United States Patent [19]

Linz

[11] **3,834,827**

[45] **Sept. 10, 1974**

[54] **VEHICLE MOUNTED VIBRATORY COMPACTOR**

[76] Inventor: **Albert Linz,** Julweg 40, Hoffnungsthal/Cologne, Germany

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[21] Appl. No.: **178,567**

[30] **Foreign Application Priority Data**

Sept. 23, 1970 Germany 2046840

[52] **U.S. Cl.** **404/133**

[51] **Int. Cl.** **E01c 19/34**

[58] **Field of Search** .... 94/48, 49, 50 R, 50 V; 404/113, 133, 102

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Assistant Examiner—Thomas J. Holko
Attorney, Agent, or Firm—Eric P. Schellin; Martin P. Hoffman

[57] **ABSTRACT**

A vehicle mounted vibratory compactor employing a pair of vibrating elements that extend laterally across the entire width of the vehicle and are mounted, as a unit, in a rigid support frame. The frame is designed as a parallelogram with parallel control arms for resisting the bending forces applied thereto by the action of the vibrating elements compressing materials, such as soils of all types, and the reactive forces of the materials. The frame is movable vertically within guides defined on the vehicle so that the vibrating elements can be elevated sufficiently to avoid contacting road obstacles as the vehicle is moved to the work site, and lowered variable amounts to alter the amplitude of the movement of the vibrating elements. The magnitude of the compression forces exerted by the vibrating elements is varied considerably by the utilization of diverse combinations of springs, reciprocal plates, pressure limit valves, pneumatic servomechanisms, pistons, and the like, which determine the natural frequency of the compactor system and thus control the resonant frequencies at which the maximum compressive forces are produced. The vibrating elements, which may assume the form of plates, rollers, beams or shoes, may be fabricated from a plurality of simplified components. If desired, "sheepsfoot" projections may be formed on the bottom surface of the vibrating elements.

24 Claims, 34 Drawing Figures

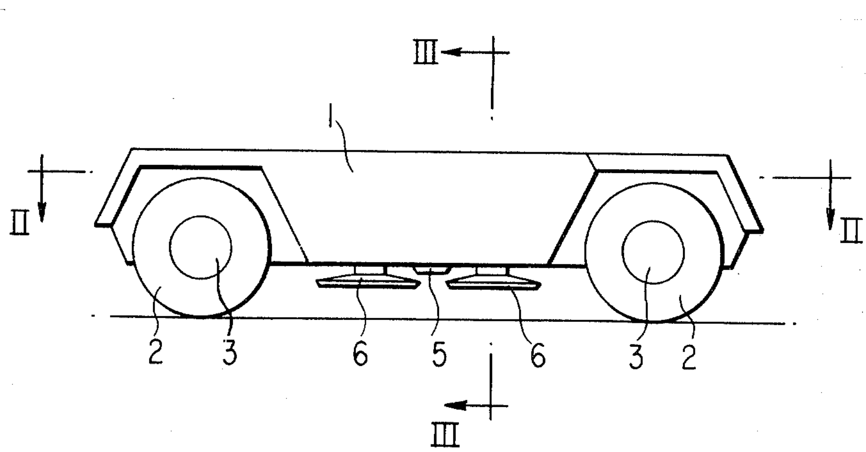


FIG. 1

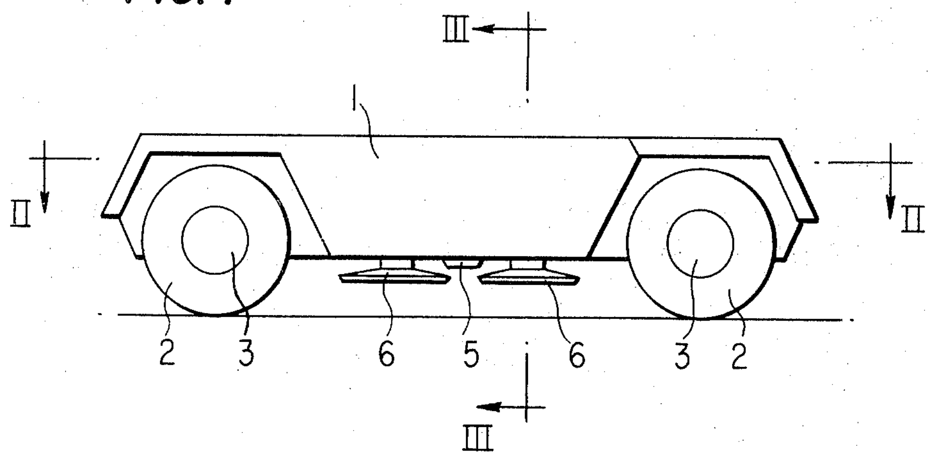
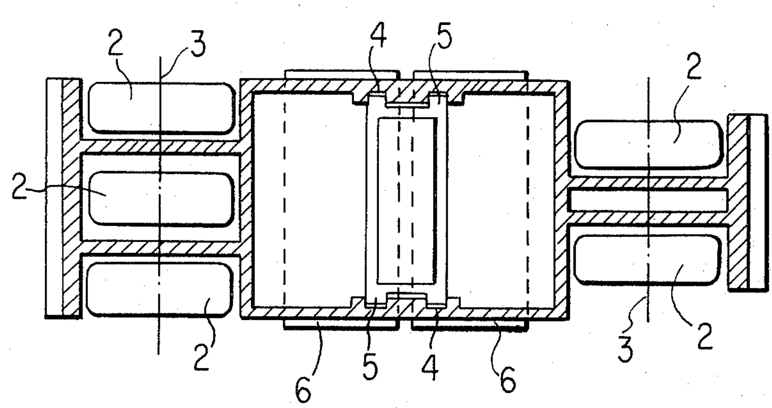
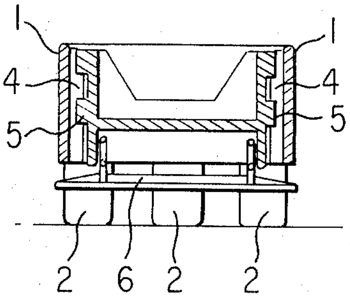


FIG. 2





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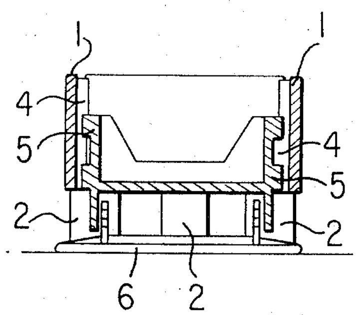

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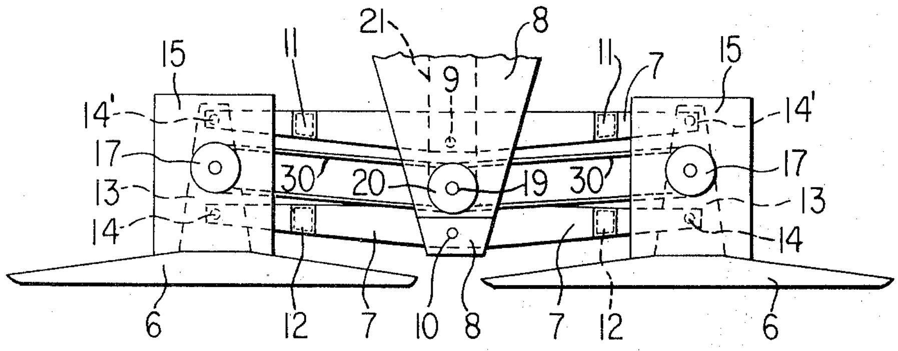

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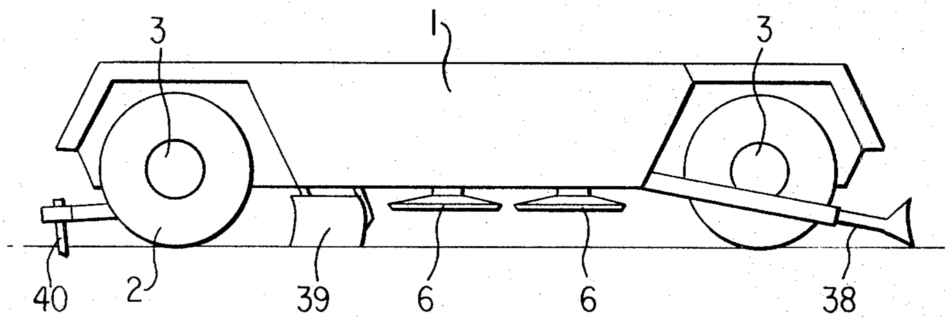

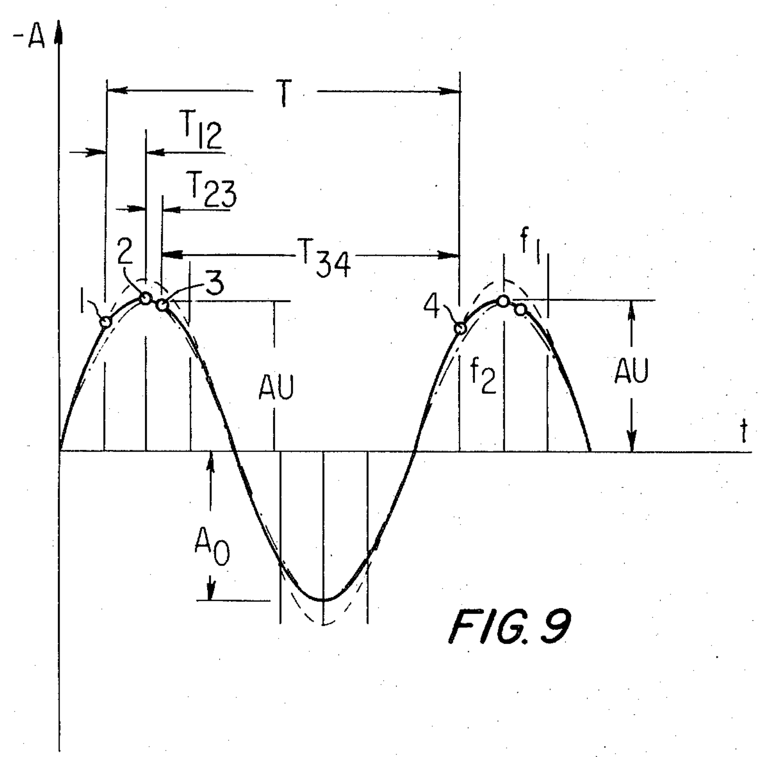

FIG. 9

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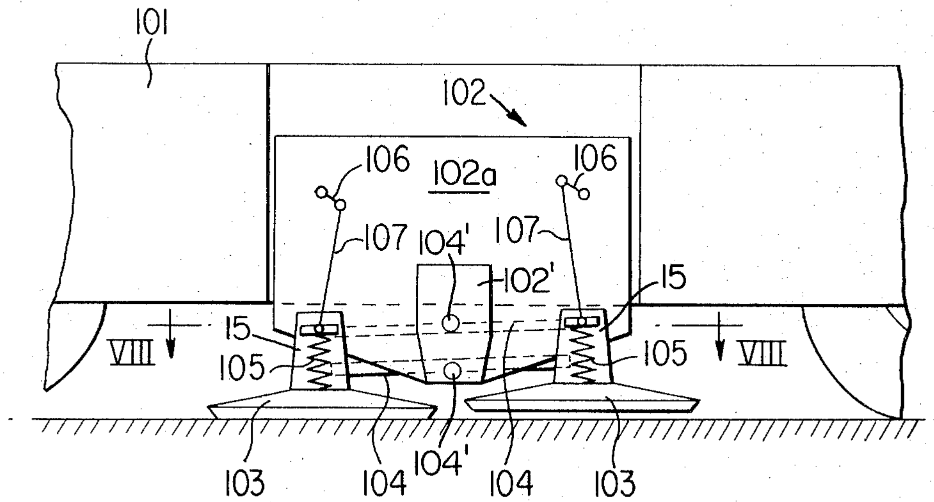
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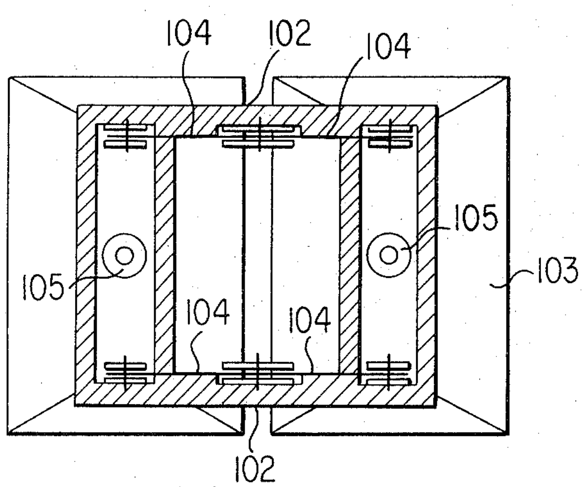
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FIG. 8

FIG. 10

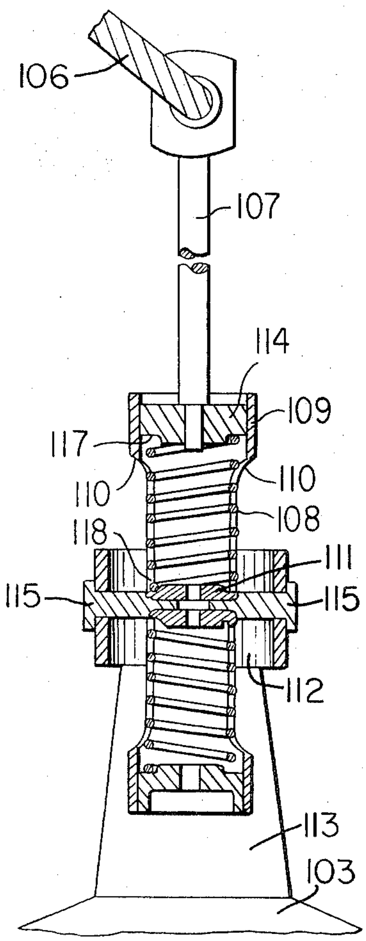
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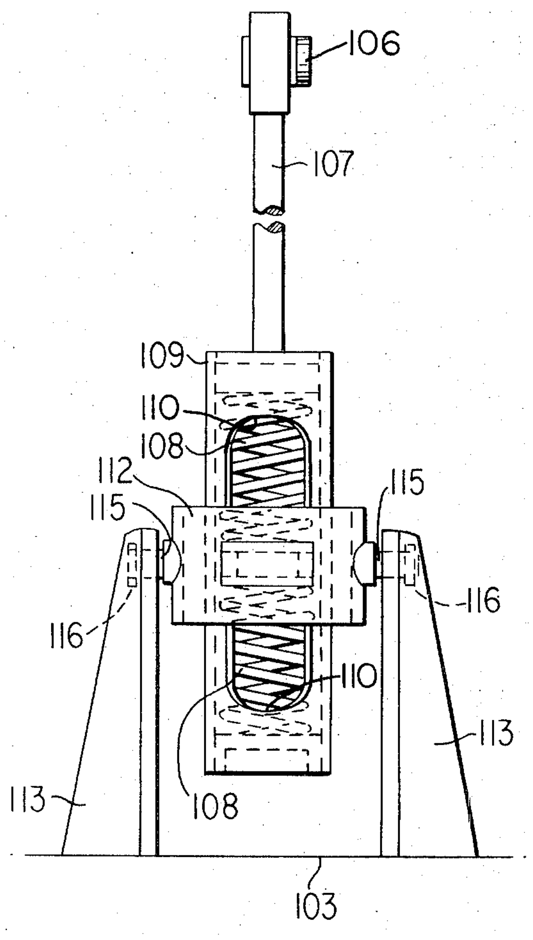
FIG. 11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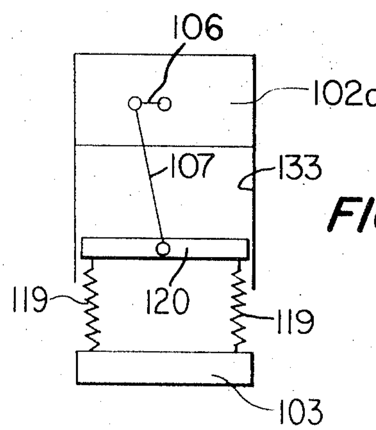

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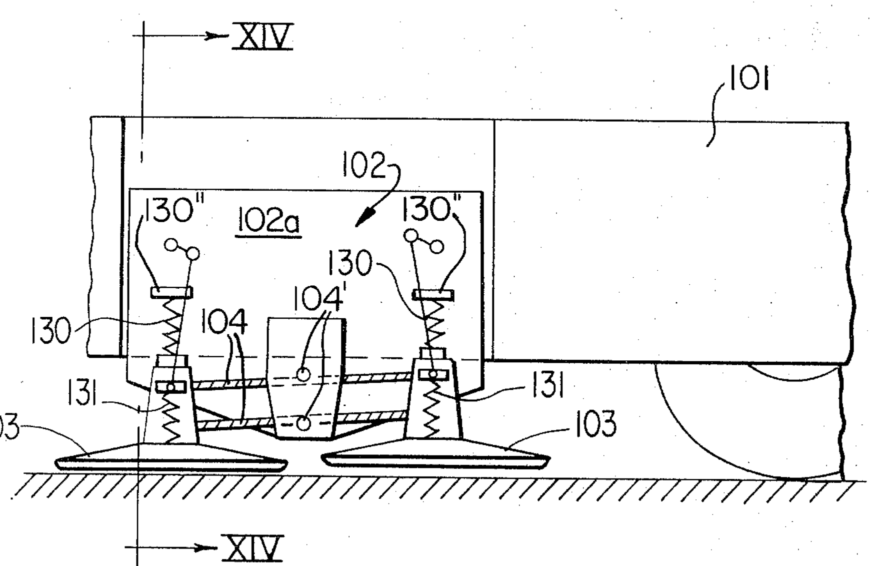

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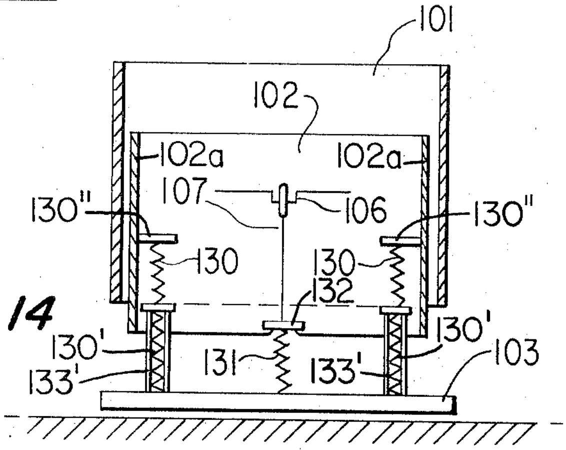

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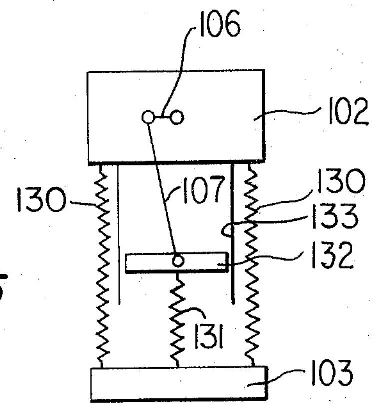

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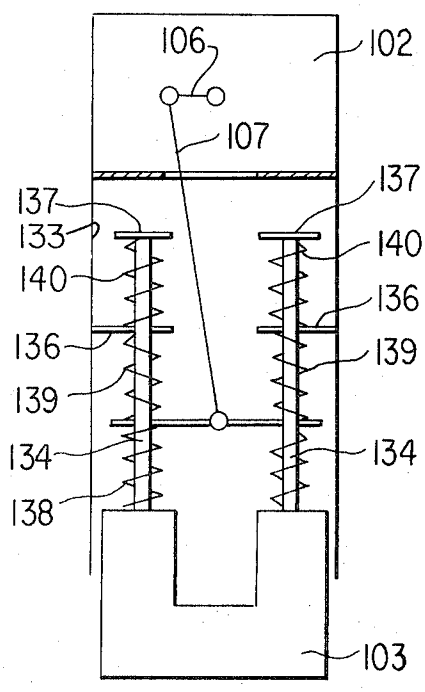

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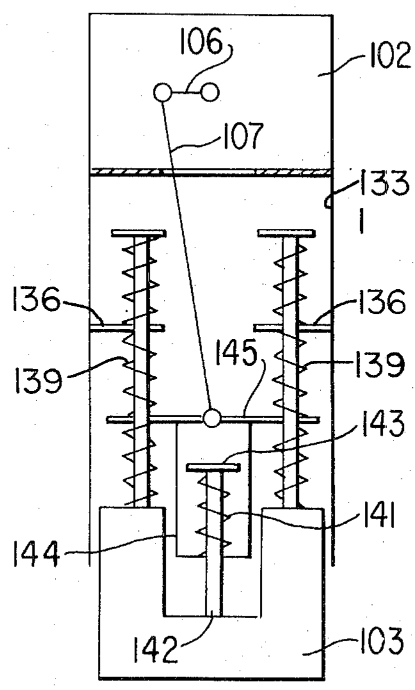

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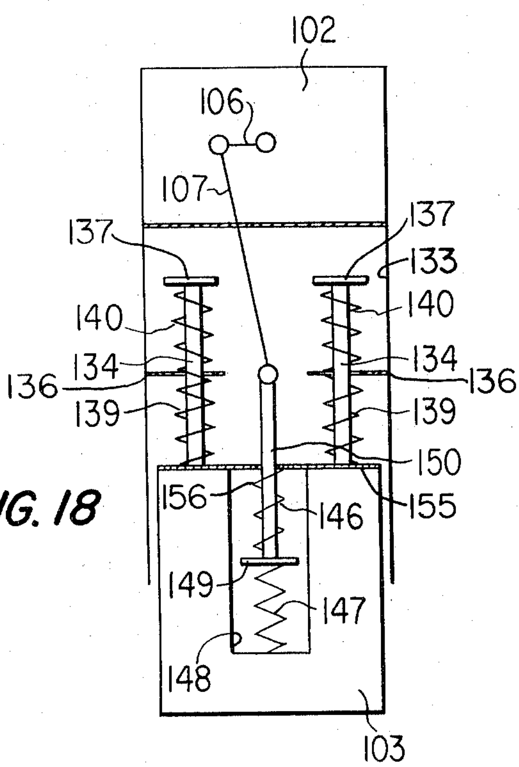

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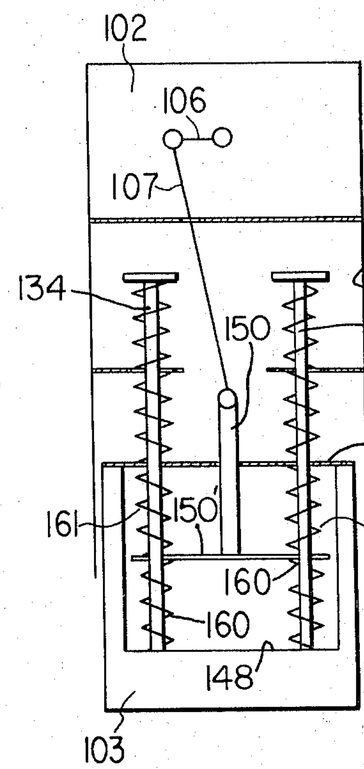

FIG. 19

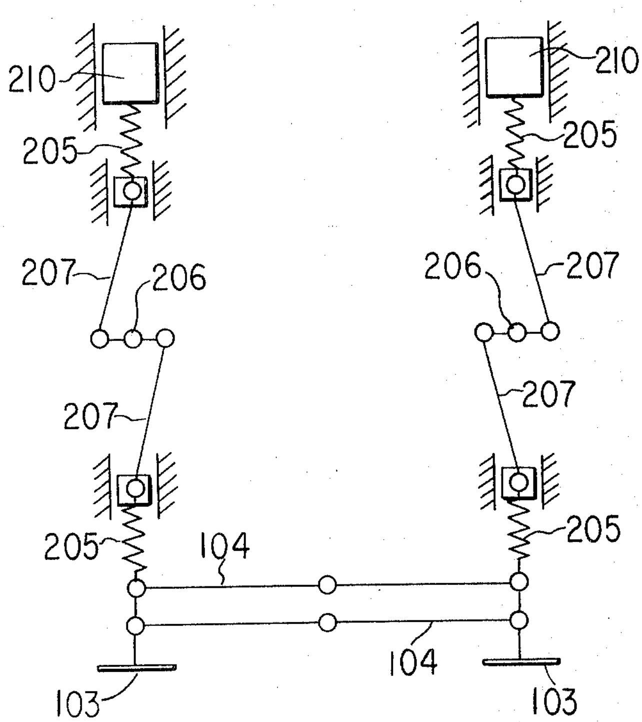

FIG. 28

FIG. 20

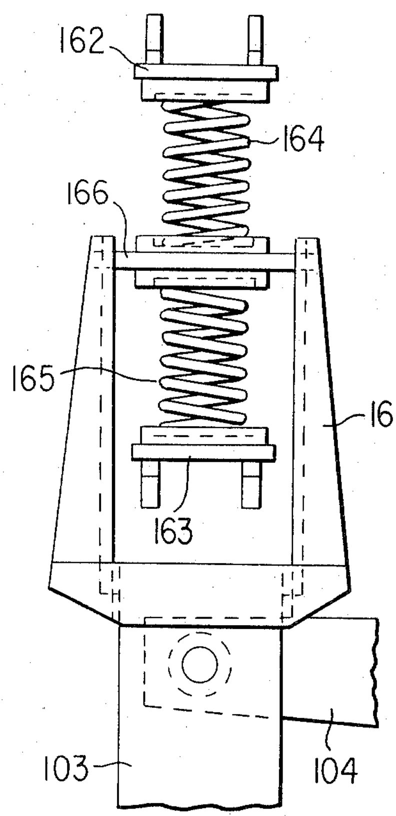
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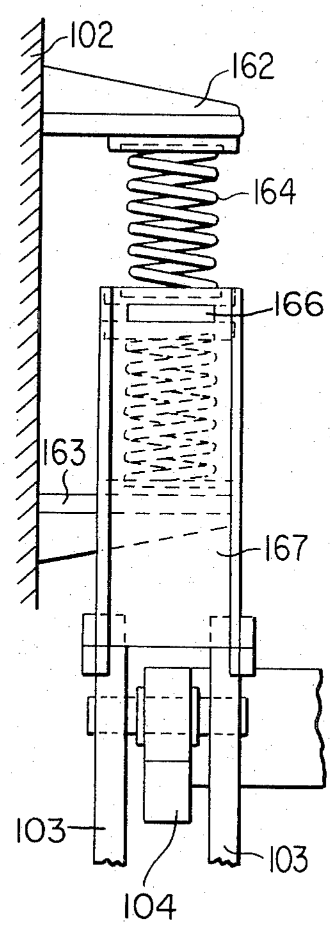
FIG. 21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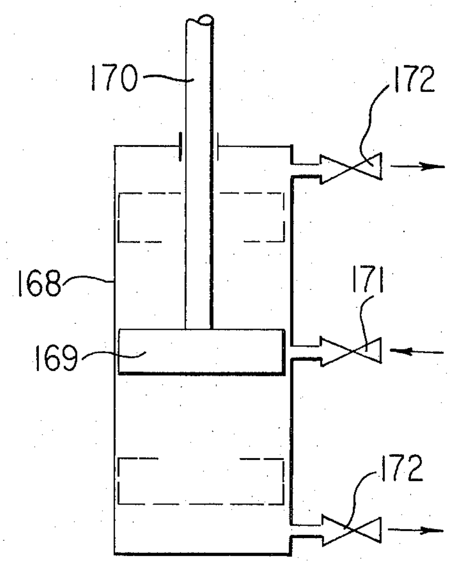
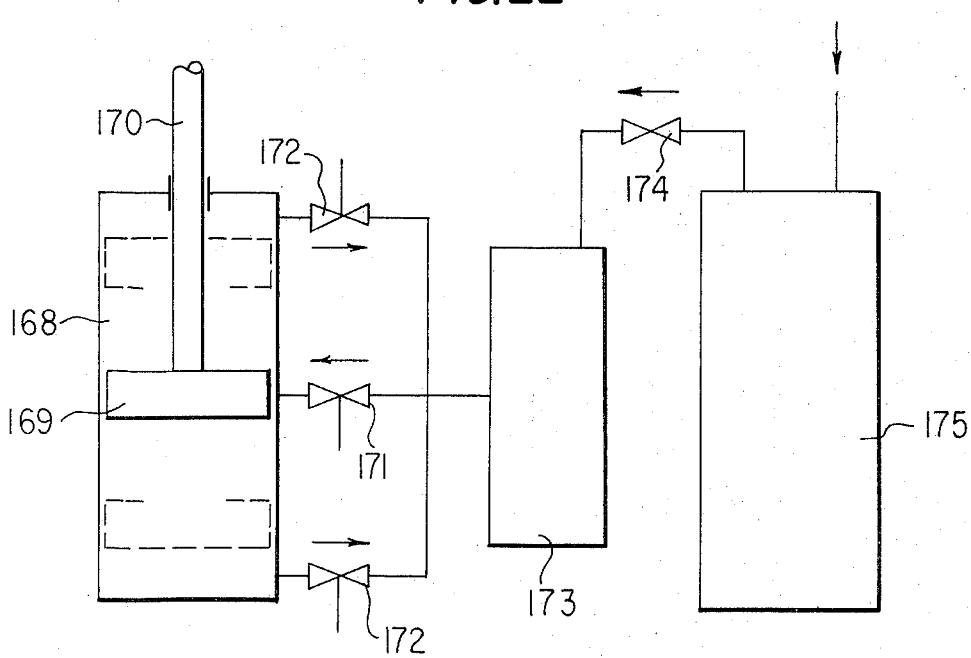


FIG. 22



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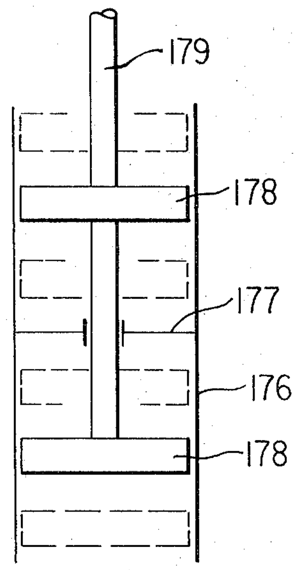
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FIG. 24

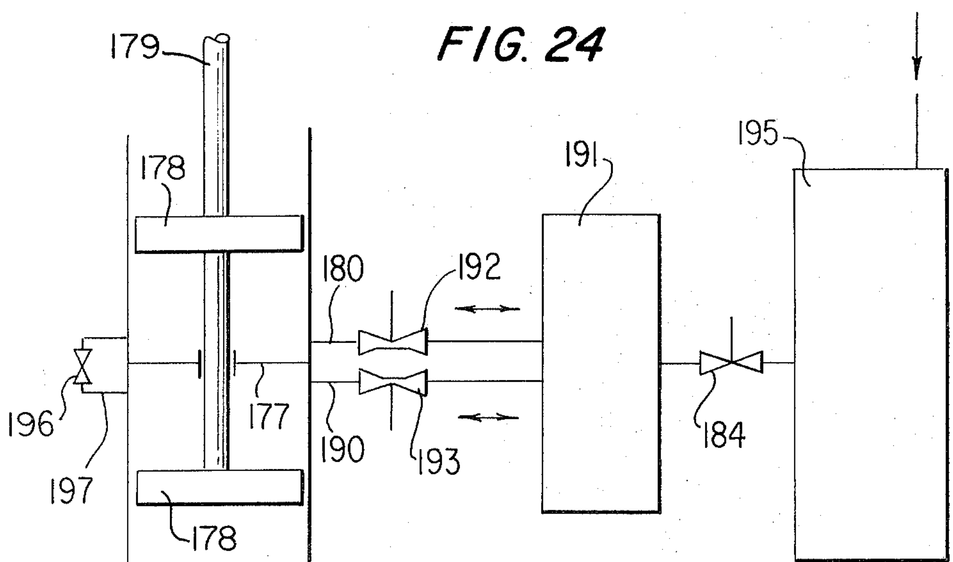
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FIG. 25

FIG. 26

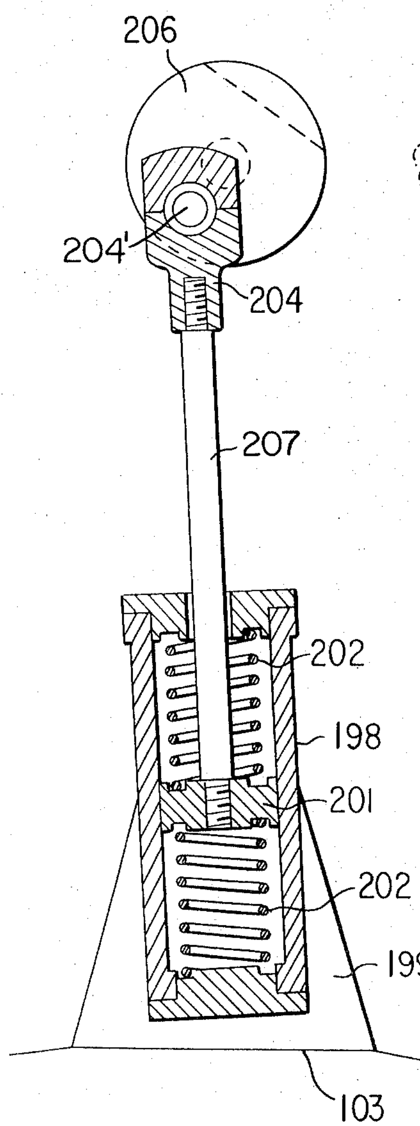
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FIG. 27

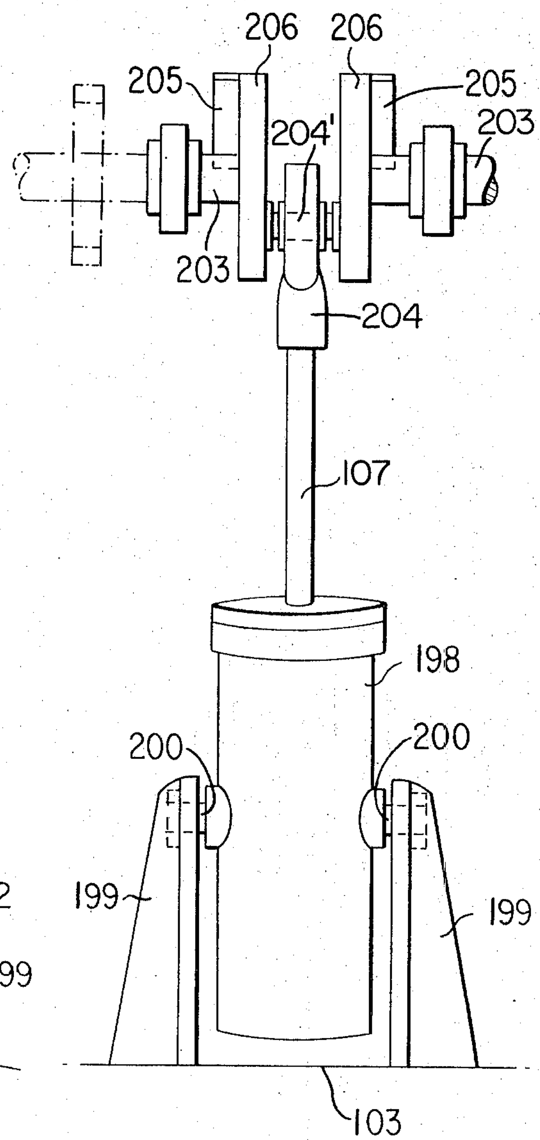
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FIG. 29

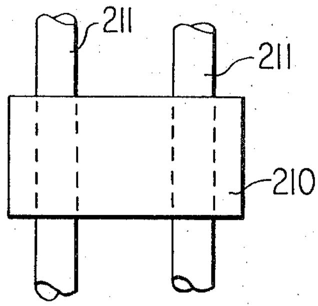


FIG. 30

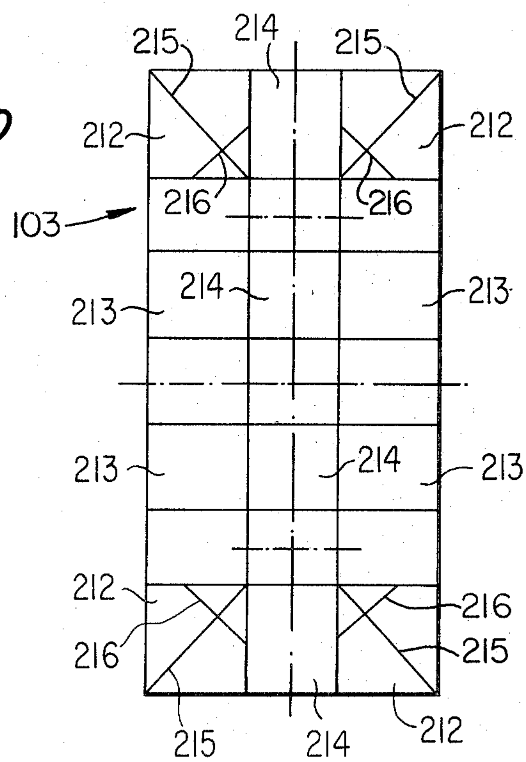


FIG. 31

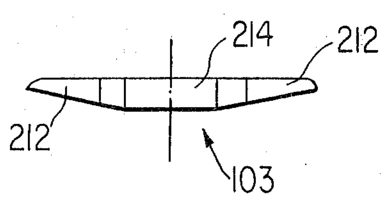


FIG. 34

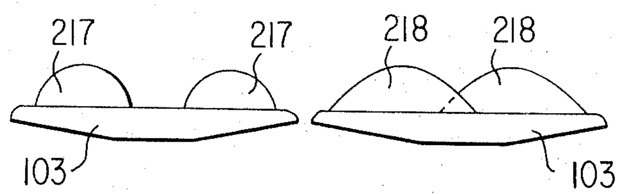


FIG. 33

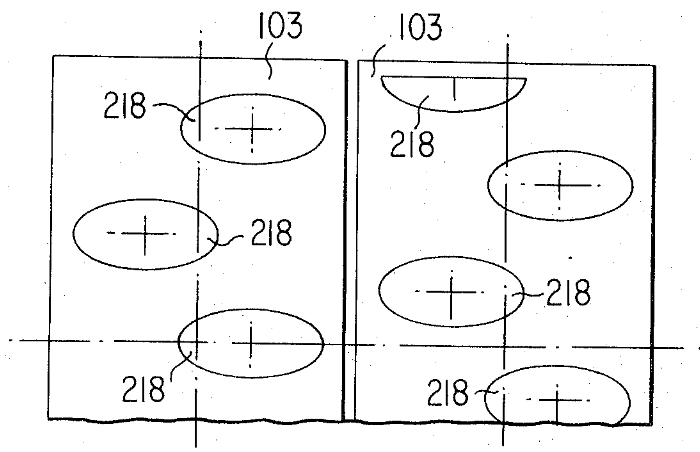
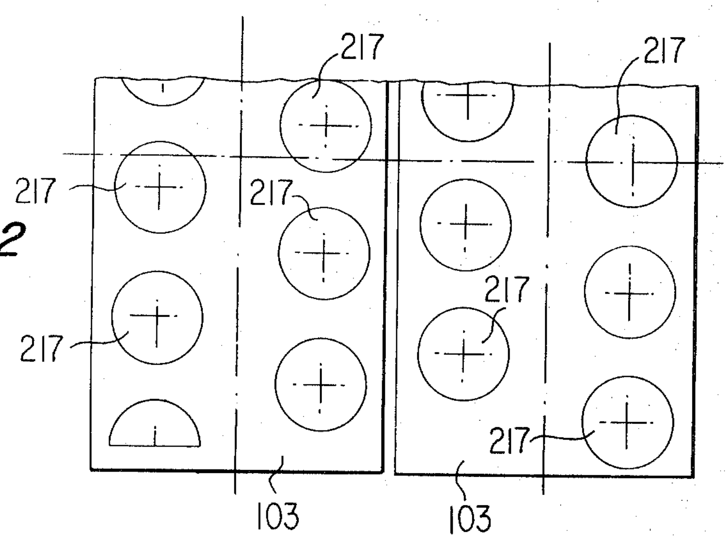


FIG. 32



VEHICLE MOUNTED VIBRATORY COMPACTOR

BACKGROUND OF THE INVENTION

1. Field of the Invention

The invention relates generally to vehicle mounted vibratory compactors, and more particularly to vertically adjustable rigid frames that support a pair of cooperating vibrating elements for efficient compacting operation. 2. Description of the Prior Art

Vibratory compactors of various designs have been widely used for compacting particulate materials, such as sand, gravel, soils of all kinds and consistencies, by repeatedly and rapidly applying vibratory compacting forces thereto. One variety of conventional vibratory compactors has included a vehicle frame, a motor for driving the frame to the work site, a transverse shaft supported on the vehicle frame, an array of heavy tamping shoes suspended from the shaft at spaced intervals, and take-off belts for delivering vibratory energy from the motor to each of the tamping shoes in sequence. Such multishoe vibratory compactors are disclosed in several United States patents, including U.S. Pat. No. 2,938,438, granted to W. L. Hamilton, U.S. Pat. No. 2,958,268, granted to Moir, and U.S. Pat. No. 3,181,442, granted to J. H. Brigel.

Several problems have been encountered with the multishoe vibratory compactors; for example, alignment between the adjacent shoes has to be carefully maintained, lest the shoes collide with attendant damage to the shoes and impairment of compacting efficiency. Also, the spacing between shoes leads to a tendency to create a series of irregularities in the compacted material disposed therebetween; and the repeated raising and lowering, and vibrating movement, of the tamping shoes leads to excessive wear at the supporting bearings and/or the drive belts that deliver the vibrating forces to the shoes.

SUMMARY OF THE INVENTION

The present invention overcomes the above noted structural shortcomings and functional deficiencies of known vibratory compactors by providing a vehicle mounted compactor of simplified design that is characterized by a rigid, parallelogram frame for supporting the vibrating elements, such frame being vertically adjustable within guides defined on the vehicle. Consequently, the vibrating elements can be elevated to pass freely over obstacles in the road bed as the compactor is driven to the work site, and the vibrating elements can be lowered upon reaching the site. Additionally, the amplitude of the stroke of the vibrating elements can easily be varied. The pair of vibrating elements, which may be plates, beam, shoes or rollers, are supported by the parallelogram frame for both pivotal and vertical movement. The driving means for imparting vibratory energy to the vibrating elements can be provided over a wide range of frequencies, thereby enabling the compactor to satisfactorily compress a wide variety of materials. Vibration dampeners deaden harmful, reactive forces and extend the useful life of the instant vibratory compactor.

It is another objective to provide at least one embodiment of the rigid parallelogram support frame lacking a natural or inherent frequency, so that the vibrations imparted thereto by the driving mechanism are transmitted with minimal losses to the vibrating elements. In other embodiments, sundry combinations of springs are utilized to establish a desired natural, or inherent, frequency for the frame. A vibration generator is positioned directly on each vibrating element, so that the driving mechanism is far simpler in design than the conventional complex power take-off belt arrangement utilized for sequentially driving each vibrating shoe in the multi-shoe compactors described above. The driving mechanism includes two cranks operating 180° out of phase so that one vibrating element is engaged with the material to be compacted while the other vibrating element is raised to its highest point of travel.

It is yet another objective to provide, in certain embodiments of the vibratory compactor, series or parallel pneumatic circuits interposed between the cranks and the compacting elements to function in the same manner as mechanical springs to influence the frequency of the compacting system. Such circuits, which include various combinations of pistons, manually adjustable pressure limiting valves, reservoirs, etc. may be used in lieu of mechanical springs, or in combination therewith.

It is yet another objective to provide series or parallel pneumatic circuits, which function in the same manner as mechanical springs interposed between the cranks and the compacting elements. Additionally, the pneumatic circuits can be utilized in combination with the mechanical springs. The pneumatic circuits may include various combinations of pistons, manually adjustable pressure limiting valves, pneumatic reservoirs or accumulators.

It is still another objective to provide compensating weights located vertically above the vibrating elements and driven by the same cranks that drive the vibrating elements, but always 180° out of phase therewith. Such compensating weights dampen the inertial forces reacting upon the frame of the vibratory compactor, and permit the utilization of a vehicle and frame of reduced tonnage without a reduction in operational efficiency.

Since the inertial reactive forces acting upon the vibrating elements are substantially reduced by virtue of the various unique constructional features noted above, the vibrating elements, per se, may be assembled from simplified, hollow components. The vibrating elements may thus be readily altered in size, and their light weight reduces the magnitude of their inertial forces. "Sheepsfoot" protuberances may be formed on the compacting surface. In contrast thereto, conventional vibrating shoes were heavy unitary members that lacked versatility in size and shape.

Other objectives and features of the instant invention will be pointed out in the following description and illustrated in the accompanying drawings, which disclose, by way of example, the principles of the instant invention, and the best modes contemplated for achieving such principles.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a side elevational view of a vehicle mounted vibratory compactor constructed in accordance with the principles of the instant invention;

FIG. **2** is a horizontal cross-sectional view of the vibratory compactor, such view being taken along line II—II in FIG. **1** and in the direction indicated;

FIG. **3** is a vertical cross-sectional view of the vibratory compactor with the vibrating elements in the elevated position, such view being taken along line III—III in FIG. 1 and in the direction indicated;

FIG. 4 is a vertical cross-sectional view identical to the view of FIG. 3, but showing the vibrating elements in a lowered position;

FIG. 5 is a side elevational view, on an enlarged scale, of a preferred embodiment of the frame for supporting the vibrating elements;

FIG. 6 is a side elevational view of a modification of the vibratory compactor of FIGS. 1–5, such view showing additional tools mounted on the vehicle frame;

FIG. 7 is a combined side elevational view and schematic representation, on an enlarged scale, of a first alternative embodiment of the frame for supporting the vibrating elements and the drive mechanism for imparting vibratory energy thereto;

FIG. 8 is a horizontal cross-sectional view of the first alternative embodiment of the frame for supporting the vibrating elements, such view being taken along line VIII—XIII in FIG. 7 and in the direction indicated;

FIG. 9 is a graphical representation of the cycle of operation for the frame and vibrating elements shown in FIGS. 7 and 8;

FIGS. 10 and 11 are side and front elevational views, respectively, of the drive mechanism for imparting vibratory energy to one of the vibrating elements of the embodiments of FIGS. 1–5, 6, or 7–8;

FIG. 12 is a schematic representation of an alternative drive mechanism for imparting vibratory energy to one of the vibrating elements of the embodiments of FIGS. 1–5, 6 or 7–8;

FIG. 13 is a combined side elevational view and schematic representation, on an enlarged scale, of a second alternative embodiment of the frame and the drive mechanism for imparting vibratory energy thereto;

FIG. 14 is a vertical cross-sectional view of the embodiment of FIG. 13, such view being taken along line XIV—XIV and in the direction indicated;

FIGS. 15–19 are schematic representations of various mechanical spring systems, including appropriate guides, for imparting vibratory energy to the vibrating elements;

FIGS. 20 and 21 are side and front elevational views, respectively, of another mechanical spring system for imparting vibratory energy to the vibrating elements;

FIGS. 22–25 are schematic views of pneumatic systems for imparting vibratory energy to the vibrating elements;

FIGS. 26 and 27 are side and front elevational views, respectively, of an alternative mechanical spring system for imparting vibratory energy to the vibrating elements;

FIG. 28 is a schematic view of a mechanical spring and compensating weight system for dampening vibrations;

FIG. 29 is a detail view of the compensating weight used in the system of FIG. 28; and

FIGS. 30–34 show constructional details of the vibrating elements.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings wherein similar reference numerals identify similar components, the preferred embodiment of FIGS. 1–5, and the variant shown in FIG. 6, depict a carrier vehicle including a chassis 1 with two axles 3. Five pneumatic tires 2 are arranged on the axles 3 in an offset fashion so that the displaced tracks exert a compacting pressure upon the surface of the road or upon the material to be compacted. The carrier vehicle can be propelled forwards and backwards at variable speeds; and can be steered with great facility.

Vertical guides 4 are defined on opposite sides of the central portion of chassis 1, and lateral extensions on the frame 5 for the vibrating elements 6 fit within the guides, as shown in FIGS. 2–4. Frame 5 may be supported for movement within the guides by rollers or guides, and the frame may be moved vertically by various conventional devices, such as worm gears, hydraulic actuators, etc.

In the preferred embodiment of FIGS. 1–5, the vibrating elements consist of a pair of vibrating plates, although vibrating beams, shoes, or rollers can be substituted therefor.

The constructional details of the frame 5 which supports the pair of vibrating elements 6 are best shown in FIG. 5. The vibrating elements are held parallel to one another by a pair of horizontally extending control arms 7; the upper control arm is secured to the end plate 8 by a pin or shaft 9, while the lower control arm is secured to plate 8 by a pin or shaft 10. The ends of the control arms 7 furthest from shafts 9 and 10 are secured to side supports 13 by bolts or shaft 14, 14′ and corresponding slots which function as hinges for the parallelogram frame 5 and permit a slight rocking movement of the vibrating elements in response to the vibration generators. The midpoints of control arms 7 pivot about shafts 9 and 10, which are situated in the same vertical plane passing through the center of plate 8. Roller bearings may be positioned about shafts 9, 10 to facilitate movement of the arms about the shafts.

Transverse braces 11, 12, seen in dotted lines and in end elevation in FIG. 5, extend across the lateral dimension of the vehicle and thus tie the parallelogram frame at the right side of the vehicle to the parallelogram frame at the left side of the vehicle. The braces further enhance the strength of the frame supporting the vibrating elements. The optimum location for the total center of mass for the means for imparting vibratory energy to vibrating elements 6 is located in the vertical plane defined by shafts 9 and 10.

The equilibrium position for parallelogram frame 5 is shown in FIG. 5. In the equilibrium position wherein the bottom surfaces of the vibrating elements lie in the same horizontal plane, the shafts 14 are situated a short vertical distance above shaft 10, and shafts 14′ are situated a short vertical distance above shaft 9. Accordingly, when one of the vibrating elements is lowered to its operative position, the frame will exert an unbalancing force with respect to the lowered element.

Vibration generators 15 of conventional design are secured directly upon compacting elements 6, as shown in FIG. 5. The main engine (not shown) for the compactor, which may be an internal combustion engine, transmits power to the generators over a system of transmission belts, including belt 21, which is entrained over pulley 20 on shaft 19, and horizontally extending power take-off belts 30, which are also entrained over pulleys 17 and 20. Shaft 19 extends laterally between cover plates 8 of frame 5, and is situated vertically intermediate shafts 9 and 10. The right hand vibration generator operates 180° out of phase with the the left hand vibration generator. Accordingly, the vibrating elements **6** will always be moving in opposite vertical directions; by virtue of the configuration of frame **5**, the vibrating elements will also always be pivoting or oscillating about shafts **9, 10** in opposite directions.

In operation, vehicle **1** is driven to the work site with frame **5** and compacting elements **6** in the elevated position shown in FIG. **3.** At the work site, the frame and the compacting elements will be lowered into the position shown in FIG. **4.** Then, the elements **6** will press against the ground and be vibrated by vibration generators **15.** Frame **5** can be vertically adjusted within guides **4** to accurately vary the amplitude, or length of travel to an extreme up or down position from a mean position of vibrating elements **6.** The oscillating frequency of the vibrating elements is adjustable by altering the rotational velocity of vibration generators **15.** Furthermore, the weight of the vehicle exerted upon the offset pneumatic tires **2** as the vehicle moves to the work site enhances the efficiency of the compactor.

In the embodiment of FIG. **6,** the carrier vehicle **1** is provided with oppositely disposed scraper blades **38, 39** and a scarifier **40.** Consequently, such embodiment may be utilized as a road working machine of increased versatility for handling bituminous or similar paving materials along a roadway.

The alternative embodiment of FIGS. **7–8,** shows a frame, indicated generally by reference numeral, **102,** and including large plates **102*a*** at opposite sides of the chassis mounted for movement relative to the carrier vehicle **101.** Two vibrating elements **103** are secured to each end plate **102'** and large plate **102*a*** of the frame by shafts **104'** which pass through the horizontally extending control arms **104.** A crank **106** and a rod **107** are connected to each end of the control arm, and a spring **105** extends between the lower end of rod **107** and vibrating element **103** and opposes the downward movement of the rod. Springs **105** may be spiral springs or torsion springs, and crank **106** and rod **107** are only schematic representations of one type of conventional vibration generator.

In operation, cranks **106** are driven from the prime mover in a conventional fashion, so that the elements **106** start to vibrate with a phase lag of 180° between one another. The frame and the vibrating elements, because they are influenced by springs **105,** are a part of an oscillating or vibratory system that has a definite natural frequency. The amplitude of vibrating elements **103** is usually very small but increases sharply when the speed, or frequency, of the crank rotation falls within the resonance range for the springs.

To illustrate, assuming that frame **102** is lowered within guides defined in the body of vehicle **101** to a position wherein the compacting elements are **2** centimeters above the ground, and further assuming that crank **106** starts to rotate, at a speed or frequency half, or less, than the natural frequency of the system, then the vibrating elements will be subjected to a forced oscillation of such a minute amplitude that they will not even completely close the **2** centimeter gap and touch the ground. However, as the frequency of cranks **106** approach the natural frequency of the system, the energy driving forces imparted by the prime mover to the vibration generators **15** increase in effectiveness. Finally, when the frequency of cranks **106** equals the natural frequency of springs **105,** all of the energy imparted by vibration generators **15** is effectively delivered to vibrating elements **103.** The matching or coincidence of the driving frequency of the cranks to the natural frequency of the system is known as resonance magnification, and the compacting process begins with the attainment of resonance magnification.

FIG. **9** graphically represents the course of the oscillatory movement of vibrating element **103** during a cycle of operation. Assuming that the movement of element **103** approximates a sinusoidal path of travel and plotting the amplitude of movement versus time, A_o designates its upper amplitude, A_u its lower amplitude, and T the period of time for execution of a complete cycle. T_{12} is the time interval during which the vibrating element compresses the ground and performs work, T_{23} is the interval during which the vibrating element releases its compressive forces and starts its upward movement away from the ground, and T_{34} is the interval during which the vibrating element does not touch the ground at all.

As noted previously, vibrating element **103** only approximates a sinusoidal function. The variations from the theoretical sinusoidal function are attributable, in part, to the effect of the natural or inherent frequency of springs **105** and the mass of frame **102** and the components suspended therefrom. Also, the movement of the oscillatory system changes rapidly at time T, because of the impact of vibrating element **103** with the ground, and the reaction forces thereby set up in the vibrating element and the frame supporting same. During the time interval T_{34}, during which the elements **103** is free of the ground, the path of oscillatory movement closely approximates the ideal or theoretical function f, for the companion element **103** only exerts a minimal influence upon the system.

During the time interval T_{12}, the vibrating element accomplishes work compressing the plastic, semi-plastic, or hard soil, or other particulate material, while overcoming the initial inertial resistance of the frame to movement. During upward movement, the vibrating element may be assisted in small measure by the resiliency of the material being compacted. However, in most instances, the resiliency of the material being compacted becomes an insignificant factor when compared to the amplitude of the vibrating element. Also, when the compacting forces applied to the material are increased significantly by resonance magnification, the magnitude of the resilient force of the material becomes insignificant by comparison.

While the embodiment of FIGS. **7–8** utilizes a single spring **105** at opposite end of each vibrating element, the embodiment of FIGS. **10** and **11** utilizes a pair of coil springs **108** in lieu of the single spring. Both springs are situated within cylinder **109** which tapers outwardly at its opposite ends, and the rod **107** is driven by crank **106** and is secured to the upper end of the cylinder. Two diametrically opposite, axially extending sections **110** are removed from the central portion of cylinder **109,** and stub shafts **115** extend through open sections **110** and maintain a spacer **111** intermediate the two springs. One end of each stub shaft extends axially through sleeve **112** which concentrically surrounds cylinder **109,** while the opposite end of each shaft is mounted for pivotal movement by a journal **116** in bracket **113.** The brackets are fastened directly to the upper surface of vibrating element **103.**

The pair of springs **108** are easily assembled within cylinder **109** in operative relationship in the following manner. The shafts **115** are removed from engagement with brackets **113** and end caps **114** are removed from the opposite enlarged ends of the cylinder. Springs **108** are then inserted into the cylinder, and end caps **114** are then pushed into position, firmly seating the springs. Aligned holes are drilled through the central portion of the first end cap, the spacer **111** and the second end cap. One end of rod **107** is then seated within the hole in the upper end cap.

Due to the oscillatory movement of crank **106**, rod **107** moves vertically and the upper end cap **114** slides within the enlarged upper end of cylinder **109**. The cylinder also pivots about stub shafts **115** in response to the movement of rod **107**. The movement of the end cap compresses the upper spring **108**, and the resultant forces are transmitted through spacer **111** and stub shafts **115** to brackets **113** and thence to vibrating elements **103**. The springs are retained in fixed position by annular projections **117** on end caps **114** and annular projections **118** on spacer **111**. The cylinder **109** prevents kinking of the springs as they are compressed.

During the cycle of operation of the system embodiment of the elements **103** shown in FIGS. **7** and **8**, the natural frequency of the oscillating system is determined by the combined mass of pivotally mounted cylinders **109** and the springs seated therein.

The embodiment of FIG. **12** utilizes a pair of spaced parallel springs **119** in lieu of the springs **105** of the embodiment of FIGS. **7** and **8**, or the opposed pairs of prestressed springs **108** of the embodiment of FIGS. **10–11**. The piston rod **107**, which is driven by the crank at one end, is secured to a beam **120** at its opposite end. The parallel springs **119** extend between the beam and the vibrating element **103**, and beam **120** is movable vertically within guides in frame **102**. Both springs act as pressure springs, and together with the other components secured to frame **102**, form an oscillatory system. Such system may have the same natural frequency as the oscillating systems of FIGS. **7–9** and **10–11**, or it may have a different natural frequency.

In the embodiment of FIGS. **13** and **14**, pressure springs **130** extend through cylinders **130'** at opposite ends of vibrating element **103**, and pressure spring **131** is connected between plate **132** and the midpoint of element **103**. Plate **132** is connected to the lower end of rod **107**, which is driven by crank **106**. The upper ends of springs **130** rest against ledges **130''** on the inner surface of frame **102**. Springs **130** and **131** have the same effect upon the natural frequency of the oscillatory system as springs **105** in the embodiment of FIGS. **7–8**. Under normal operating conditions, a portion of the reactive forces caused by the vibrations of element **103** striking the ground is transferred through springs **130** directly to the frame **102**, while by-passing the vibratory drive mechanism **106, 107, 131** and **132**. At resonance, the ratio of the reactive forces transferred directly to the frame **102** to the reactive forces transferred back to the oscillation drive mechanism, is directly proportional to the ratio of the force of springs **130** to that of spring **131**. Phrased in another manner, the larger the ratio of the force of springs **130** to that of spring **131**, the greater the amount of reactive force transferred directly to the frame **102**. The frame, obviously, is of sturdy construction and can easily withstand forces of such magnitude while increasing the operational life span of the other components of the oscillating system. Additionally, the pair of vibrating elements are supported for vertical movement, as well as pivotal movement, by a rigid parallelogram support frame including spaced side supports, a pair of control arms **104**, and a pair of spaced shafts **104'** about which the control arms can pivot.

FIG. **15** is a detailed view of the vertical guide **133** that controls the movement of beam **132**, which is connected to the lower end of rod **107**. Springs **130** and **131** are also provided with suitable guides, such as cylinders **133**, to prevent sidewards kinking during compression thereof.

FIG. **16** illustrates another guide adapted to restrain the movement of two, or more, springs. The guide, which is indicated generally by reference numeral **133**, includes a pair of rods **134** extending upwardly from compacting element **103**, and a beam **135** which moves vertically along both rods. Rods **134** extend upwardly through guide surfaces **136** which are secured to the guide **133**; the guide, in turn, is secured directly to the frame **102**. A first spring **138** is located on each rod between the upper end of compacting element **103** and beam **135**; a second spring **139** is located on each rod between beam **135** and guide surface **136**; and a third spring **140** is located on each rod between guide surface **136** and enlarged stop **137** atop the rod. All of the springs are prestressed in the position shown in FIG. **16.**

When crank **106** is rotating in a clockwise direction, springs **139** are compressed while springs **138** are relaxed, and vibrating element **103** follows the vertical movement of beam **135** due to the action of springs **140** and guide **133**. At the midpoint in the cycle the process is reversed, and springs **138** are compressed while springs **139** are relaxed, so that vibrating element **103** follows the movement of beam **135** against the resistance of springs. **140**. The vibrating element **103** oscillates due to the vertical reciprocation of beam **135**. When the resonance frequency of the system is reached, the system is damped only by air resistance, internal friction in springs **138, 139** and **140**, and by the work done during compacting the ground.

FIG. **17** illustrates another guide adapted to restrain the movement of three parallel springs. The guide, which is against indicated generally by reference character **33**, includes an additional spring **141** that is positioned about a rod **142** secured to the upper surface of element **103**. Spring **141** extends between a stop **143** at the upper end of rod **142** and a supporting surface on the cylindrical member **144** that is mounted beneath plate **145**. The cylindrical member is concentric with rod **142**. Consequently, as plate **145** is vertically reciprocated by crank **106** and rod **107**, the movement of cylindrical member **144** compresses, and relates spring **141**. In all other essential details, guide **233** functions in the same manner as guide **133** in FIG. **16**.

FIG. **18** depicts another mechanical spring system for effectively transmitting vibratory energy from an internal combustion engine, or other conventional prime mover to the vibration generator, and thence to the vibrating elements **103**. A beam **155** is secured across the open upper end of element **103**, so that the beam and the interior walls of the element define a chamber **148** therebetween. An aperture is formed in the center of the beam, and a piston rod **150** with an enlarged lower end **149** extends therethrough. A first spring **146** extends between end **149** and the bottom surface of chamber **148**. A pair of spaced rods **134** are secured to the upper surface of beam **155**, and a pair of guide surfaces **136** extend inwardly from guide **133**. A first spring **139** is located on each rod **134** between beam **155** and surface **136**, and a second spring **140** is located on each rod **134** between surface **136** and the enlarged stop **137** atop rod **134**. During a portion of the cycle of operation of the vibratory energy transmitting system, crank **106** and lever **107**, beam **155**, and thus vibrating element **103** is influenced by the compression of springs **146** and **139** while the springs **140** and **147** are relaxed. After passing the midpoint in its cycle of operation, beam **155** is influenced by the compression of springs **147** and **140** while springs **139** and **146** are relaxed.

FIG. **19** shows a variant of the mechanical spring system of FIG. **18** wherein the chamber **148** defined between beam **155** and the internal walls of vibrating element **103** has been enlarged. Also the springs **146, 147** operatively associated with piston rod **150** have been eliminated, rods **134** have been lengthened, and beam **150'** is secured to the lower end of rod **150** and moves vertically along rods **134**. Springs **160** are tensioned, when springs **161** are relaxed, and vice versa. Springs **160** act in concert with springs **139**, while springs **161** act in concert with springs **140**. The net result is that energy imparted by crank **106** and rod **107** is effectively transmitted by the springs and beam **150'** to parallel beam **155** and vibrating element **103**.

FIGS. **20** and **21** disclose a mechanical spring system for transmitting a portion of the spring forces directly to the frame supporting elements **103**, such as frame **102** shown in the embodiment of FIGS. **7** and **8**. Brackets **162** and **163** are secured directly to the heavy plate **102**(a) of frame **102**. A first spring **164** has one end centered and seated firmly in bracket **162** and the rest of the spring extends downwardly toward bracket **163**. A second spring **165** has one end centered and seated firmly in bracket **163**, and the rest of the spring extends upwardly toward bracket **162**. A bar **166** extends horizontally across the upper end of supports **167**, and the lower end of spring **164** and the upper end of spring **165** are centered and seated firmly therein. The lower end of supports **167** is connected to one end of control rod **104**. Alternatively, the lower end of the support could be connected directly to vibrating element **103**.

If the maximum amplitude of the vibrating element is assumed to be 40 mm., and the total travel of the element is 80 mm., the length of the control rod is selected such that the maximum lateral deflection of springs **164, 165** amounts to only 0.8 mm. Thus, there the danger of lateral kinking of the springs is obviated. FIG. **22** shows a pneumatic servo-piston that may be used in combination with mechanical spring systems shown in FIGS. **10–21**, or in lieu thereof. Such pneumatic servo-piston may be used as a solitary control device or as an integral part of a pneumatic control circuit. The pneumatic servo-piston includes a cylindrical housing **168** and a piston **169** mounted piston rod **170** so as to movable within the housing. A first pressure limiting valve **171** is operatively associated with the mid-section of the housing, while a pair of pressure limiting valves **172** are situated at opposite ends of the housing. Valves **172** limit the maximum pressure built-up within the housing, while valve **171** determines the minimum pressure level within the housing. Consequently, the pneumatic servo-piston operates with a particular volumetric efficiency over a selected range of pressures, and closely approximates the action of a mechanical spring. These characteristics can be adjusted by varying the settings of valves **171** and **172** so that the power transmitted to compacting elements **103** from crank **106** and rod **107** can be adjusted. Also, the valve settings may be altered to overcome the effects of inertia when starting crank **103** in motion or when attempting to stop same once it is in motion.

FIG. **23** shows a pneumatic circuit including the pneumatic servo-piston of FIG. **22** and further including a container **173** of pressurized fluid that is connected through pressure reducing valve **174** to reservoir **175**. The pressurized fluid in container **173** influences the movement of piston **169** and piston rod **170** through valves **171** and **172**. By utilizing the closed circuit of FIG. **23**, the servo-piston operated independently of ambient air conditions and pressure changes. Additionally, the valves may be adjusted so that the servo-piston may be adjusted into harmony with the operation of the various mechanical spring systems.

FIG. **24** shows another pneumatic servo-piston that can be utilized within the power transmitting system of the instant invention. Such servo-piston includes a cylindrical housing **176** which may be open at both ends, and a middle partition **177** which extends perpendicular to the axial dimension of the cylinder and divides same into two chambers. A pair of pistons **178** are spaced along piston rod **179** at opposite sides of partition **177**. The rod passes through an aperture in the center of partition **177**, and the pistons move along the inner wall of housing **176**.

FIG. **25** shows a pneumatic circuit including the pneumatic servo-piston of FIG. **24** and further including a container **191** of pressurized fluid that is connected through pressure reducing valve **184** to reservoir **195**. The chamber formed above partition **177** is connected to container **191** via conduit **180** and adjustable valve **192**, while the chamber formed below partition **177** is connected to container **191** via conduit **190** and adjustable valve **193**.

An equilibrium condition occurs during the operation of pistons **178** and piston rod **179** wherein the fluid entering housing **176** from container **191** on each piston rod stroke is equal to the fluid leaving housing **176** and returning to container **191**. Such condition is related to the volumetric effeciency of the servo-piston, and influences the power transmitted to compacting element **103**. The equilibrium condition can be altered by varying the settings of valves **192, 193** and **194**.

The motion of elements **103** can be stopped even through crank **106** and rod **107** are running by the simple expedient of venting valves **192, 193**, to relieve any pressure build-up. Similarly, crank **106** and rod **107** can be brought up to full speed before connecting housing to the source **191** of pressurized fluid. In both instances, the crank can not transmit power to the vibrating elements. When power transmission is desired, then valves **192, 193** are adjusted accordingly.

Alternatively, an adjustable valve **196**, or a variable stop member, or the like, is positioned within conduit **197** which communicates with opposite sides of partition **177**. Valve or stop **196** can readily be adjusted, and valves **192, 193** can be set once and such setting can be maintained constant, if so desired.

FIGS. **26** and **27** show a combined pneumatic servo-piston and spring system for transmitting energy to compacting element **103**. The servo-piston includes a cylindrical housing **198** that is pivotally connected to supports **199** mounted upon compacting elements **103** by hinge pins **200.** A piston rod **207** extends through the end cap of housing **198** and is secured at its lower to piston **201.** The piston is movable within housing **198,** and is normally retained in a central position by opposed biasing forces of a pair of springs **202** for one spring is situated on each side of the piston. The upper end of rod **207** is secured to head **204** which is mounted upon shaft **204'** and is operatively associated with crankshaft **202.** By virtue of this arrangement, inertial forces are minimized. At least one compensating weight **205** and at least one oscillation disc **206** is mounted on crank shaft **203** on opposite sides of piston rod **207.** Weights **205** compensate for the crank forces, and the discs **206** compensate for, or dampen the reactive forces transmitted to the drive mechanism that occur as the crankshaft accelerates and as each compacting element reverses direction. The compensating weights and oscillation discs may be well used in harmony with the pneumatic servo-piston shown in FIG. **22** and the pneumatic circuit shown in FIG. **23.**

In addition to the inertial forces attributable to the movement of crankshaft **203,** rod **207,** etc., inertial forces are attributable to the movement of elements **103.** FIG. **28** depicts a combined mechanical spring and compensating weight system for overcoming these undesirable inertial forces. Large compensating weights **210** are utilized to overcome the inertial forces attributable to compacting elements **103;** such weights are secured in operative position in frame **102** above the vibrating elements **103.** Weights **210** are connected via springs **205,** rods **207** and cranks **206** to the opposite ends of control rods **104,** and the weights are actuated 180° out of phase. Additional weight may be added to, or substracted from, weights **210** until the optimum response is achieved.

The compensating weights **210** may be formed so as to slide along a pair of rods **211,** as shown in FIG. **29.** Alternatively, the weights may be formed simply as cylindrical sleeves, as represented by sleeve **112** as shown in FIG. **11.** In another alternative configuration, the weights may be formed as discs that can be readily secured to the exterior of cylinder **109** shown in FIG. **11;** if desired, additional light adjusting weights can be added thereto, or removed therefrom, to accommodate minor variations in the operation of the compacting machine.

FIGS. **30–31** show constructional details of the elements **103,** which are fabricated from several components. Due to such construction, the weight of the compacting element itself is reduced and the undesirable inertial effects are minimized. The metal elements so formed are rigid in nature, and are admirably suited for compressing or tamping flat surfaces, such as streets.

Element **103** comprises edge members **212,** longitudinal spacers **213,** and lateral spacers **214.** Edge members **212** are reinforced by ribs **215** and **216.** The longitudinal extent of element **103** is governed by the number of spacers **213** utilized, while the lateral extend of element **103** is governed by the number of spacers **214** utilized. The element may be formed to include one, or several, layers of the various components, depending upon the material to be compressed. Components **212, 213** and **214** may be welded or bolted together so that the size of compacting element **103** can readily be varied, particularly if bolts are utilized.

Components **212, 213** and **214** may be integrally cast or formed of standard sized steel stock welded together. Spacers **214** are easily formed as boxes having an open upper end, while spacers **213** are formed with a bottom having a slight upward taper and a rectangular sectional area. Edge members **212** are shaped in much the same fashion as spacers **213,** but include diagonal reinforcing rib **215** and intersecting reinforcing rib **216.**

The working face of element **103** that contacts and compresses the material may have semi-circular projections evenly distributed thereabout, as shown in FIG. **32** and the left hand view of FIG. **34.** The working face may have oval-shaped projections evenly distributed thereabout, as shown in FIG. **33** and the right hand view of FIG. **34.** Alternatively, the working face may have both semi-circular and oval projections, or projections of other configurations may be used. The projections enhance the effectiveness of the compacting element in a manner similar to the action of a "sheepsfoot" on a heavy duty roller.

While the foregoing specification has set out the preferred mode of practicing the present invention, it will be apparent that various modifications and changes may be made thereto by the artisan to which this invention appertains without departing from spirit and scope of the invention as expressed in the following claims.

I claim:

1. A compacting machine comprising, in combination,

a. a vehicle with a chassis transportable to the material to be compacted,

b. a compacting unit suspended from said chassis,

c. said compacting unit including first and second vibrating elements extending parallel to one another across the chassis of said vehicle, a first vibration generator means connected to said first vibrating unit and a second vibration generator means connected to said second vibrating element, and vibratory drive means connected to said vibration generators for alternately driving said elements against the material to be compacted,

d. support means for securing said compacting unit to said chassis, said support means comprising first and second end plates located on opposite sides of said chassis,

said support means further comprising first and second spaced supports positioned parallel to one another at opposite sides of said chassis, a first side support connected to said first vibrating element and a second side support connected to said second vibrating element, a pair of control arms, each of said control arms being pivotally secured at its opposite ends to said side supports, said arms being vertically spaced and substantially parallel to one another and said side supports being parallel to one another to define a parallelogram frame,

f. each of said control arms being pivotally joined at its midpoint to one of said plates so that said parallelogram frame can be oscillated relative thereto.

2. The compacting machine as defined in claim **1** further comprising:

g. guide means defined between said compacting unit and the chassis of said vehicle so that said compacting unit can be vertically adjusted relative to said chassis.

3. The compacting machine as defined in claim 2 wherein said guide means comprises lateral extensions on opposite sides of said compacting unit and said chassis has vertical slots defined therein to receive said lateral extensions of said compacting unit.

4. The compacting machine as defined in claim 1 wherein said control arms are joined to said side supports by pivotal connections so that said vibrating elements can rock slightly relative to said control arms.

5. The compacting machine as defined in claim 1 wherein said machine includes a plurality of control arms, at least one pair of control arms being disposed at opposite sides of the chassis, and braces extending horizontally between said pairs of control arms for enhanced rigidity.

6. The compacting machine as defined in claim 1 wherein each control arm tapers gradually outwardly to define a thickened center portion, and the points at which each control arm is joined to one of said plates lies in a straight, vertically extending line.

7. The compacting machine as defined in claim 1 wherein the vibratory drive means includes a vibration generator for each vibrating element including a driven shaft with cranks secured thereto, the vibration generator for the first vibrating element being operated 180° out of phase with the vibration generator for the second vibrating element.

8. The compacting machine as defined in claim 7 wherein said vibratory drive means for each vibrating element further includes a rod secured at one end to a crank, and spring means interposed between the other end of said rod and said vibrating element.

9. The compacting machine as defined in claim 8 wherein said spring means for each vibrating element includes pneumatic spring means having a housing with an opening at one end, a piston rod extending through the opening and connected to the opposite end of the crank rod, at least one piston secured to the piston rod for movement within said housing, and valve means to adjust the pressure within said housing, and thus alter the energy transmitted to the vibrating element.

10. The compacting machine as defined in claim 9 wherein the pneumatic spring means is pressurized from a closed system including a reservoir, a hydraulic pump, a conduit extending between the reservoir and the pump, and said pump is connected via conduits to said housing to pressurize same.

11. The compacting machine as defined in claim 10 wherein said pneumatic spring means further includes a partition wall dividing the housing into an upper and lower chamber, an aperture in said partition wall to allow said piston rod to pass therethrough, a first and a second piston secured to the piston rod, one piston movable within the upper chamber and the second piston movable within the lower chamber.

12. The compacting machine as defined in claim 7 wherein said vibratory drive means for each vibrating element further includes a rod secured at one end to a crank, a pair of spaced, vertically extending brackets secured to said element, a cylindrical member disposed between said brackets and secured thereto by shafts so that said member can pivot relative to said brackets, the opposite end of said rod being connected to said cylindrical member, and spring means axially disposed within said cylinder for transmitting vibratory energy from the driven shaft to the vibrating element while preventing kinking of said spring means.

13. The compacting machine as defined in claim 12 wherein a lateral spacer divides the cylinder into an upper and a lower chamber, and said spring means comprises a first spring located within the upper chamber, and a second spring located within the lower chamber, one end of each spring being seated on opposite faces of said spacer.

14. The compacting machine as defined in claim 12 wherein a compensating weight is positioned in operative relationship to said cylinder to dampen out undesirable vibrations caused by the compacting action of the compacting element.

15. The compacting machine as defined in claim 7 wherein the vibratory drive means for each vibrating element further includes a rod connected at one end to a crank, a beam spaced parallel to the vibrating element and connected to the opposite end of the crank rod, and spring means interposed between the beam and the compacting element for transmitting vibratory energy thereto.

16. The compacting machine as defined in claim 15 wherein said spring means includes springs connecting each end of the beam to the compacting element, and a spring connecting the midportion of the beam to the compacting element.

17. The compacting machine as defined in claim 15 wherein the vibratory drive means further includes a guide for assisting the vertical movement of the beam.

18. The compacting machine as defined in claim 15 wherein the vibratory drive means further includes at least a pair of elongated rods connected at their lower ends to each vibrating element, said rods having enlarged stops formed at their upper end; said spring means encircling said rods and extending therealong between said stops and said vibrating element.

19. The compacting machine as defined in claim 7 wherein each vibrating element has an upwardly opening U-shaped chamber, and the drive means for each vibrating element further includes a rod connected to one end of a crank, a beam secured to the vibrating element across the open end of the chamber, the beam having an aperture formed therein that communicates with the chamber, a piston rod passing through the aperture being secured to the opposite end of the rod connected to said crank, a piston secured to said piston rod for movement within said chamber, and spring means interposed between the piston and the beam for transmitting vibratory energy to the vibrating element.

20. The compacting machine as defined in claim 7 wherein the vibratory drive means for each vibrating element includes spring means interposed between an upstanding laterally spaced plate and a depending control rod of the support frame.

21. The compacting machine as defined in claim 7 wherein the vibratory drive means for each compacting element further includes a rod connected at one end to a crank, brackets connected to each compacting element, a hollow cylindrical member mounted for pivotal movement with respect to said brackets, end caps for sealing opposite ends of said cylindrical member, one of said caps having an opening extending axially therethrough, a spacer movable within said cylindrical member and dividing the member into an upper and lower chamber, spring means disposed in said upper and lower chambers and urging said spacer in opposite directions, said rod extending downwardly through the

opening in the end cap of said cylindrical member and being secured at its lower end to said spacer so that the spacer can be moved within said housing in opposite directions for transmitting vibratory energy from said crank to said compacting element.

22. The compacting machine as defined in claim **21** wherein the vibratory drive means further includes eccentric discs and compensating weights for dampening undesirable vibrations caused by the compacting action of the compacting elements.

23. The compacting machine as defined in claim **1** wherein each vibrating element comprises a metal shoe comprising a plurality of lateral spacers, longitudinal spacers, and edge members, the dimensions of the vibrating element being determined by the number of lateral and longitudinal spacers utilized between the edge members, said spacers and edge members being joined together so that they form a working surface for the vibrating element.

24. The compacting machine as defined in claim **6** wherein a horizontal line drawn between the points at which said control arms are connected to said side supports will be vertically spaced from the midpoint of the thickened center portion of each control arm in the equilibrium position for said parallelogram frame.

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