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[54] FREE PISTON POWER SOURCE

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[52] **U.S. Cl.** **91/232,** 92/85, 92/143
[51] **Int. Cl.** **F01l 21/02,** F01b 11/02
[58] **Field of Search** 91/234, 232; 92/85, 143, 8; 417/341

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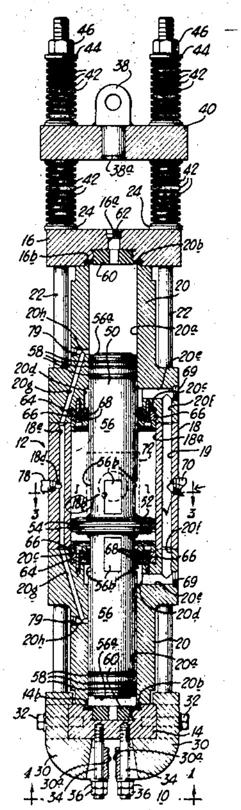
[57] **ABSTRACT**

A free piston vibrator power source for applications wherein control of vibration frequency, force, and amplitude is important and useful in underwater applications as a sonic generator. The vibrator includes a cylinder and a fluid piston slidably mounted in the cylinder for vibrating reciprocal movement between opposite ends. Valve means is provided for introducing pressurized activating fluid into the cylinder on opposite sides of said piston in response to the position of said piston between opposite ends of said cylinder and a discharge port is provided for exhausting fluid from opposite sides of the piston in response to the position thereof. Fluid storage chambers are provided at opposite ends of the cylinder and rod-like piston members on opposite sides of the piston are slidable in the chambers which contain compressible fluid at pressures independent of the activating fluid pressure.

5 Claims, 8 Drawing Figures



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SHEET 1 OF 2

Fig. 1

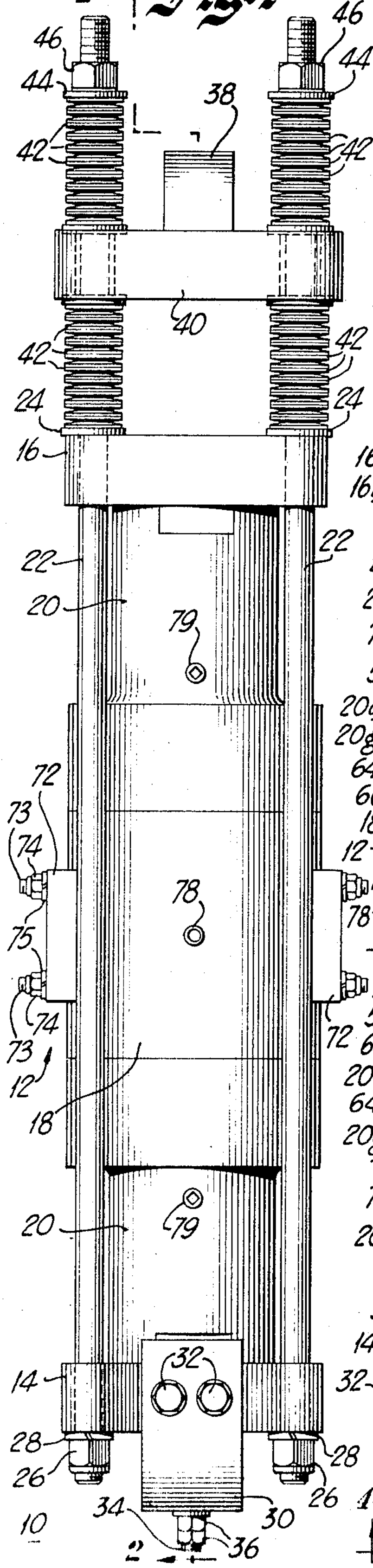


Fig. 2

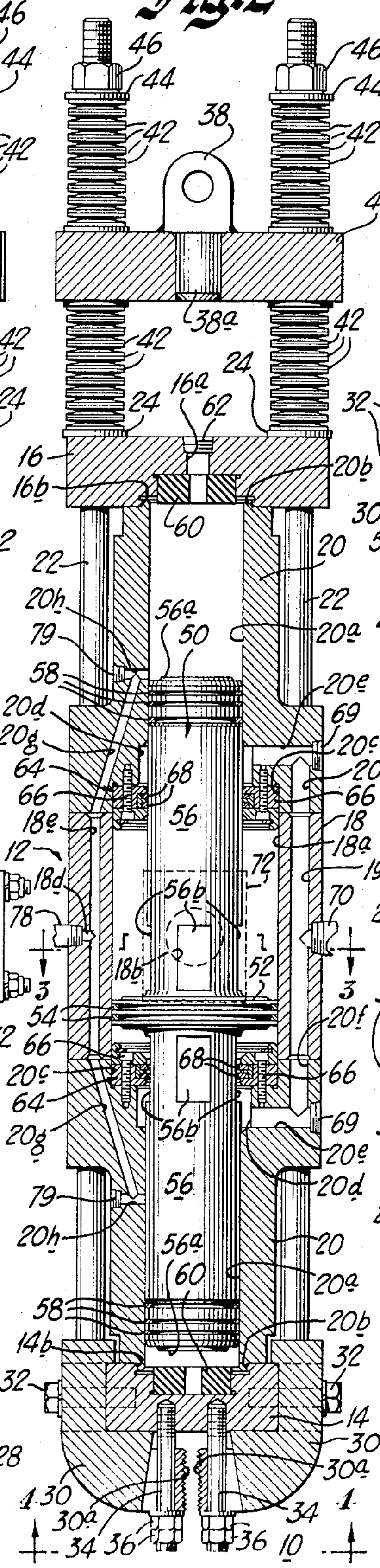


Fig. 3

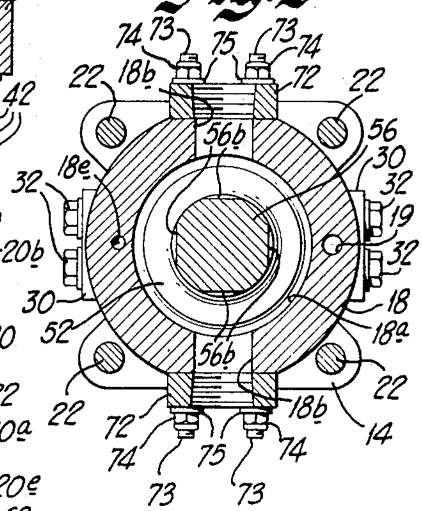
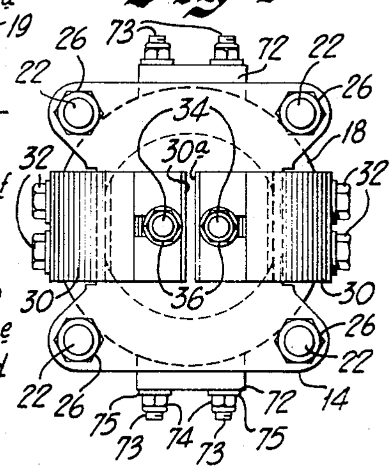


Fig. 4



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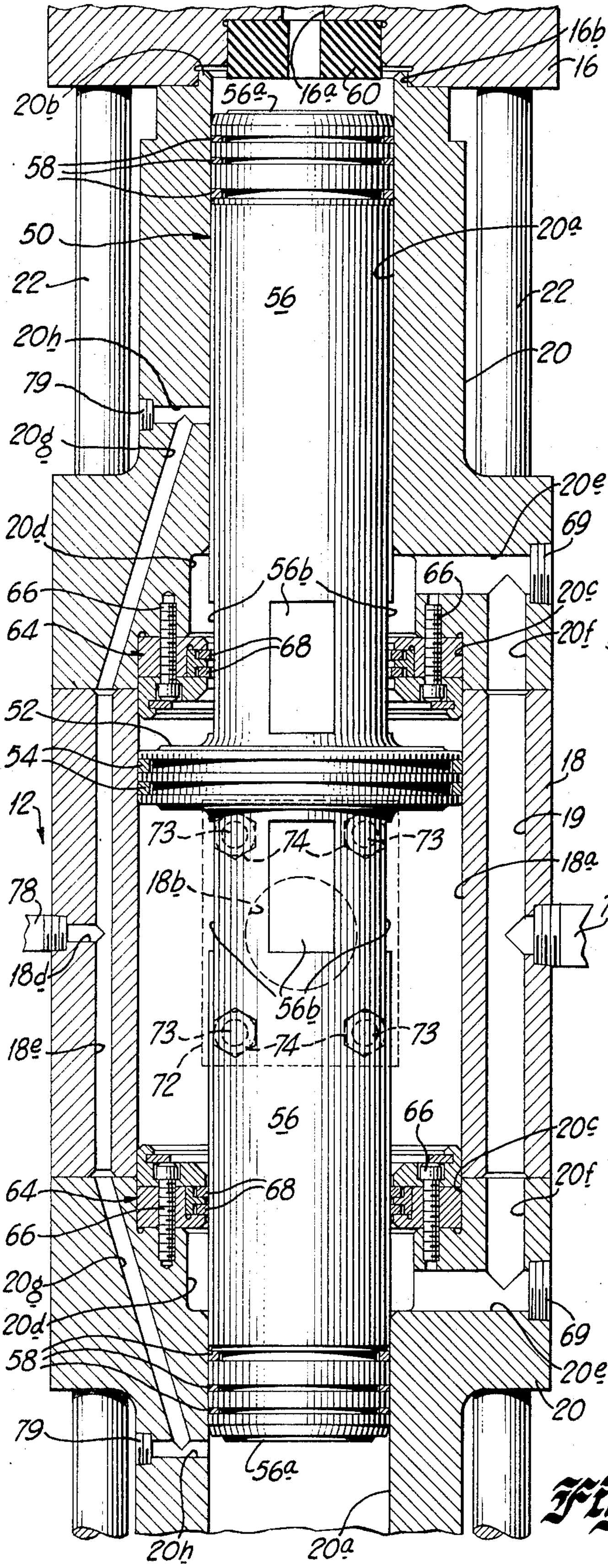


Fig. 5

Fig. 6

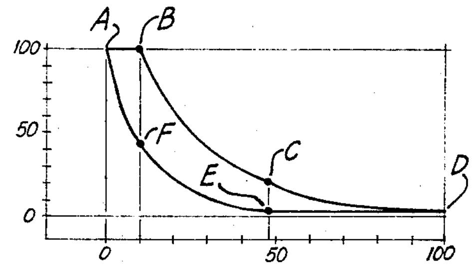


Fig. 7

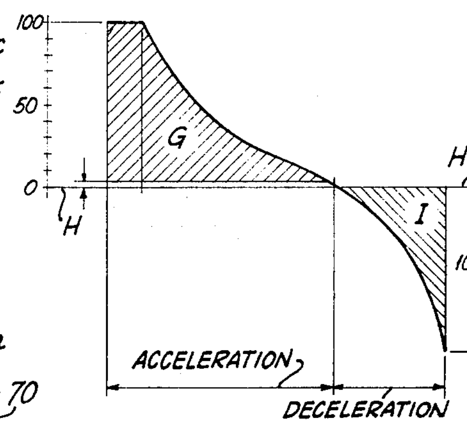
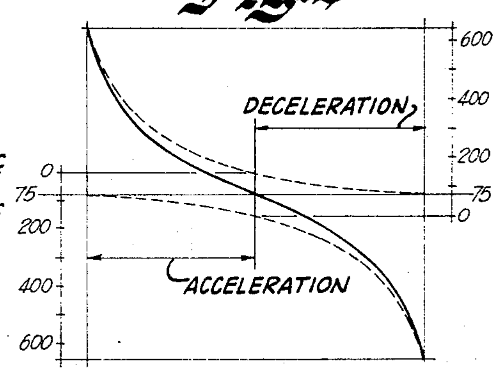


Fig. 8



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FREE PISTON POWER SOURCE

The present invention relates to a new and improved free piston vibratory power source useful in applications wherein selective control of the vibration frequency, the force, and amplitude is desirable.

The free piston power source is extremely useful in underwater applications for a variety of purposes including use as a hydrosonic generator and is also well suited to more conventional uses as a power source for pile drivers, tampers, tool pullers, oil well sinking, and other mechanical shaking applications.

In underwater applications, as a hydroacoustic horn, the free piston vibrating power source of the present invention is especially well suited because no rotating shafts are required and no electrical motors or associated electrical control apparatus is needed. Maintenance of shaft bearings and electrical machinery and controls in an underwater environment is difficult and costly, and the vibrator eliminates these items yet still provides means for easy, selective control of the frequency of vibration, the force magnitude, and amplitude of vibration.

It is an object of the present invention to provide a free piston vibratory power source which is driven by compressible fluids, such as compressed air or steam, and which can be readily controlled to provide the desired frequency, amplitude, and force of vibration desired.

Another object of the present invention is to provide a free piston vibrating power source of the character described in which both the frequency and the force can be easily controlled and selected and in which the magnitude of inertial forces developed can be selectively varied and controlled as desired.

Another object of the present invention is to provide a new and improved linear vibrator which is extremely efficient and provides a high output with a minimum of fluid consumption.

Another object of the present invention is to provide a free piston power source of the character described which is valveless in the sense that no complex, conventional valve mechanisms are required to control the reciprocation or the vibration frequency or stroke.

Another object of the present invention is to provide a free piston linear vibrator of the character described which is relatively low in cost, high in operating efficiency, and requires a minimum of maintenance.

Another object of the present invention is to provide a new and improved free piston powered linear vibrator of the character described which does not require rotating shafts and eccentrics, electric motors and associated controls and, accordingly, is well suited for applications undersea wherein extreme conditions of compression, corrosion, and maintenance inaccessibility prevail.

Yet another object of the invention is to provide a new and improved free piston linear vibrator of the character described which does not require mechanical springs for deceleration of a vibrating mass.

Still another object of the invention is to provide a new and improved free piston linear vibrator for generating aquatic sonic impulses.

Yet another object of the invention is the provision of a new and improved free piston power source useful as a hydroacoustic horn and sonar screen source.

The foregoing and other objects and advantages of the present invention are accomplished in an illustrated embodiment thereof which comprises a free piston linear vibrator having a body defining an expansion cylinder therein. A piston member is slidably mounted in the cylinder for reciprocal movement between opposite ends thereof and first valve means is provided for introducing pressurized fluid into the cylinder on one side of the piston for movement of the piston in one direction. Second valve means is provided for introducing pressurized fluid into the cylinder on the opposite side of the piston for moving the piston in the opposite direction. A discharge port is provided in the cylinder for exhausting fluid from respective opposite sides of the piston in response to the position thereof in said cylinder with respect to the port. The piston reciprocates within the cylinder providing inertial force vibrations of the desired frequency and strength. Fluid compression chambers are provided at opposite ends of the expansion cylinder for cushioning the end of each stroke.

Briefly, the foregoing and other objects and advantages of the present invention will best be understood by reference to the following detailed description, taken in conjunction with the drawings, in which:

FIG. **1** is a side elevational view of a free piston linear vibrator constructed in accordance with the features of the present invention;

FIG. **2** is a cross-sectional view taken substantially along line **2—2** of FIG. **1**;

FIG. **3** is a transverse cross-sectional view taken along line **3—3** of FIG. **2**;

FIG. **4** is an end elevational view looking in the direction of the arrows **4—4** of FIG. **2**;

FIG. **5** is an enlarged, fragmentary, longitudinal, sectional view similar to FIG. **2** but showing the piston member in an opposite position;

FIG. **6** is a pressure-versus-stroke diagram in schematic form for an expansion cylinder of the linear vibrator with theoretical adiabatic expansion and compression;

FIG. **7** is a theoretical composite pressure-versus-stroke diagram for the expansion cylinder; and

FIG. **8** is a theoretical isothermal pressure-versus-stroke diagram for a compression chamber.

Referring now, more particularly, to the drawings, a free piston power source comprising a linear vibrator, designated generally by the numeral **10** in FIGS. **1** and **2**, is therein illustrated and is constructed in accordance with the features of the present invention. The vibrator **10** includes an elongated cylindrical body generally indicated as **12**, terminating in a lower end plate or head **14** at one end and an upper end plate or head **16** at the opposite end. The head plates are mounted at opposite ends of the generally cylindrical body section which comprises a large diameter, centrally positioned, expansion cylinder **18** and a pair of outer end body sections **20** providing compression chambers disposed on opposite sides of the central cylinder. The heads **14** and **16** close against the opposite ends of the cylindrical body **12** and the body sections and heads are secured together by a plurality of elongated tension rods **22** having intermediate thrust collars **24** which bear against the outside surface of the upper head **16**. The lower ends of the rods are threaded for locknuts **26** and washers **28** which bear against the lower head **14**.

In order to couple vibratory energy from the linear vibrator **10** to do useful work, the lower head plate **14** is provided with a pair of coupling jaws **30** which are secured to the head **14** by means of lateral cap screws **32** and longitudinal studs **34** and nuts **36**. As best shown in FIG. **4**, the generally L-shaped jaws or brackets **30** are bifurcated and are threaded along facing edge surfaces **30***a* (FIG. 2) for clamping engagement with devices to be vibrated by the device **10**.

The vibrator **10** is adapted to be supported from a cable or pin, and a clevis **38** having a shank **38***a* (FIG. 2) is attached to a support plate **40** mounted on the rods **22** above the head **16**. Stacks of Belleville-type spring washers **42** or coil springs are provided in order to space the support plate **40** away from the upper head plate **16**, yet permit relative movement between the plate **40** and the vibrator body **12**. Compression washers **44** and nuts **46** are provided adjacent the outer end of longitudinal rods **22** to maintain the desired amount of compression on the stacks of spring washers **42**. The linear vibrator **10** is thus resiliently supported from the clevis **38** and support plate **40** and is shock mounted in relation to the support by means of the stacks of Belleville spring washers on the rods. Suitable coil springs may be used instead of the stacks of spring washers if desired, and the number of spring washers can be changed to suit the specific spacing and resiliency requirements.

In accordance with the present invention, the linear vibrator **10** includes an elongated, generally cylindrical piston member **50** which is reciprocal back and forth within the bores defined in the cylindrical body **12**. The piston member **50** comprises an elongated rod with an enlarged diameter radial portion **52** located midway between opposite ends of the rod. A pair of piston rings **54** are mounted on the piston **52** which reciprocates in the large diameter bore **18***a* of the central expansion cylinder **18** of the body. Extending outwardly in opposite directions, the piston member includes rod end portions **56**, each having a plurality of rings **58** adjacent the outer end and adapted to slide in the bore **20***a* compression chamber end sections **20**.

In order to cushion the stroke of the piston member as it approaches the end plates **14** and **16** and reverses direction, each end portion **56** of the piston member is formed with an end projection **56***a* having a diameter slightly smaller than the diameter **20** of the compression chamber. The projecting end bosses **56***a* are designed to engage annular shock absorbers **60** formed of resilient material and seated in annular recesses provided in the heads **14** and **16**, respectively. The head **16** is provided with a cylindrical clean out bore **16***a* axially aligned with the center axis of the piston member **50**, and a suitable bore closing plug **62** is provided to close the outer end of the bore.

In order to insure accurate coaxial alignment between the compression chamber sections **20** and the respective heads **14** and **16**, the heads are formed with annular recesses **14***b* and **16***b* to receive outer end flanges **20***b* on the compression chamber cylinder sections. The inner end of the cylinders **20** are formed with annular recesses **20***c*, which are of a diameter equal to that of the bore **18***a* of the central expansion cylinder **18**. Sealing ring assemblies **64** are mounted in the recesses **20***c* to seal opposite ends of the expansion cylinder bore **18***a*, and the rings are held in place by a plurality of longitudinal cap screws **66**. Each sealing ring assembly **64** includes one or more inner sealing rings **68** which bear and seal against a rod portion **56** of the piston member **50**. The cylinder sections **18** and **20** are dimensioned in length with respect to the length of the piston member **50** so that the end boss **56***a* will strike the cushion or shock members **60** at opposite ends of the compression chambers **20** before the central piston portion **52** strikes the sealing ring assemblies **64** to prevent damage to the rings.

In order to introduce pressurized fluid into either end of the bore **18***a* of the expansion cylinder **18** without the use of conventional valves or valve members, the rod portions **56** of the piston member **50** are provided with flatted lands **56***b* of a predetermined longitudinal dimension and width. These flats or lands **56***b* cooperate with the rings **68** to provide inflow passages for compressed fluid flow into the expansion cylinder **18** during a particular range of longitudinal positions of the piston member **50** in the cylinder body **12**.

While the flats or lands **56***b* are adjacent the sealing rings **68** the fluid passages are defined and open, but when the flats move beyond the rings, sealing off of the expansion cylinder **18** is accomplished. In order to supply pressurized fluid to the ends of the expansion cylinder **18**, the outer compression chamber sections **20** of the body **12** are provided with annular passages **20***d* coaxially and outwardly adjacent the sealing ring recesses **20***c*. The annular fluid feed passages **20***d* are supplied with compressed fluid through radial passages **20***e* closed at their outer ends by clean out plugs **69**. Longitudinal fluid passages **20***f* are provided for communication between the passages **20***e* and the opposite ends of a longitudinal supply or feed passage **19** defined in the expansion cylinder **18**. The passage **19** is supplied with pressurized fluid through a supply line **70** which is connected to a suitable source of pressurized fluid at a selectively controlled pressure.

When the piston member **50** is in a lower position, as shown in FIG. **2**, the flats **56***b* on the lower rod section **56** are adjacent the lower sealing ring assembly **64** defining an inflow passage for fluid into the lower end of the expansion cylinder bore **18***a* below the piston portion **52**. The upper end of the expansion cylinder above the piston portion **52** is at a lower pressure and is exhausted through a pair of relatively large exhaust ports **18***b* defined on opposite sides of the cylinder wall midway between opposite ends. In order that pipe connections may be made with the exhaust ports, enlarged bosses **72** having threaded internal bores aligned with the ports are attached to the cylinder body **18** and cap screws or studs **73**, nuts **74** and washers **75** are provided to facilitate pipe connections with the ports. The expansion cylinder **18** below the piston portion **52** is being supplied with pressurized fluid and the cylinder above the piston portion **52** is being exhausted through the ports **18***b* so that the piston member **50** moves upwardly in the cylinder body **12**. When the piston member **50** moves upwardly so the lower flats **56***b* are above the lower sealing rings **68**, the supply of pressurized fluid to the expansion cylinder **18** is cut off and the fluid in the cylinder further expands in a theoretical adiabatic expansion process until the piston portion **52** begins to uncover the exhaust ports **18***b*. As the ports **18***b* are uncovered during an upward stroke of the piston member, the expanded pressurized fluid beneath the piston portion **52** begins to exhaust to the atmosphere and the upward forces acting on the piston member **50** decline. Moreover, during an upward stroke, the fluid entrapped in the upper compression chamber **20***a* above the upper end piston rod **56** is being compressed and acts as an air spring with a cushioning effect. This entrapped fluid within the upper compression chamber **20** acts as a spring or shock absorber and decelerates the upward movement of the piston member **50**. When the fluid being compressed in the upper chamber **20***a* reaches a sufficient pressure, upward movement of the piston member **50** is stopped and the compressed cushioning fluid begins to expand and move the piston member in the opposite or downward direction. In addition, when the piston member **50** approaches the upper position, as shown in FIG. **5**, the upper flats **56***b* on the upper piston rod **56** are adjacent the upper sealing rings **68**, thereby defining fluid inlet passages for the entry of fluid into the upper end of the expansion cylinder **18** above the center piston portion **52**. The pressurized fluid introduced into the upper end of the expansion cylinder **18** above the piston portion **52** provides the main force for moving the piston in a downward direction, and fluid is supplied only during the time that the upper flats **56***b* are adjacent the upper sealing rings **68** in inlet passage defining relation. When the flats **56***b* move downwardly below the rings **68**, the supply of pressurized fluid to the upper end of the expansion cylinder is cut off but fluid expansion continues in an adiabatic process until the piston portion **52** begins to uncover the exhaust ports **18***b* and begin exhausting fluid from the upper end of the cylinder. During a downward stoke, the entrapped fluid in the lower compression chamber **20***a* is compressed to act as a cushioning device and shock absorber.

The operating cycle, including an upward and downward stroke of the piston member **50** is repeated, and the acceleration and deceleration of the piston member during each stroke and the resultant inertial forces developed are based on a number of factors including the width and the length of the flats **56***b*, the pressure of the fluid supplied to the system through the inlet line **70**, the difference in diameters between the central piston portion **52**, and the rod end portions **56**, the diameter of the exhaust ports **18***b*, the length of the cylinder **18** between the seal ring assembly **64**, and the mass of the piston member **50**.

In addition to the factors mentioned, the rate of reciprocation and inertial or acceleration forces developed by the vibrator **10** is also controllable by means of supplying pressurized fluid to the opposite ends of the compression chambers **20**. The extent of the pressurization determines the amount of the cushioning action and opposing forces developed by compression of the entrapped fluid in the compression chambers and controls the length of the stroke of the piston member **50**. Cushioning fluid is supplied through a pipe **78** threaded into the outer end of a short radial bore **18***d* which intersects a longitudinal bore **18***e*. Opposite ends of the bore **18***e* are in communication with angular bores **20***g* which intersect short radial bores **20***h* spaced intermediate the ends of the compression chamber cylinder sections **20**. Clean-out plugs **79** are provided in the outer, threaded end portions of the bores **20***h*. The difference in mean effective pressure between the power fluid supplied to the expansion cylinder **18** and the cushioning fluid supplied to the cushioning chambers **20** affects the length of stroke, the rate of reciprocation of the piston member **50**, and the resultant vibratory forces developed as the piston member reciprocates. The pressurized cushioning fluid supplied to the opposite ends of the compression chamber sections **20** for shock cushioning and slowing down the piston member **50** at the end of a stroke, is an effective air spring and requires little maintenance or attention, as compared with a mechanical spring. The annular, resilient material shock absorbers **60** are used only for safety purposes in case too great a pressure is used on the power side of the system. When the fluid pressure of the cushioning fluid in the outer ends of the cushioning chambers **20** is close to that of the power fluid supplied to the central expansion cylinder **18**, the length of the stroke of the piston member **50** is relatively short and the speed of reciprocation is relatively changed because of the relatively high strength of the air cushioning effect. If a great pressure differential is provided, the piston member **50** moves greater distances at higher speeds and develops much greater inertial vibrating forces.

FIG. **6** is a diagram illustrating the theoretical pressure-versus-stroke relationship occurring in the expansion cylinder **18** on one side of the piston portion **52** as the piston member **50** moves from bottom dead center to top dead center and returns. The straight line A to B indicates the relatively constant inlet pressure in the cylinder as power fluid is admitted during the period that the flats **56***b* are adjacent the seal rings **68** in inlet passage defining relation. When the flats **56***b* move on past the seal rings, the supply of power fluid is cut off and adiabatic expansion of the fluid takes place from B to C. At point C the piston portion **52** starts to uncover the exhaust ports **18***b* and the pressure drops down to the ambient exhaust pressure at point D at the end of the power stroke. As the return stroke commences, the pressure remains relatively constant at the exhaust ambient pressure until the piston portion **52** moves past the exhaust ports **18***b* at point E. From point E to point F work is done by the piston on the entrapped fluid in the cylinder until the flats **56***b* move into inlet passage defining relation with the rings **68** at point F. From point F back to starting point A the pressure builds up rapidly to that of the inlet power supply system. The work done by the power fluid acting on one side of the piston portion **52** during a power stroke is represented by the area under the curve A-B-C-D, and the work expended on the fluid by the piston on a return stroke is represented by the area under the curve D-E-F-A, and the net work available is the area within the closed loop of the curve.

It can be seen from FIG. **6** that some cushioning effect is provided in the expansion cylinder **18** for slowing down the moving piston member **50**. This cushioning effect is represented by the area under the curve E-F as the entrapped fluid is compressed and cushioning effect is also achieved by the power fluid itself as represented by the area under the curve F-A.

FIG. 7 is a diagram of the theoretical composite pressure-versus-stroke in the expansion cylinder. The shaded area G above the neutral pressure line H—H represents the acceleration portion of the power stroke while the shaded area I below the line represents the deceleration portion of the power stroke. Even though the maximum pressure values above and below the neutral line H—H are equal approximately the area G is larger than the area I, and the difference represents the net work done or the available vibratory energy. The average pressure or MEP during the acceleration phase of the power stroke is considerably greater than the MEP of the deceleration phase of the power stroke.

FIG. **8** is a diagram similar to FIG. **7** but based on the theoretical consideration of isothermal rather than adiabatic expansion and compression in the cylinder **18**. The acceleration and deceleration phases of the power stroke are more nearly equal in length than in the adiabatic process of FIG. **7**. The dotted lines represent the theoretical isothermal compression curves and the solid line is a composite of the two.

Although the present invention has been described by reference to only a single embodiment thereof, it will be apparent that numerous other modifications and embodiments will be devised by those skilled in the art which will fall within the true spirit and scope of the principles of the present invention.

What is claimed as new and desired to be secured by Letters Patent of the United States is:

1. A linear vibrator comprising a body defining a cylinder; a piston slidably mounted in said cylinder for reciprocal vibratory movement between opposite ends of said cylinder; valve means responsive to the position of said piston in said cylinder between said opposite ends for introducing pressurized activation fluid into said cylinder alternately on opposite sides of said piston; discharge port means in said cylinder for alternately exhausting fluid from opposite sides of said piston in response to the position of said piston between said opposite ends of said cylinder; a pair of fluid storage chambers having a cross-sectional area different than that of said cylinder and capable of storing compressible fluid at pressure independent of the pressure of said activating fluid in said cylinder; a pair of rod-like extensions on opposite sides of said piston slidably disposed as piston members in said chambers; said valve means comprising a pair of annular valve seats at opposite ends of said cylinder and passage means of finite length on said piston members movable with said piston and cooperating with said seats to open and close in response to the position between said opposite ends of said cylinder; and seal means between said piston members and said chambers for high pressure sealing between said chambers and opposite ends of said cylinder.

2. The linear vibrator of claim **1** wherein said vibrator includes passage means in said body for supplying compressed fluid to both said chambers and interconnecting the same externally independent of said cylinder.

3. The linear vibrator of claim **1** wherein said piston and piston members are cylindrical and are coaxially aligned with each member being formed with at least one flatted recess forming an activating fluid inlet passage for one side of said piston while said recess is adjacent one of said annular valve seats.

4. The linear vibrator of claim **1** including a pair of annular, activating fluid inlet chambers spaced outwardly of said annular valve seats for directing said pressurized activating fluid against opposite sides of said piston in said cylinder, during the periods when one or the other of said passage means is extended to communicate between an inlet chamber and the adjacent end of said cylinder.

5. The linear vibrator of claim **1** wherein said passages on said piston rod members are defined by flatted segments having a dimension extending longitudinally thereof less than the length of said members whereby said valve means is open to admit activating fluid to one side of said piston through a flatted segment on one rod member while a flatted segment on the other rod member is adjacent said port means to exhaust said cylinder on the opposite side of said piston.

* * * * *