descent into the position shown in Fig. 11, and then suddenly removes his force from the handles. The constant pneumatic pressure at the top of bell chamber 32 symbolized by the three arrows in that position in Fig. 11, will thereupon accelerate the tool casing 20 upward, causing closure member 22 to strike upon shoulder 26 on spike 25 with a powerful upward impact, as indicated in Fig. 9, now considering spike 25 to be stationary and casing 20 to be in upward motion, thus knocking the spike out of the material in which it is impacted. If the first blow fails to extract the spike, this process can be repeated until successful so that in no case will it be necessary for the operator to pull upward on the tool against the opposition of the work-object. However, since in the average tool design, this upward impact in energy terms will be about equivalent to a 6-foot downward blow with an 8-pound sledge, it may be expected that, in nearly every case, the spike will be successfully extracted by the first such upward impact.

It should also be noted that when the tool is being carried from place to place by the handles, the anvil, spike, and piston will be forced by the constant pressure in the upper portion of cylinder 21 to assume their extreme downward position, with the result that slots 93 will not register with ports 76 and accidental actuation of lever 38 cannot cause the tool to start operating inadvertently. This feature is most important from the standpoint of safety, since one of the common source of accidents with conventional pneumatic tools is their being accidentally actuated during carriage from one place to another on a job.

The preceding description of the operation of my invention has assumed that throttle valve 83 had been set to cause rotation of valve member 70 at the desired speed. The effect of variation in the speed of rotation of valve 70, accomplished by adjustment of throttle valve 83, is to change the frequency of impact and, simultaneously, the length of the piston stroke. If valve 70 is made to rotate more slowly, a greater quantity of high-pressure air is admitted to the space between anvil 27 and piston 29 during each intake interval; as a result, piston 29 will rise higher before its accumulated momentum is overcome by the constant force operating on it from above, and thus on its down-stroke the piston will travel further and have greater velocity at the time it strikes anvil 27. Conversely, if the speed of rotation be increased, the piston stroke will be shortened and the frequency of impacts increased. Thus this feature accords to my invention a degree of versatility far greater than that of any tool made by conventional methods in which the lenght of the piston stroke is fixed. Adjustment of the speed of rotation of valve 70 will not affect the synchronism of the inlet, expansion, and exhaust portions of the cycle, since this is controlled entirely by the relative pressures of the

two pressure sources and the design of cuts 74 and 75 on the rotary valve 70.

During the operation of my invention as described in the foregoing paragraphs, the ring member 78 oscillates in the axial direction during the rotation of rotary valve member 70; its function is to compensate for small changes in the pressure above the piston 29 and for the effects of friction between the piston and the wall of cylinder 21, and, by such compensation, to reduce the net vibration of the casing substantially to the vanishing point.

It will be recalled that the theoretical discussion of the conditions under which a pneumatic tool would be entirely free from casing vibration reached the conclusion that the force on the top of the piston should be at all times constant. Likewise, that theoretical treatment did not take into account the effects of friction between the piston and the cylinder wall.

It will be apparent that as piston 29 rises in cylinder 21, it reduces the volume within which the air above the cylinder is confined, and as a result a very slight increase in pressure may be expected. This increase will be very small if the pressure source to which hose 50 is connected has good regulation, but even so it will, if not compensated, cause a slight amount of casing vibration. Similarly, a certain amount of friction between piston 29 and the wall of cylinder 21 is unavoidable in a practical tool, and that friction likewise tends to produce slight casing vibration.

It will be seen that as the piston rises the increased pressure above piston 29 will tend to accelerate the casing upward and at the same time the energy transmitted to the cylinder wall by friction will likewise tend to accelerate the casing upward. During the down-stroke of the piston, these two effects operate in the reverse direction.

In order to compensate these forces tending to cause vibration, I have provided the ring member 78. The curved track 77 should be cut so as to cause upward acceleration of the ring member 78 during the time that the piston is rising and to cause downward acceleration of member 78 during the down-stroke of piston 29. It will be seen, by study of Fig. 4, that this result is achieved by the track 77 as thereon illustrated. At the 0° rotation point, when the piston starts upward, the member 77 is moving downward but its rate of downward travel begins to diminsh as the 0° point is passed. Thus the downward acceleration of the ring member 78 is negative, which is the equivalent of positive upward acceleration. This positive upward acceleration continues until the 90° point is reached, at which time the member 78 is moving upward at maximum velocity, and during the down-stroke of the piston, from the 90° to the 160° points of rotation of valve member 70, the acceleration of the ring member 78 is positive in the downward direction. It will be understood that the motion of the ring member between the 180° point and the 360° point of rotation of valve member 70 will repeat the 0° to 180° cycle.

The positive upward acceleration of the ring member 78 during the up-stroke of piston 29 causes a reaction on the casing 20 tending to accelerate it downward, thereby counteracting the tendency to upward acceleration caused by the friction and pressure variation heretofore mentioned. Similarly, during the down-stroke of the piston, the reaction on the casing caused by the positive downward acceleration of member 78 tends to accelerate the casing upward thus compensating for the tendency to downward acceleration caused by friction and pressure change during the piston's down-stroke.

In the design of a particular tool, well-known methods of vibration analysis may be used to determine the ideal shape to be given to track 77; its shape will normally approximate that shown in Fig. 4.

I have learned that the use of the ring oscillator 78, properly guided by track 77, will reduce substantially to the vanishing point casing vibration in my invention, with the result that my invention can be operated at full power with only barely perceptible vibration being transmitted to the operator's arms from the casing.

I desire to emphasize that the casing vibration produced by my invention, even in the absence of the compensating means just described, is only a small fraction of the vibration inherent in pneumatic tools made according to the prior art.

The design of the cuts 74 and 75 in rotary valve member 70 must take into account the magnitudes of the pressures supplied by the pressure sources connected to hoses 49 and 50 respectively. The down-stroke of the piston will normally consume a time equivalent to that required for 70° to 80° of rotation of the valve member 70; hence cuts 75 are normally positioned to begin at the 90° and 270° points respectively and to end at approximately the 170° and 350° points.

Similarly, the cuts 74 will normally be commenced at the 0° and the 180° points respectively; the gaps of 10° or thereabouts between the end of the cuts 75 and the beginning of cuts 74 are desirable to provide a "rest period" after each down-stroke within which the piston can fully dissipate its kinetic energy on the anvil and come to a rest before starting the next upward stroke. Accordingly, the major design consideration in constructing rotary valve member 70 lies in determining the proper angular width to be given to cuts 74. It will be understood that for any given speed of rotation the quantity of air injected into the space between anvil 27 and piston 29 will be proportional to the width of cuts 74 and to the pressure of the source connected to hose 49. Given particular values of pressures for the two pressure sources, it is possible to calculate, from well-known compressed air formulas, the appropriate angular width for cuts 74, the object of the calculation being to provide for admission, during the inlet interval, of a quantity of air such that it will, during the expansion interval, raise the piston against the constant force above it and cause its upward velocity to approximate zero at the time cuts 75 register with ports 76 to begin the down-stroke. The valve illustrated in Figs. 1 and 4 has cuts 74 and 75 designed for operation with pressure sources having pressures approximately equal to fifty pounds per square inch in absolute pressure as the constant pressure above the piston and one hundred pounds per square inch absolute as the pressure of the source connected to hose 49.

While exact adherence to theoretically ideal values for the position and width of cuts 74 and 75 is desirable to achieve maximum utilization of the energy of compression in the air from hose 49, a considerable range of latitude in these respects can be permitted without affecting seriously the operation or efficiency of my invention.

It should be noted that the constant pressure source connected to hose 50 has no work to perform except to overcome leakage and thereby

to maintain the pressure in hose 50 at a constant value. The constant pressure source, therefore, can be a compressor of relatively small size. The pressure source connected to hose 49 can in practice be a compressor of perhaps one-third the horsepower rating normally used with a pneumatic tool of conventional design. The greatly improved efficiency of my invention over pneumatic tools of the prior art permits the use of much smaller compressors for a given task than could be used with previously existing pneumatic tools.

The Figure 12 modification

Fig. 12 shows, in a fragmentary view showing the upper portion of a pneumatic tool according to my invention, a modification whereby a means is provided in the tool itself for holding constant the pneumatic pressure in the upper end of cylinder 21 above piston 29.

In the Fig. 1 embodiment, it will be recalled, it was assumed that the pneumatic pressure supplied to the tool through hose 50 would be held at a constant value. In practice, this would normally be accomplished by providing a relatively large air reservoir on the hose 50 source, coupled with a suitable pressure-regulating means of conventional design. In some cases, however, as where hose 50 is necessarily of considerable length, a regulating means on the tool itself is desirable.

Fig. 12 shows the bell chamber 32 as in the Fig. 1 embodiment, and on one side of chamber 32, below handle 33, a cylindrical coupling member 121 is provided, coupling member 121 having internal threads adapted to receive a constant-pressure valve 122.

Valve 122 comprises a cylindrical casing having at its inner end a machined valve seat 123. A poppet valve member 124 is carried within the cylindrical casing, the shaft of valve member 124 being slidably carried in a central aperture within a cap member 125. The outer end of the cylindrical casing of valve 122 has internal threads adapted to cooperate with corresponding threads on cap 125, as shown. A spring 126 is compressed within the valve cylinder so as to bear against the head of poppet valve member 124 and cap member 125. An escape port 127 is provided on the under side of the cylinder of valve 122. It will be seen that valve 122 will, according to the force exerted by spring 126, yield at a predetermined pressure and permit air to escape from the interior of bell chamber 32, thus maintaining the pneumatic pressure within bell chamber 32 at a constant critical value. Should it be desired to change the value of the pressure within bell chamber 32, the tension on spring 126 may be varied by appropriate rotation of cap 125.

The use of a pressure-regulating means on the bell-chamber 32, as shown in Fig. 12, would, if desired, permit the use of a single hose and single pressure source, since hose 50 and the constant-pressure source attached thereto could be replaced by a restricted passage into the bell chamber from hose 49, the valve 122 being then effective to maintain the pressure within bell chamber 32 at a substantially constant value lower than the pressure within hose 49.

The Figure 13 modification

Fig. 13 shows, somewhat diagrammatically, a modification of the Fig. 1 embodiment which can be made to function in a substantially similar manner without the necessity for two sources of pneumatic pressure.

It has been shown that a condition necessary to be met in a pneumatic tool having no casing vibration is that the force imposed on the top of the piston be at all times constant. In Fig. 13, this condition is approximated by the use of two coil springs adapted to urge the piston downward, in lieu of the constant pneumatic pressure bearing upon the top of the piston in the other embodiments of the invention herein shown.

In the Fig. 13 embodiment, a pair of slots 131, positioned on opposite sides of casing 20, run axially downward from a point near the top of the casing as shown. A piston 139 is carried in the cylinder 21; near the upper end of piston 139 a transverse bore provides accommodation for a rod 132 which passes through the piston and the slots 131, extending therebeyond on each side. To the ends of rod 132 there are affixed the upper ends of a pair of coil springs 133 which are coiled downward around casing 20 and secured at its lower end by clamps 134 which are suitably affixed to the casing 20. The springs 133 may be of the pre-stressed type wherein the force imposed is largely independent of length over a substantial range of expansion.

In the Fig. 13 embodiment, the high-pressure pneumatic intake hose 49 is coupled to casing 20, as shown, at a point below the lower ends of springs 133. The high-pressure inlet bore 59 may run from the coupler 135, which receives hose 49, to the valve mechanism as in the other embodiments of the invention, and to carry the compressed air from bore 59 to the opposite side of the casing, an annular recess 136 is provided in the wall of casing piece 20*b*. Annular recess 136 is adapted to register with bore 59*a* on the side of the casing opposite bore 59. Closure for annular recess 136 is provided by shoulder 137. Shoulder 137 is homologous to shoulder 65 in the Fig. 1 embodiment, it being, like shoulder 65, an upward extension of casing piece 20*d*.

Any suitable means (not shown) may be provided to enable the operator to turn on the compressed air fed through hose 49 for operation of the tool and to turn it off when desired.

While it has been found that springs 133 cannot be made to maintain the force on the upper side of piston 139 as nearly constant as will the pneumatic pressure source employed in the other embodiments, suitable compensation for variations in that force can be achieved by increasing the mass of ring oscillator 78, with the result that the embodiment of Fig. 13 can be made to operate with only slightly more vibration than the embodiments of Figs. 1 and 12, and in all cases very much less vibration than is characteristic of tools made according to the prior art.

While I have in this specification, for purposes of illustration, described in considerable detail certain specific embodiments of my invention, it will be understood that many changes in details can be made by those skilled in the art without departing from the spirit of my invention.

I claim:

1. In combination, a cylinder, a piston reciprocable therein, means for establishing between the cylinder and the piston throughout the cycle of piston reciprocation a substantially constant axial force, said constant force being the only substantial axial force developed between the cylinder and said piston during said cycle, and valve means adapted to connect a source of fluid pressure to said cylinder operative to admit compressed gas thereto to impose axial force on said piston and to cause movement thereof in opposi-

tion to said substantially constant force, and to shut off and vent such gas thereafter.

2. In a percussive tool, a cylinder, a piston reciprocable therein, means for establishing throughout the cycle of piston reciprocation a substantially constant force on the piston tending to accelerate the same in one direction, fluid-pressure means for reciprocating the piston by imposing thereon a varying force alternately greater and less than said substantially constant force, a residual-vibration-compensating member mounted on said cylinder for reciprocation along an axis parallel to the axis of the cylinder, and means operative to reciprocate the compensating member in a continuously predetermined relation with the reciprocating motion of the piston.

3. A combination according to claim 2 which includes a vibration-compensating inertia member mounted on the cylinder for reciprocation along an axis parallel to the axis of the cylinder, and means operative to reciprocate the inertia member in a continuously predetermined relation with the reciprocating motion of the piston and in substantially quadrature phase thereto.

4. A pneumatic tool having a cylinder, a piston reciprocable therein, means for establishing a substantially constant force on said piston tending to accelerate the same in one direction, means adapted to connect a source of pneumatic pressure to the cylinder including a rotary valve operative alternately to admit gas under pressure to the cylinder to create a force opposing said substantially constant force and to vent said cylinder, a residual-vibration-compensating inertia member mounted coaxially with the rotary valve and means coupling the rotary valve to the inertia member operative to reciprocate the same in controlled time relation with the rotation of the valve.

5. A pneumatic tool comprising a cylinder, a member slidably carried in one end of the cylinder and adapted to transmit force to an external object, means operative to limit the outward movement of said slidable member, a piston freely reciprocable in the cylinder between the other end thereof and said slidable member, means closing off from the atmosphere said other end of the cylinder, means for introducing into said other end of the cylinder a super-atmospheric gas pressure operative to establish between said piston and said other cylinder end throughout the cycle of piston reciprocation a substantially constant force tending to accelerate the piston away from said other end of the cylinder, means for introducing between said slidable member and the end of the piston adjacent thereto a varying gas pressure operative intermittently to move said piston against said substantially constant force, and valve means operative responsively to movement of the slidable member to vent the space between the piston and the slidable member when the slidable member approaches said limiting means.

6. A pneumatic tool comprising a cylinder, a work member slidably carried in one end of the cylinder and adapted to transmit force to an external object, an anvil slidably carried within said cylinder adapted to rest on said work member, said anvil having substantially the same cross-sectional area as the cylinder, a freely reciprocable piston in the cylinder between its other end and the anvil, means for closing off from the atmosphere said other end of the cylinder, means for establishing between said other end of the cylinder and the end of the piston adjacent thereto throughout the cycle of piston reciprocation a super-atmospheric gas pressure of substantially constant magnitude, and means for introducing intermittently between the piston and the anvil compressed gas at a pressure greater than said substantially constant pressure.

7. A pneumatic tool comprising a cylinder, a work member slidably carried in one end of the cylinder and adapted to transmit force to an external object, a pneumatic-seal element within said cylinder adapted to rest on said work member, means carried by the cylinder adapted to limit outward movement of the work member and the seal member, a freely reciprocable piston in the cylinder between the other end of the cylinder and the seal member, means for closing off from the atmosphere said other end of the cylinder, means for introducing intermittently between the piston and the seal member compressed gas for moving the piston away from said seal member, and valve means operative responsively to movement of the seal member to vent the space between the piston and the seal member when the seal member approaches said limiting means.

8. A pneumatic tool comprising a cylinder, a member slidably carried in one end of the cylinder and adapted to transmit force to an external object, means operative to limit the outward movement of said slidable member, a piston freely reciprocable in the cylinder between the other end thereof and said slidable member, and means for introducing between said slidable member and the end of the piston adjacent thereto a varying gas pressure operative intermittently to move said piston, said last-mentioned means including a pair of valves operative responsively to movement of the slidable member, the first of said valves being operative to cut off said varying gas pressure from the space between the slidable member and the piston when the slidable member approaches said limiting means and the second of said valves being operative to vent said space when the slidable member approaches said limiting means.

9. A percussive tool comprising a member adapted to be hand-held, spring means seated on said member, a blow-striking element of substantial mass seated on said spring means and mounted for oscillatory motion along a single axis with respect to said member, means for maintaining said blow-striking element in oscillatory motion relative to said member, a vibration-compensating part having substantial mass and mounted on said member for oscillatory movement relative thereto along said axis, and means for oscillating said part with respect to the hand-held member in timed relation to the oscillatory motion of the blow-striking element.

10. In a percussive tool, a cylinder member, a free piston reciprocable therein, means for reciprocating the same, a vibration-compensating mass member constrained by said cylinder member for reciprocation thereon in an orientation coaxial with the axis of the piston, and means operative to reciprocate said mass member in controlled time relation with the reciprocating motion of the piston and to maintain at substantially all times during tool operation reactive force on the cylinder opposite in direction to the variable component of the force on the cylinder derived from movement of the piston.

11. In a percussive tool, a rigid member adapted to be hand-held, an impact-delivering member constrained by said rigid member to

movement in a straight line relative thereto but free to move therealong in either direction, a vibration-compensating mass member supported by the rigid member and provided with a locus thereon for reciprocatory movement along the line of movement of the impact-delivering member, means for reciprocating the impact-delivering member relative to the rigid member, and means controlled in synchronism with the last-mentioned means operative to accelerate the mass member according to a continuously predetermined pattern throughout the full cycle of reciprocation of the impact-delivering member.

12. In a percussive tool, a rigid member adapted to be hand-held, an impact-delivering member constrained by said rigid member to movement in a straight line relative thereto but free to move therealong in either direction, a vibration-compensating mass member supported by the rigid member and provided with a locus thereon for reciprocatory movement along the line of movement of the impact-delivering member, means for reciprocating the impact-delivering member relative to the rigid member, and means controlled in synchronism with the last-mentioned means operative to reciprocate the mass member according to a predetermined smoothly varying velocity pattern throughout the full cycle of reciprocation of the impact-delivering member.

13. In a percussive tool, a casing member adapted to be hand-held, a plurality of mass members mounted for oscillation thereon along a single axis, at least one of said members being a blow-striking piston constrained within the casing member, and means for reciprocating said members in a continuously predetermined relation to one another operative to maintain substantially constant at all times during tool operation the total force imparted to said casing by said mass members.

14. In a percussive tool, a cylinder, a free piston reciprocable therein, a vibration-compensating member mounted on said cylinder for reciprocation along the axis of the cylinder, means for reciprocating the piston, and means operative to reciprocate said member in a continuously predetermined relation with the reciprocating motion of the piston, said relation being characterized by a phase intermediate coincident phase and opposed phase.

15. In a percussive tool, a cylinder, a free piston reciprocable therein, a vibration-compensating member mounted on said cylinder for reciprocation along the axis of the cylinder, means for reciprocating the piston, and means operative to reciprocate said member in a continuously predetermined relation with the reciprocating motion of the piston, said relation being characterized by substantially quadrature phase.

16. In a percussive tool, a cylinder, a piston reciprocable therein, means for establishing throughout the cycle of piston reciprocation a continuously acting force on the piston tending to accelerate the same in one direction, a fluid-pressure means for reciprocating the piston by imposing thereon a varying force opposite in direction and alternately greater and less than said first-mentioned force, a vibration-compensating member having substantial mass and mounted on said cylinder for reciprocation along the axis thereof, and means operative to reciprocate said member in a continuously predetermined relation with the reciprocating motion of the piston, said relation being characterized by a phase intermediate coincident phase and opposed phase.

17. The combination of claim 1 in which said cylinder is equipped with handles and is provided with a work member slidably carried in one end thereof, said work member being adapted to transmit force to an external object.

18. The structure of claim 17 in which an anvil member is slidably carried in said cylinder adjacent the end thereof toward said work member and is adapted at one end to receive blows from said piston and at its other end to rest upon said work member, and in which said valve means is adapted to connect a source of fluid pressure to said cylinder adjacent the end thereof toward said anvil.

19. The structure of claim 4 in which said means coupling the rotary valve to the inertia member comprises a generally sinusoidal cam track extending entirely around the outer surface of said rotary valve and a pin carried by said inertia member and engageable with said cam track, and in which means are provided to constrain said inertia member against rotation.

20. The precussive tool of claim 9 wherein said means for oscillating said part includes a rotary member arranged coaxially with the axis of said blow-striking element, and a cam and cam follower are operatively arranged with said rotary member and said part.

21. The structure of claim 10 in which said mass member is ring shaped and is circumjacent said cylinder.

22. The structure of claim 13 in which said means for reciprocating said mass members includes rotary cam track means.

23. The structure of claim 13 in which said means for reciprocating the mass members includes fluid pressure means for reciprocating the blow-striking mass member, cam means for reciprocating another of said mass members, and a movable member, said fluid pressure means including a valve element carried by said movable member and controlling the motion of the blow-striking mass member and said cam means including a cam surface also carried by said movable member and controlling the motion of said last-mentioned mass member.

24. The structure of claim 13 in which said means for reciprocating the mass members includes fluid pressure energizing means and a single movable valve member controlling the respective motions of at least two of said mass members.

25. The structure of claim 14 in which said percussive tool is adapted for manual operation and equipped with handles therefor and is also equipped with an axially slidable work member adapted to deliver impact force to an external object and an axially slidable anvil member adapted at one end to receive blows from said piston and at its other end to rest upon said work member.

26. In combination, a cylinder, a piston reciprocable therein, mechanical elastic means producing a force operative axially between the cylinder and the piston tending to propel the piston in one axial direction and to cause the cylinder to recoil in the opposite axial direction, said force remaining substantially constant throughout the cycle of piston reciprocation and comprising substantially the entire axial force active between said piston and cylinder, and fluid-pressure means for applying to the piston in opposition to said first-mentioned force there-

upon a cyclically varying axial force serving to energize the reciprocatory cycle of the piston by varying in magnitude to alternately become greater than said first-mentioned force to overcome the same and propel the piston in the direction opposite to the aforesaid one axial direction, and less than said first-mentioned force whereby the piston is then propelled in that one axial direction.

27. A hand-held percussive tool comprising a cylinder, a piston reciprocable therein, a slidable member carried in one end of the cylinder and adapted to transmit to an external object impact force developed by arresting the movement of the piston, means for establishing throughout the cycle of piston reciprocation a substantially constant total axial force between the cylinder and the piston tending to propel the piston in the direction toward said slidable member and to cause the cylinder to recoil in the opposite direction, said constant force being substantially the entire axial force active between said piston and cylinder, and fluid-pressure means for applying to the piston in opposition to said constant force thereupon a cyclically varying axial force serving to energize the reciprocatory cycle of the piston by varying in magnitude to alternately become greater than said constant force to overcome the same and propel the piston in the axial direction away from said slidable member, and less than said constant force whereby the piston is then propelled in the axial direction toward said slidable member.

28. In combination, a cylinder and a piston reciprocable therein, fluid-pressure means producing a force operative axially between the cylinder and piston tending to accelerate the piston in one axial direction and to cause the cylinder to recoil in the opposite direction, said axial force produced therebetween remaining substantially constant throughout the cycle of piston reciprocation and comprising substantially the entire recoil-producing force acting upon the cylinder, and fluid-pressure means for applying to the piston a cyclically varying axial force intermittently exerting at certain cyclically repeated times a sufficient push on the piston in opposition to the said first-mentioned force thereupon to overcome the same and propel the piston in the direction opposite to the aforesaid one axial direction in which it is urged by said first-mentioned force, whereby the piston is at other cyclically repeated times positively accelerated in that one axial direction to assist in maintaining it in a state of reciprocating motion.

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