A pneumatically percussive tool having a handle isolated from the body of the tool so that vibrations of a reciprocating hammer in the body are not transmitted to the handle. This is accomplished by arranging the handle in axial slidable relation to the body so as to permit the body to vibrate relative to the handle; and by maintaining a cushion of pressurized air in a chamber between the handle and the body which chamber is connected with a pressure air source and with a relief check valve. The pressure developing in the chamber is determined by a spring load on the check valve, which load varies automatically with the pressure exerted by the operator in holding the tool pressed against the work. The pressurized air cushion constantly counterbalances the opposing force exerted by the operator so that the body of the tool is maintained at all times in spaced relation to the handle as it vibrates. A pair of air feed ports to the hammer are adapted to be brought into or out of register with each other accordingly as the tool is pressed against or removed from the work.

13 Claims, 2 Drawing Figures
PNEUMATICALLY PERCUSSIVE TOOL HAVING A VIBRATION FREE HANDLE

BACKGROUND OF THE INVENTION

This invention relates to pneumatically powered percussive tools of a type designed to be held in pressed relation to the work under pressure exerted by the operator on the handle of the tool.

In tools of this nature, a piston hammer is pneumatically reciprocated at a rapid rate to pound a work implement. The force of the blows transmitted to the work varies with the pressure exerted by the operator upon the handle of the tool.

A problem with conventional tools of this general nature arises from the shock and vibration accompanying their operation which subjects the operator to undesirable discomfort.

A general objective of this invention is to improve the handle structure for such tools so as to substantially isolate the vibration of the tool from its handle, and thereby avoid its transmission to the operator.

This objective is accomplished by arranging the handle in slidable relation to the body of the tool so as to allow the body to vibrate relative to the handle, and by maintaining a cushion of pressurized air between the body of the tool and the handle in such a way that the pressure of the air cushion will vary automatically with the pressure exerted by the operator in holding the tool pressed to the work, and will be constantly held in substantially counterbalancing relation to the opposing pressure exerted by the operator irrespective of the vibrating frequency of the body of the tool.

In this arrangement, the body of the tool vibrates at a high frequency relative to the handle as the latter is maintained substantially stationary in the hands of the operator and the handle becomes substantially free of, and isolated from, the vibration.

While the principles involved in obtaining this isolation of the vibration of one member from a substantially stationary member with which it is connected are conceivably subject to various industrial applications, they are especially applicable to pneumatically powered percussive tools of the handle type. Accordingly, the invention is here illustrated as embodied in a pneumatically powered percussive tool, such as a chipping hammer.

BRIEF DESCRIPTION OF DRAWING

In the accompanying drawing:

FIG. 1 is a view in longitudinal section of a pneumatically powered percussive tool embodying the invention; and

FIG. 2 is a similar view of a modified embodiment of the invention.

DESCRIPTION OF PREFERRED EMBODIMENT

The embodiment illustrated in FIG. 1 includes a body defined by a barrel or cylinder 10 having a chamber 11 in which a piston hammer 12 is pneumatically reciprocable at a high frequency to pound a work element, such as a chisel 13.

A backhead 14, threadedly fixed at 9 to the cylinder, provides a valve chamber 15 in which a conventional air-blown distributing valve 16 is arranged. The valve functions automatically in response to live air entering the valve chamber to direct the air alternately to opposite ends of the hammer chamber 11 to reciprocate the hammer 12.

The backhead is supported to a handle 17 for relative axial sliding movement by means of a guide shaft 18 and by means of a piston extension 19 of the backhead. The piston 19 extends axially from a rear face of the backhead into a piston chamber A defined in a block portion 22 of the handle. The block is rigidly bolted to a mating flange portion 23 of the handle. The piston 19, piston chamber A, and hammer chamber 11 are axially aligned.

Shaft 18 extends parallel to the piston 19 through the backhead 14 and block 22 into a recess 24 formed in a neck 25 of the handle. A lock screw 26 secures the shaft rigidly to the handle. A stop ring 27 upon an end of the shaft projecting beyond the backhead above the level of cylinder 10 is cooperable with a forward face 29 of the backhead to stop the backhead from sliding free of the shaft. Abutment of the backhead against the stop ring is cushioned by means of a resilient washer 28 adhered to the face of the backhead.

The length of that portion of shaft 18 extending forwardly beyond block 22 is greater than the axial dimension between the front and rear faces 29, 30 of the backhead, as indicated by the separating gap 31. This gap is greater than the normal range of axial movement along the shaft developing in the backhead during normal operation of the tool.

During normal operation of the tool, a forwardly directed pressing force—exerted by the operator upon the handle of the tool in maintaining the work end 13 of the tool pressed against the hard surface of the work—displaces the backhead 14 rearwardly from the stop ring 27; and a counter air-pressurized condition that is developed in chamber A against the piston 19 resists full movement of the backhead rearwardly across the gap 31. These opposing forces serve to maintain the rear face of the backhead about midway of the gap 31 and to maintain the opposed faces of the backhead clear of both the block 22 of the handle and the stop ring 27.

Vibrations occurring in the backhead during operation of the tool are of high frequency and relatively short range; and are dampened by the opposing counterbalancing forces created by the operator and the pressurized condition of chamber A so that in this arrangement the backhead does not, during normal operation of the tool, impact against either the block 22 of the handle nor against the stop ring 27. The vibrations of the backhead are accordingly substantially isolated from, and have minimal effect upon, the handle of the tool and upon the operator.

A manipulative throttle valve 32 is provided in the tool to control live air flow from an external source into an inlet passage 33. The latter extends into an axial bore 34 of guide shaft 18.

To connect the inlet passage 33 with the air distributing valve chamber 15, it is required that a side port 35 in the guide shaft be brought into communication with a port or passage 36 in the backhead leading to the valve chamber 15. When the backhead abuts the stop ring 27, as in FIG. 1, ports 35 and 36 are out of register and blocked off from one another. When the backhead is displaced rearwardly from the stop ring, these ports communicate with each other.
The inlet passage 33 is connected at all times with chamber A through a restricted opening or orifice 37; and is also connected through a similar second restricted opening or orifice 38 with a chamber B. The latter chamber is located in the handle in opposed axial alignment with chamber A. Chamber B is of greater diameter and larger volume than chamber A. Coaxial with chamber A is an annular groove C, separated from chamber A by an annular beveled edge 39 of groove C. Groove C is in direct opposed relation to chamber B and has a corresponding outer diameter.

An elastomeric diaphragm 41, sandwiched about its outer portion in the handle, seals chamber A and groove C from chamber B. It is normally seated under pressure of air admitted to chamber B upon the annular edge 39 so as to normally close chamber A to groove C. Groove C is constantly vented to atmosphere by a venting passage 42, the diameter of which is greater than that of the restricted inlet port 37 to chamber A. Chamber B is subject to being connected to atmosphere through a normally closed relief port 43. The relief port is of greater diameter than that of the restricted inlet port 38 to chamber B; and is normally closed by a ball check valve 45 under the load of a spring 44.

The spring 44 extends parallel to the piston 19 across the gap 31 and is seated at its forward end in a recess of the backhead. It can be seen that, as the backhead 14 moves axially relative to the handle, the spring load 44 upon the check valve is accordingly increased or relaxed.

It is apparent that the degree of air pressure permitted to build up in chamber B is determined by the spring load on the check valve 45. Pressure developing in chamber B in excess of this load will be relieved with escape of air through the check valve. Further, the pressure build-up in chamber A is determined by the counterpressure in chamber B over the diaphragm 41 holding the latter seated over groove C.

In summary of the operation of the tool: the operator grips the handle 17 with one hand and, before pressing the work end of the tool against the work, actuates the throttle valve 32. Source air having a regulated pressure then flows through ports 37 and 38 to pressurize chambers A and B; and flows to the distributing valve chamber 15 unless ports 35 and 36 are blocked from one another, as in FIG. 1. If these ports are not blocked from one another, the hammer 12 will be momentarily moved by pressure air entering the valve chamber 15 to pound the bottom 46 of the cylinder. As chamber A expands upon being pressurized, the backhead 14 will be forced against the stop ring 27 to cut port 35 off from port 36 causing air feed to the hammer to be blocked.

Chambers A and B become pressurized to the extent determined by the spring load 44. This pressurized condition will be initially indicated by movement of the backhead into abutment with the stop ring 27 and by air leakage from chamber B through the relief valve 45. Leakage at this time through the check valve before the tool is pressed against the work occurs because the source pressure acting upon the relief valve 45 is regulated to correspond substantially to the force that would normally develop under the spring load when the tool is pressed by the operator against the work.

Next, the operator applies the work end 13 of the tool to the hard surface of the work and pushes forwardly upon the handle sufficiently to carry the stop ring 27 forwardly and clear of the backhead and to carry ports 35 and 36 into register to allow feed of inlet air to reciprocate the hammer.

The pressure exerted by the operator on the handle in pressing the tool against the work moves the piston portion 19 of the backhead inwardly of chamber A and causes the backhead 14 to contract the spring 44 to increase its load upon the check valve 45. The consequent pressure increase developing in chamber A under compression of the piston momentarily exceeds the counterpressure in chamber B so as to momentarily unseat the diaphragm 41 and allow relief of the excess pressure from chamber A with escape of air through the vented groove C. The increased load on the spring 44 allows the pressure in chamber B to rapidly build up to a counterbalancing value, and the diaphragm is reseated. The pressurized condition of the contracted volume in chamber A backed by the counterpressure in chamber B counterbalances the usual biasing force exerted by the operator on the tool and normally resists further retraction of the piston in chamber A. The gap 31 is sufficiently elongated so that the backhead does not at any time come into contact with the block portion 22 of the handle as the operator exerts varying pressure forwardly upon the handle.

While the tool is held pressed in this manner and the reciprocating hammer is repeatedly pounding the work element, the cylinder 10, together with the backhead 14, vibrate as a unit at high frequency. With each rearward movement in chamber A of the piston portion 19 of the vibrating backhead, the pressure in chamber A momentarily exceeds that in chamber B and is relieved through groove C. In this rearward movement the energy of the vibrating backhead is substantially dissipated; it is partly relieved with escape of air through groove C and is partly dissipated in compressing the volume in chamber B. The latter action tends to unseat the check valve 45.

With each forward or outward vibrating movement of the piston of the backhead in chamber A, inlet air feeds to chamber A to replenish or compensate for any previously escaped air; and the forward thrust of the vibrating backhead is dampened by the opposing pressure caused by the pressing action of the operator. The range of movement of the backhead as it vibrates at high frequency is relatively short and at no time during normal operation of the tool does the back head impact against either the block 22 of the handle or the stop ring 27. The biasing pressure in chamber A upon the piston 19 and diaphragm 41; and the counterpressure in chamber B are maintained substantially constant during operation of the tool.

When the operator withdraws the tool from the work, or should the tool suddenly break through the work into a void area, the biasing force in chamber A responds to slide the backhead 14 forwardly into abutment with the stop ring 27. This carries the feed port 35 and valve feed passage 36 out of register to stop operation of the hammer as further air feed to the valve chamber 15 is cut off. This action avoids undesirable reciprocating of the hammer which would otherwise occur were it not for the particular arrangement of the feed ports 35 and 36. As the volume of chamber A expands in this action, it is replenished with inlet air.
through the restricted port 37 to maintain the pressure balance between chambers A and B.

The embodiment in FIG. 2 is identical to that of FIG. 1 except that the restricted feed port 37, the venting groove C and its vent passage 42, and the diaphragm 41, all shown in FIG. 1, have been omitted. In this embodiment, chambers A' and B' connect with each other. They are both served by one restricted inlet port 38 and by the one relief passage 43 normally closed by the spring loaded ball check valve 45.

In this FIG. 2 embodiment, when the tool is pressed by the operator against the work, excess pressure developing in the combined chambers A' and B' under compression of the piston 19 is relieved with escape of some of the pressure air through the check valve 45. During vibration of the backhead, some of the energy of vibration is dissipated in compressing the air in the larger volume chamber B' and, in doing so, tends to unseat the check valve. The backhead 14 vibrates similarly to the FIG. 1 embodiment, along the guide shaft 18, remaining clear at all times of the block 22 of the handle and the stop ring 27.

It is understandable that in the embodiment of FIG. 1, the inlet ports 37 and 38 for chambers A and B may be connected to a separate regulated source of supply independently of the inlet feed passage 33.

What is claimed is:

1. In a pneumatically percussive tool including a cylinder having a hammer pneumatically reciprocable therein to repeatedly pound a work element carried in a front end of the cylinder, and including a handle separable from the cylinder adapted to be held by the operator, a backhead closing over the rear of the cylinder having a portion raised above the level of the cylinder, a shaft rigid with the handle slidably received through a bore in the raised portion of the backhead and extending axially beyond a forward end of the backhead, a stop ring on a forward end of the shaft preventing slidable escape of the backhead from the shaft, a throttle controlled pressure air inlet feed passage extending through the handle into the shaft having connection with a side port in the shaft, an air distributing valve in the backhead having response to pressure air fed to it to direct the air alternately to opposite ends of the hammer, a valve feed passage through the backhead communicating with the air distributing valve, the backhead having a forwardly moved position on the shaft wherein the valve feed passage is located in unregistered relation to the side port, and the backhead having a position displaced away from the stop toward the handle in which position the valve feed passage is registered with the side port.

2. In a pneumatically percussive tool as in claim 1, wherein the backhead has a piston extending axially from its rear into a piston chamber in the handle in parallel relation to the shaft.

3. In a pneumatically percussive tool as in claim 2, wherein a chamber of larger volume than the piston chamber is formed in the handle in coaxial relation to the piston chamber, restricted passage means is provided connecting both the piston chamber and the larger volume chamber with the air inlet passage, and the larger volume chamber is connected with a relief passage having a normally closed spring loaded check valve, the spring being loaded in parallel relation to the shaft between the check valve and the rear of the backhead.

4. In a pneumatically percussive tool as in claim 3, wherein an annular groove is formed in the handle coaxial with the piston chamber and has a vent connection, a diaphragm seals the groove from the larger volume chamber and normally blocks communication of the piston chamber with the groove.

5. In a pneumatically percussive tool including a cylinder having a hammer pneumatically reciprocable therein to repeatedly pound a work element carried in a front end of the cylinder, and including a handle to be held by the operator; a shaft rigid with the handle supporting the cylinder for limited axial sliding movement of the cylinder along the shaft relative to the handle, a piston extending axially rearwardly from a rear face of the cylinder, a piston chamber in the handle in which the piston is received, the piston being adapted to be retracted into the piston chamber and the cylinder adapted to be moved along the shaft toward the handle upon the operator applying the work element to a hard stationary surface and exerting a forwardly pressing force upon the handle, and means for exerting upon the piston a pneumatic force substantially counterbalancing the pressing force that might be applied by the operator to the handle.

6. In a pneumatically percussive tool as in claim 5, wherein means is provided for automatically varying the counterbalancing pneumatic force accordingly as the pressing force applied by the operator is varied.

7. In a pneumatically percussive tool as in claim 6, wherein a spring is loaded between the check valve and an opposed face of the cylinder and is disposed in parallel relation to the axis of the shaft and that of the cylinder.

8. In a pneumatically percussive tool as in claim 6, wherein manipulative air feed control means is provided for feeding from a common source pressure air concurrently to the cylinder to reciprocate the hammer, and to the piston chamber rearwardly of the piston.

9. In a pneumatically percussive tool as in claim 6, wherein control means is provided for feeding pressure air concurrently to the cylinder to reciprocate the hammer, and to the piston chamber rearwardly of the piston.

10. In a pneumatically percussive tool as in claim 9, wherein a pressure air chamber of larger volume than the piston chamber is formed in the handle in opposed aligned and coaxial relation to the piston chamber, the pressure air chamber is connected with a vent passage, a spring loaded check valve in the vent passage blocks escape of air from the pressure air chamber, means is provided for automatically varying the spring load accordingly as the cylinder is moved axially relative to the handle, a groove is formed in the handle coaxially with the piston chamber and facing the pressure air chamber, the groove is connected with a vent and is separated from the piston chamber by means of an annular shoulder, a diaphragm seals the groove off from the pressure air chamber and is seated under pressure of air in the pressure air chamber upon the annular shoulder so as to normally block communication of the piston chamber with the groove, a restricted passage means is connected with the control means for flow of...
pressure air to the piston chamber, and a separate similar restricted passage is connected with the control means for flow of pressure air to the pressure air chamber concurrently with the flow to the piston chamber.

11. In a pneumatically percussive tool as in claim 9, wherein a pressure chamber of larger volume than the piston chamber is formed in the handle in opposed aligned and coaxial relation to the piston chamber and in direct communication with the latter, the pressure chamber being connected with a vent passage, and there being a spring loaded check valve in the vent passage normally blocking escape of air from the pressure chamber.

12. In a pneumatically percussive tool as in claim 11, wherein the spring load on the check valve determines the degree of pressure development in the pressure chamber, and means is provided for automatically varying the spring load accordingly as the cylinder is moved axially relative to the handle.

13. In a pneumatically percussive tool including a cylinder body having a hammer pneumatically reciprocable therein to repeatedly pound a work element carried in a front end of the body, and including a handle to be held by the operator; a shaft rigid with the handle supporting the body for limited axial sliding movement of the body relative to the handle, a piston extending axially from a rear face of the body received in a piston chamber provided in the handle, means for continuously exerting a pneumatic force upon the piston biasing it outwardly of the piston chamber, and stop means limiting the extent of movement of the body away from the handle as a consequence of the biasing force upon the piston so that the piston remains at all times at least in part within the piston chamber.

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