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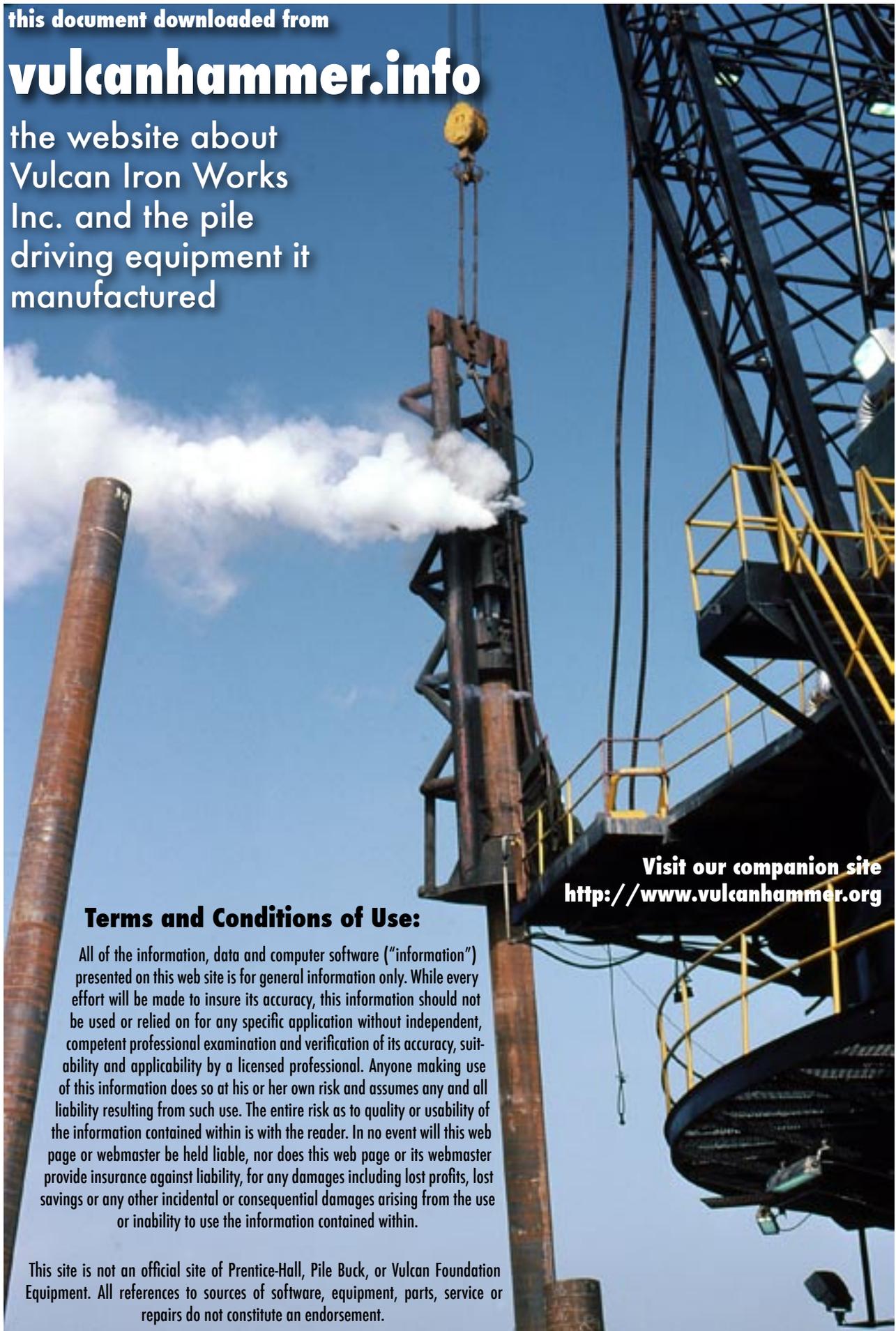
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- [54] **FREE PISTON POWER SOURCE**
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- [51] Int. Cl. **F011 21/02, F01b 11/02**
- [58] Field of Search **91/234, 232; 92/85, 143, 8; 417/341**

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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

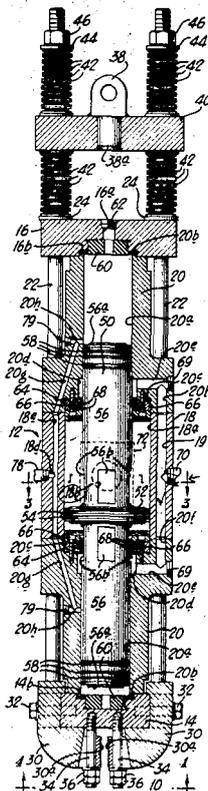
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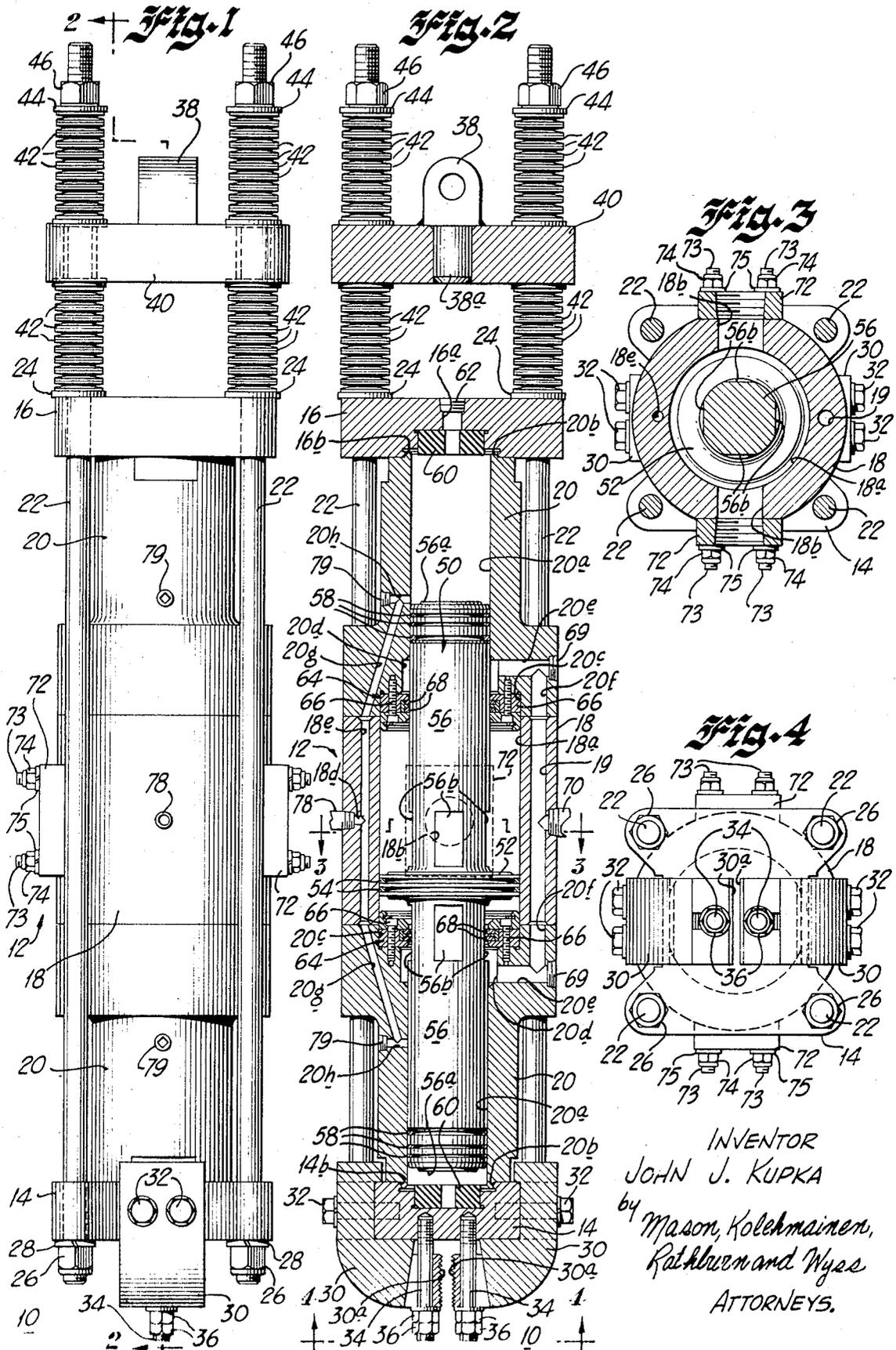
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FREE PISTON POWER SOURCE

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 30a (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 38a (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 18a 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 20a 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 56a having a diameter slightly smaller than the diameter 20 of the compression chamber. The projecting end bosses 56a 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 16a 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 14b and 16b to receive outer end flanges 20b on the compression chamber cylinder sections. The inner end of the cylinders 20 are formed with annular recesses 20c, which are of a diameter equal to that of the bore 18a of the central expansion cylinder

18. Sealing ring assemblies 64 are mounted in the recesses 20c to seal opposite ends of the expansion cylinder bore 18a, 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 56a 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 18a 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 56b of a predetermined longitudinal dimension and width. These flats or lands 56b 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 56b 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 20d coaxially and outwardly adjacent the sealing ring recesses 20c. The annular fluid feed passages 20d are supplied with compressed fluid through radial passages 20e closed at their outer ends by clean out plugs 69. Longitudinal fluid passages 20f are provided for communication between the passages 20e 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 56b 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 18a 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 18b 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 18b so that the piston member 50 moves upwardly in the cylinder body 12. When the piston member 50 moves upwardly so the lower flats 56b 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 18b. As the ports 18b 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 20a 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 20a 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 56b 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 56b are adjacent the upper sealing rings 68 in inlet passage defining relation. When the flats 56b 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 18b and begin exhausting fluid from the upper end of the cylinder. During a downward stroke, the entrapped fluid in the lower compression chamber 20a 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 56b, 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 18b, 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 18d which intersects a longitudinal bore 18e. Opposite ends of the bore 18e are in communication with

angular bores 20g which intersect short radial bores 20h 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 20h. 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 56b are adjacent the seal rings 68 in inlet passage defining relation. When the flats 56b 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 18b 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 18b 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 56b 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 stor-

ing 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.

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