

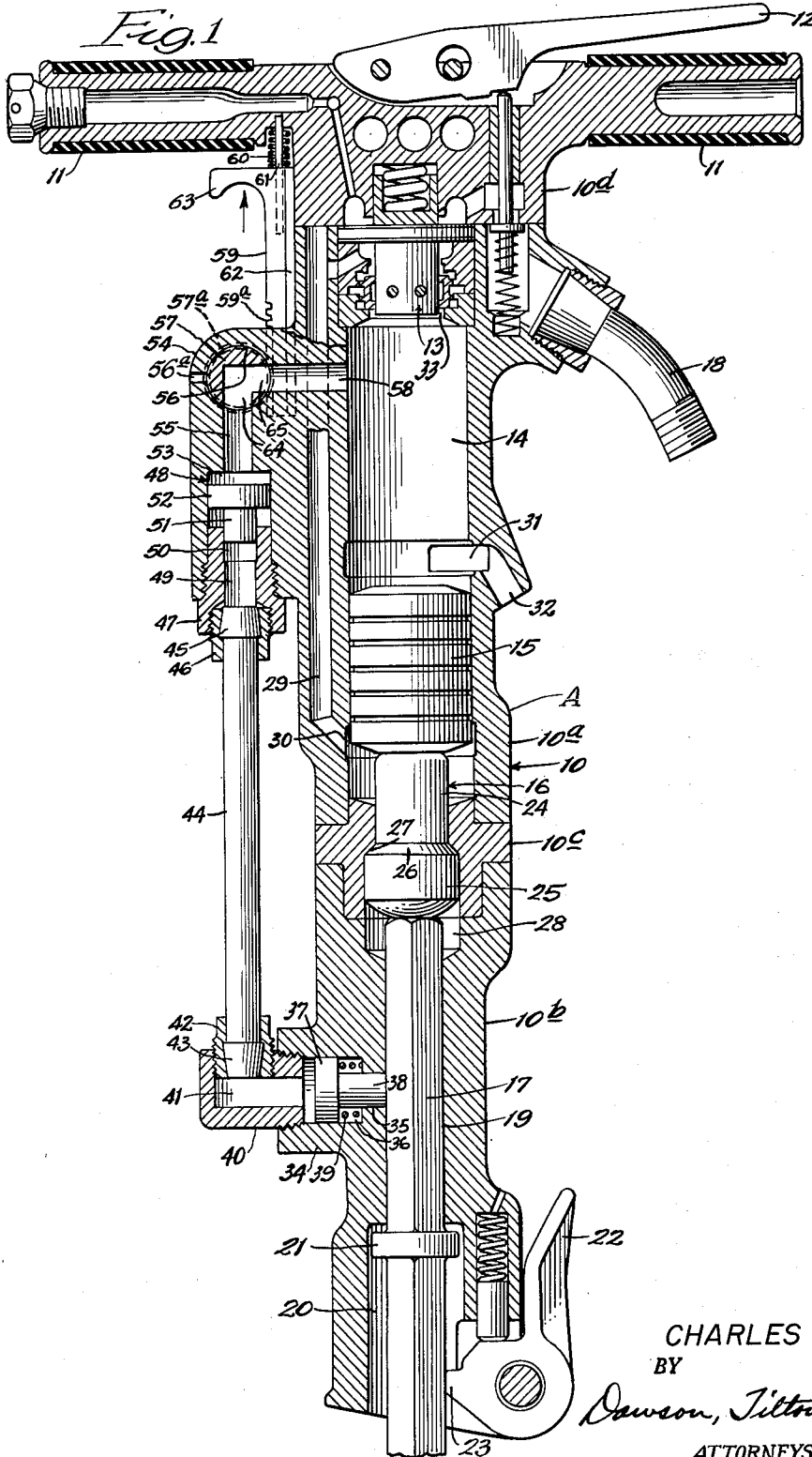
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PERCUSSION TOOLS

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PERCUSSION TOOLS

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This invention relates to percussion tools, and particularly to pressure fluid operated tools, such as pneumatic paving breakers, and the like.

It is well known that the operation of pressure fluid actuated percussion tools such as pneumatic paving breakers is attended with a great amount of vibration that is extremely fatiguing for the operator of the tool. The very existence of the vibration alone is sufficient to quickly tire a workman since the vibratory energy of the tool is absorbed by him, but in addition, in pneumatic tools the very cause of the vibration requires that the workman exert an active force against the tool, to the end that it is operated efficiently, to urge it into the material being broken. This pushing against the tool by the operator is in direct opposition to one direction of movement of the vibrating tool and, thus, the workman's burden is increased, adding to his fatigue and rapidly rendering his operation of the tool inefficient. Of course, a worker, due to lack of skill or contrariness, may not push against the tool with sufficient force to overcome the waste component of the forces and pressures causing the tool vibration, and his operation of the tool will be inefficient from the beginning.

Another phenomenon associated with pneumatic paving breakers and similar devices that is quite apart from the effects of vibration upon the workman but stems directly from the causes of vibration is a relatively low work efficiency inherent in such tools. That is, though the driving or working impact is repeated hundreds of times each minute, a portion of the impact force driving the spike of the tool is unproductive because of the mechanical relation between the driving anvil and the bottom closure of the hammer cylinder. When the head of the anvil is adjacent the bottom of the cylinder, the falling hammer traps a mass of air therebelow that has no escape and as the lower cylinder volume decreases with the falling of the hammer the pressure of the trapped air increases and a retarding force is exerted against the hammer so that not all of the kinetic energy of the hammer is converted into impact against the anvil.

It is accordingly an object of this invention to provide fluid pressure operated percussion tools wherein the operation thereof is substantially vibrationless. Another object of the invention is to provide in pneumatic paving breakers and similar tools means for substantially minimizing muscular fatigue of the tool operator by eliminating vibration from the operation thereof and by rendering unnecessary the exertion by the operator of an active pushing force against the tool, as is required for efficient operation of conventional paving breakers. A further object is that of improving the working efficiency and consequently the working speed of tools in the class described.

Still another object is to equip a pneumatic paving breaker with a friction brake automatically operable to lock the tool casing against movement with respect to the spike or work member during certain phases of the tool operation and thereby eliminate vibration. Still a fur-

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ther object is to prevent relative movement between the casing of the tool and the work member by means of a friction brake during phases of the tool operation while allowing free relative movement between the casing and work member during another phase of the operation to the end that pneumatic pressure is utilized to push the tool towards the work to obviate the customary need of the operator's push. Yet another object is that of providing a friction brake operating sequentially and in timed relation with the movement of a paving breaker actuating hammer and the application of fluid pressure thereto to move the tool casing with respect to the work member and tappet to establish a mechanical relationship therebetween whereby the hammer attains maximum momentum on the power stroke and maximum impact is imparted to the tappet and work member with the result that the working efficiency and working speed of the tool is maximized. Further objects and advantages will appear as the specification proceeds.

In the accompanying drawing, a longitudinal sectional view of a conventional paving breaker incorporating my invention is illustrated in Figure 1.

Illustrated in the drawing is a conventional pneumatic paving breaker A, having a casing 10, a pair of laterally-extending handles 11 at the top, an actuating valve handle 12 adjacent one of the handles 11, a valve assembly 13, a cylinder 14, a hammer or piston 15 mounted for reciprocating movement within the cylinder 14, an anvil or spike head 16 (usually referred to as a tappet) extending upwardly into the cylinder 14 and adapted to receive blows from the hammer 15, a spike or work member 17 equipped with the usual work point (not shown), and a fitting 18 for connecting a suitable supply of pressure fluid to the cylinder 14 through the valve assembly 13 and suitable passages in the casing 10.

Customarily the spike or work member 17 is hexagonal in shape and is slidably mounted within an appropriate passage 19 provided in the casing 10. The passage is enlarged at 20 to provide clearance for an annular flange 21 with which the work member is equipped. A pivotally-mounted, spring-biased key or retainer 22 having a protuberance or ear 23 extending into the enlarged passage 20 prevents the work member 17 from accidentally falling from the casing 10, while pivoting of the retainer in a clockwise direction withdraws the ear 23 from the passage 20, permitting free removal and replacement of the work member. For the purposes of this invention, the work member 17 and anvil or tappet 16 may be a unitary structure, but a damaged spike is economically replaced when the parts are separate since the work member need only be finished roughly while the anvil must have a fine finish.

The tappet 16 has a head 24 integrally formed with an enlarged body portion 25. A downwardly- and outwardly-sloping shoulder 26 connects the head 24 and body 25 and this shoulder is adapted to mate with a corresponding shoulder 27 formed near the upper end of an enlarged passage 28 provided in the casing 10 as an enlarged upper extension of the work member passage 19. The upper end of the passage 28 above this shoulder 27 is restricted and the head portion 24 of the tappet is slidably and snugly received therein while the body portion 25 is snugly received within the larger lower part of the passage 28. In the illustration, the casing 10 is shown in four parts—an upper or cylinder casing portion 10a and a lower casing portion or front head 10b joined by a connector member or tappet seat 10c, and a back head 10d. This construction facilitates assembly of the paving breaker and permits the tappet 16 to be easily positioned within the passage 28.

The valve assembly 13 is of usual construction and is automatically operable upon depressing of the valve handle 12 to alternately supply the pressure fluid to the upper

and lower portions of the cylinder 14 for reciprocating the hammer 15. In one position of the valve incorporated in the assembly 13, pressure fluid passes through the fitting 18 and to the lower end of the cylinder 14 through a longitudinally-extending passage 29. Thus the pressure fluid discharges through the orifice 30 adjacent the lower end of the cylinder 14 and pushes against the bottom of the hammer 15 to force it upwardly in the cylinder 14. The valve automatically operates upon the passing of the hammer 15 beyond the exhaust port 31, communicating with the exterior by passage 32 extending through the side wall of the casing 10, to reverse the flow of pressure fluid into the cylinder and to deliver it adjacent the upper end of the cylinder 14 through an annular inlet 33. Conversely, when the hammer 15 is moved downwardly and has passed the exhaust port 31 the valve in the assembly 13 reverses itself and delivers the pressure fluid through the passage 29 to the lower end of the cylinder while the pressure fluid in the upper portion of the cylinder is exhausted through the port 31 and passage 32. In the position illustrated, the work member 17 is resting upon or embedded in the work and the casing 10 is in its most downward position with respect to the work member, and the tappet and tappet seat shoulders 26 and 27 are therefore in abutting relation. It is obvious that with these parts in this relationship no air will be trapped below the hammer just before the moment of impact to retard its motion in the manner already explained, wherefore this is the ideal relationship between these parts to obtain their most efficient cooperation during demolition work. But it is never maintained (except through use of my invention) when a conventional paving breaker is in operation.

So far the structure described is conventional and well known in the art and a further description thereof in detail is believed unnecessary. In the operation of this structure, a workman grasps the handles 11 and positions the working end of the spike or work member 17 in proper position with respect to the material to be broken. The workman then depresses the valve handle 12 and pressure fluid is delivered alternately to the upper and lower ends of the cylinder 14 to actuate the hammer or piston 15. The control valve automatically reverses itself to bring about a reciprocating motion of the hammer 15 within the cylinder.

When the hammer 15 is positioned as illustrated in the drawing, the control valve will operate to connect the pressure fluid supply fitting 18 with the passage 29 and the actuating pressure fluid will enter the cylinder 14 through the lower port 30 and the hammer 15 will be driven upwardly. The moment the lower edge of the hammer 15 passes the lower edge of the exhaust port 31, the cylinder portion below the exhaust port is exhausted to atmosphere through the passage 32 and the control valve reverses itself and the pressure fluid is delivered to the upper end of the cylinder 14 through the supply inlet 33. The pressure fluid operating against the hammer 15 drives it downwardly and just prior to the impact of the hammer against the tappet head 24, the hammer passes the limits of the exhaust port 31 and the upper portion of the cylinder 14 is immediately exhausted to atmosphere. The impact between the hammer 15 and the tappet 16 is operative to drive the work member 17 a very small distance into the material being broken. The cycle is then repeated and the hammer 15 is driven upwardly in the cylinder 14 and then again downwardly into impact relation with the anvil 16. The cycle is repeated rapidly and occurs about 1200 times each minute in a typical tool. The rapid blows imparted to the work member 17 are effective to break the material upon which the work is being done.

It is believed that the invention may best be understood by first describing the operative features of the paving breaker that create the particular problems the present invention provides a solution for and such a description will now follow. When the hammer 15 has moved upwardly in the cylinder 14 and has passed the exhaust port

31 so that the flow of pressure fluid is delivered to the upper end of the cylinder, the pressure fluid therein exerts forces in all directions and is not only operative against the hammer 15 to drive it downwardly in the cylinder but is also operative against the upper end of the cylinder 14 and thereby lifts the casing 10, and since the work member 17 and tappet 16 are freely slidable within the casing, the casing moves upwardly with respect to these members and the tappet shoulder 26 descends relative to the tappet seat shoulder 27 causing a separation therebetween. On the second phase of the cycle of operation, immediately after the hammer 15 in moving downwardly passes below the exhaust port 31, thereby exhausting the upper portion of the cylinder, the hammer 15 strikes the tappet 16. The pressure on the lower end of the cylinder 14 resulting from the entry of the pressure fluid into the lower portion of the cylinder through the port 30 after the tappet has been struck by the hammer forces the casing downwardly. Obviously this up and down movement or reciprocation of the casing 10 with respect to the work member and tappet continues repetitiously during the movement of the hammer 15, and since the hammer movements are very rapid, the casing 10 reciprocates at a very rapid rate or vibrates. The vibration, as previously indicated, is absorbed by the workman who is operating the paving breaker and reacts upon him with strain and fatigue.

Moreover, a portion of the energy of the pressure fluid made available by the entrance of the fluid into the upper end of the cylinder 14 is wasted, since it acts to lift the casing 10 with respect to the work member and tappet and is as a result ineffective to push the hammer 15 downwardly. Therefore, if the paving breaker is to be operated efficiently and at top working speed, the workman must exert an active push or force against the casing 10 in an attempt to hold it against upward movement and in its most downward position in which, as already explained, retardation of the hammer by air trapped below it is prevented. A force or push of sufficient magnitude to fully accomplish this result cannot be applied through muscular effort of a workman since the recoil or upwardly-directed forces operating against the casing 10 in the manner described are in the nature of 400 pounds (after subtracting the casing weight, which however is only available to aid the workman when the paving breaker is operated in a downward direction). A workman is incapable of exerting such a push and the ideal operating condition is never established, wherefore not all of the kinetic energy created by the pressure fluid is utilized to drive the hammer 15 into the tappet 16. Further, the necessity of pushing against the handles 11 understandably fatigues the workman rapidly and his ability to push against the handles of the casing with a force of any real value decreases rapidly with the length of time he operates the paving breaker. Further, the requirement that an active force be exerted by a workman against the paving breaker means that the full extent of the vibratory movements of the casing must be absorbed by the workman and this further adds to his fatigue.

The undesirable feature of retardation of the hammer by air trapped below it is particularly accentuated when the paving breaker is operated in a horizontal position because the weight of the casing 10 is then made ineffective to assist the worker in applying pushing force, so that the casing 10 never completely restores itself to its ideal position relative to the work member 17 and tappet 16, and eventually only a very small portion of the tappet head 24 extends into the cylinder 14. To explain this retarding action in more detail, after a very short period of operation the upper end of the tappet head 24 is at a position slightly below the port 30. Therefore as the hammer 15 is driven downwardly, a mass of air is driven before it into the pocket in the lower portion of the cylinder below the port 30. As its volume decreases with the descent of the hammer the pressure of the trapped air of course increases and it exerts an upward force against

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the hammer 15 that tends to retard its descent. In other words, under these conditions the trapped air within the pocket tends to cushion the descent of the hammer 15 before it strikes the head of the tappet, thus reducing the force of the blow (as compared with its force when the upper end of the tappet head 24 extends above the port 30 in the ideal relationship of the parts already mentioned, and shown in the drawing). The driving force of the work member 17 is thereby lessened and the working efficiency and working speed of the paving breaker is considerably reduced.

My invention is operative to eliminate vibration and dispense with the necessity of an active downpush exerted by the workman against the paving breaker casing, as soon as the work member 17 becomes frictionally gripped by the material it has penetrated. In the case of concrete, a frictional force of from 500 to 2000 pounds may be holding the work member in the material. The principle of my invention is that of a friction brake that is cyclically operable in timed sequence with the rise and descent of the hammer 15 and the corresponding descent and rise of the casing 10 to lock the casing 10 and work member 17 (assumed to be locked in the material it has penetrated) so that relative movement of the casing with respect to the work member is prevented during the part of the cycle of operation when, as previously explained, the casing has a tendency to rise, but the casing is permitted to follow the work member downwardly during another part of the operative cycle. That is to say that upward movement of the casing is inhibited while downward movement of the casing is permitted, with the result that vibration is eliminated and an effective downward force of between 300 and 400 pounds—a force that no man can approximate—is utilized to push the casing downwardly and establish the ideal mechanical operative relationship between the casing and work member.

The structure by which this is accomplished consists of a boss or friction member housing 34 that is shown integrally formed with the lower casing portion or front head 10b but may be added to pneumatic tools already in use by any suitable means such as welding, etc. Axially aligned with the center of the boss 34 is a passage 35 communicating at one end with the spike or work member passage 19 and at its other end with an enlarged passage 36 that has a finished inner surface that slidably receives the enlarged head 37 of a friction member 38 in liquid-sealing engagement therewith. The reduced portion of the friction member 38 is slidably received in the passage 35 and is adapted to engage at its inner end a lateral face of the hexagonal work member 17. A coil spring 39 normally urges the friction member 38 and head 37 thereof in an outward direction so that the friction member does not interfere with the insertion of the work member 17 in the casing.

The outer end of the enlarged passage 36 is threaded and threadedly receives an end of an L-shaped fitting 40 having a passage 41 extending therethrough. It is desired that a fluid-tight connection exist between the boss 34 and fitting 40, and suitable gaskets or other arrangements may be utilized to prevent leakage of a fluid under pressure. In the same sense, a fluid-tight sliding engagement between the head 37 of the friction member 38 and passage 36 is required and any customary arrangement to accomplish this result may be used, including the accurately machined finish illustrated, or suitable O-ring structures, etc.

The upper end of the L-shaped passage 41 is threaded and threadedly receives a connector 42 having a suitable passage therethrough for receiving the flared head 43 of an elongated tube 44. The opposite end of the tube 44 is also flared as at 45 and it is received within an appropriate passage provided by a connector 46 threadedly received within the threaded outer end of a fitting 47. Since the tube 44 constitutes a part of a fluid pressure transfer means, it is desired that a fluid-tight coupling

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between the flared ends 43 and 45 and the connectors 42 and 46 be provided.

A pressure multiplier is designated generally with the numeral 48, and the multiplier is a further part of the fluid pressure transfer means. The pressure multiplier 48 includes the fitting 47 which has a longitudinally-extending passage 49 therethrough aligned with the tube 44 and communicating therewith. The upper end of the passage 49 is provided with an accurately- and finely-finished interior surface designated with the numeral 50. Slidably received within the upper end of the passage 49 is a piston 51 that sealingly engages the finished interior 50. The piston 51 is equipped at its upper end with an enlarged head 52 that is slidably received within a cylinder 53 provided in an enlargement or pressure multiplier casing 54 formed integrally with or otherwise rigidly secured to the casing 10. The lower end of the cylinder 53 is threaded interiorly and receives a threaded portion of the fitting 47. Suitable gaskets or sealing materials may be provided to insure a fluid seal between the fitting 47 and the threaded portion of the cylinder 53. Further, both the piston 51 and its head 52 should be in fluid-sealing relation with the passage portion 50 and the cylinder surface 53, respectively, and this may be accomplished by carefully finishing the engaging surfaces or by the use of O-rings or packing glands, etc.

A passage 55 communicates with the cylinder 53 and extends upwardly therefrom to communicate with a valve chamber 56 equipped with a valve 57. Extending from the valve chamber 56 at substantially right angles to the passage 55 is a passage 58 communicating with the cylinder 14 near the upper end thereof. Preferably the passage 58 opens into the cylinder 14 as close to the upper end thereof as is physically practical. The valve 57 may be equipped with a sector gear 57a and is actuated by a trigger 59 having a rack at its lower end equipped with teeth 59a adapted to mesh with the teeth of gear 57a. A spring 60 normally biases the trigger 59 in the down position to normally hold the valve 57 in closed position. Coaxial with the spring 60 is a pin 61 secured at one end in the handle 11 and slidably received in a suitable recess or passage in the trigger 59. The trigger 59 may be equipped on either side with laterally-extending flange portions 62 slidably received within suitable recesses in the enlarged casing portion 54. The upper end of the trigger 59 may be equipped with a fingerhold 63 to facilitate movement of the trigger by the operator of the paving breaker and, if desired, a customary latch assembly (not shown) may also be provided to lock the trigger 59 in upward position against the biasing action of the spring 60. It is evident that the valve 57 is equipped with outlets 64 and 65 disposed at right angles to each other and when in the position illustrated the valve is open and establishes communication between the passages 55 and 58. However, when the trigger 59 is lowered, the valve 57 is rotated through 90 degrees, thereby blocking communication between the passages 55 and 58 and thereby between the top of the cylinder 14 and the pressure multiplier cylinder 53, and vent 56a exhausts cylinder 53.

Preferably, the entire space between the lower face of the piston 51 and the enlarged head 37 of the friction member 38 is completely filled with a liquid such as oil; however, water and other liquids may readily be used. Liquid of course is only slightly compressible, so that even the slight downward movement of the piston 51 will increase the pressure of the liquid against the head 37 of the friction member 38 and push it against the work member 17 so as to frictionally lock the same against movement with respect to the casing 10. When the valve 57 is open and during the part of each reciprocatory cycle when pressure fluid is being supplied to the upper portion of the cylinder 14 through the inlet 33, the pressure thus created therein is almost instantaneously transmitted through the short passages 58 and 55 to exert an

actuating pressure against the head 52 of the piston 51. The piston is thereby forced downwardly and the pressure is transferred through the liquid column and to the friction member 38 to actuate the same into strong frictional contact with the work member 17 almost simultaneously with the development of the pressure within the upper portion of the cylinder 14.

The force required to be exerted by the friction member 38 against the work member 17 to frictionally prevent relative movement between the casing 10 and work member will depend to a great extent upon the particular tool, and the dimensions of the pressure multiplier unit 48 may be varied as necessary with respect to the needs of particular tools. In paving breakers of the customary size it has been found that a force of approximately 1200 pounds exerted by the friction member 38 against the work member 17 (understood to be rigidly immobilized by the frictional grip of the material it has penetrated) is more than sufficient to effectuate proper operation of my friction brake assembly and to lock the work member and casing against relative movement, and thereby the casing with respect to the material, so as to eliminate vibration. After determining the force necessary to achieve the proper locking of the casing to the work member in any particular design, the dimensions of the pressure multiplier may easily be determined. Generally the pressure within the cylinder 14 when pressure fluid is being supplied to the upper end thereof through the inlet 33 will be almost simultaneously present at the upper end of the pressure multiplier cylinder 53 and this intermittent pressure will therefore be exerted against the upper end of the head 52. The resultant force will be equal to the pressure times the area of the head 52 and this force will be exerted against the liquid within the tube 44 and passage 49, etc., by the reduced lower end of the piston 51. The pressure, then, within the tube 44 will be much greater than the pressure within the cylinder 14 since the diameter of the lower end of the piston 51 is less than the diameter of its head 52 and since the pressure will be equal to the force divided by the area. This resultant pressure will of course be exerted against the enlarged head 37 of the friction brake or member 38 and a resultant large force will be applied by the member 38 against the spike or work member 17.

This may be summed up mathematically as follows:

$$P_{44} = P_{14} \times \frac{A_{52}}{A_{51}}$$

and

$$F_{38} = P_{44} \times \frac{A_{37}}{A_{51}} \times A_{37}$$

where P_{44} is equal to the pressure of the liquid column in the tube 44, P_{14} is the pressure exerted by the pressure fluid in the upper portion of the cylinder 14, A_{52} is the area of the enlarged head 52 of the pressure multiplier or differential piston 51, A_{51} is the area of the reduced portion 51 of the pressure multiplier piston, A_{37} is the area of the enlarged head 37 of the friction member 38, and F_{38} is the resultant force exerted by the friction member 38 against the work member 17. It will be apparent from this that the effective areas of the enlarged portion 52 and reduced portion 51 of the pressure multiplier piston and the effective area of the enlarged head 37 of the friction member may readily be determined for any design.

Operation

In operation, it is preferred that the paving breaker A be operated in the usual manner until the work member becomes embedded in the material that is being broken. That is, until the work member has become frictionally gripped by the material it has penetrated, the trigger 59 will not be raised and the valve 57 will block the communication of fluid pressure from the cylinder 14 to the pressure multiplier cylinder 53. But after the

work member has become properly embedded, the workman raises the trigger 59 to rotate the valve 57 and establish communication between the cylinder 14 and the pressure transfer means that will actuate the friction brake 38. Because the passage 58 communicates only with the upper end of the cylinder 14, the operation of the friction member 38 will be intermittent or cyclical and will be in timed relation with the application of pressure fluid to that end of the cylinder, which will be in timed relation to the movement of the hammer 15.

During one phase of the operating cycle soon after the workman has depressed the handle 12 and the work member 17 has become embedded in the material, and the trigger 59 has been raised to open the valve 57, the valve assembly 13 will cause pressure fluid, for the sake of definiteness sometimes herein specified as air, to be passed from the supply connector 13 through passage 29 and into the lower end of the cylinder 14 through the inlet or port 30. The pressure thus created within the lower end of the cylinder 14 will be exerted in all directions and will push downwardly upon the tappet seat member 10c (comprising the bottom closure of the cylinder) and also upwardly against the hammer. Two actions result. The casing 10 is driven downwardly until the shoulders 26 and 27 of the tappet 16 and tappet seat member 10c, respectively, are brought into abutting relation, with the enlarged bottom portion 25 of the tappet abutting the upper end of the spike or work member 17. In other words, the casing 10 has been pushed downwardly by the fluid pressure within the lower portion of the cylinder 14 as far as it can possibly be moved with relation to the work member 17. At the same time the fluid pressure drives the hammer 15 upwardly so that it passes over the exhaust port 31 to close the same and compress the air thus trapped by it in the upper portion of the cylinder 14 during the first part of its upward movement, until its continued upward movement has uncovered the exhaust port to permit the lower end of the cylinder to exhaust to atmosphere through the passage 32. The valve within the assembly 13 then responds to the almost instantaneous loss of pressure within the space comprising the lower end of the cylinder 14 and the passage 29, and breaks communication between the pressure fluid supply and the passage 29 and at the same time establishes communication through inlet 33 between the pressure fluid supply and the upper end of the cylinder 14 to admit live air thereto during the remainder of the upward movement of the hammer 15 and also during its succeeding downward movement until it again uncovers and passes below the exhaust port 31, when the valve will again reverse to supply live air to the lower end of the cylinder. Thus it will be seen that from the time when the exhaust port 31 is closed by the hammer 15 in upward movement until it is again opened by the hammer in downward movement, a positive pressure greater than the pressure exerted by the atmosphere is developed in the upper portion of the cylinder 14 above the hammer 15. Further to this point, the initial value of this positive pressure, just after the upper edge of the upwardly-moving hammer has reached the upper end of the exhaust port so that it is fully closed will be only about 16 or 17 pounds per square inch in absolute terms, or approximately 1 or 2 pounds per square inch in net terms or gauge pressure (pressure in excess of the approximately 15-pound-per-square-inch pressure exerted by the atmosphere). From such initial values, this "positive pressure" may increase to a value somewhat higher than the fluid supply or hose pressure of say 90 pounds per square inch gauge pressure as the upwardly-moving hammer imposes maximum compression upon the "trapped air," just before the admission of "live air" at the hose pressure as the hammer nears its uppermost position, and then as the hammer descends may continue to decrease in value to perhaps one-half of the hose pressure or 45 pounds per square inch just as opening of the exhaust port commences with the

upper edge of the descending hammer coincident with and passing across the upper edge of this port.

Now since this varying pressure, which is exerted equally in all directions in the upper portion of the cylinder during the interval of the operating cycle between the closing of the exhaust port by the upwardly-moving hammer and the opening of that port by the downwardly-moving hammer, is thus seen to remain positive (i. e., greater than atmospheric pressure) at all times during this interval, it follows that from the commencement to the end of this interval, a lifting force, resulting from the application of this varying positive pressure to the closure at the top of the cylinder and equal at any instant to the area of this closure multiplied by the value of the pressure at that instant, will tend to raise the casing 10 with respect to the work member 17. In one typical construction, the area of such upper closure is 5.4 square inches, so that as this pressure varies as described during the specified operating interval through initial, intermediate, and final values of 2, 90, and 45 pounds per square inch (gauge), the lifting force on the casing will correspondingly vary from an initial value of 10.8 pounds, through an intermediate value of 486 pounds, to a final value of 243 pounds.

At the same time, this varying pressure which remains continuously positive throughout this interval is also exerted against the head 52 of the pressure multiplier piston 51 through passages 58 and 55, and the resultant force is translated through the liquid within the tube 44 and into a frictional force always exerted during the interval by the member 38 against the work member 17, which as already indicated is embedded in and fixedly gripped by the concrete or other work material. Due to the pressure multiplying action of the piston 51, this frictional force is sufficient to lock the casing 10 very strongly with respect to the work member 17 so that the pressure exerted upwardly on the closure at the upper end of the cylinder 14 is made ineffective to lift or raise the casing.

Thus, in addition to the development by this pressure of a lifting force applied to the casing, the described structure provides for the simultaneous development of a frictional force acting between the casing and embedded work member and tending to prevent this "lifting force" from being effective to raise the casing, and assuming that a sufficient pressure-multiplying factor has been incorporated in the differential piston 51 by proper dimensioning thereof, this frictional force will be at all times during the specified operating interval sufficient to prevent any upward movement of the casing whatever.

For example, by taking the diameter of the enlarged head of the differential piston 51 to be 3 times the diameter of its reduced lower portion, and the diameter of the head 37 of friction member 38 as 1.40 inches, and assuming that the coefficient of friction applicable to sliding movement between the work member 17 and work member passage 19 is 20% (a usual value for dry steel on dry steel) and therefore that this same value of 20% applies as the coefficient of friction between friction member 38 and work member 17, it is found that the pressure-multiplying factor is 9, that the area of head 37 is 1.54 square inches, and that the effective coefficient of friction defining the resistance to movement of the casing 10 relative to work member 17 for a given transverse push imposed by member 38 against the work member is 40% (the sum of the 20% coefficient of friction applicable to the force developed by this push between the left side of the work member and the abutting surface of member 38, and of the 20% coefficient independently applicable to the resulting and equal force transmitted transversely through the work member and simultaneously developed between the right side thereof and the abutting portion of the work member passage

19); whence the frictional force tending to prevent the casing 10 from being raised by the "lifting force" developed by the varying pressure, already noted as having initial, intermediate, and final values of 2, 90, and 45 pounds per square inch, is found to have a corresponding initial value of $2 \times 9 \times 1.54 \times 0.40$ or 11.09 pounds. And this initial value of the frictional force, thus calculated from the initial 2-pound-per-square-inch value of the pressure, is seen to exceed the simultaneous 10.8-pound initial value of the lifting force, already noted as corresponding to this same initial value of the pressure. Obviously, then, as between these respective 11.09- and 10.8-pound frictional and "lifting" forces simultaneously obtained from the aforesaid initial 2-pound-per-square-inch pressure value, the preponderance of this frictional force over this "lifting" force will absolutely prevent the application of this latter force to the casing 10 from causing any raising movement whatever of the casing 10 and rigidly related handles 11. Similarly, the frictional force corresponding to the intermediate 90-pound-per-square-inch value of the pressure is found to be $90 \times 9 \times 1.54 \times 0.40$ or 498 pounds, and therefore to preponderate over the lesser "lifting" force of 486 pounds previously noted as corresponding to this same intermediate pressure value, wherefore the casing and handle structure will be absolutely immobilized against the tendency of this lesser force to raise or displace this structure away from the work material in which the work member is embedded. And likewise, the casing and handle structure will still be rigidly immobilized against displacement away from the work material and toward the operator when the frictional force has decreased to its final value of $45 \times 9 \times 1.54 \times 0.40$ or 249.5 pounds corresponding to the final 45-pound-per-square-inch value of the pressure, because at the same time the "lifting force" will have decreased to the yet smaller final value of 243 pounds already noted as corresponding to this same final pressure value.

These several examples clearly show how the frictional force remains continuously preponderate over the "lifting force" throughout the specified operating interval, hereinbefore described as extending from the closing to the opening of the exhaust port by the upper edge of the hammer and as succeeding the forcing of the casing into its downmost position upon the shoulder 26 of the tappet by downwardly-acting fluid pressure applied to the tappet seat member 10c during a preceding portion of the operating cycle. It will assist convenient summary of the explanation of the operational behavior of my invention to designate this described interval as the "up-pressure" interval. It should be noted that because of the continuous preponderance of the frictional force over the "lifting force" during any such up-pressure interval, the casing will remain throughout and until the end of that interval in its downmost position into which it was forced in an earlier part of the cycle in the manner explained.

Furthermore, the opening of the exhaust port by the upper edge of the descending hammer not only terminates the described up-pressure interval in which a positive fluid pressure within the upper portion of cylinder 14 applied the "lifting force" to the casing which has been seen was continuously nullified by the action of friction member 38, but also at the same time causes the deactuation of the friction member to commence, so that the cycle can continue in the efficient manner hereinafter to be detailed.

It will be understood that the critical requirement applying to this deactuating operation is that it occur with sufficient rapidity to remove the now unnecessary frictional inhibition of relative movement between the work member and casing, before such relative movement is irresistibly impelled by the blow of the hammer on the upper surface of the tappet, or in other words, before the instant of impact. Retention of such fric-

tional inhibition after this instant would result in excessive wear of the mutually frictioning surfaces of the member 38 and work member 17, and of work member 17 and work member passage 19.

A numerical example will provide the best demonstration that this critical requirement can be realized in the operation of my invention. In a typical paving breaker structure, the distance through which the hammer will move after its upper edge commences to descend below the upper edge of the exhaust port 31 and until its lower surface impacts the upper surface of the tappet is approximately 1 inch. The average speed of the hammer in traversing this final inch before impact varies somewhat with the air pressure used to operate the tool, but will usually be between 30 and 36 feet per second. Employing 33 feet per second as the mean value between these two values, and converting feet to inches, gives an approximate hammer speed of 396 inches per second, whence the hammer may be regarded as descending from the position in which it commences to uncover the exhaust port and through this final inch to its impact position, when it is required that friction member 38 be deactuated, in about one four-hundredth of a second, or in numerals, 0.0025 sec.

Now, in physical terms, deactuation of the friction member 38 results from the delivery through the passages or tubes 58, 55, 44, and 41, and to the pressurized surface of the enlarged head 37 of the member 38, of a pressure drop originating with the sudden decrease in pressure in the region of the upper portion of cylinder 14 just above the exhaust port, which immediately follows the uncovering of the exhaust port by the upper edge of the descending hammer. And, as is well known in physics, the speed with which a pressure drop is delivered through a fluid-filled tubular channel of any sort is equal to the speed of transmission of sound through the fluid contained in the channel; whence the time required for the delivery of a pressure drop through any such fluid-filled channel is equal to its length divided by such speed of sound transmission through it.

Therefore the deactuation time which will elapse between the instant when the upper edge of the descending hammer commences to uncover the exhaust port 31 and the slightly later instant when the resulting pressure drop arrives at the pressurized surface of the friction member 38 to commence deactuation of that member can be approximately calculated as

$$t = \frac{l}{s} + \frac{L}{S}$$

where l is the length of the path from the exhaust port 31, upwardly through cylinder 14, horizontally through passage 58, and downwardly through passage 55 to the upper pressurized surface of differential piston 51, which in a typical embodiment of my invention may be about 9 inches or 0.75 foot; s is the speed of sound in the fluid distributed along this path, which, assuming the fluid to be compressed and somewhat heated air from an air compressor, may be about 1500 feet per second; L is the length of the path from the lower pressurized surface of differential piston 51, downwardly through tube 44, and horizontally through passage 41 to the pressurized surface of friction member 38, which in a typical embodiment of the invention may be a little less than 13 inches or 1.10 feet; and S is the speed of sound in the fluid distributed along this second path, which, assuming this fluid to be an air-free oil, may be 4500 feet per second. Substitution of these numerical values in the equation just stated then gives

$$t = \frac{0.75}{1500} + \frac{1.10}{4500}$$

whence it is found that the deactuation time is 0.00074 sec. On comparing this figure with the previously stated requirement that deactuation of friction member 38 be

accomplished within 0.0025 sec. after the instant when the upper edge of the descending hammer 15 reaches coincidence with the upper edge of exhaust port 31 to commence uncovering this port, it is found that the deactuation time 0.00074 sec. is only 34% or about one-third of this 0.0025-sec. limitation on the deactuation time.

To state this conclusion in terms of distances, it will be recalled that this limit on the length of the deactuation time is simply an expression of the mechanical requirement that deactuation of the friction member 38 be accomplished in the last inch of movement of the descending hammer, between its position when it commences to uncover the exhaust port and its final position at the instant of impact; whence it follows that the deactuating pressure drop, which originates when the upper edge of the hammer reaches coincidence with the upper edge of the exhaust port, will travel through the upper portion of cylinder 14, and the passages or tubes 58, 55, 44, and 41 to arrive at the pressurized surface of the friction member 38 while the upper edge of the relatively slow-moving hammer is descending to a position 0.34 or about one-third of an inch below the upper edge of the exhaust port.

It will be understood that the deactuation of the friction member merely commences when the pressure drop thus arrives at its pressurized surface, and that the duration of the brief interval required for the completion of its deactuation depends on the cross-sectional area of the exhaust port or ports comprised between the position of the upper edge of the hammer at the instant of impact and the upper edge of the exhaust port. In the preferred embodiment of my invention, this cross-sectional area is made sufficiently large so that the deactuation of the friction member 38 is completed, and the frictional force developed by this member between the casing 10 and work member 17 is fully removed, simultaneously with the occurrence of the impact of the hammer 15 upon the tappet 16. Correspondingly, it will be convenient to define an additional interval in the operational cycle, regarded as commencing with the termination of the aforesaid up-pressure interval as the exhaust port is opened by the upper edge of the descending hammer and as ending with the impact of the hammer upon the tappet, and to designate the interval so defined as the deactuation interval.

And now in more particular reference to the previously mentioned portion of the operational cycle, hereinafter to be termed the down-push interval, in which the casing is forced into its downmost position by downwardly-acting fluid pressure applied to the tappet seat member 10c before the commencement of the next up-pressure interval, it should be noted that because of the continuous preponderance of the frictional force over the "lifting force" throughout that up-pressure interval as hereinbefore discussed in detail, and because no similar lifting force can be applied to the casing during the succeeding deactuation interval, which as just explained commences with the escape of the positive pressure from the upper portion of cylinder 14 through the exhaust port, it follows that the casing will remain throughout these consecutive up-pressure and deactuation intervals continuously seated in its downmost position. And since this downmost position is rigidly determined by the abutment of the shoulder 27 of the casing composition upon the shoulder 26 of the tappet 16 at the same time that the bottom surface of the tappet abuts the top surface of the work member 17 which is rigidly embedded in the work material at its lower end, it is therefore obvious that the casing cannot be pushed any farther down by the next succeeding application of fluid pressure to tappet seat member 10c until after the next impact of the hammer upon the tappet has forced the shoulder 26 thereof to a lower position by driving the tappet and work member combination farther down against the resistance of the work ma-

terial. For this reason, the aforesaid "down-push interval" will not be considered to commence until the moment of impact. Furthermore this interval will be considered to terminate at the later moment, after the tappet shoulder 26 has been thus forced to a lower position by the impact of the hammer, when the shoulder 27 of the casing composition has been moved downward through the same distance and again into abutment with the tappet shoulder by the action of fluid in the lower portion of cylinder 14 upon tappet seat member 10c as aforesaid, because after such reestablishment of abutment between the shoulders 27 and 26 (with the tappet resting on the work member and the work member again rigidly gripped by the work material), no additional down-push movement of the casing can be impelled by the action of fluid pressure in the lower portion of the cylinder until the next cycle of operation. That is to say, the "down-push interval" is regarded as comprising only that portion of the operating cycle during which the casing is actually being moved downward by such action of fluid pressure in the lower portion of the cylinder.

In more particular reference to the actions occurring during this down-push interval, inasmuch as the mass of the tappet and work member combination (for brevity in this explanation sometimes loosely referred to as the work member unit) is much less than the mass of the casing (about 10 pounds as compared to about 85 pounds in a heavy duty paving breaker) and because the force of the hammer blow which drives the work member unit through the distance comprising each of its downward advances into the work material is much greater than the fluid pressure force upon member 10c which immediately thereafter accelerates the casing downward through the same distance (of the order of 20,000 pounds as compared to 300 pounds), it follows that the duration of the descent of the work member unit is much shorter than the duration of the descent of the casing (the ratio of these durations in a typical case being about 1 to 25). Moreover, the distance through which the work member unit is driven by each impact (and through which the casing is accelerated immediately afterward) is known to be of the order of one two-hundredth of inch for ordinary concrete. Hence the duration of the down-push interval (equal to the time required for the acceleration of an 85-pound weight through $\frac{1}{200}$ of an inch by a 300-pound force, or by 3.5 g) is very brief. And since, as indicated above, the descent time of the work member unit may be as small as $\frac{1}{25}$ of this very brief down-push interval, it will be convenient, and satisfactory for all practical purposes, to regard this descent time as instantaneous, or substantially instantaneous.

It may be stated in summary, then, that the down-push interval of the operating cycle is a very brief interval commencing at the instant of impact of the hammer upon the top surface of the tappet, with the working point and top surface of the work member respectively embedded in the concrete and supporting the bottom surface of the tappet, while the shoulder 26 of the tappet supports the shoulder 27 of the casing composition; that this interval embraces an initial motion consisting in the almost instantaneous propulsion of the tappet and work member combination by the impact through a descent of about $\frac{1}{200}$ of an inch, and a second and final motion consisting in the slower propulsion of the casing thus left momentarily without support through an equal descent by a 300-pound fluid-pressure force; and that the interval therefore terminates with the working point and top surface of the work member respectively embedded in the concrete and supporting the tappet, and the tappet supporting the casing, in exactly the same relative configuration as at the beginning of the interval, but with this configuration of parts located downwardly about $\frac{1}{200}$ of an inch as compared to its location at the beginning of the interval.

Finally, this delineation of the operational cycle of a

paving breaker embodying my invention will be completed with the description of the actions occurring in the portion of the cycle commencing with the termination of the down-push interval and ending with the beginning of the up-pressure interval. This portion of the cycle may otherwise be said to come to an end with the closing of the exhaust port by the upwardly moving hammer, since the beginning of the up-pressure interval was defined in these terms. Or, in terms of position, this portion of the cycle may be said to end when the upper edge of the hammer in its rising motion comes into coincidence with the upper edge of the exhaust port. Also, therefore, this portion of the cycle may otherwise be said to commence with the hammer located below this specified position, and, more particularly, only slightly risen from contact with the tappet, from which it started to rebound immediately after its impact thereon at the beginning of the preceding down-push interval, because the very brief duration of the down-push interval is insufficient to permit a substantial upward movement of the hammer before the commencement of the immediately following portion of the cycle now being discussed.

Now since the valve in assembly 13 will not reverse the live air flow being fed underneath the rising hammer, until after its upper edge closes the exhaust port, or in other words, until after the end of this portion of the cycle, it follows that the fluid pressure between the lower face of the hammer and the tappet seat member 10c whereby the casing was accelerated downwardly about $\frac{1}{200}$ of an inch during the preceding push-down interval, will be maintained by the continuation of this live air flow at a positive value, not greatly inferior to the hose pressure, from the beginning to the end of this portion of the cycle.

Furthermore, since the highest position of the hammer during this portion of the cycle is its final position therein, when its upper edge coincides with the upper edge of the exhaust port, it follows that the pressure in the upper portion of the cylinder 14 could not rise above atmospheric pressure at any time during this portion of the operating cycle.

Therefore, throughout this portion of the cycle, the casing is continuously subject to a strong pressure force acting downwardly against its bottom closure 10c, and continuously free of any effective pressure acting upwardly against its upper closure. Consequently the casing will not be raised at any time during this portion of the cycle above its position at the commencement thereof, or, what is the same thing, its position at the termination of the preceding down-push interval, already described as its downmost possible position as rigidly determined by the abutment of shoulder 27 of the casing composition upon the shoulder 26 of the tappet 16 at the same time that the bottom surface of the tappet abuts the top surface of the rigidly embedded work member 17.

Now with reference to a convenient designation for this portion of the operational cycle, choosing the term to be employed particularly with respect to the overall utility of the paving breaker as a demolition tool, this last-described portion of its operational cycle will be called the initial interval thereof, for the reason that the description hereabove set forth of the direction and range of movement of the hammer and of the relative configuration of the casing, tappet, and work member elements precisely applies to the first phase of operation of the tool, in its normal vertical use against concrete road or floor, immediately following depression of the valve lever 12 to commence its actuation.

Accordingly, from the foregoing detailed description of the several phases of the operational cycle of the pneumatic paving breaker embodying the "intermittent friction brake" which is the specific subject matter of my invention, the complete cycle may be defined as consisting of the four consecutive intervals hereinbefore described, and having their proper cyclic order indicated

by the numerals before their respective names, as follows: (1) the initial interval, (2) the up-pressure interval, (3) the deactuation interval, and (4) the down-push interval.

Having thus described the structure and operational cycle of a paving breaker incorporating my novel intermittent friction brake, I wish now to direct attention specifically to the manner in which my invention accomplishes the several valuable objects set forth at the outset of this specification.

First of all, it will be noted in the light of the explanation of structure which follows the enumeration of these several objects that they will all be satisfactorily attained immediately upon the frictional gripping of the work member by the concrete or other work material, if the structure functions successfully thereafter to (a) substantially prevent vibratory motion of the casing, and (b) provide entirely "non-manual" actuation of the casing, before each impact, into its downmost possible position relative to the "work member unit" (consisting of the tappet and work member in rigid abutment) which was shown to be the optimum position of the casing to secure the greatest effect of the impact in driving the work member unit downward against the resistance of the work material.

And since both of these required conditions (a) and (b) relate to motions of the casing member—either relative to the motionless slab of work material or to the work member unit descending with respect to the slab—it will suffice, in turn, to check the satisfactory attainment of these two conditions by reviewing only the part or parts of the complete operational cycle in which the casing exhibits either or both of these specified motions.

Now it has just been shown in the description of the initial interval of the cycle that the casing will remain throughout this interval in its downmost possible position, as rigidly determined by the abutment of shoulder 27 of the casing composition upon shoulder 26 of the work member unit, which has its lower portion rigidly embedded in the motionless slab. And (as was previously shown in the detailed discussion of the down-push interval) these several elements will remain in exactly this relative configuration throughout the succeeding up-pressure interval and throughout the then succeeding deactuation interval. Consequently, this relative configuration will continue without change through these consecutive initial, up-pressure, and deactuation intervals. Furthermore, the work member unit cannot have moved at any time during these three intervals, because the impact occurs only in the down-push interval.

It follows that the casing cannot have moved relative to either the work member unit or the motionless slab except during the down-push interval, so that the attainment of the required conditions (a) and (b)—the substantial elimination of vibratory motion of the casing, and the provision of non-manual actuation of the casing into its downmost position before each impact—by my novel structure can be fully checked by reviewing the actions occurring in the down-push interval only.

Attention is therefore directed back to the italicized summary of the down-push interval. With respect to condition (a) it is seen that the casing undergoes no movement whatever during this interval except for its described downward movement through a distance of about $\frac{1}{200}$ of an inch. Therefore throughout this interval, and consequently throughout the complete operational cycle, the casing exhibits no upward vibratory movements whatever, while this limitation of $\frac{1}{200}$ of an inch is imposed on its single downward vibration in the cycle. Hence in the continuous operation of the tool, the vibration of its casing is limited to a rapid sequence of these 0.005-inch downward movements, which I have found experimentally to be so nearly undetectable that the tool is, for all practical purposes, vibrationless. And with respect to condition (b), the described actuation of the casing through this descent of 0.005" immediately after each impact, into

its ideal downmost position for the next impact, is seen to be entirely non-manual, because it will be compelled to occur by the described 300-pound fluid-pressure force acting downwardly on the bottom closure of the cylinder, even in the entire absence of any downward push on the handles of the tool contributed by muscular effort on the part of the operator.

And it will be seen that in the continuous operation of the tool, this non-manual downward actuation of its casing has the same practical effect as would be derived from a continuous 300-pound downpush maintained upon its handles by a giant operator: in either case, the casing 10 is always held down in its optimum position for maximum impact efficiency.

Since this reference to a tremendous, continuous downpush supplied by a giant operator leads to a very simple understanding of my invention, I will explain more accurately that such a giant operator would actually have to apply a steady downward force of about 400 pounds, equal to the "lifting force" on the casing during the up-pressure interval which the intermittent friction brake completely inhibits, in order for the tremendous muscular force thus provided to be the full useful equivalent of the action provided by this "brake," but if raised to such a high value approximating 400 pounds, the supposed muscular force would actually serve as a complete substitute for the intermittent action provided by the "brake." That is to say, a continuous down-push of 400 pounds on the handles of a paving breaker is fully adequate to suppress its vibratory motion for all practical intents and purposes, and at the same time such a very great continuous force would at all times compel the casing to descend with the descending tappet and work member combination, in a relation of substantially continuous abutment therewith, so as always to remain in its optimum downmost position for the attainment of maximum impact efficiency and highest working speed.

After these remarks, the following physical explanation of my invention will be readily appreciated: by applying to a powerful 2-directional vibration synchronous means for suppressing the vibratory movements in one direction but not the other, the unsuppressed vibrations, all acting in this second direction, are made the useful equivalent of a powerful continuous force acting in this second direction.

In terms of the paving breaker in question, the "powerful 2-directional vibration" is the vibration of the casing, alternately in upward and downward directions; the "synchronous means" is the intermittent friction brake which operates to suppress the upward but not the downward vibratory movements of the casing; and these unsuppressed vibratory movements, all acting in the downward direction, are with respect to the useful results accomplished equivalent to a powerful force continuously exerted downward upon the casing.

While in the foregoing specification I have set forth an embodiment of my invention in considerable detail for purposes of illustration, it will be apparent that changes may readily be made by those skilled in the art without departing from the spirit of my invention.

I claim:

1. In combination with a pressure fluid actuated percussion tool having a casing and a work point movably mounted therein, a friction brake carried by said casing and adapted when actuated to frictionally grip said work point to restrain relative movement between the casing and work piece, and means for actuating said friction brake sequentially and in timed relation with the operation of said tool to restrain relative movement between said casing and work point during the periods of actuation.

2. In combination with a percussion tool having a casing, a work point movably mounted in said casing, a cylinder and a pressure fluid operated hammer reciprocable therein for driving said work point, a friction brake carried by said casing adjacent said work point and adapt-

ed upon actuation to frictionally engage the same, and fluid means arranged with said friction brake for actuating it intermittently in timed relation with the movement of said hammer to hold said work point and casing against relative movement.

3. In a tool having a pressure fluid actuated hammer adapted to repeatedly impart impact to a work point movably carried by the tool casing, a friction brake movably mounted in said casing and adapted upon actuation to frictionally grip said work point to hold said casing from movement relative thereto, and fluid means responsive to the pressure fluid when applied to move said hammer in point-striking direction for actuating said brake, whereby said brake is intermittently actuated to lock said casing and work point against relative movement in timed relation with the movement of said hammer.

4. In a paving breaker or the like wherein a casing provides a cylinder having a hammer adapted to reciprocate therein upon application of a pressure fluid alternately adjacent the opposite ends of the cylinder to deliver impact to a work point slidably carried by said casing, a friction member movably carried in said casing and adapted upon actuation to engage said work point to lock the casing against movement with respect to said point, and fluid pressure transfer means for cyclically actuating said member and communicating at one end with said friction member and at its other end with said cylinder adjacent the end thereof most removed from said work point, whereby said friction member is actuated for substantially the period that pressure fluid is applied to the end of said cylinder most removed from said work point.

5. The structure of claim 4 wherein said transfer means includes a pressure multiplier piston in circuit therewith.

6. The structure of claim 4 wherein said transfer means includes a conduit communicating at one end with said friction member and being filled with a liquid, and means at the other end of said conduit responsive to the pressure within the end of said cylinder most remote from said work point for exerting a pressure upon said liquid to actuate said friction member.

7. The structure of claim 4 in which said transfer means comprises a conduit communicating at one end with the end of said cylinder most remote from said work point and at its other end with a pressure multiplier means, said multiplier means being arranged with one end of a liquid column to establish pressure therein when energized by pressure fluid in said conduit to actuate said friction member, said friction member being arranged with the other end of said column.

8. In a friction brake assembly for use in pneumatic paving breakers of the type wherein the alternate automatic connection of pneumatic pressure to opposite ends of a cylinder provided by the paving breaker casing reciprocates a hammer to impact a work member arranged with a head extending into said cylinder adjacent an end thereof and having an enlarged portion abutting said casing when extending to its maximum position in the cylinder, the combination comprising, a friction member slidably mounted in said casing and adapted when actuated to grip said work member with sufficient force to restrain relative movement between said casing and work member, and means for actuating said friction member when pneumatic pressure is supplied to said cylinder at the end most remote from said work member to drive said hammer into impact with said head while releasing said member when pneumatic pressure is supplied to said cylinder to move said hammer in the opposite direction comprising a conduit communicating at one end with said friction member and at its other end with a pressure multiplier, said conduit being filled with liquid, and a passage communicating at one end with said multiplier and at its other end with the end of the cylinder most removed from said work member to energize the multiplier

when pneumatic pressure is supplied at that end of the cylinder.

9. The structure of claim 8 in which the paving breaker casing is adapted to move into abutting relation with said head when pneumatic pressure is supplied to said cylinder adjacent the end thereof receiving said head, and said multiplier is a piston having an enlarged head communicating with said passage and a reduced portion communicating with the liquid in said conduit.

10. The structure of claim 9 wherein said friction member is spring biased in a direction away from engagement with said work member, and said passage is equipped with a manually-operated valve to provide selective actuation of said friction member.

11. A friction brake assembly adapted for use in pressure fluid operated percussion tools, comprising a friction member adapted to extend through the tool casing and frictionally engage the work point thereof, a liquid-filled conduit communicating at one end with said friction member and at its opposite end with a pressure multiplier piston, said piston being equipped with an enlarged head, and a support for said piston adapted to be secured to the casing of the tool and providing a passage therethrough adapted to establish connection between the actuating cylinder of the tool and said enlarged head.

12. In a pneumatic paving breaker, a casing, a spike slidably mounted in said casing, an anvil at the upper end of said spike having a head portion extending into an actuating cylinder provided by said casing, a hammer mounted for reciprocation within said cylinder, passages communicating with said cylinder adjacent the upper and lower ends thereof, a valve assembly automatically operable for controlling said passages and adapted to alternately connect a pressure fluid input to each of the passages, a friction member slidably mounted in said casing adjacent said spike and adapted when actuated to engage the same, said friction member having an enlarged head communicating with a liquid-filled conduit whereby said friction member is forced into frictional engagement with said spike by pressure exerted thereagainst by the liquid within the conduit, a pressure multiplier piston arranged with said conduit and adapted to apply a force to the liquid therein, said multiplier piston being equipped with an enlarged head in open communication with a passage provided by an enlargement carried by said casing, said passage being in open communication at its other end with the upper end of said actuating cylinder, and a manually-operable valve for opening and closing said passage.

13. In combination with a percussion tool having a casing providing a cylinder therein, a piston reciprocable within said cylinder and a workmember carried by said casing for axial movement relative thereto and being adapted to receive impact from said piston, a brake member operatively arranged with said casing and workmember for applying a force therebetween to restrain relative movement between the casing and workmember during preselected phases of the reciprocatory cycle of said piston, and means for cyclically energizing said brake in timed relation with the reciprocatory movement of said piston to establish said force.

14. In a percussive tool having a casing normally vibrating during tool operation, a hammer member mounted for reciprocatory movement within said casing and a workmember carried by said casing for limited axial movement relative thereto and arranged to receive impact from said hammer in its reciprocatory movement, locking means coactive, when actuated, between said casing and workmember for rigidly locking the same together to prevent relative movement therebetween, and means for actuating said locking means cyclically in timed relation with the reciprocatory movement of said hammer so that the component of the vibratory movement of said casing relative to said workmember in one direction is inhibited while the component of the vibratory movement in the opposite direction is uninhibited.

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15. The apparatus of claim 14 in which said locking means comprises a friction brake member operative between said casing and work member to apply, when actuated, a frictional resistance to relative movement therebetween.

16. The structure of claim 14 in which said hammer member is reciprocated by the application of pressure fluid thereagainst alternately in opposite directions, and in which said means for actuating said locking means comprises a fluid pressure column arranged therewith and responsive to the application in one direction of the fluid pressure against said hammer member.

17. In combination, a casing defining a cylinder equipped with a piston reciprocable therein, fluid pressure means for reciprocating the piston within said cylinder, the reciprocatory movement of the piston being operative to effectuate vibratory movement of the casing in directions along the path of reciprocation of the piston, a work

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member, locking means coactive, when actuated, between said casing and work member for rigidly locking the same together to prevent relative movement therebetween, and means for actuating said locking means cyclically in timed relation with the vibratory movement of the casing so that the component of vibratory movement thereof relative to said work member in one direction is inhibited while the component of the vibratory movement in the opposite direction is uninhibited.

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