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(54) **LINEAR PUMP WITH SOUND ATTENUATOR**

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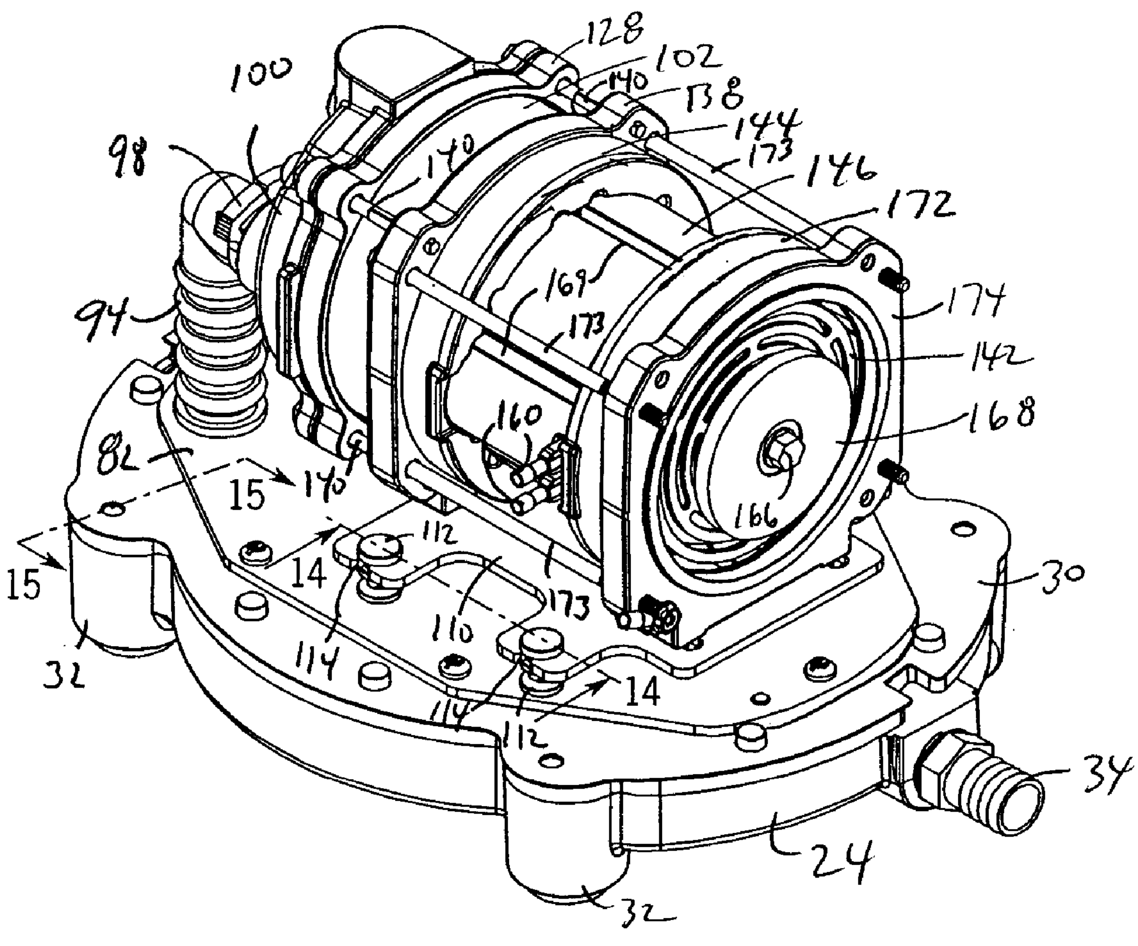
(57) **ABSTRACT**

A linear piston pump has a multi-cavity air intake providing improved sound attenuation characteristics. The air intake is formed between a cover mounted in spaced relation to a pump housing so as to define an air intake passage leading around the periphery of the cover to an outer cavity separated from an inner cavity by a partition. Flow orifices are formed in the partition as well as in the inner cavity for air to flow to the intake chamber. Intake air re-expands in the inner cavity to dissipate pulsations in the air flow from the rapidly action of the intake valve, and thereby reduces intake noise. A filter and air baffle filter retainer may also be included to further reduce intake noise.



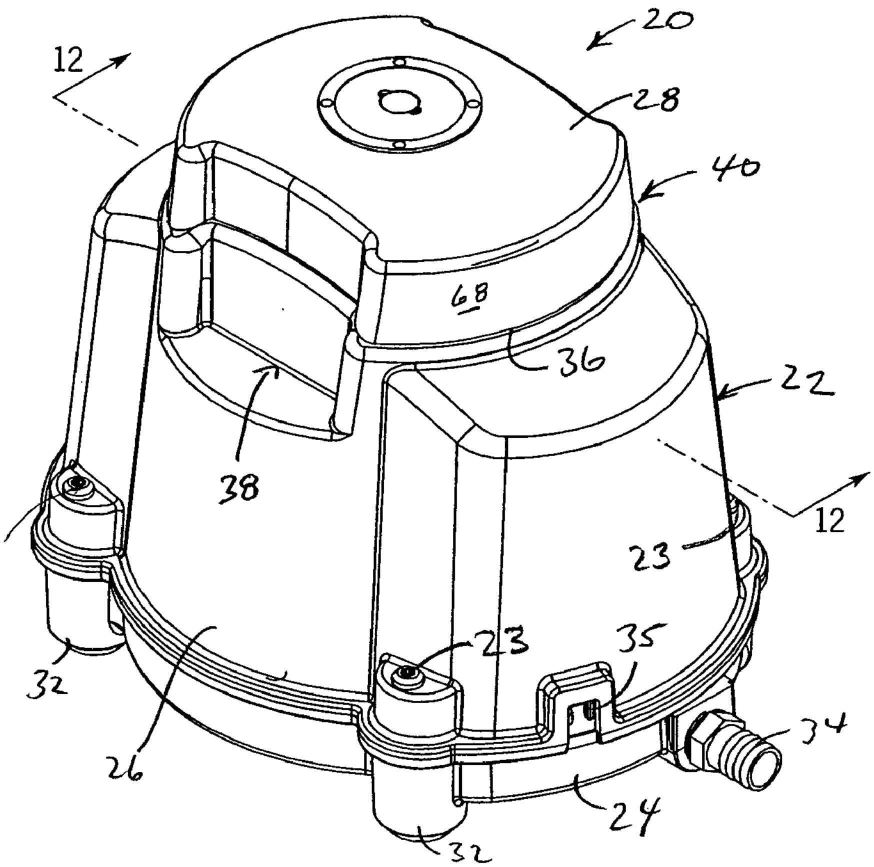

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FIG. 1

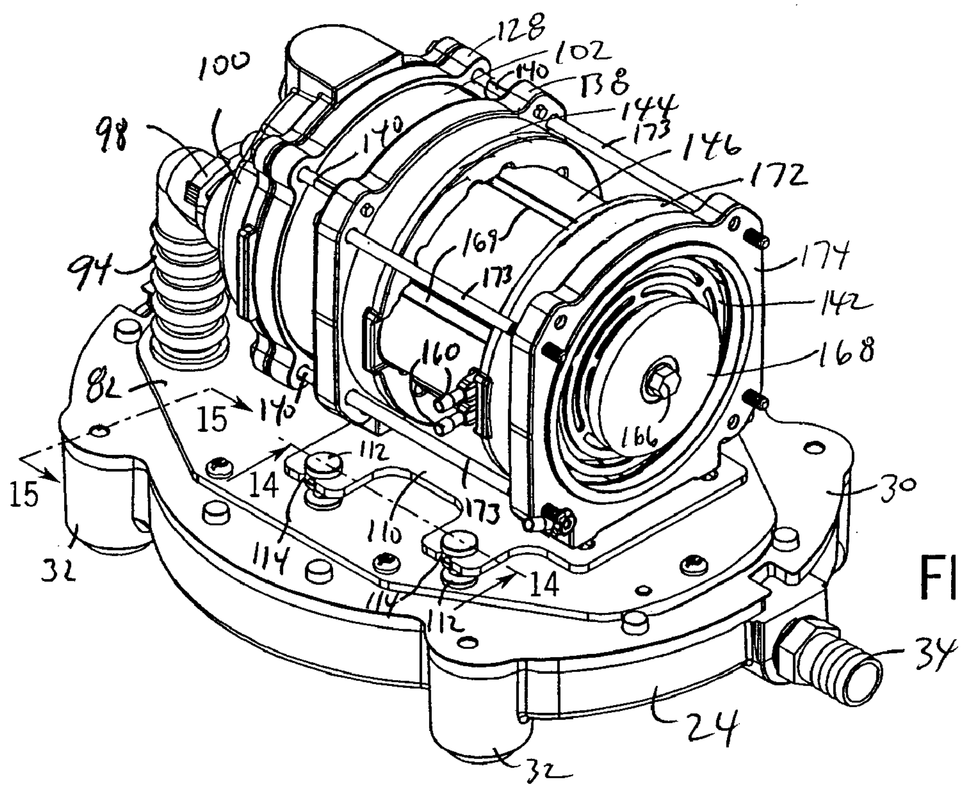
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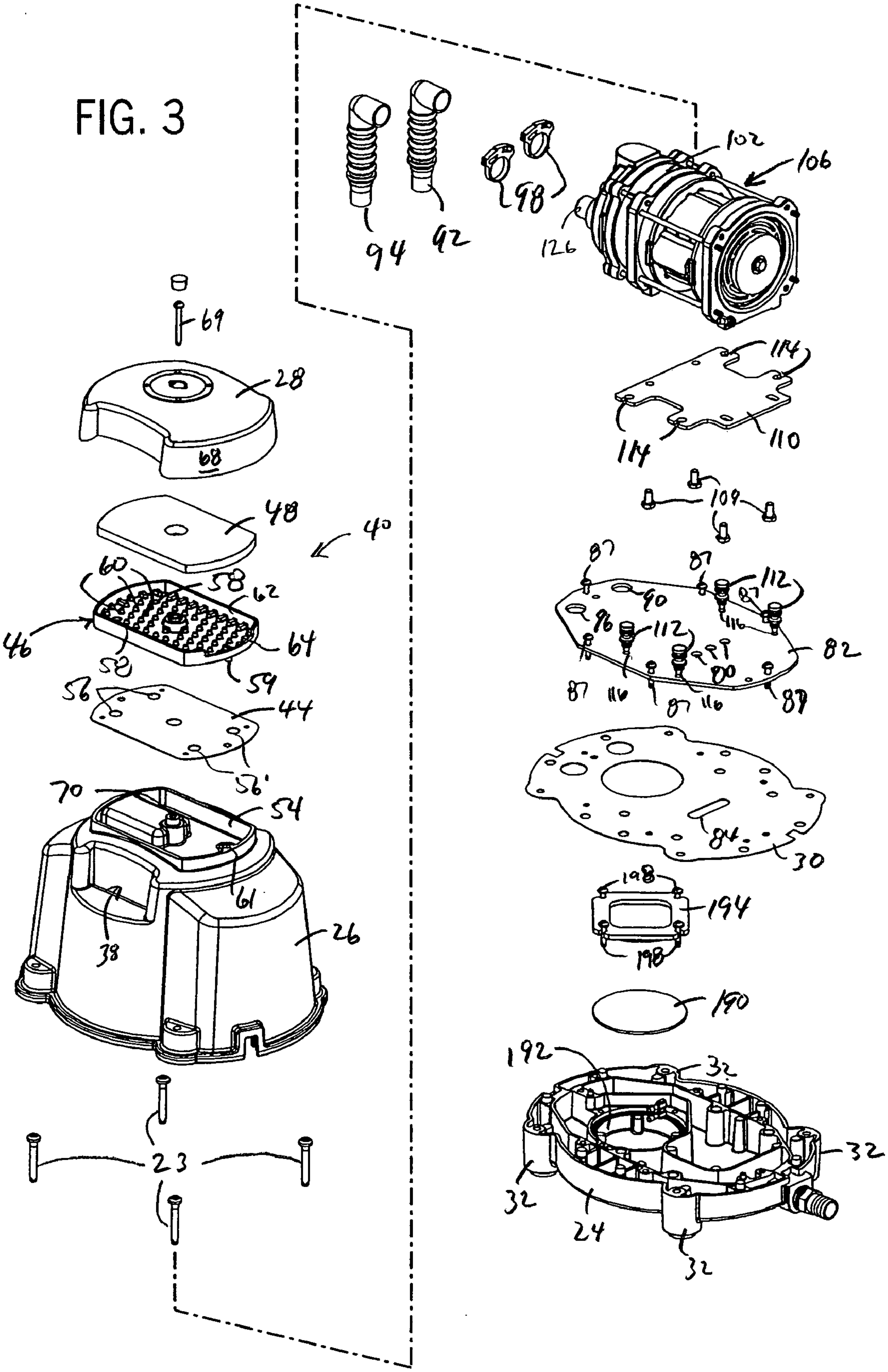
FIG. 2

FIG. 3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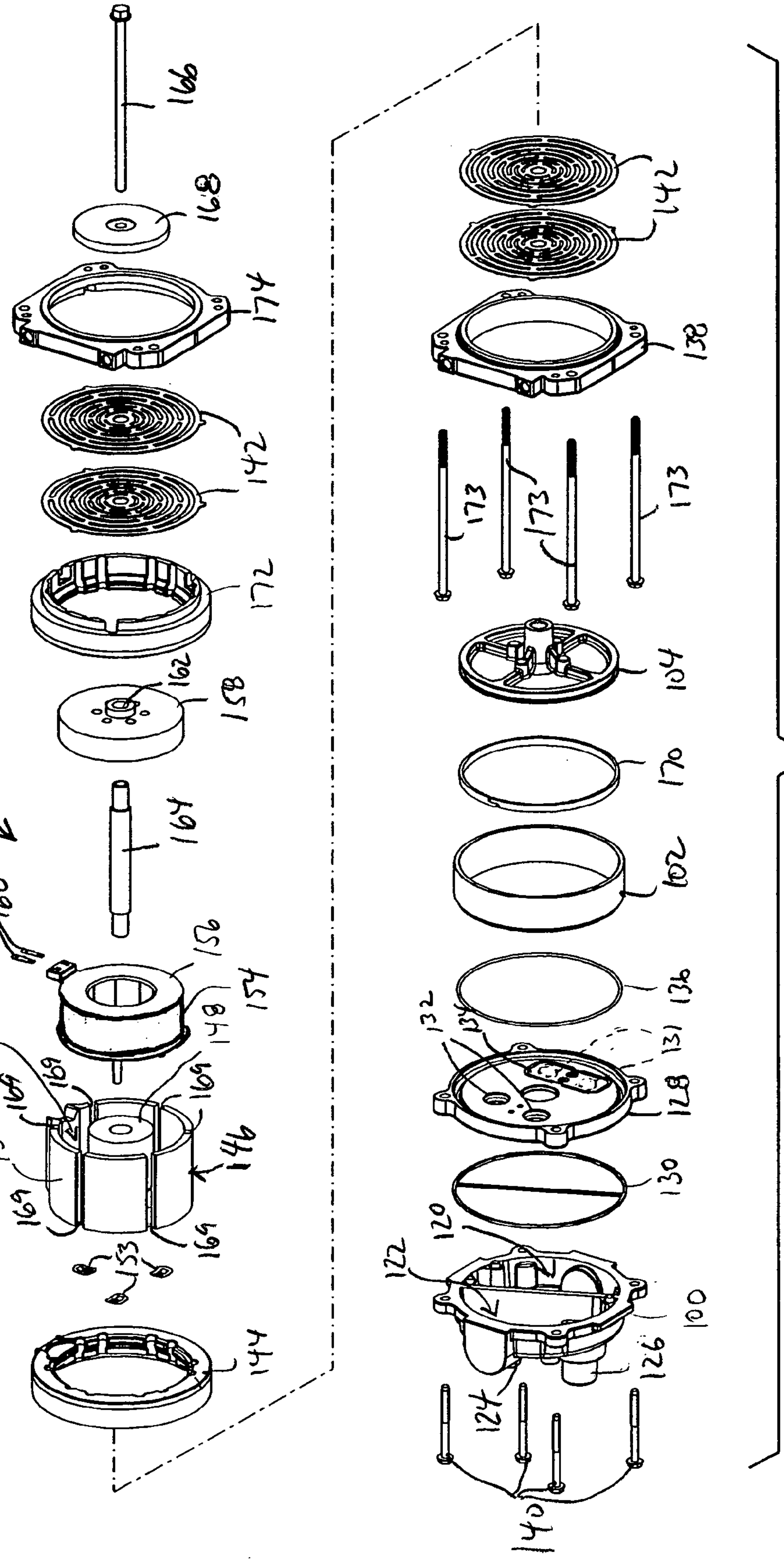
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FIG. 4

FIG. 5

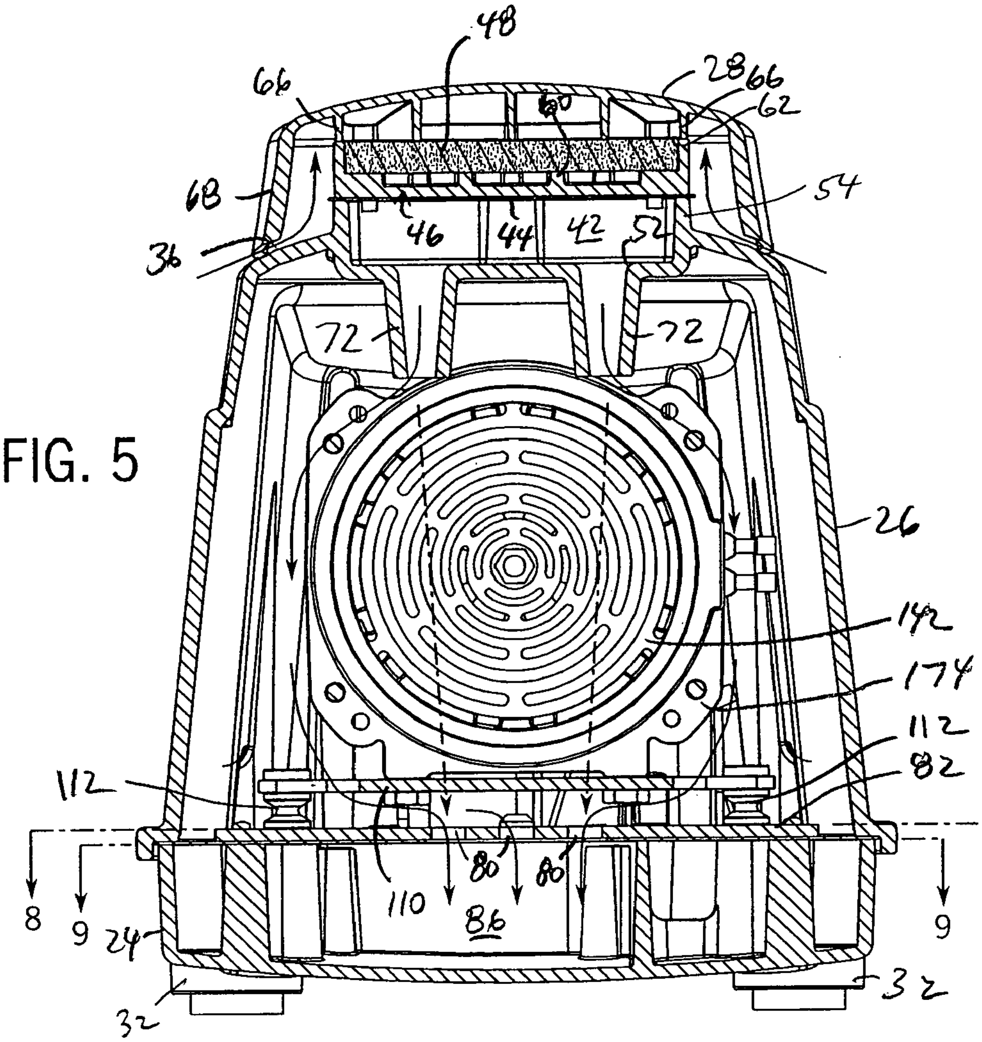
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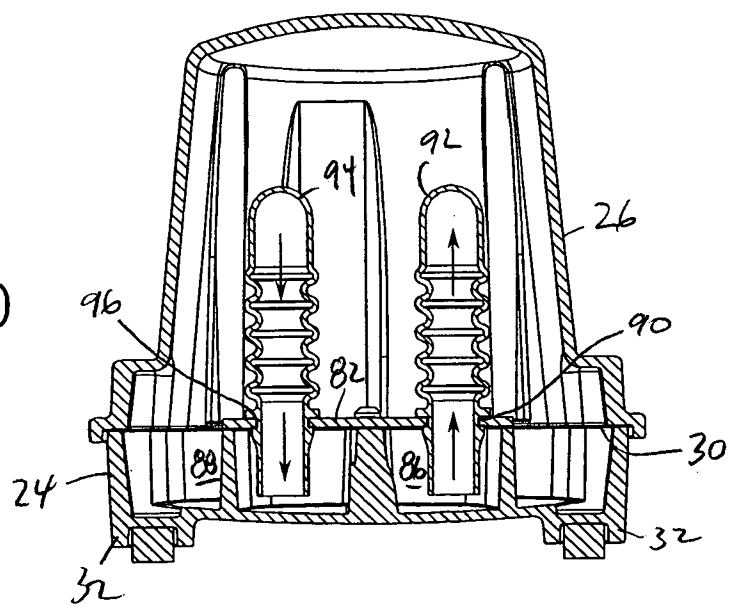
FIG. 10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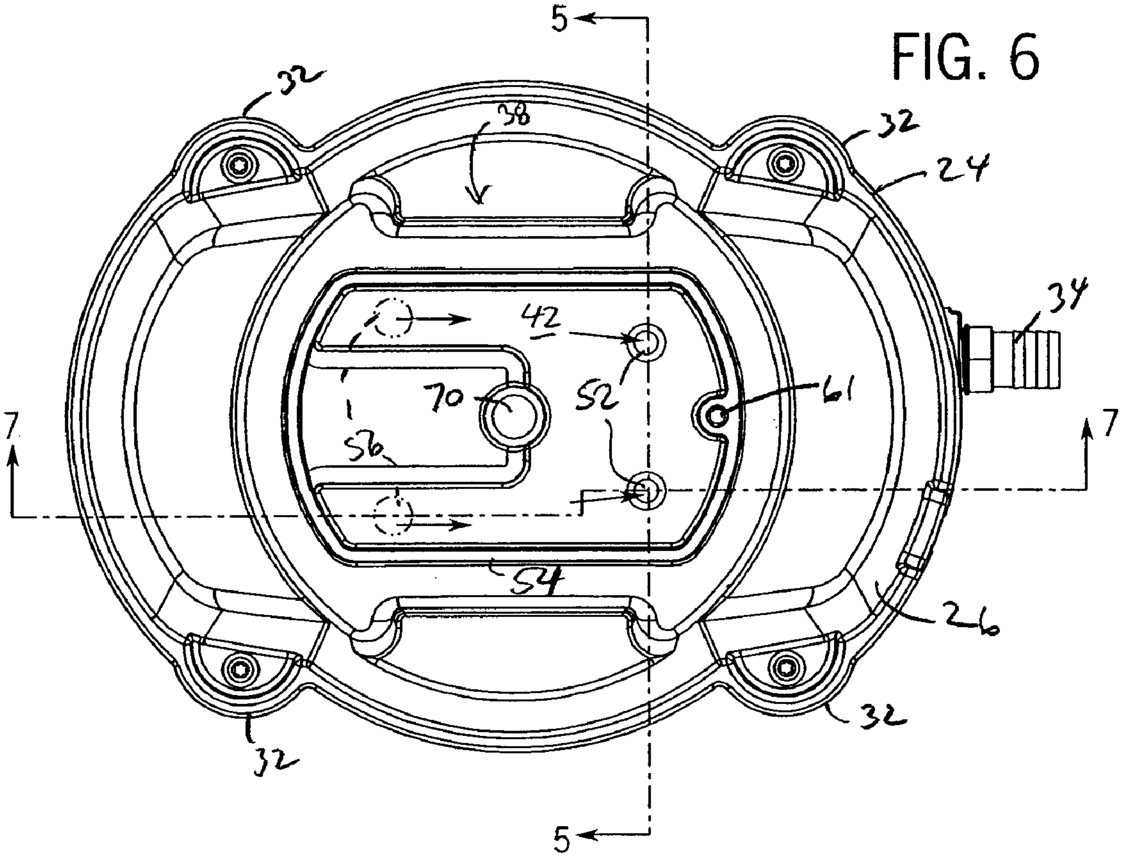
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FIG. 6

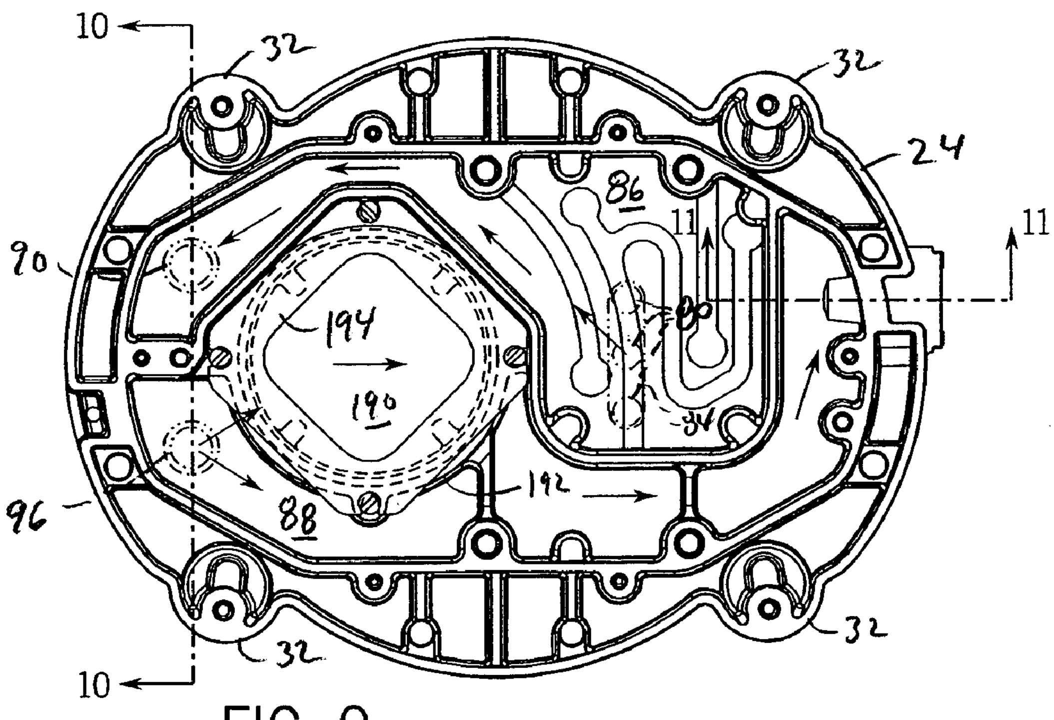
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FIG. 9

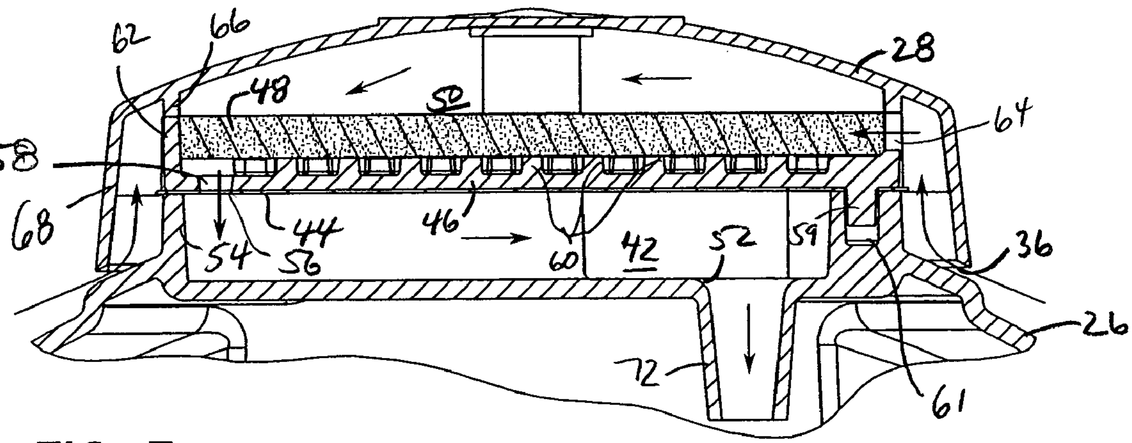

FIG. 7

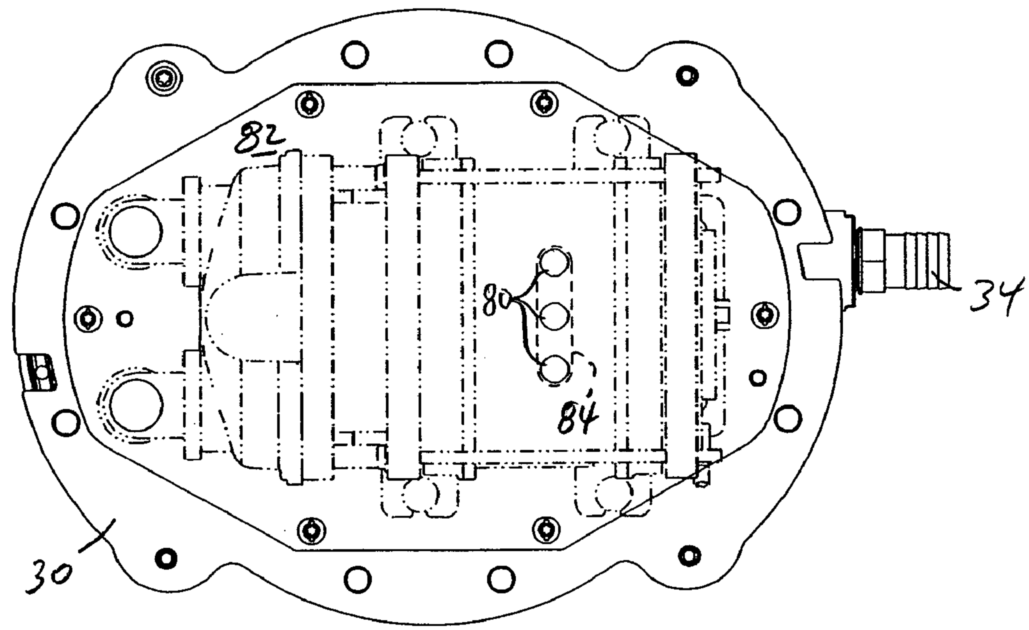

FIG. 8

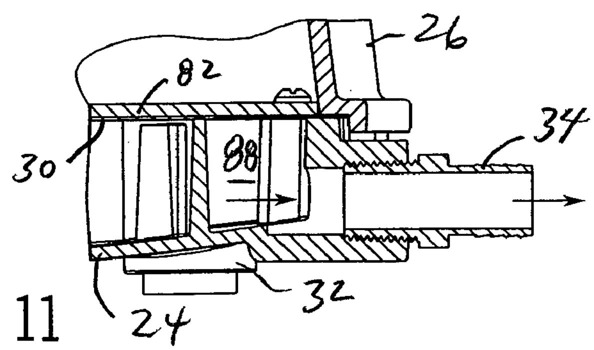

FIG. 11

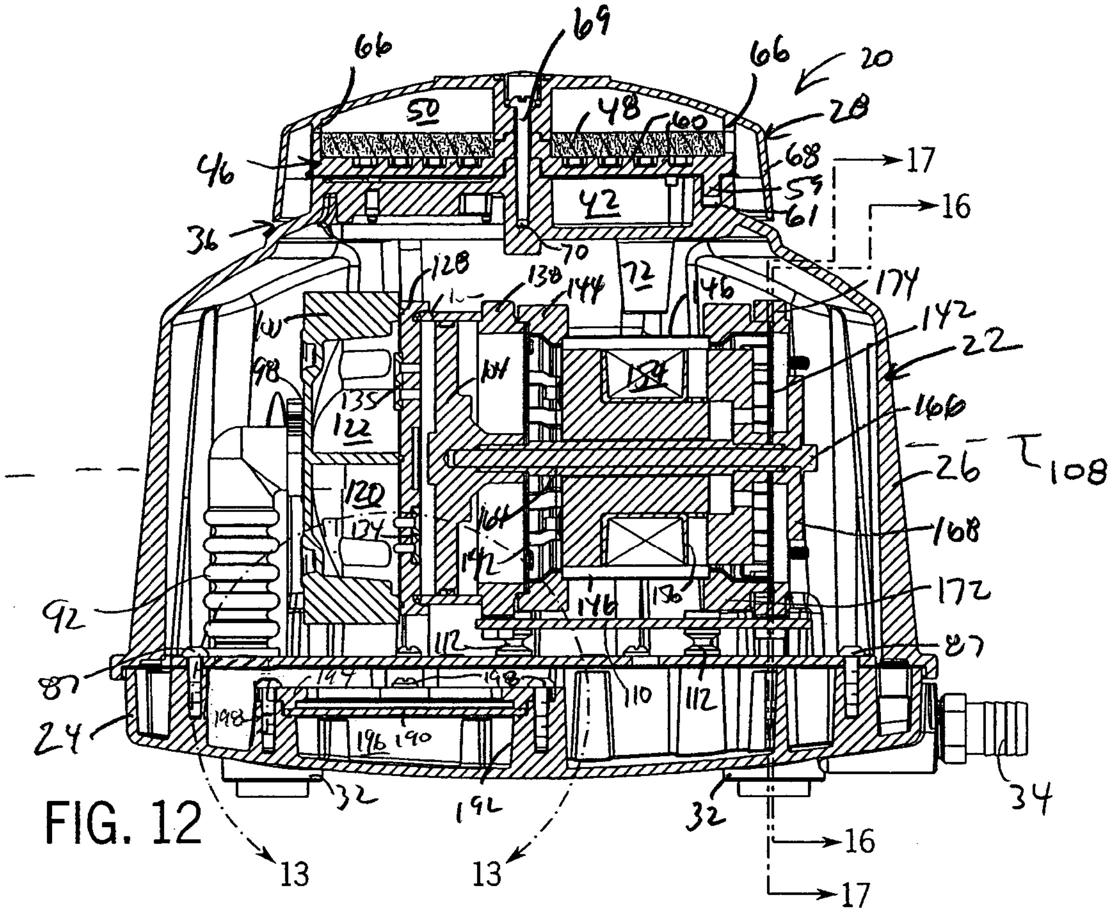

FIG. 12

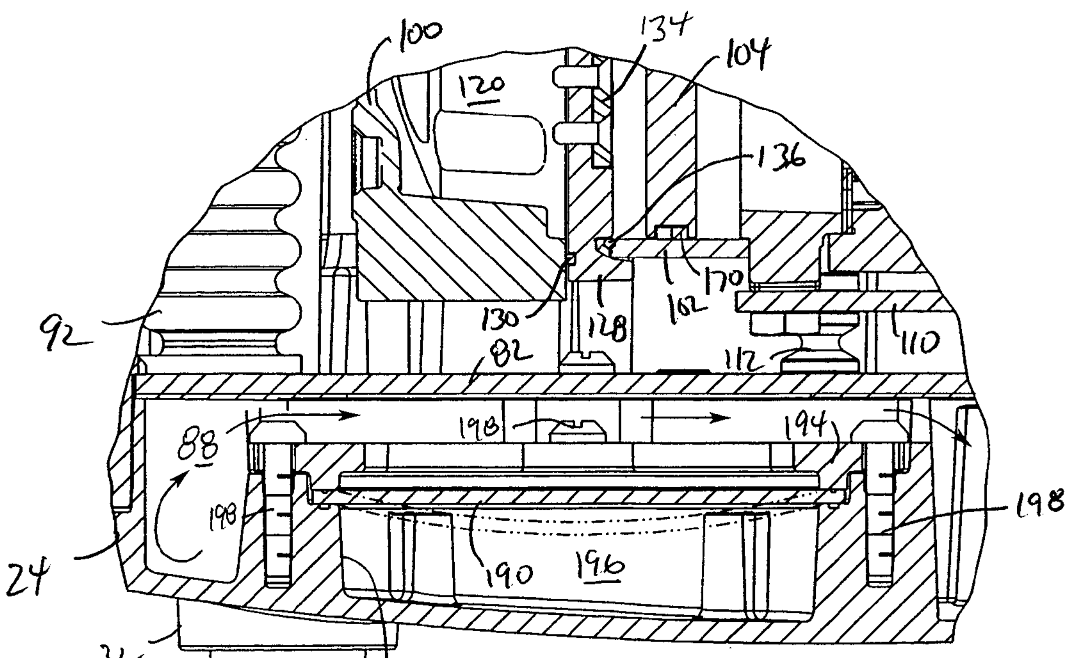

FIG. 13

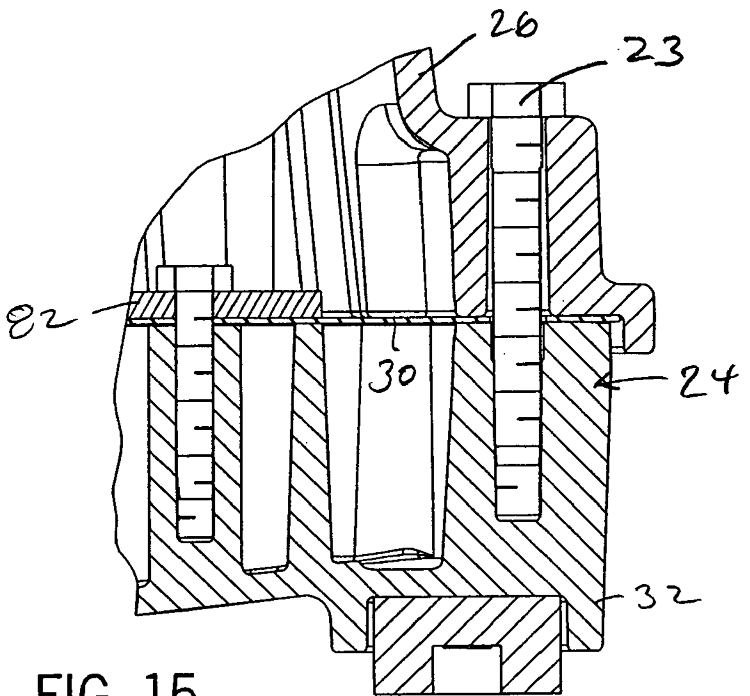
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FIG. 15

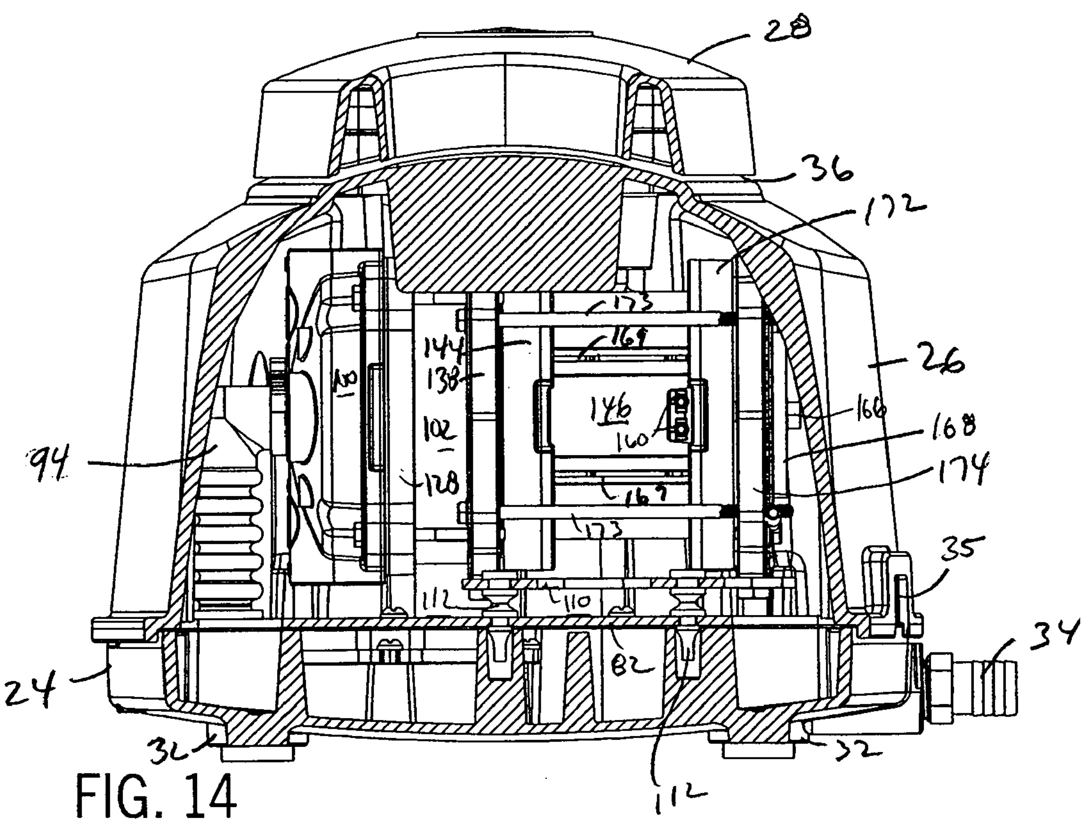
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FIG. 14

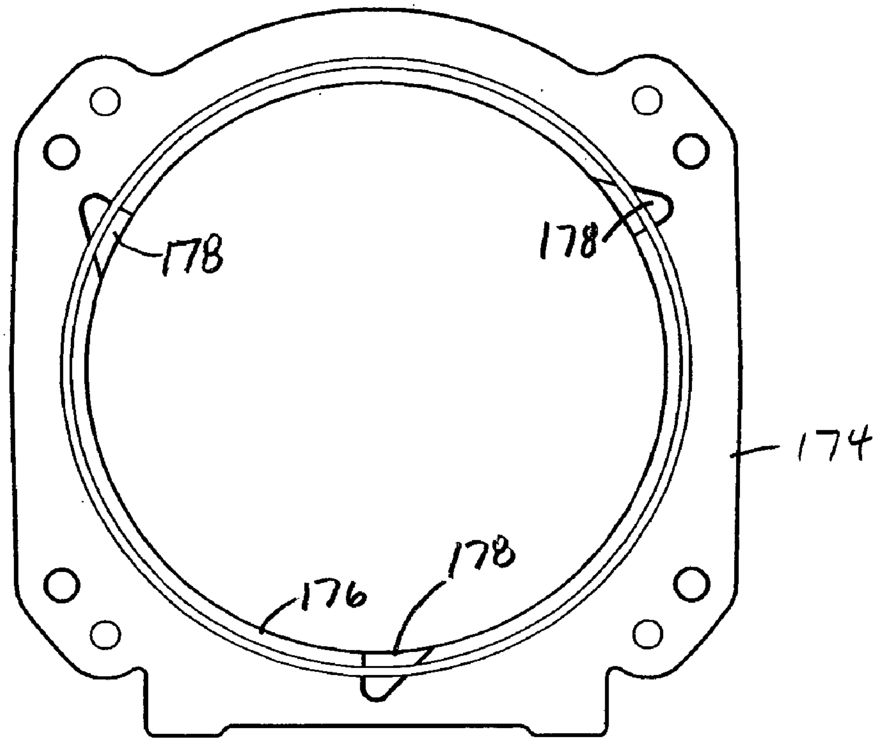

FIG. 16

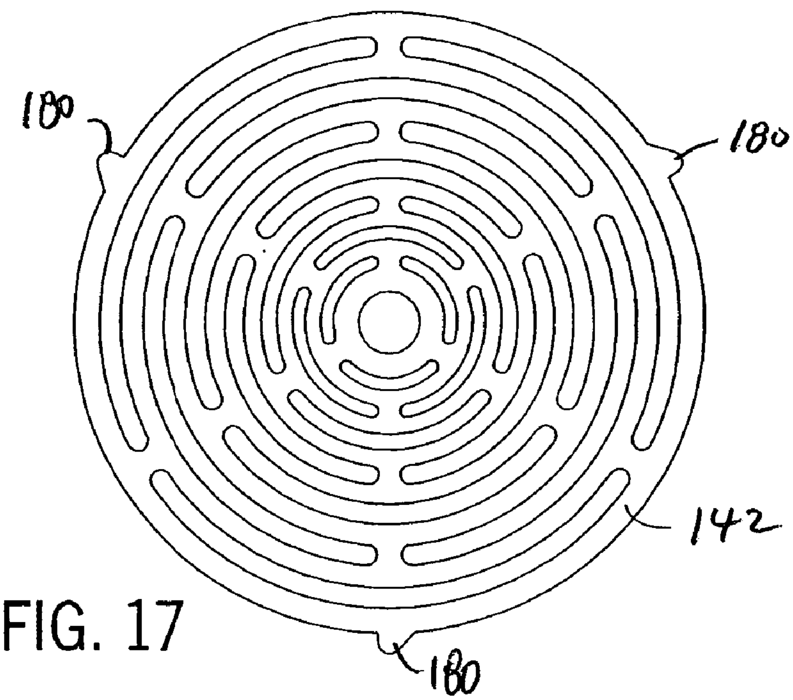

FIG. 17

LINEAR PUMP WITH SOUND ATTENUATOR

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] Not applicable.

STATEMENT OF FEDERALLY SPONSORED RESEARCH OR DEVELOPMENT

[0002] Not applicable.

BACKGROUND OF THE INVENTION

[0003] The present invention relates to pumps and in particular to compact linear piston pumps.

[0004] Pumps for certain duties, such as oxygen concentration and sewage aeration, generally need to be compact and operate discreetly. It is thus important to properly muffle the working air as well as reduce vibration during operation of the pump without relying on a large, thick-walled housing to attenuate the sound and vibration. Discreet operation of the pump can be obtained by insulating the housing, however, this adds bulk and can cause cooling problems. Mufflers can be added at the output, however, this adds hardware and cost.

[0005] Such compact linear pumps/compressors are often single cylinder devices with a small piston that reciprocates rapidly within a small cylinder to pressurize the air. The rapid movement of the single piston generates considerable vibration. These vibrations are often transferred directly to the pump housing, via a direct rigid mounting connection.

[0006] To facilitate reciprocation of the piston with less vibration, it is known to suspend the drive member, such as the armature of an electromagnetic motor, by springs or like flexible members. Stacks of thin metal leaf springs are well-suited for this. However, when multiple springs are used, it can be difficult to achieve the prescribed spring rate to which the piston drive components of the pump have been tuned. Changes in the angular (about the piston axis) and/or axial (along the piston axis) orientation of the springs relative to one another can effect the spring rate. To ensure that the pump operates efficiently, it is thus important to achieve the intended spring rate and thus ensure the consistent orientation of the suspension springs, which can make pump assembly difficult.

[0007] Another problem is that the intake and exhaust valves of the valve head must open and close rapidly for each stroke of the piston. Typically, thin metal flapper valves are used for this purpose because of their ability to seat and unseat very rapidly. Since the exhaust port opens under the force of the compressed air, a valve stop is used to support the valve and prevent it from being hyper-extended beyond its elastic range. The rapid contact between the intake valve and the valve head or the exhaust valve and the valve stop can generate tapping or clicking sounds. Another problem is that the rapid opening and closing of the intake valve can cause pressure fluctuations or pulsations in the air flow upstream from the valve head. These air pulsations can generate a low-frequency, rumbling noise.

[0008] Another problem confronting the design of compact linear piston pumps is eliminating pulsations in the output air stream. Pulsations in the air downstream from the outlet has been found to alter the resonant frequency of pump when different lengths and/or diameters of output lines are attached to the pump. Changing the operational frequency of the pump causes inefficiencies that can ultimately render the pump unusable for particular applications. It can also exacerbate noise and vibration issues.

[0009] Yet another persistent problem in compact linear pump design is cooling. To decrease noise, or perhaps to make immersible or suitable for outdoor use, the working components of these pumps are often enclosed in a pump housing. With operation of the pump, friction and the current in the electromagnet coil generate heat. As is well understood, heat adversely affects the pump efficiency and life. Many times the need to keep the pump operating efficiently requires the housing to be vented or to have other measures taken which destroy, or at least significantly reduce, the noise retarding features of the housing or other components.

[0010] Accordingly, an improved linear pump is needed that addresses the aforementioned problems.

SUMMARY OF THE INVENTION

[0011] This invention is a linear pump with improved sound attenuation characteristics. Lower overall sound is accomplished by introducing the air through a multi-cavity intake in which the intake air is allowed to re-expand before passing further downstream into the pump.

[0012] In particular, the linear pump includes an electromagnetically driven piston reciprocating within a cylinder in communication with the multi-cavity air intake. The intake has at least two cavities. The first cavity is in communication with ambient air and is in communication with the second cavity through a first orifice. The second cavity has a second orifice allowing the intake air to pass downstream to an intake chamber and ultimately to the cylinder. The second cavity is sized to allow the intake air to re-expand before exiting the second orifice.

[0013] In preferred forms, the linear pump includes a cover mounted to a pump housing to define the intake air passage, which preferably extends essentially around the entire periphery of the cover. The cover encloses (but for the intake air passage) the two cavities, one being located inward of the other outer cavity. The pump housing at least in part defines both the inner and outer cavities. A partition is mounted beneath the cover to seal off the inner cavity, except for at a pair of orifices in the partition. The outer partition orifices are preferably located near to one side of the intake than are the pair of orifices in the inner cavity. This offset arrangement causes the intake air to turn, approximately 90 degrees, in a serpentine path as it passes from the outer to the inner cavity.

[0014] The intake also preferably include a filter. The filter can be any suitable foam, paper or mesh screen air filter commonly used to screen particulates from the intake air. Preferably, the filter is disposed in a retainer having a plurality of spaced apart riser elements spacing the filter from a base surface of the retainer, which defines the outer orifice, such that intake air can pass between the inner and outer cavities through the filter and around the riser elements.

[0015] The present invention thus provides a quieter linear air pump. The unique intake design attenuates the noise, particularly a low pulsating rumbling noise, that is generated by the air rushing into the intake valve as the valve is opened and closed rapidly. The pulsations in the intake air are diminished in the cavities of the intake, particularly the inner cavity where the intake air is allowed to re-expand, and the associated noise is thus reduced. The offset arrangement of the orifices, which baffles and directs the intake air in a serpentine path, also aids in sound reduction. Similarly, passing the air through an air filter and routing the air around the raised projections of the air filter retainer further reduces noise. Still further, the pump housing enclosure reduces the noise of the pump as does the remote location of the working components and the intake chamber of the pump from the intake.

[0016] These and other advantages of the invention will be apparent from the detailed description and drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0017] **FIG. 1** is a perspective view of a linear pump according to the present invention;

[0018] **FIG. 2** is a perspective of the pump of **FIG. 1** shown with a housing shroud removed;

[0019] **FIG. 3** is an exploded assembly view of the pump;

[0020] **FIG. 4** is an exploded assembly view of a drive components of the pump;

[0021] **FIG. 5** is a sectional view taken along line **5-5** of **FIG. 6**;

[0022] **FIG. 6** is a top view showing the pump with an inlet filter and top cover removed;

[0023] **FIG. 7** is a partial sectional view taken along line **7-7** of **FIG. 6** albeit with the inlet filter and top cover in place;

[0024] **FIG. 8** is a top view of the pump with the housing shroud removed and the drive components shown in phantom;

[0025] **FIG. 9** is a top view of a pump base showing the intake and exhaust chambers thereof;

[0026] **FIG. 10** is a sectional view take along line **10-10** of **FIG. 9**;

[0027] **FIG. 11** is a partial sectional view taken along line **11-11** of **FIG. 9**;

[0028] **FIG. 12** is a sectional view taken along line **12-12** of **FIG. 1**;

[0029] **FIG. 13** is an enlarged partial sectional view taken along arc **13-13** of **FIG. 12**;

[0030] **FIG. 14** is a sectional view taken along line **14-14** of **FIG. 2** albeit shown with the housing shroud in place;

[0031] **FIG. 15** is a partial sectional view taken along line **15-15** of **FIG. 2** albeit with the outer housing in place;

[0032] **FIG. 16** is a view of a retainer ring shown in isolation as viewed from line **16-16** of **FIG. 12**; and

[0033] **FIG. 17** is a view of a leaf spring shown in isolation as viewed from line **17-17** of **FIG. 12**.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

[0034] The present invention provides an axial or linear piston pump. The term pump used herein includes a device for providing either positive or negative pressure, and thus either acting as a vacuum pump or a compressor. The pump has a compact form factor, with a preferred operating range of 2-30 psi depending upon the application (however, the pump could be designed to operate at other pressures) with low external vibration and noise and less sensitivity to pump attachments (lines, hoses, tubing, etc.) downstream from the outlet.

[0035] Referring to **FIGS. 1-3**, the pump, generally referred to in the drawings by reference number **20**, has a compact housing **22** including a base **24**, a shroud **26** and a top cover **28**. The shroud **26** is bolted to the base **24** (via bolts **23**), with a gasket **30** therebetween (see **FIG. 15**), and the cover **28** is bolted (via bolt **69**) to the shroud **26**. The base has four feet **32** to support the pump **20** and provides an opening for connecting a fitting **34** (see **FIG. 11**). The shroud **26** has an opening **35** for an electrical socket (or power cord). The cover **28** is spaced off of the shroud **26** slightly to define an air intake passage **36** along the periphery of the cover **28**, as will be described in greater detail below. The top of the shroud **26** and the cover **28** are recessed on opposite sides to provide a hand gripping area **38**.

[0036] Referring to FIGS. **3**, **5-7** and **12**, the pump **20** breathes through an air intake assembly generally designated **40**. The air intake **40** is configured to reduce the low, rumbling noise associated with pulsations in the intake air caused by rapid movement of the intake valve. In particular, the air intake **40** includes an inner cavity **42** defined in part by a recess in the top of the shroud **26**, a seal partition **44**, a filter tray **46**, a filter **48** and the cover **28**, which provides the upper boundary for an outer cavity **50**.

[0037] As mentioned, the recessed top of the shroud **26** defines the inner cavity **42**, in the floor of which are two spaced apart orifices **52** near one end (the right end in the drawings). The inner cavity **42** is bounded at the top by the partition **44** which seals against a peripheral wall **54** of the inner cavity **42**. The partition **44** has a pair of spaced apart orifices **56** located at a (left) end of the inner cavity **42** opposite the orifices **52**. To facilitate assembly the partition **44** includes another set of orifices **56'** at the opposite end, however, these are not used when the partition **42** is oriented as shown in the drawings. In any event, intake air flows through only the one set orifices **56** (or **56'**), which is located opposite the orifices **52** in the floor of the inner cavity **42**. Resting on the partition **44** is the filter tray **46** which holds the filter **48** in the outer cavity **50**. The filter tray **46** has a bottom wall with a pair of openings **58** which align with orifices **56** in the partition **44**. A small alignment feature **59** extends down from the underside of the filter tray **46** and fits into an opening **61** in the shroud **26** to ensure that the filter tray **46** is assembled in the proper orientation. The filter **48** is held spaced off of the bottom of the filter tray **46** by a number of small spaced apart risers **60**, and is retained by a short peripheral wall **62** ringing the filter tray **46**. The wall **62** has a cut-out **64** at one end (opposite the openings **58**) allowing intake air to flow laterally into the filter **48**. The cover **28** fits onto the shroud **26** over the filter **48** to define the outer cavity **50**. Ribs **66** on the inside of the cover **28** contact the wall **62** of the filter tray **46** to keep the bottom edge of the peripheral wall **68** of the cover **28** spaced slightly from the shroud **26**, and also to funnel the intake air into the outer cavity **50** through the cut-out **64**. The space between the top cover **28** and the shroud **28** defines the intake air passage **36**, which extends around the periphery of the cover **28**. Aligned center openings in the cover **28**, filter **48**, filter tray **46** and partition **44** allow a bolt **69** to screw into a threaded opening **70** in the shroud **26** to secure the assembly.

[**0038**] Referring now to **FIGS. 5**, **7**, **8** and **12**, the shroud **26** also integrally defines a pair of vent tubes **72** which extend down form the orifices **52** in the floor of the inner cavity **42**. The vent tubes **72** inject the intake air passing from the intake **40** to the inside of the shroud **26** past the drive assembly, described in detail below, through three openings **80** in a base plate **82** (and elongated opening **84** in the gasket **30**), which is bolted to the base **24**, and into an intake chamber **86** in the base **24** of the housing, as shown in **FIG. 5**. Note that the base plate **82** is bolted to the base **24** by bolts **87** (see **FIGS. 2 and 3**).

[**0039**] As shown in **FIG. 9**, the intake chamber **86** is defined in the base **24** with an exhaust chamber **88** in a yin-yang-like configuration, each with a wide portion and a narrow portion in opposite orientation. The intake air is drawn from the intake chamber **86** up through an intake opening **90** in the base plate **82** (and an associate opening the gasket **30**) and an intake tube **92** connected to the intake side of the drive assembly, as shown in **FIG. 10**. Exhaust tube **94** runs between an exhaust opening **96** in the base plate **82** (and an associated opening in the gasket **30**) to the exhaust side of the drive assembly. Importantly, the intake **92** and exhaust **94** tubes are of bellowed construction allowing the tubing to flex in response to vibrations of the drive assembly, without transferring the vibration to the housing base **24**. The intake **92** and exhaust **94** tubes simply snap into the base plate **82** and are clamped (via clamps **98**) to an associated nipple in the drive assembly.

[**0040**] Referring now to **FIGS. 2**, **4**, **5**, **12** and **14**, the drive assembly generally includes a valve head **100**, a cylinder **102**, a piston **104** and an electromagnet motor **106**, all aligned concentrically about a piston axis **108** (see **FIG. 12**). The entire drive assembly is bolted (via bolts **109**) to a motor mount **110** and mounted to the base plate **82** via resilient mounts **112**. The four resilient mounts **112** have narrowed necks between enlarged heads and bodies that slideably snap into four open ended slots **114** in the motor mount **110**. Slightly enlarged bottom ends of the resilient mounts **112** can be pushed straight down through openings **116** in the base plate **82**, thereby, the motor mount **110** is captured by the resilient mounts **112** and flexibly mounted to the base plate **82**. The resilient mounts **112** are preferably made of a suitable rubber, such as an EPDM with a durometer of about 60, so as to provide enough stiffness to securely mount the drive assembly while allowing enough flexibility to isolate the vibrations of the drive assembly.

[**0041**] With primary reference to **FIGS. 2, 4** and **12**, the drive assembly will now be described in detail. The valve head **100** defines an intake side **120** and an exhaust side **122** each having a respective nipple **124** and **126** to which the intake **92** and exhaust **94** tubes connect via clamps **98**. A valve plate **128** mates with the valve head **100**. The valve plate **128** has a groove for a double D-shaped (bisected circle) o-ring **130** sealing against the valve head **100** to isolate the intake side **120** from the exhaust side **122**. The valve plate **128** is generally disk-shaped and defines a pair of intake ports **131** (shown in phantom in **FIG. 4**) and a pair of exhaust ports **132**. Each pair of ports is covered by respective thin metal flapper valves **134** and **135** (the flapper valves **134** and **135** can be supported by valve stops (not shown)). The intake **131** and exhaust **132** ports are in communication with the associated sides of the valve head **100** and the inside of the cylinder **102**, which fits into another groove in the back side of the valve plate **128** sealed with another o-ring **136** (see **FIG. 13**). The opposite side of the cylinder **102** fits around a hub of a retainer collar **138**. The retainer collar **138** has four threaded openings, which receive four bolts **140** fit through four ears of each of the valve head **100** and valve plate **128** to clamp the cylinder **102** tightly together with these components.

[**0042**] The other side of the retainer collar **138** clamps one or more leaf springs **142** with a recessed groove (having alignment features as discussed below) in a spacer ring **144**. The opposite side of the spacer ring **144** receives a stator **146** of the electromagnet motor **106**. The stator **146** is a slotted annular member having a circular base and concentric inner **148** and outer **150** cylindrical walls (with axial slots **169** in outer wall **150**), which define an annular channel **152** therebetween. A wire coil **154** is disposed in a bobbin **156** within the channel **152**. The bobbin **156** has three posts that extend through openings in the base of the stator **146** and are engaged by retainers **153** to retain the bobbin **156** and coil **154**. A diode (not shown) may be electrically coupled to the coil **154** to rectify the alternating current input signal so that it drives an armature (or shuttle) **158** in only one direction, preferably toward the stator **146**. Conductive tabs **160** for coupling the coil **154** to the power are also included.

[**0043**] The armature **158** has a series of axial bores therethrough and slides in and out of a side (right in the drawing) of the stator **146** when the coil **154** is energized. The armature **158** has a short hub with an axial bore **162** that receives a bottom end of a connecting rod **164**. The connecting rod **164** is suspended along the piston axis **108** by the leaf springs **142** and passes through the center bore in the stator **146**. The connecting rod **164** is secured to the armature **158** and the piston **104** by a long bolt **166** threaded into the piston **104** and mounting a mass disk **168** under its head.

[**0044**] The stator **146** is clamped between the spacer ring **144** and another spacer ring **172**. That spacer ring **172** clamps one or more additional leaf springs **142** against a second retainer collar **174**. Four tie rods **173** extending through ears in the first retainer collar **138** are threaded into openings in ears of the second retainer collar **174** to unite the components of the motor **106**. The retainer collars **138** and **174** also have threaded openings receiving bolts **109** to connect the motor mount **110** and thereby mount the entire drive assembly to the base plate **82** via the resilient mounts **112**, as described above.

[**0045**] As shown in **FIGS. 16 and 17**, at the side opposite the cylinder **102**, the retainer collar **174** (as well as spacer ring **144**) has a circular groove **176** about its inner periphery with three alignment pockets **178**, the sides of which taper asymmetrically away from the groove **176**. One or more leaf springs **142** fit into the groove **176** and their asymmetrically tapered alignment tabs **180** fit into the pockets **178**. Due to the asymmetric configuration of the pockets **178** and tabs **180**, the leaf springs **142** can seat properly into the retainer collar **174** (and the spacer ring **144**) in only one angular and axial orientation (relative to the piston axis **108**). The alignment tabs **180** are spaced apart about 120 degrees so that the springs can be mounted in one of three angular orientations. If a single angular orientation is desired, the alignment tabs **180** could be spaced asymmetrically. This facilitates and ensures the assembly of multiple leaf springs **142** in the same orientation at both ends of the connecting rod **164**. In particular, the web pattern (and arcuate slots) of the leaf springs **142** are aligned, and any curvature or bowing of the leaf springs **142** out of the plane perpendicular to the piston axis **108** (which can occur from the die cutting process) will be in the same direction for each leaf spring **142**.

[**0046**] This is important to ensure that the motor **106** has the spring rate for which it was designed. Specifically, during development the pump is tuned to operate at a frequency at or near its natural resonant frequency. In particular, the pump is operated with a load applied and using a calculated spring-mass system (i.e., the combination of spring rate of the springs **142** and mass of the moving components, namely the piston **104**, armature **158**, connecting rod **164** and any mass disk **168**). The frequency of the input signal to the motor **106** is varied as various parameters are measured. For example, because power consumption goes up as the input frequency strays from the resonant frequency of the spring-mass system, power consumption measurements can be used to adjust the spring-mass system so that its natural frequency will be at or near that of a typical input signal, for example 60 Hz. Operating the pump at the resonant frequency improves efficiency, and reduces vibration, and thereby noise. The spring-mass system can also be adjusted to operate efficiently at different pressures. For example, by increasing mass or spring rate the spring-mass system can be made to operate at or near resonant frequency while the pump is providing increased pressure output. It should be noted that the mass disk **168** is used as a cost effective alternative to increasing or decreasing the mass of the piston, armature and/or connecting rod.

[**0047**] As is well understood, the piston **104** is driven by movement of the armature **158**, when energy is supplied to the wire coil **154**, to reciprocate within the cylinder **102**. The piston **104** has an enlarged head with a peripheral groove holding a split piston ring **170** that seals against the cylinder **102** when pressure is developed. The stroke length is approximately 8 mm (4 mm in each direction) and is positioned approximately 1 mm from the top of the cylinder when at top dead center.

[**0048**] Given the single cylinder arrangement of the pump **20**, the reciprocating piston **104**, armature **158** and connecting rod **164** can cause the drive assembly inside the housing to vibrate. The leaf springs **142** absorb much of the energy from these moving components. The number, size and thickness of the leaf springs **142** are selected to achieve a spring rate determined primarily according to the mass of the piston **104** and the input frequency. The leaf springs **142** are selected so that in combination (between the two stacks) they result in a resonant frequency of the piston **104** and springs **142** (i.e., the spring-mass system) approximately equal to the input frequency, which is typically 50 or 60 Hertz. For example, in one preferred embodiment there is a stack of two springs in the second retainer collar **174** and a stack of two springs in the spacer ring **144** near the piston **104**. If the stroke length were to be increased, for example if the pump to be used in an application requiring more air flow, the springs **142** could be of a thinner gauge, in which case the number of springs may be increased to three in each stack to achieve the same spring rate.

[**0049**] With reference to **FIGS. 5-7**, **9-11** and **13**, air flow through the pump will now be described in detail for a preferred compressor embodiment of the pump. When the drive assembly is operating, ambient air is drawn into the pump intake through the intake air passage **36** around the periphery of the top cover **28**. As shown in **FIGS. 5-7**, the intake air is drawn through the intake air passage **36** flowing upwardly along the ribs **66** and makes its way into the outer cavity **50** through the cut-out **64** in the peripheral wall **62** of the filter tray **46**. Intake air then moves from that end of the outer cavity **50** through the filter **48** and around the risers **60** in the filter tray **46** to the openings **58** in the filter tray **46** and the partition **44** where it enters the inner cavity **42**. The pressure of the intake air drops as it passes through the small openings into the inner cavity **42**. The inner cavity **42** is effectively larger than the outer cavity **50**, which allows the intake air to expand. The expansion of the intake air in the inner cavity **42** helps to dissipate the pulsations in the intake air arising from the operation of the intake valve. After entering the inner cavity **42**, the intake air turns through a bend of about 90-180 degrees and travels to the other end of the inner cavity **42** to the orifices **52** where it travels down the vent tubes **72** and into the interior of the housing shroud **26**. As shown in **FIG. 5**, the vent tubes **72** are located to direct the intake air into the slots **169** in the stator **146** of the motor **106** so as to convectively cool the coil **154**. After passing through and around the motor **106**, the intake air passes through the openings **80** in the base plate **82** into the wide part of the intake chamber **86**. The air is routed through the narrowed part and up through the intake opening **90** in the base plate **82** and the intake tube **92** to the intake side **120** of the valve head **100**, as shown in **FIG. 10**. Reciprocation of the piston **104** draws air into the cylinder **102** and compresses it. The pressurized air is then passed through the exhaust side **122** of the valve head **100** and down through the exhaust tube **94** in the base plate **82** to the wide part of the exhaust chamber **88** through exhaust opening **96**. As shown in **FIGS. 9 and 11**, the pressurized air flows through the narrow portion of the exhaust chamber **88** and out through the outlet opening where fitting **34** is attached to suitable hose or tubing (not shown).

[**0050**] Since the pump has only a single cylinder, the pressurized air is pulsed at the rate of the input frequency, for example 60 Hz. The inventors have determined that pulsations in the output air can adversely effect the operation of the pump. In particular, the output lines act as resonant chambers, having their own natural frequency. If the pulsations in the output air are at a different frequency, the air will effectively encounter increased resistance going through the output lines. This creates excessive back pressure on the pump so that the spring-mass system can be made to operate at a different (non-resonant) frequency, thereby decreasing the efficiency of the pump. This makes the pump more sensitive to variations in input frequency which can further decrease efficiency. By reducing the amplitude of the pulsations in the air before leaving the pump, this problem can be avoided.

[0051] To that end, as shown in **FIGS. 3, 9** and **13**, the pulsations in the pressurized output air leaving the pump are dampened by a diaphragm **190** in the exhaust chamber **88**. In particular, the diaphragm **190** is preferably a rubber disk mounted to the top of a cup **192** defined by the housing base **24** in the wide part of the exhaust chamber **88** by a support ring **194** bolted (via bolts **198**) to the base **24**. The cup **192** defines a trapped air pocket **196** below the diaphragm **190**. As the pressurized air flows through the exhaust chamber **88**, the pulsations act against the diaphragm **190**, tending to make it flex into the cup **192**. The air pocket **196** will compress slightly, but tend to resist inward movement of the diaphragm **190**. This reactive force on the diaphragm **190** will tend to counter, and thus cancel or reduce the amplitude of, the pulsations in the exhaust chamber air. The diaphragm **190** and trapped air pocket **196** act similar to an accumulator, and as a result, allow the pump to output smoother, relatively non-pulsed, air through the output lines. As a result pump inefficiencies are avoided that may otherwise arise from changes in the length or diameter of the output lines or from changes to the input frequency to the motor. As an example of one advantage, the inventors have determined that use of the diaphragm in this way allows a single mass-spring system to be used for both 50 and 60 Hz applications.

[0052] An illustrative embodiment of the present invention has been described above in detail. However, the invention should not be limited to the described embodiment. To ascertain the full scope of the invention, the following claims should be referenced.

What is claimed is:

1. A linear pump, comprising an electro-magnetically driven piston reciprocating within a cylinder in communication with a multi-cavity air intake, the intake including a first cavity and a second cavity, the first cavity being in communication with ambient air and being in communication with the second cavity through a first orifice and the second cavity being in communication with a downstream section of the pump through a second orifice, wherein intake air re-expands in the second cavity before passing through the second orifice.

2. The pump of claim 1, wherein the intake creates a first pressure drop through the first orifice and a second pressure drop through the second orifice.

3. The pump of claim 1, wherein the second cavity is inward of the first cavity.

4. The pump of claim 1, wherein at least the second cavity is defined by a pump housing.

5. The pump of claim 1, further including partition defining the first orifice and sealing the second cavity from the first cavity.

6. The pump of claim 1, wherein the first orifice is offset from the second orifice so that the intake air turns directions between the first and second orifices.

7. The pump of claim 6, wherein there are a pair of first orifices and a pair of second orifices.

8. The pump of claim 1, further including a cover mounted to a pump housing over the first cavity and defining an air passage from ambient to the first cavity between the cover and the pump housing.

9. The pump of claim 8, wherein hold offs keep the cover spaced from the pump housing.

10. The pump of claim 8, wherein the air passage extends along the periphery of the cover.

11. The pump of claim 1, wherein the intake includes a filter member.

12. The pump of claim 11, wherein the filter member is disposed in the first cavity.

13. The pump of claim 11, wherein the filter member is held by a retainer having a plurality of spaced apart riser elements spacing the filter member from a base surface of the retainer such that intake air can pass from the first cavity through the filter member around the riser elements and into the second cavity through an orifice in the retainer and the first orifice.

14. The pump of claim 1, wherein the pump includes a pump housing enclosing the piston and cylinder arrangement.

15. The pump of claim 1, wherein the second orifice is spaced from the cylinder.

16. The pump of claim 1, wherein the cylinder and an intake chamber are spaced from the intake.

17. A linear pump having a pump housing containing an electro-magnetically driven piston reciprocating within a cylinder in communication with a multi-cavity air intake, the intake including:

a cover mounted to the pump housing to define an intake air passage;

an inner cavity defined in part by the pump housing having an inner orifice in communication with an intake chamber of the pump; and

an outer cavity defined between the cover and a partition mounted to the pump housing to seal off the inner cavity, the outer cavity being in communication with ambient air through the intake air passage and in communication with the inner cavity through an outer orifice in the partition which allows intake air to re-expand in the inner cavity before passing through the inner orifice.

18. The pump of claim 17, wherein the outer orifice is offset from the inner orifice so that the intake air flow turns approximately 90 degrees between the outer and inner orifices.

19. The pump of claim 18, wherein there are a pair of inner orifices and a pair of outer orifices

20. The pump of claim 17, wherein the intake air passage extends along the periphery of the cover.

21. The pump of claim 17, wherein the intake further includes a filter.

22. The pump of claim 21, wherein the filter member is disposed in a retainer having a plurality of spaced apart riser elements spacing the filter member from a base surface of the retainer defining the outer orifice such that intake air can pass between the inner and outer cavities through the filter and around the riser elements.

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