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Elliston

[11] **3,866,480**

[45] **Feb. 18, 1975**

[54] **ORBITAL VIBRATOR**

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[73] Assignee: **Martin Concrete Engineering Company**

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[52] **U.S. Cl.** **74/87,** 259/DIG. 42, 308/9, 308/184

[51] **Int. Cl.** **F16h 33/10**

[58] **Field of Search** ..... 74/61, 87; 259/DIG. 42; 418/61 B, 63; 308/9, 184

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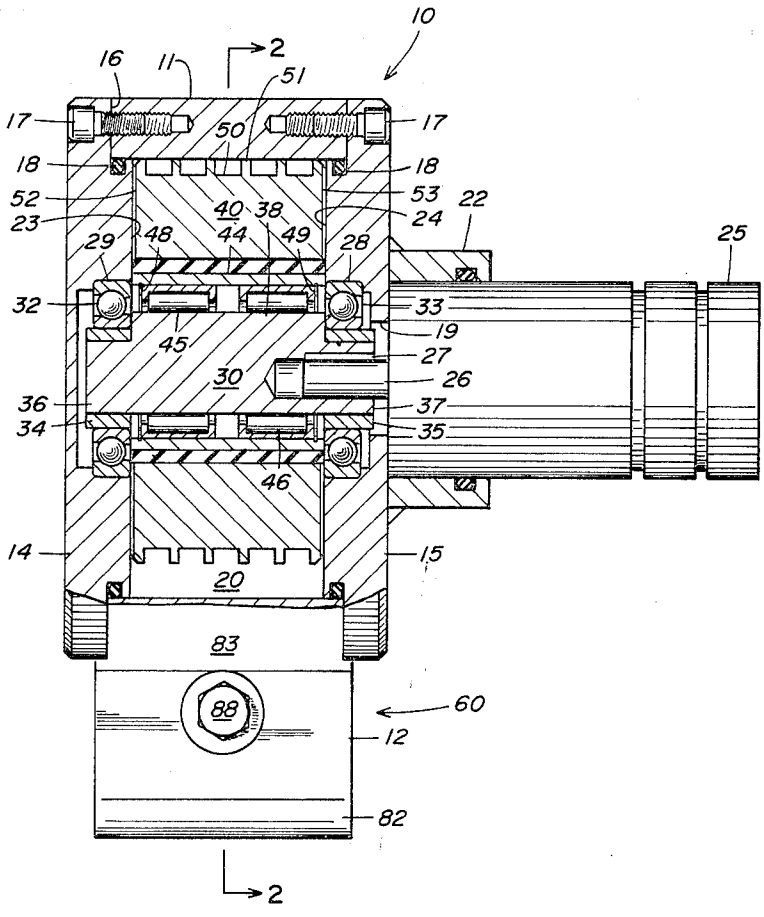
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Primary Examiner—Allen D. Herrmann
Attorney, Agent, or Firm—Hubbard, Thurman, Turner & Tucker

[57] **ABSTRACT**

An orbital vibrator and a method of using the same is disclosed. The vibrator includes an orbiting rotor driven by the output shaft of a hydraulic motor through a positive drive connection. The positive drive connection may comprise an eccentric lobe on the drive shaft on which the rotor is mounted or a differential cycloidal gear connection between the rotor and shaft which imparts multiple orbital vibrations for each drive shaft revolution. The motor is driven by a hydraulic pump which is pressure compensated and responsive to the requirements of the vibrator operating at the highest pressure. As the rotor oscillates, relief means in the rotor surface maintains orbital contact between the rotor and housing and radial bearing slots in the rotor faces form hydrodynamic bearing surfaces. A hydraulic actuable piston lock for securing the vibrator unit in a bracket secured to the surface to be vibrated is also shown. The method includes monitoring the fluid flow rate and pressure to the drive motor to determine the effectiveness of the vibrational energy transfer to the surface to which the vibrator is attached. When applied to compaction of concrete forms, several vibrators may be selectively operated and alternatively disengaged and advanced along the concrete form and reenergized at a new position. This is continued until the entire length of the form has been vibrated.

14 Claims, 12 Drawing Figures



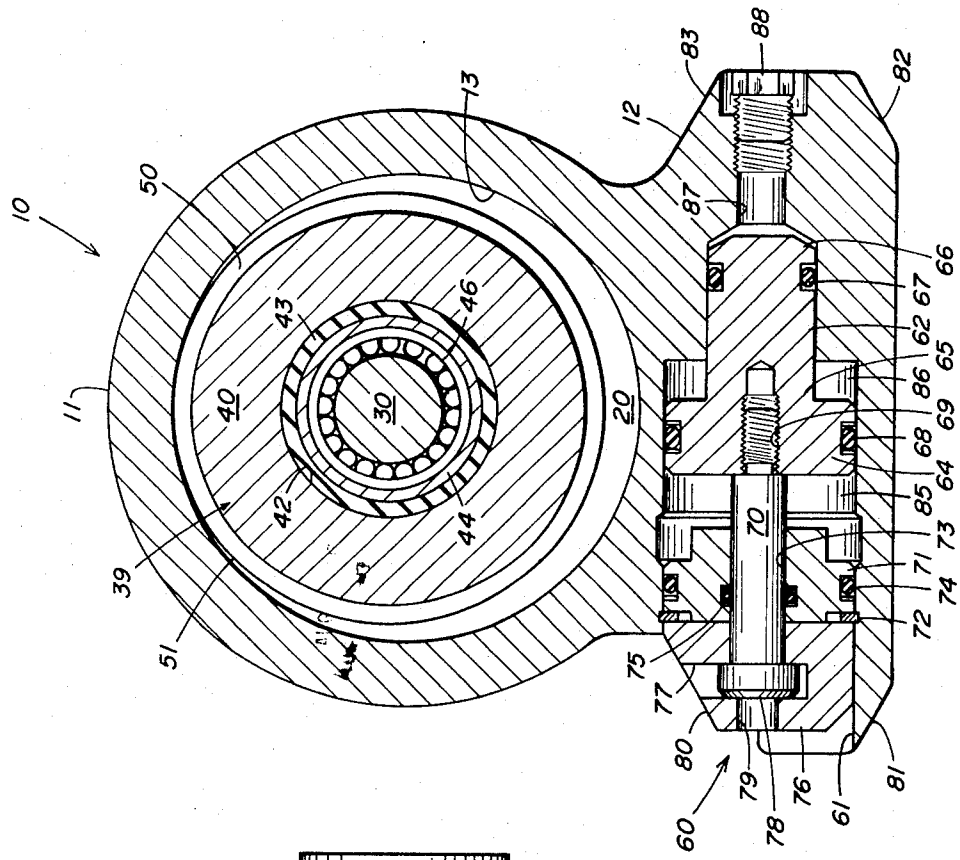


FIG. 2

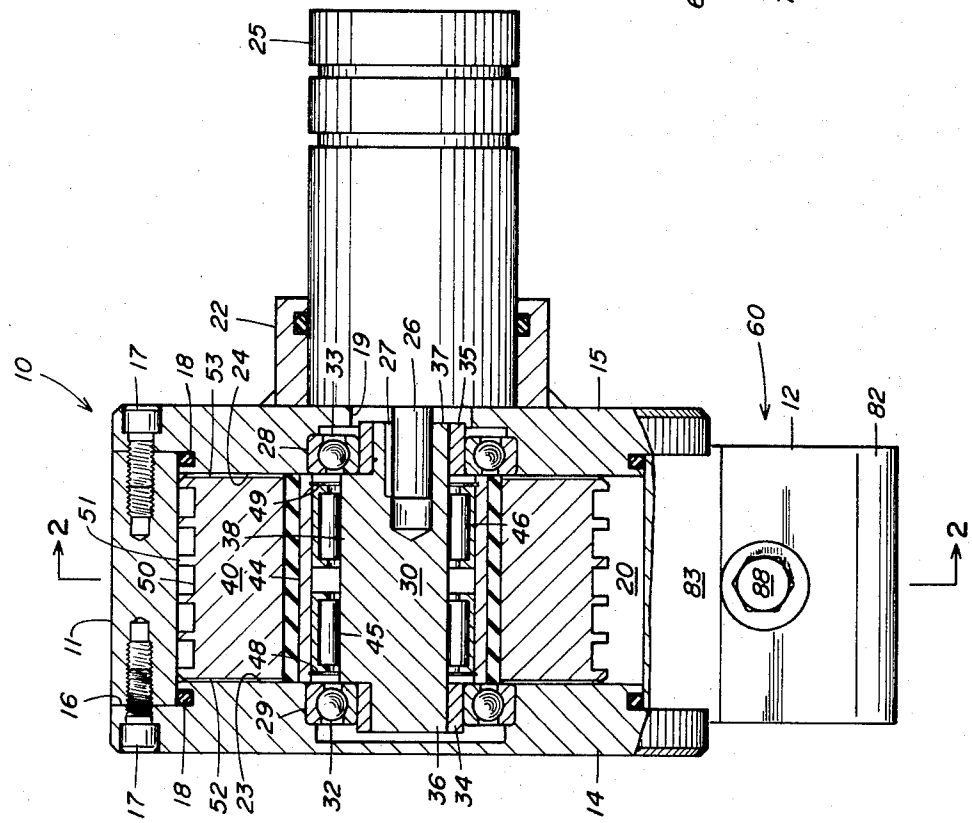


FIG. 1

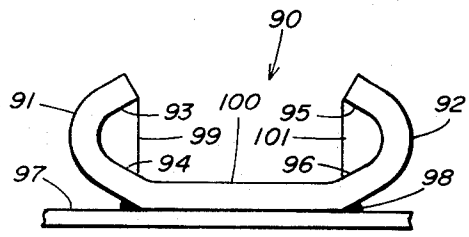


FIG. 3

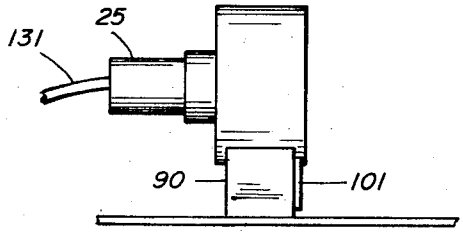


FIG. 3A

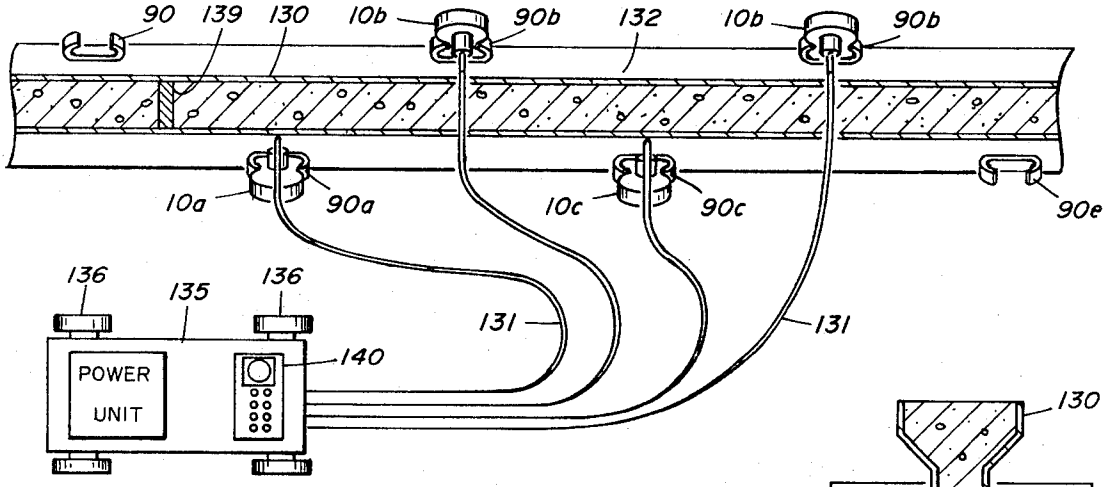


FIG. 7

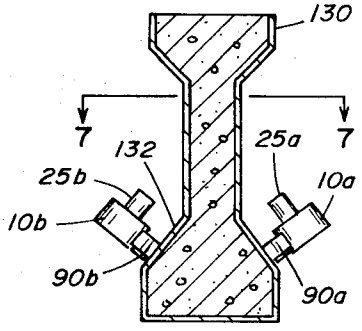


FIG. 6

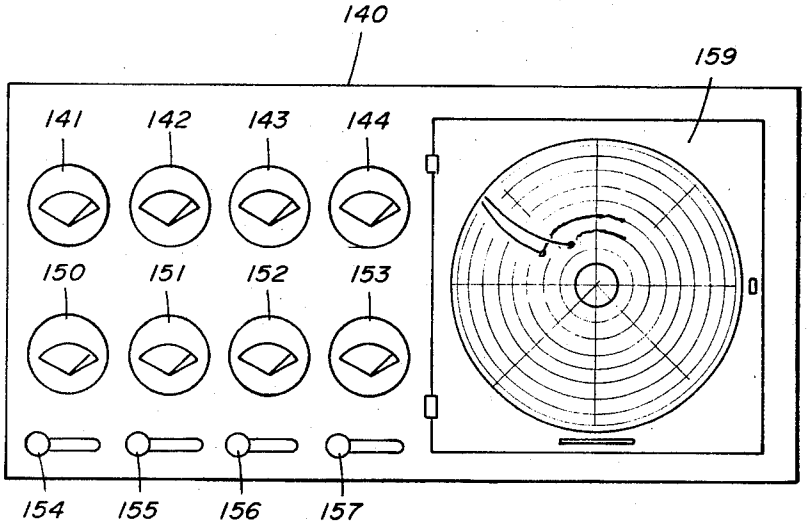


FIG. 8

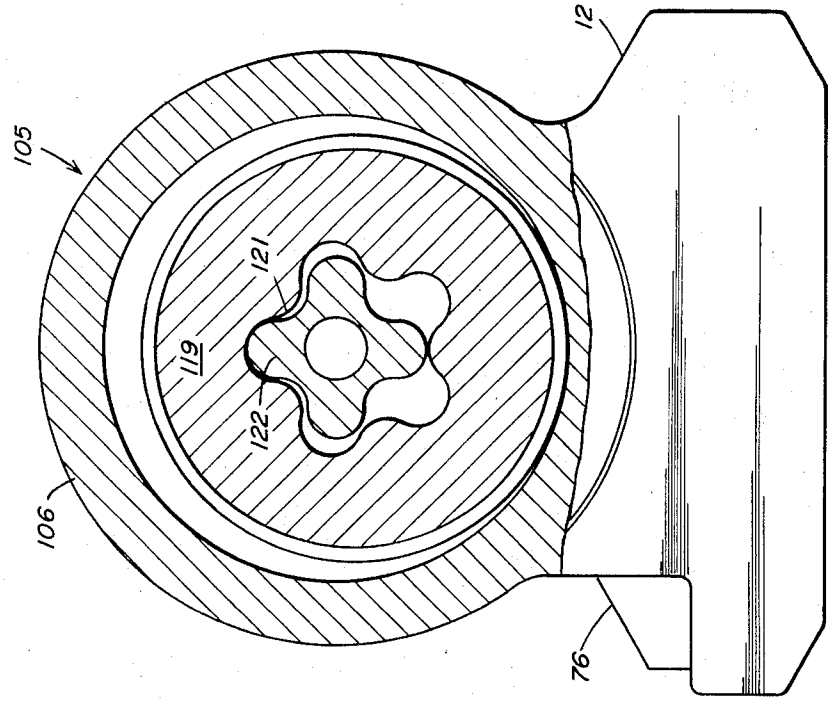

12
105
121
119
122
106
76

FIG. 5

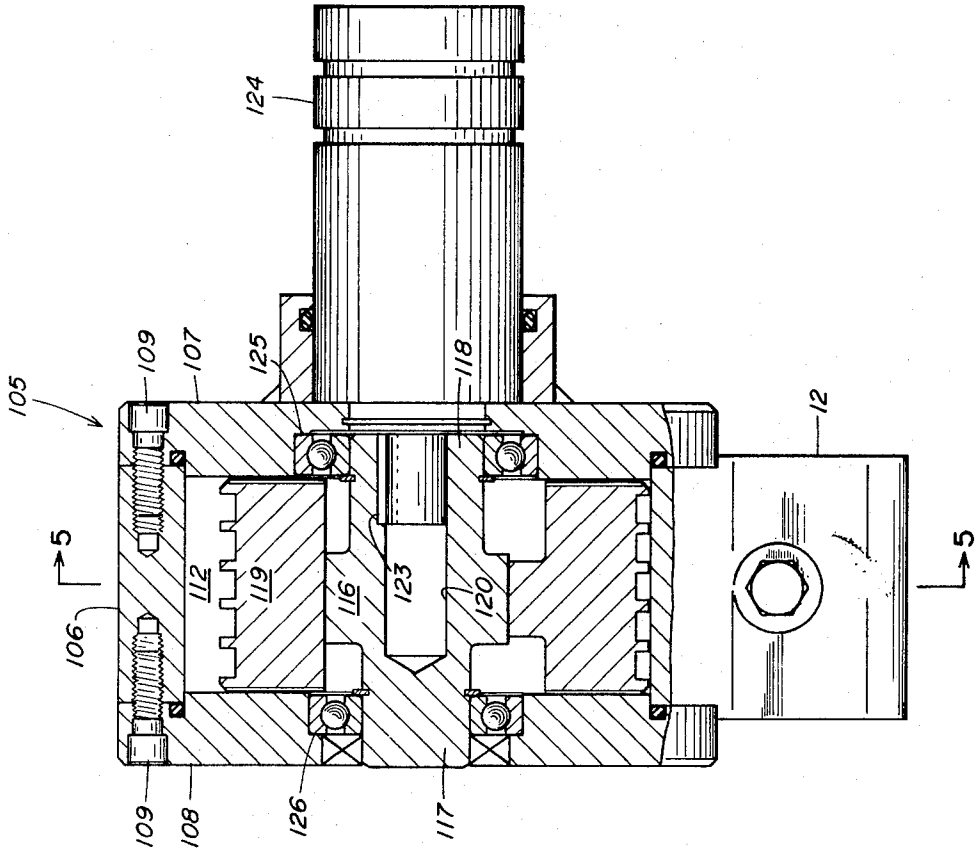

124
105
109
107
125
118
12
5
5
112
119
116
123
120
106
109
108
126
117

FIG. 4

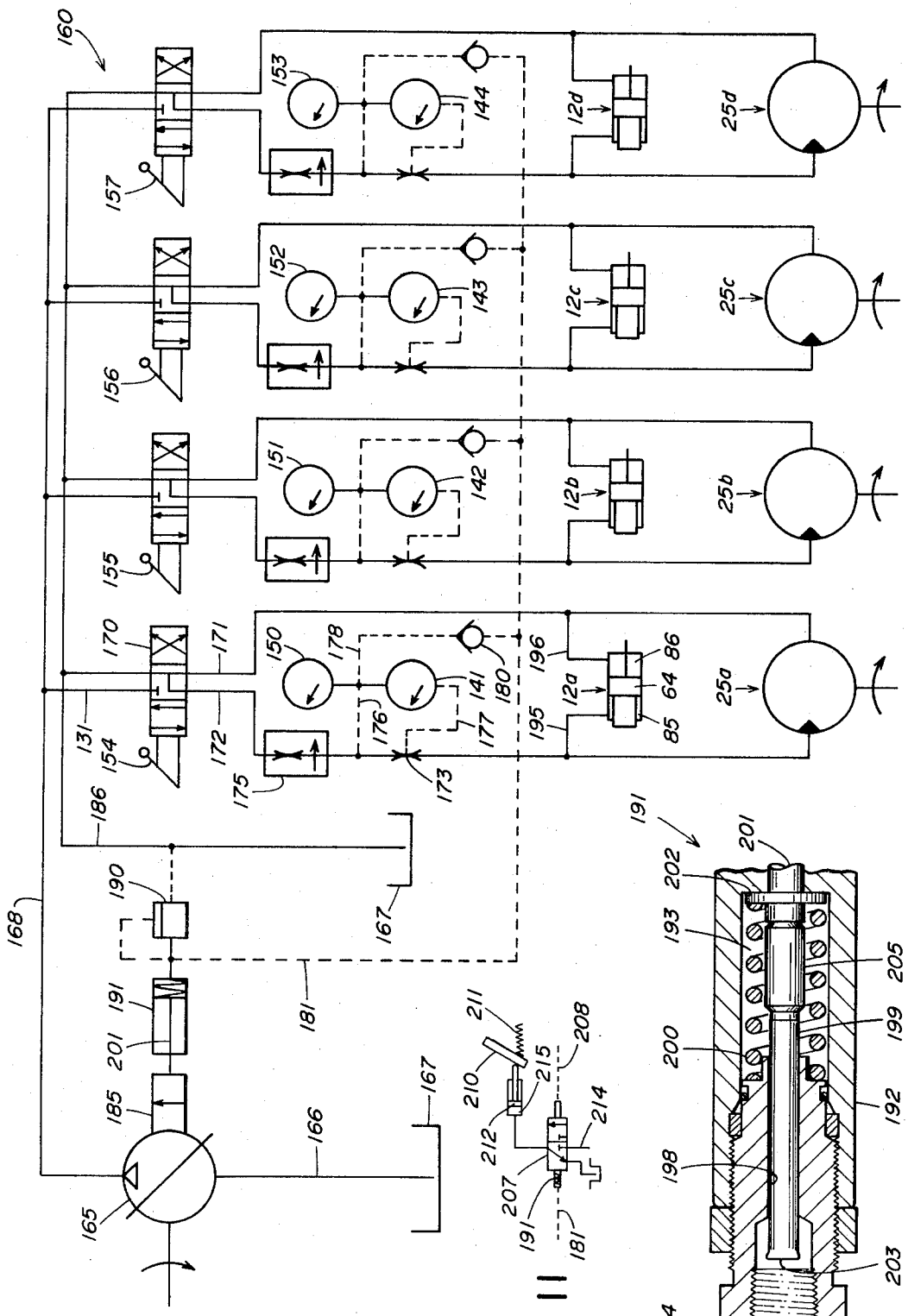


FIG. 10

FIG. 11

FIG. 9

ORBITAL VIBRATOR

The present invention relates generally to a mechanical vibration mechanism and more particularly to a positively driven orbital vibrator for generating high frequency orbital vibrations and a method of operating the orbital vibrator.

The use of mechanical vibrators for imparting energy to materials such as concrete to aid in compaction is well known. Vibrators are also often used with storage bins and conveyor systems to control the loading, unloading, and conveying of particulate materials such as cement, flour, grain and the like. Numerous other applications of vibrators can be found in industry.

Most prior art vibrators are of the type that utilize the rotation of an eccentric rotor to produce vibrational energy. The rotor is in the form of a cylindrical weight rotatably driven around a hard circular race formed in the housing such that the weight follows an orbital path. High pressure air is introduced into the housing to drive the rotor. This fluid driven type of vibrator, although having found some acceptance, has the serious disadvantage that rotor slippage can occur because the rotor is not positively driven. The internal orbiting member may begin to lift away from the housing race or "hydroplane" due to a pressure differential existing across the rotor. As a result, the efficiency of the vibration unit is seriously impaired as the amplitude of the vibrations generated is reduced.

The apparatus of the present invention provides a highly effective and simple vibrator which overcomes the deficiencies of prior art orbiting mass vibrators. The drive always remains in positive contact with the rotor. The vibrator is designed so that rotational forces are transferred to the rotor housing at the line of contact with the rotor so that loads imposed on the drive shaft bearings are substantially reduced. Pressure relief means associated with the rotor serve to equalize pressure forces on the rotor periphery to prevent rotor lift-off or hydroplaning. A hydraulically actuated locking device may be included with the vibrator which cooperates with a special mounting bracket, as for example for use with a concrete form. When the bracket is firmly secured by the hydraulic lock, maximum transfer of energy from the vibrator through the bracket into the form is achieved. A novel hydraulic system for operation of the vibrator and lock permits the operator to monitor flow and pressure at the vibrators to visually determine the operating efficiency of the vibrator and the effectiveness of the engagement of the lock and bracket.

As well as providing a novel vibrational apparatus, it is an important object of this invention to provide a method for vibrating concrete forms, chutes and the like in a manner to assure high energy transfer into the device being vibrated which requires a minimum number of vibrators and auxiliary equipment.

Other objects of this invention will become apparent from the following description taken in connection with the accompanying drawings in which:

FIG. 1 is a side elevational view, partly in section, of one form of the vibrator of the present invention;

FIG. 2 is a sectional view taken along line 2—2 of FIG. 1;

FIG. 3 is a top view of the vibrator mounting bracket;

FIG. 3A shows an end view of the mounting bracket with a vibrator secured in position in the bracket;

FIG. 4 is a vertical sectional view of an alternate form of the vibrator shown in FIG. 1;

FIG. 5 is a sectional view taken along line 5—5 of FIG. 4;

FIGS. 6 and 7 illustrate a novel method of using a vibrator according to the present invention for vibrating a concrete form; FIG. 7 showing the form in section as taken along line 7—7 of FIG. 6;

FIG. 8 illustrates in detail the control panel shown in FIG. 6;

FIG. 9 is a detail view of the modified pump compensator control used with the system pump;

FIG. 10 is a schematic of a hydraulic control system connected to the vibrators; and

FIG. 11 is a schematic diagram to further show the operation of the pump compensator of FIG. 9.

As outlined, the use of an orbiting rotor to introduce vibration and its general operation is well-known and understood. A weight or rotor member is mounted within a cylindrical housing for rotation about a shaft concentrically mounted with respect to the housing cavity. The rotor is mounted eccentrically with respect to the drive shaft and axial center of the housing and, therefore, as the drive shaft is rotated, the rotor will rotate about the axis of the drive shaft and orbit around the periphery of the housing cavity.

With this general background in mind and referring now to FIGS. 1 and 2, a vibrator according to the present invention is illustrated and represented by the numeral **10.** Vibrator **10** includes upper circular housing portion **11** which encloses the orbiting rotor and lower portion **12** which serves as a mounting or locking device for the vibrator. Housing portion **11** has a circular bore **13** therein which serves as a raceway for the rotor. The housing is steel or other suitable material and finished on the interior at **13** to the desired hardness to withstand the high speed operation of the rotor **39** within the housing. The sides of housing **11** are closed by opposite side plates **14** and **15** having inner surface **23** and **24,** respectively, which define with race **13** the working chamber **20.** As is seen in FIG. **1,** these plates are provided with annular peripheral grooves **16** which mate with the periphery of the housing to enclose the working chamber **20.** Suitable fastening means such as cap screws **17** secure the side plates **14** and **15** to housing member **11.** Annular O rings or seals **18** at the interface between the side plates and the housing **11** prevent leakage of fluid from the housing.

Side plate **14** is counter bored at **29** and ball bearing assembly **32** is pressed therein. Opposite side plate **15** is provided with counter bore **28** which receives ball bearings **33** similar to bearing assembly **32.** Concentric bore **19** extends through side plate **15** for insertion of a drive shaft. Circular flange **22** is affixed to the exterior of side plate **15** and adapted for mounting of an external drive means **25.** Drive means **25** may be any suitable type of power unit and preferably would be a high speed fluid operated gear motor of the type which operates in the range of 10,000 rpm to positively drive the rotor.

Output shaft **26** of drive unit **25** extends through opening **19** into working chamber **20** of the housing. Eccentric shaft **30** is internally affixed to drive shaft **26** at key **27** and has eccentric lobe **38.** Lobe **38** has its center radially displaced from the axial centerline of shaft **26.** Opposite journals **36** and **37** on eccentric shaft **30** extend respectively into bearing inserts **34** and

35. The inserts are mounted for rotation within ball bearings 32 and 33. Inclusion of annular inserts 34 and 35 between the eccentric journals and the bearings facilitates assembly of the unit and allows use of the proper size bearings while permitting the required eccentricity of the shaft 30 to be maintained.

The rotor 39 consists of outer annular weight member 40 having concentric through bore 42 therein. Annular rubber insert 43 is pressed into bore 42. In the assembly of rotor 39, rubber insert 43 is actually preloaded or compressed slightly prior to insertion. Inner journal or sleeve 44 is inserted within rubber sleeve 42. Sleeve 44 is provided with an outer rubber backing or membrane so that it may be vulcanized or adhesively bonded to the rubber insert 43. Annular roller bearing assemblies 45 and 46 are pressed into the cavity within insert 43 and held in place by snap rings 48 and 49 engaging annular grooves provided in sleeve 44.

As seen in FIGS. 1 and 2, the eccentric or lobe portion 38 of eccentric shaft 30 is received within roller bearings 45 and 46. The compressed rubber sleeve 43 insures that the centrifugal force of the rotor is evenly distributed on the roller bearings 45 and 46 as the sleeve expands under load and serves to take up any tolerance differences. Roller bearings 45 and 46 serve only to transmit the drive torque from the eccentric shaft 30 to the rotor 40 and primarily support only rotating torque. In operation, rotor 40 will orbit about chamber 20 on race 13 in the direction of rotation of drive shaft 26 and will rotate about the centerline of lobe 38 in a direction opposite the rotation of shaft 26 and at a lesser proportional speed.

An important feature of the invention resides in providing means for maintaining orbital contact between the rotor and the race. The thrust imposed by centrifugal force is mainly absorbed at the race 13 of the chamber as the rotor weight rotates. The design of the rotor insures that metal-to-metal contact between the interior wall of housing and the outer periphery of rotor 40 is maintained. Supporting the centrifugal loads at this surface substantially reduces the loads on the bearings as well as provides for the efficient generation of vibrational energy. To establish and maintain good contact between the rotor and race 13, pressure relief means serve to equalize pressure about the rotor periphery. The pressure relief means are in the form of one or more grooves 50 provided in the peripheral surface 51 of the rotor. The grooves 50 are preferably annular and axially spaced and extend entirely around the rotor. Other configurations are possible such as grooves extending diagonally across the rotor face or grooving the housing which would functionally operate in the same way to overcome the tendency of the rotor to hydroplane as it oscillates about the rotor cavity by equalizing the pressure on the rotor periphery. The phenomenon of hydroplaning is known and in this instance is basically due to high relative speed between the rotor and cavity in the presence of fluid such as air or oil within the cavity which causes a pressure force at the line of rotor contact tending to lift the rotor away from the housing race. The efficiency of the vibrator becomes impaired because as the rotor lifts away from the race 13, the amplitude of the vibrations are reduced. It has been found that the inclusion of the annular grooves 50 serves to equalize pressure around the rotor periphery and the pressure force tending to lift the rotor from contact with the race is substantially reduced. The inclusion of pressure relief grooves may be applied to prior art mechanical and fluid driven vibrators to improve their efficiency in the manner described above.

The axial or side thrust imposed by the moving rotor is absorbed at interior side walls 23 and 24 at a hydrodynamic bearing surface formed by radially extending slots 52 and 53 in the opposite rotor faces. Typically four slots are provided in each rotor face extending radially at 90° relative to one another from the bore 42 outwardly to the rotor peripheral surface 51. Each slot is of relatively shallow depth, approximately on the order of one-sixteenth inch, and has a width exceeding the depth. Functionally, these slots moving at high speed relative to the inner housing side walls 23 and 24 form a hydrostatic bearing between the housing wall and rotor face that serves to stabilize the rotor against axial movement. Preferably, a small amount of lubricating fluid such as a silicon oil or similar lubricant is placed in the chamber 20. The fluid is distributed by the slots between the rotor face and housing wall, creating a lubricating film between the two, and maintains the hydrostatic bearing at the side walls. The inclusion of the radial bearing slots 52 and 53 significantly reduces operational noise and chatter and dampens small, high speed axial movements of the rotor.

The lower portion 12 of the housing 10 contains a hydraulically actuated locking device 60 for positively securing the vibrator in an operational position in a specially designed cooperating bracket. Bore 61 extends into section 12 of the housing at a right angle relative to the axis of shaft 30. Counter bore 62 extends concentrically from its lower end. Piston 64 is reciprocably mounted and has main portion 65 within bore 61 and an axially extending stub shaft 66 which extends into counter bore 62 and serves as a guide for the piston. Annular seal members 67 and 68 surround sections 66 and 65 of the piston, respectively, and serve to prevent fluid leakage across the surfaces of the piston. Internally threaded bore 69 in piston 65 is adapted to receive the end of allen head bolt 70.

Bearing member 71 encloses bore 61 to define the cylinder chamber and is held in place by annular snap ring 72. Axial bore 73 through bearing 71 receives the shank portion of bolt 70. Appropriate seals 74 and 75 may be provided to prevent fluid leakage from within the chamber defined by bearing member 71.

Lock plug 76 is slidable along the lower extension of bore 61 and is provided with slots 77 which receive the head 78 of allen bolt 70. Axial bore 79 permits access to recess head 78 of bolt 70 to permit engagement with an appropriate wrench.

Surfaces 80, 81, 82, and 83 of portion 12 are provided with flats to form the mounting wedge and serve to transfer the vibrations from the vibrator to the mounting bracket.

It will be seen that piston section 64 divides bore 61 into two chambers 85 and 86 on opposite sides of annular member 64. Fluid inlets or ports in section 12, not shown, provide for flow of fluid to and from chambers 85 and 86. Passage 87 connecting with counter bore 66 provides a leakage path for high pressure fluid which may escape from chamber 86 around seal 67. Plug 88 may be provided in passage 87.

It will be seen that if high pressure fluid is introduced into chamber 86, piston 65 will be caused to move to the left, thereby urging lock member 76 leftwardly via the mechanical connection of the bolt 70. Conversely,

if chamber **85** is connected to a source of pressure fluid, the the piston will be caused to retract within bore **61** and, accordingly, lock or plug **76** will be moved rightwardly to an unlocked condition. The hydraulic operation of the lock will be further described with reference to FIG. **10.**

FIG. **3** illustrates the mounting bracket which accommodates the hydraulic wedge **12** of the vibrator. The bracket **90** is shown as being affixed to plate **97** by welds **98.** Plate **97** may be, for example, a concrete form. The bracket is generally designated by the numeral **90** and includes opposite end portions **91** and **92** joined at intermediate section **100** to form a general C shape. The inner side of the bracket is configurated at **93, 94, 95,** and **96** to correspond and receive flats **80** through **83** of the lock portion **12** of the vibrator. Retainer plates **99** and **101** extend across the bottom of bracket **90** at opposite ends **91** and **92** to engage one side of locking portion **12** of the vibrator. For example, locking section **12** of the vibrator would be inserted within bracket **90** with the axis of motor **25** parallel to plate **97.** Plates **99** and **101** serve as stops to prevent the unit from slipping entirely through the bracket until the lock is actuated.

With the piston **64** of locking wedge **12** retracted, the lock may be easily engaged within the interior of bracket **90** by inserting the lock into the bracket from the lower side of the bracket as seen in FIG. **3.** Upon extension of piston **64** by pressurization of chamber **86,** face **80** of the wedge bears against flattened portion **93** of the bracket forcing the corresponding surfaces of the wedge and bracket into tight engagement. Four point metal-to-metal contact exists at the engaging surfaces of the wedge and mounting bracket and insures that vibrations generated by the rotor will be efficiently transferred through the bracket to the surface being vibrated. The wedge and bracket combination are self-aligning, that is, the wedge is self-seating within the bracket **90.** With the wedge properly in place, most of the energy is transferred at surfaces **81** to **83** of the wedge, thereby minimizing wear and tear on the O rings of the cylinder assembly of the lock. Rapid deterioration of O rings and sealing rings because of high cyclic loads placed on them has been a common problem encountered with prior art hydraulic mountings. However, such a problem is avoided with the vibrator of the present design as the locking cylinder is substantially isolated from the vibrational force.

FIGS. **4** and **5** show an alternate embodiment to the vibrator shown in FIGS. **1** and **2.** The embodiment is generally designated by a numeral **105** and includes a circular housing **106** enclosed by side plates **107** and **108** secured thereto by bolts **109** to define chamber **112.** Roller bearings **125** and **126** are mounted in side walls **107** and **108,** respectively, in recesses provided therein. Drive shaft **116** having opposite journals **117** and **118** which are received within bearings **126** and **125,** respectively, drives rotor **119** within chamber **112.** The drive shaft **116** is provided with a central bore **120** which receives the output shaft of a low speed hydraulic motor **124** and is secured thereto at keyway **123.** It will be understood that the drive motor mounting can be similar to that shown with respect to the embodiment **10.** The construction described so far conforms generally to that described with reference to FIGS. **1** and **2.**

The apparatus shown in FIGS. **1** and **2** provides a positive driving connection between the drive shaft and rotor by means of the eccentric lobe described above. A gear connection may be utilized instead of the eccentric lobe to positively drive the rotor in an orbit within the housing. Prior art machines using gear drives of various configurations have a common drawback. One of the main difficulties has been the maintenance of contact between the gear teeth, particularly during starting and stopping operations. The gears of prior art arrangements tend to fall out of engagement with one another when the rotor is stopped and upon starting are caused to be jammed against one another causing damage and excessive wear to the gear teeth.

In the embodiment of FIGS. **4** and **5,** eccentric motion is imparted to the rotor through the interconnection of the drive shaft **116** and the rotor **119** by a direct engagement gear drive without the attendant problem described in the preceding paragraph. The gear drive is of the type known in the field as a "gerotor" type gear arrangement and which consists of internal gear cavities **121** in the rotor and external gear lobes **122** on the drive shaft of a differential epicycloidal gear form which interconnect to drive the rotor **119** in an orbital path within chamber **112.** The gerotor gear connection assures that continuous engagement is always maintained between the external and internal gear lobes. As seen in FIG. **5,** the upper external gear is fully engaged, while the opposite lower external gear is maintained in contact in the land area between adjacent internal teeth. Therefore, the teeth, during periods of rest, cannot fall out of engagement which could result in damage to the gear teeth when the unit is started up again.

Another advantage of the gear drive is that the gear connection steps up the orbital speed of the rotor and a high number of orbital vibrations per revolution of the drive shaft are generated. This permits use of a relatively low speed drive motor to generate high frequency vibrational energy. The formula for determining the number of orbits of the rotor per input revolution is as follows:

$$\text{Vibrations}/_{\text{rev}} = \frac{1}{\frac{T_H - T_R}{T_R} + \left(\frac{D_H - D_W}{D_W}\right)}$$

where

T_H = number teeth on rotor
T_R = number of teeth on drive shaft
D_H = diameter of housing
D_W = diameter of rotor weight

The unit shown in FIGS. **4** and **5** has five teeth in the rotor and the drive shaft has four. Assuming a typical unit dimensioned having a rotor housing diameter of 4.30 inches and a rotor weight with a 4.00 inches diameter, the formula resolves as follows:

$$\text{vib}/_{rev} = 1/1/4 + 0.30/4.00 = 3.08$$

From this it can be seen that a 3000 RPM drive motor input will cause the weight to be driven at over 9000 orbital vibrations per minute.

The remainder of unit **105** is similar to that shown with reference to FIGS. **1** and **2** and includes the hydraulically actuated locking section **12** to permit insertion and locking in a clamp of the type shown in FIG. **3.**

The present invention also provides a unique method of operating vibrators to insure the integrity of the vi-

brational transfer to any element to which the vibrator may be attached. As discussed above, vibrators are commonly used for the compaction and transfer of fluid and particulate materials. For example, vibrators may be attached to the exterior concrete forms to obtain the required degree of compaction and eliminate voids in the concrete. In the fabrication of prestressed concrete beams, often the form is several hundred feet long and the inclusion of the vibrators every lineal ten feet or so results in a large number of expensive vibrators and a complex system to power and control the vibrators. Providing an effective mounting for the vibrators on the forms has also presented problems. The vibrators may be working properly; however, unless they are properly secured to the forms, a defective concrete beam may result, as little or no energy will be transferred into the concrete contained in the form. Obviously, such a defective beam would mean a serious economic loss to the fabricator.

FIGS. **6, 7** and **8** show the method of the present invention as applied to vibrating a concrete form or other container of material. Looking at FIG. **6,** an elongated concrete form **130** is shown. Form **130** may be several hundred feet long and may be of any suitable structural cross-sectional shape as, for example, a T- or I-beam as is shown. In the manufacture of prestressed beams it is common to first string a number of steel cables (not shown) longitudinally within the mold, placing them under tension. The concrete is then poured into the mold and consolidated by various methods. After the concrete has set up, the concrete beam is removed from the forms ready for use. It is also common to subdivide the form **130** into shorter length sections at, for example, longitudinally adjustable bulk head **139** to allow several shorter beams to be fabricated in a single pouring.

Prior art attempts to vibrate the mold to consolidate the concrete therein have proven unsatisfactory because of the inability of prior art vibrator arrangements to efficiently pass energy into the mold. In accordance with the present invention, a series of mounting brackets **90, 90***a*, **90***b*, etc., as shown in FIG. **3,** are welded to the exterior of form **130** at spaced intervals, as for example, every ten feet to lockingly engage a vibrator. As seen in FIG. **7,** brackets **90** are arranged on both sides of the form and staggered so one is effectively spaced every five feet along the length of the form **130.** FIG. **6** shows the elevational location of vibrator brackets **90.** Preferably for best compaction, the brackets are affixed near the bottom of the form and for the I-beam shown a location on the upper angular surface **132** of the lower flange is preferable. The brackets are positioned on the flange so that the axis of the vibrator is parallel to the flange surface **132** with the bottom side of the brackets facing outwardly. This location and orientation is preferred for beams of this cross section since satisfactory filling and compaction of the lower flange area is obtained. With vibrators of this type, best results are usually obtainable when the axis of one vibrator is parallel to the surface being vibrated since the primary vibrational energy is emitted in the form of a sine wave along the longitudinal axis of the locking section **12** of the unit. For other shapes of forms, other mounting orientations are desirable. For example, when the vibrator is to be used with a rectangular pan-type form, the vibrator should be affixed to the exterior side of the form floor with the motor axis parallel to the plane of the floor.

A mobile power unit **135** is carried on a chassis by wheels **136** to supply the motive power to the vibrators and can be manually or self-propelled along the form. Power unit **135** would include, as is well known in the art, a power source such as an internal combustion engine connected to a hydraulic pump mounted on the chassis of the power unit **135.** Also contained within the power unit would be a reservoir of hydraulic fluid for the pump and suitable interconnecting piping, control valves and the like. The details of the hydraulic system are shown and discussed with reference to FIG. **10.**

Four vibrators **10***a*, **10***b*, **10***c*, and **10***d* are shown inserted in clamps affixed to the form **130.** Hydraulic lines **131** interconnect the hydraulic motors **25** of the vibrators to the pump contained in the power unit **135.** The motors are individually controlled and can be shut off at control panel **140** by appropriate control means affixed to a control valve in the hydraulic system. To compact the concrete within the form, the operator would position the vibrators as shown beginning at one end of the form and first pressurize the locking cylinders **12** to cause the locking elements to tightly engage the mounting brackets **10***a* to **10***d* and secure their respective vibrators in place. Hydraulic pressure is then introduced into the hydraulic motors driving the vibrators, causing them to rotate, bringing the orbiting rotor mass in the vibrators up to speed. As pointed out above, the vibrators are rotated at speeds of approximately 10,500 RPM so that good penetration into the mass of concrete is achieved. When the length of the form defined between end vibrators has been sufficiently compacted, the operator simply will deenergize the motor on vibrator **10***a* and pressurize chamber **85** of the locking cylinder associated with vibrator **10**a to cause locking member **76** to retract within its associated mounting bracket **90***a*. Vibrator **10***a* then can be manually removed from its bracket by a workman and advanced along the form to the next open bracket which is **90***e* and then inserted into the associated bracket. The locking device is then actuated to secure the vibrator in the bracket **90***e*. The control valve is operated to energize the hydraulic motor associated with the vibrator, causing it to start up. Since the other vibrators have been operating continuously as the transfer occurred, vibrator **10**a will quickly accelerate to the operating frequency of the operating vibrators and lock in phase with the running vibrators. This permits the repositioned vibrator to pass through the resonant frequency with minimum pressure and easily achieve the frequency necessary for proper consolidation and compaction of the concrete. The opposite vibrator **10**b can next be manually advanced down the form to the next bracket and the procedure repeated until the entire length of the form has been compacted. In this way, only several vibrators are needed to compact an extremely long form during the pouring operation, two being adequate in many installations. This reduces the equipment costs considerably because, as for example, a standard 300 foot form could require as many as 60 separate vibrators if each were permanently affixed within its mounting bracket. Mobile power unit **135** and the associated vibrators can be moved to an adjacent form and used during another pour. Thus, the most efficient utilization of the equipment is made.

As seen in FIG. **8**, mobile unit **135** houses panel **140** on which gives the operator a visual readout of the vibrations and mount efficiency. Prior methods of compaction of forms by the use of vibrators affixed to the exterior of forms and molds have often proven unsatisfactory because of the inability to reliably pass energy into the form. Although the vibrators may be active, unless securely attached to the form, the energy will be dissipated and not transferred to the material within the form. Gauges **141** to **144** are provided to give a visual indication of the frequency of operation of the vibrators. For instance, gauge **141** is connected to a flow measuring device such as a fixed orifice or venturi in the hydraulic supply line **131** to the motor **25***a* of vibrator **10***a*. Because of the positive drive connection, the operating speed of the drive motor and the vibrator is determined by the rate of flow to the drive motor and, accordingly, by visually monitoring the flow rate, the operator with certainty knows that the vibrator has achieved the required RPM.

Gauges **150** to **153** indicate that the individual vibrators are operating efficiently. Each of these gauges is essentially a pressure gauge connected to the supply line directed to the individual motors driving the vibrators. If a vibrator is improperly or loosely mounted within its bracket, the pressure in the supply line to the motor to that vibrator will become unstable and rapidly increase. Gauges **150** to **153** are calibrated so that when an inordinate pressure increase occurs, an indication of poor mounting efficiency will be registered on the face of the gauge and the operator can take the proper steps to rectify the situation. Control levers **154** to **157** are connected to operate the control valves to the individual vibrators to actuate or deactuate the motors and locking mechanisms as required by the operator.

Recorder **159**, provided in panel **140**, records the pressure and volume of flow to each of the vibrators. The recorder may be rotary, as shown, or strip chart recorder and will provide a permanent record of the effectiveness of the compaction method during a pour.

Thus, the above method of advancing the vibrators along the form during the pour materially reduces the number of vibrators required and substantially decreases the complexity and cost of the system required to consolidate the concrete. Further, monitoring the pressure and flow to the individual vibrators and providing a visual indication gives the operator a continuous reading of the efficiency of the vibrating operation. A low flow rate to a particular vibrator will be reflected on the upper row of gauges and will give an indication of a low frequency reading. If the pressure to the vibrator exceeds the predetermined maximum, the operator is then advised that the vibrator did not break through the resonance frequency of the form and thus did not achieve the necessary frequency needed to insure good consolidation of the concrete. Also, by turning only one vibrator off and leaving the remaining vibrator running at any one time, energy is continually being introduced to the form and to the contained material and the removed vibrator, when reintroduced into the system, will quickly accelerate and lock in phase with the running vibrator.

FIG. **10** shows in detail a schematic of the hydraulic system in which four vibrators are operably connected to a hydraulic pump **165**. Pump **165** is preferably a variable displacement axial piston unit having an inclinable swash plate which may be varied to change the displacement of the pump. Compensator **191** is operably connected to the swash plate of the pump to vary the displacement in response to the pressure requirements of the system. Pressure regulator **190** limits the system pressure to a preselected maximum.

FIG. **9** shows the details of the compensator **191** which is a modified version of standard pressure compensator units con mercially available to maintain the output pressure in accordance with the system requirements. Cylindrical body **192** would be affixed to the housing of pump **165**. Control rod **201** is connected to control the position of the swash plate yoke of pump **165** to vary the pump development. Blind bore **193** in body **192** is closed at its upper end by threaded plug **194**. Axial bore **198** extends through plug **194** and receives plunger **199** therein. Threaded port **197** is adapted to receive a pilot pressure signal through line **181** from the motor operating at the highest pressure. Spring **200** acts against washer **202** and the inner end of plug **194**. Plunger **199** also bears against washer **202** through insert **205**. Thus, the rightward force acting to move rod **201** is the total of spring force **200** and the pressure force acting on the end **203** of rod **199**. When the pilot pressure at **203** increases, rod **201** will move rightward destroking the pump and when the pressure signal reduces rod **201** will move leftwardly to increase the displacement.

FIG. **11** illustrates schematically the operation of the pressure compensator. The control rod **201** of the compensator control **191** is connected to the spool of hydraulic compensator valve **207** to urge the spool to the right. Pressure from the discharge of pump **165** is continually applied at the opposite end of the spool of valve **207** at **208**. Swash plate **210** of pump **165** is biased by spring **211** to full delivery. Piston **212** is actuable to destroke the pump. It can be seen that when pressure in line **181** decreases, valve **207** will shift to the left to decrease the pump displacement by pressuring piston chamber **215** to move piston **212** to the left. Conversely, when the biasing pressure and spring force at **191** overcomes the pressure force at **208**, valve **207** will move to the right, venting chamber **215** so that spring **211** will move the swash plate to full displacement. Thus, the compensator **191** controls the position of swash plate **210** to automatically maintain a desired output pressure under varying flow requirements. FIG. **9** shows the compensator control **191** in a position of minimum pump displacement. In the absence of a pressure signal, spring **200** will determine the pressure output of the pump and then the pump will compensate itself out.

The output of the pump is connected through line **168** to individual lines **131** to each of the hydraulic motors **25***a*, **25***b*, **25***c*, and **25***d*. Common return line **186** connects with reservoir **167**. Suction line **166** depends from motor **165** into reservoir **167**. Most of the system components such as the pump, reservoir, valving, for convenience, would be contained within the mobile unit **135** described with reference to FIG. **7**.

Since each of the individual motor circuits is identical, a description of one is sufficient and will apply equally to all. Looking for example at the portion of the circuit controlling the operation of motor **25***a*, line **131** is intercepted by directional control valve **170** which is a threeposition, four-way control valve. Control handle **154** operable at panel **140** serves to position the spool

of valve **170.** When control handle **154** is actuated, causing the spool of valve **170** to be placed in the extreme left hand position (straight through), flow of pressure fluid from line **131** will be directed across valve **170** to line **172** delivering pressurized fluid to motor **25***a* and causing it to be actuated.

Return fluid from motor **25**a flows through line **171** across valve **170** to line **186** and to reservoir **167.** Pressure compensated flow control valve **175** is interposed in line **172.** Flow venturi **173** in line **172** imposes a pressure drop. Valve **175** operates as a pressure compensated fixed flow control to limit overspeed. Valve **175** is set to limit the motor **25***a* to a predetermined speed, say 10,500 RPM. Venturi **173** serves to give a pressure differential signal which is converted to flow.

Line **176** is connected to line **172** on the upstream side of venturi **173** and communicates with pressure gauge **150** and flow meter **141.** Similarly, line **177** connects flow meter **141** to the lower or downstream side of venturi **173.** Line **178** connects line **176** across one-way check valve **180** with common return line **181.** Thus, the pressure in the motor circuit operating at the highest pressure will also be communicated across the corresponding check valve to pilot line **181.** Locking device **12**a, which serves to secure the vibrator to the mounting bracket as described above, has chamber **85** connected via line **195** to line **172.** Similarly, chamber **86** is connected via line **196** to line **171.** Piston **64** is reciprocal within the cylinder, as has been described above. When valve **177** is actuated to the right as described, piston **64** will also move to the right as fluid is introduced in the chamber **85.** This will cause locking element **79** to move into tight engagement in the mounting bracket and secure the vibrator in place. When it is desired to unlock a vibrator, valve **170** is moved by handle **154** to the extreme left, causing pressure fluid to be delivered across valve **170** to line **171** and line **196** to cylinder chamber **86.** This will cause piston **64** to move to an unlocking position. Fluid from cylinder chamber **85** is returned to reservoir via lines **195, 172,** and **186.** During unlocking, the inertia of the motor will cause some flow to discharge from the motor through line **171,** the motor acting as a pump. Further, some vibrational energy is still being generated and, therefore, the unlocking action is aided and easily accomplished. Nevertheless, in some instances, it may be desirable to incorporate a one-way check valve in the motor discharge line **171** to assist in a rapid pressure increase to unlock the vibrator when valve **170** is shifted leftwardly.

In the neutral position shown, the motors are all interconnected to the reservoir **167** by line **186.** Similarly, it will be observed that when one or more motors are placed in operation through actuation of their respective control valves, the pressure in line **181** will correspond to the pressure at the highest pressure operating motor. Check valve **180** at the motor, operating at the highest pressure operating will be held open and the corresponding check valves **180** in the remaining individual motor control systems will be held closed. The high pressure signal is sent to the compensator **191** so that the pump **165** compensates to this requirement. The system is simple and efficient in that the pump is not operating to supply fluid at pressure or flows beyond the immediate system requirements. The control permits selective operation of any combination of vibrators and the system of gauges permits a quick visual check of the operating efficiency of the individual vibrators.

Thus, a novel vibrator and method of using the same has been disclosed. The vibrator and the method may be put to numerous uses other than those described. Similarly, while two embodiments of the vibrator structure have been disclosed in detail, other forms may become apparent to those skilled in the art, and such modifications as do not depart from the scope and spirit of the present invention are included herewith.

What is claimed is:

1. An orbiting mass vibrator comprising:

rotor assembly means including an enclosure defining a rotor chamber and cylindrical rotor means having a circular periphery and flat side faces including means associated with at least one side face defining hydrodynamic bearing means, said rotor means being constrained by said chamber,

drive means adapted to drive said rotor in an orbital path about said chamber,

fluid pressure relief means associated with said rotor assembly means, said pressure relief means adapted to relieve the fluid pressure between said rotor and chamber as the rotor orbits within the chamber thereby reducing the dynamic separation between the rotor and chamber.

2. The vibrator of claim **1** wherein said bearing means are defined by at least one slot radially extending in said rotor side face.

3. An orbiting mass vibrator comprising:

rotor assembly means including an enclosure defining a rotor chamber and rotor means constrained by said chamber,

drive means adapted to drive said rotor in an orbital path about said chamber, said drive means comprises a drive shaft mounted for rotation in said rotor chamber drivingly engaging said rotor through a gear connection, and

fluid pressure relief means associated with said rotor assembly means, said pressure relief means adapted to relieve the fluid pressure between said rotor and chamber as the rotor orbits within the chamber thereby reducing the dynamic separation between the rotor and chamber.

4. The orbiting mass vibrator of claim **3** wherein said gear connection comprises mating differential epicycloidal gear teeth on said rotor and said shaft.

5. The vibrator of claim **4** wherein said chamber has opposite sidewalls, said drive shaft is mounted in said opposite sidewalls for rotation by means of journals concentric with the axis of said chamber, and said rotor has a predetermined number of internal epicycloidal gear teeth in engagement with a predetermined number of external epicycloidal gear teeth on said drive shaft, the number of teeth in said rotor exceeding the number of teeth on said shaft.

6. An orbiting mass vibrator comprising:

rotor assembly means including an enclosure defining a rotor chamber and rotor means constrained by said chamber,

drive means adapted to drive said rotor in an orbital path about said chamber, said drive means includes a drive shaft mounted for rotation in said chamber having eccentric means thereon and wherein said rotor means is mounted on said eccentric means and is positively driven through said eccentric means and resilient means are provided between

said rotor and said eccentric means with bearing means inserted between said resilient insert and said eccentric means, and

fluid pressure relief means associated with said rotor assembly means, said pressure relief means adapted to relieve the fluid pressure between said rotor and chamber as the rotor orbits within the chamber thereby reducing the dynamic separation between the rotor and chamber.

7. The vibrator of claim **6** wherein said housing has opposite sidewalls, and said drive shaft is mounted in said housing sidewalls for rotation by means of journals concentric with the axis of said chamber.

8. The vibrator of claim **6** wherein said resilient insert means comprises an annular rubber insert radially compressed to preload said insert when the rotor is in a non-operating condition.

9. The orbiting mass vibrator of claim **6** wherein said fluid relief means comprises at least one groove extending in the direction of movement of the rotor means.

10. The orbiting vibrator of claim **9** wherein the groove is formed in the rotor means.

11. An orbiting mass vibrator comprising:

housing means defining a rotor chamber having an outer race and opposite side walls,

drive shaft means mounted for rotation in said housing,

a cylindrical rotor connected to said drive shaft having a cylindrical outer surface and opposite side faces,

a driving connection between said rotor and shaft whereby said rotor is caused to orbit around said race with said side faces in close proximity to said housing side walls, and

means associated at said rotor and housing side wall interface defining hydrodynamic bearing means.

12. The vibrator of claim **10** wherein said bearing means are defined by at least one radially extending slot in said rotor side wall.

13. The vibrator of claim **10** wherein said bearing means are defined by slot means in the housing side walls.

14. The vibrator of claim **10** including lubricating fluid in said housing distributed at said rotor and housing side wall interface upon orbiting of said rotor.

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