

July 25, 1961

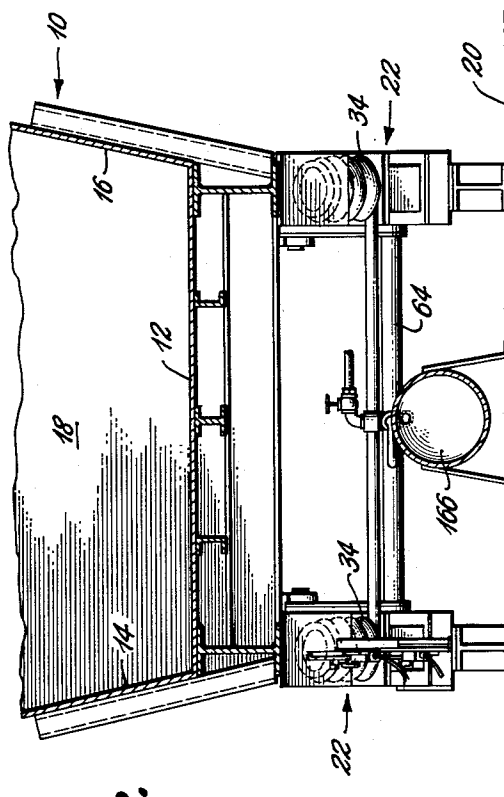
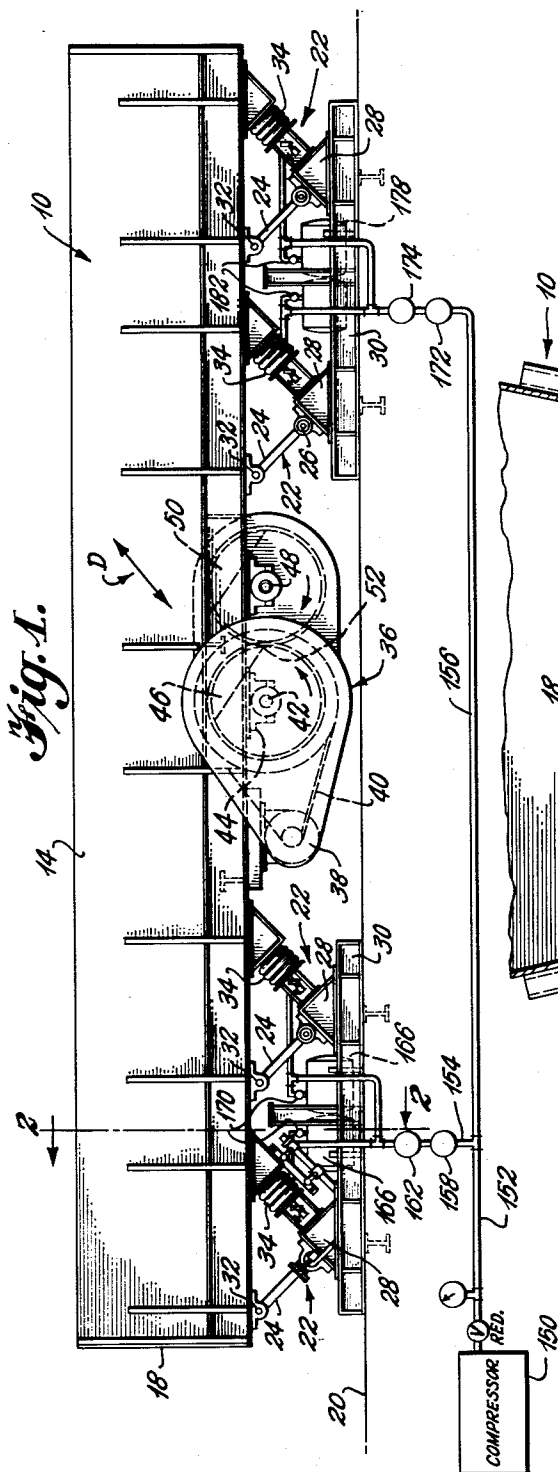
A. MUSSCHOOT

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MATERIALS HANDLING VIBRATORY APPARATUS

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MATERIALS HANDLING VIBRATORY APPARATUS

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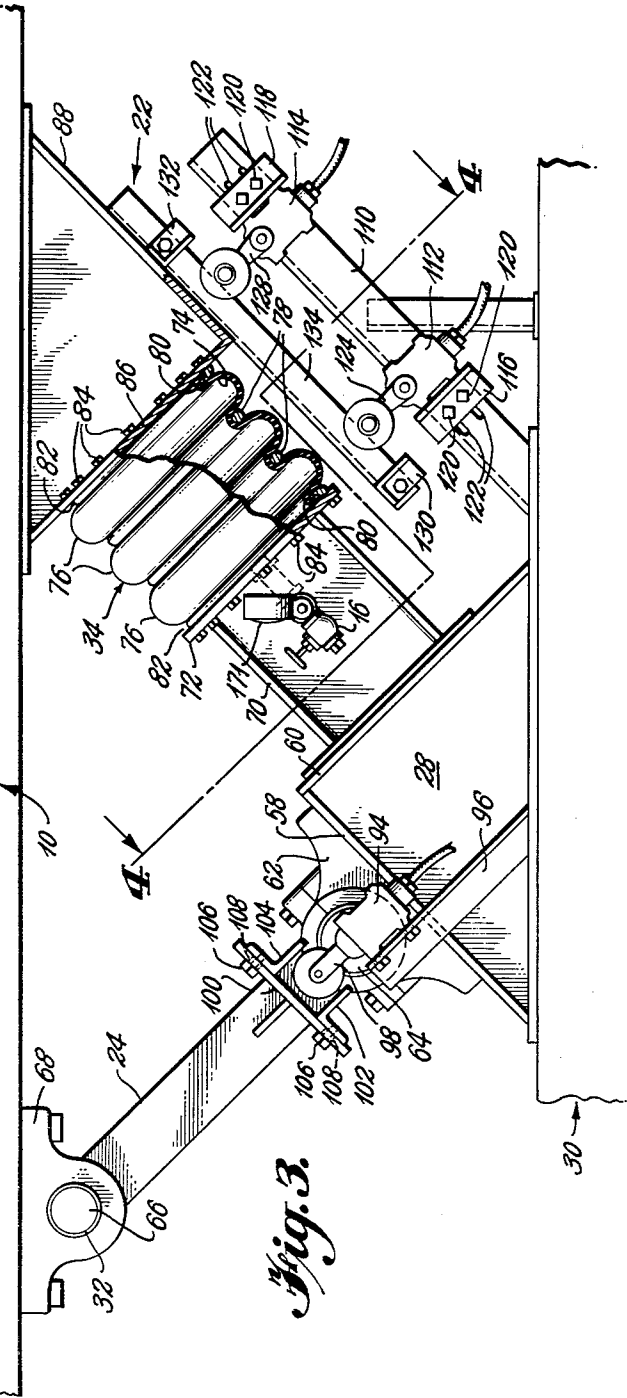


Fig. 3.

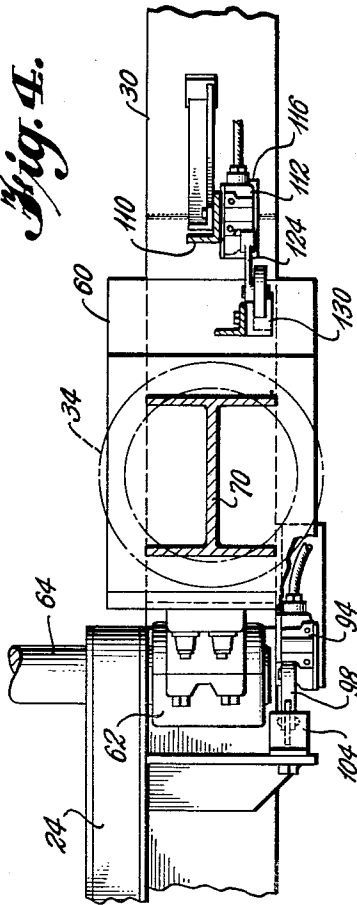


Fig. 4.

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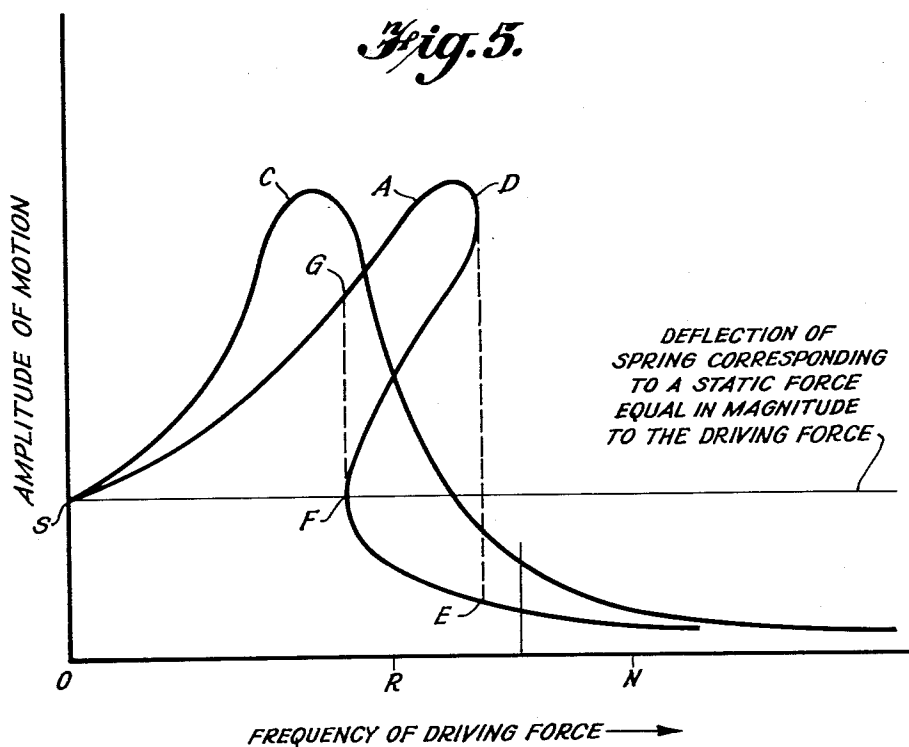
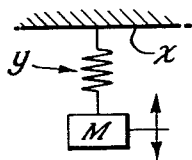


Fig. 5A.



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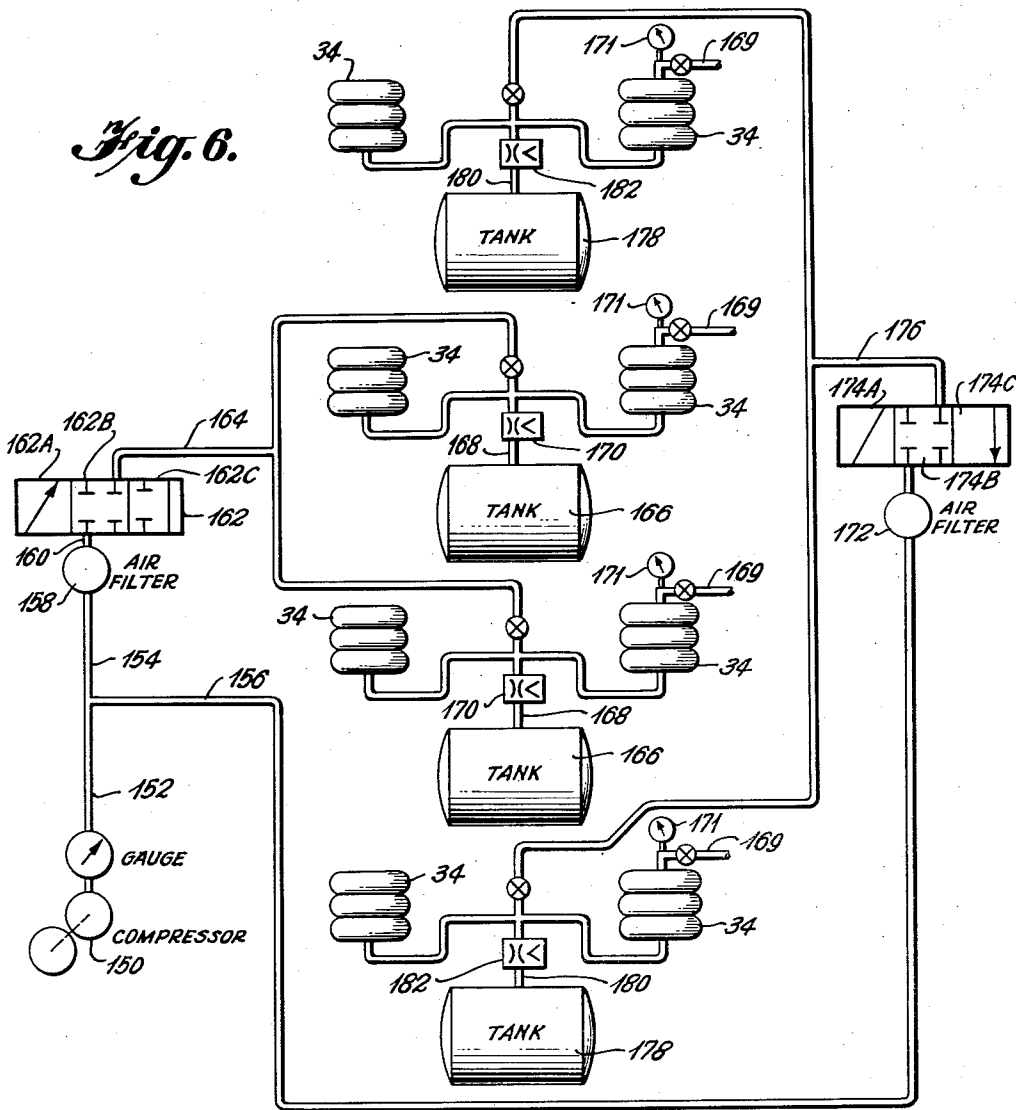
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Fig. 6.

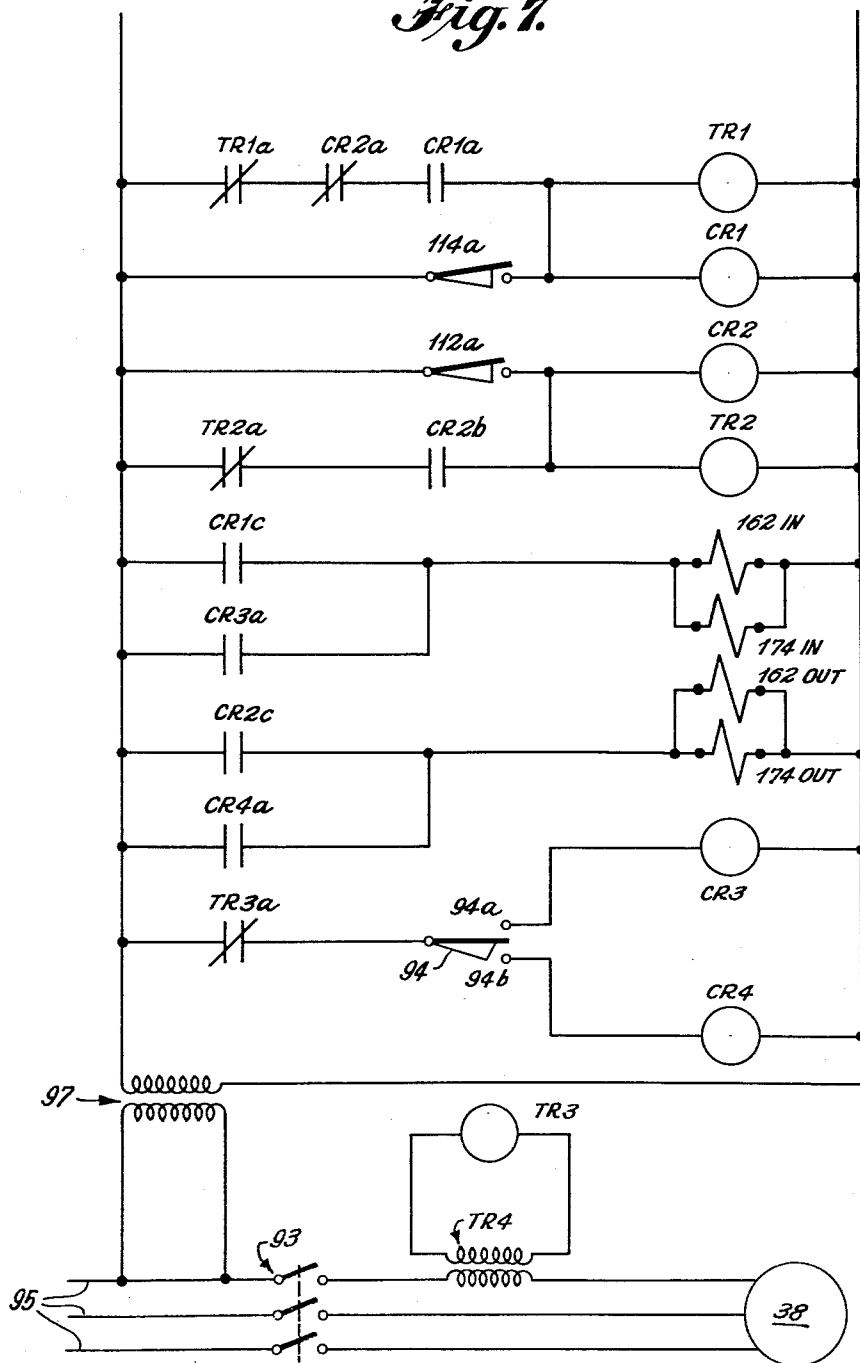


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Fig. 7.



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MATERIALS HANDLING VIBRATORY APPARATUS

Albert Musschoot, Anchorage, Ky., assignor to Link-Belt Company, a corporation of Illinois
Filed Feb. 9, 1959, Ser. No. 792,133
10 Claims. (Cl. 198—220)

This invention relates to materials handling vibratory apparatus, and more particularly to resilient support or reactor assemblies for the vibrating members of such apparatus.

The invention is of general application to materials handling apparatus wherein a materials supporting member, such as a trough, deck, hopper or table is mounted upon a fixed base for vibratory movements relative to the latter to feed, screen, discharge, stratify, etc., materials contained within or supported by said member. In conventional apparatus, the materials supporting member is provided with mechanism which applies a vibratory driving force to the member and the support means includes one or more resilient elements, usually in the form of mechanical coil or leaf springs, which resiliently oppose or cushion the vibratory movements of the materials supporting member.

The theory of the mechanical coil or leaf spring is well known, and it is quite easy to design one which has almost any desired response characteristic. However, mechanical springs are limited in that a given spring has a fixed response characteristic which cannot be changed without altering its properties or structure.

The primary object of the invention is to provide a materials handling vibratory apparatus having resilient support or reactor assemblies which are self-adjusted, during use, to compensate for wide variations in the mass of material handled by the apparatus.

Another object of the invention is to provide a materials handling vibratory apparatus with resilient support or reactor assemblies having response characteristics which may automatically vary while the apparatus is in use to maintain the vibratory movements within a selected region relative to fixed parts of the apparatus.

Still another object of the invention is to provide a materials handling vibratory apparatus including a spring system having a natural or resonant frequency which is substantially constant irrespective of the mass supported by the spring system.

Other objects and advantages of the invention will become apparent by reference to the following specification taken in conjunction with the accompanying drawings.

In the drawings:

FIGURE 1 is a side elevational view of a vibratory trough feeder embodying the invention;

FIGURE 2 is a transverse sectional view of the feeder taken on the line 2—2 of FIG. 1;

FIGURE 3 is an enlarged detail view, partially broken away, of one of the reactor assemblies of FIG. 1;

FIGURE 4 is a sectional view of the reactor assembly taken on line 4—4 of FIG. 3;

FIGURE 5 is a graph illustrating the differences between the frequency response curves of a mechanical spring mass system and an air spring mass system;

FIGURE 5A is a schematic view illustrating a simple mechanical spring mass system;

FIGURE 6 is a schematic diagram of a pneumatic system employed with the structure of FIGS. 1 through 4; and

FIGURE 7 is a schematic diagram of an electrical system employed to control the pneumatic system of FIG. 6.

In FIGS. 1 and 2, the invention is shown as being applied to a vibratory feeder assembly. In this example, the vibratory feeder includes an elongated deck or trough

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designated generally 10 having a bottom 12, side walls 14 and 16, and a rear end wall 18. The deck or trough 10 is open at its top to receive material and is open at its right-hand end, see FIG. 1, from which material is discharged from the interior of the trough. The material to be handled will, in general, be dumped into the trough 10 adjacent the left-hand end from any suitable source such as a chute, hopper or the like (not shown). In the usual case, some suitable means (not shown) will be provided to receive material discharged from the right-hand end of the hopper to convey the material away.

The deck or trough 10 is mounted upon a fixed supporting surface, or floor, 20 by a plurality of symmetrically located reactor assemblies 22. Each reactor assembly is identical and includes an inclined rigid link 24 which is pivotally coupled at its lower end 26 to a bracket 28 which is fixedly secured to the surface 20 by a suitably rigid beam 30. The upper end of the link 24 is coupled at 32 to the deck or trough 10. In FIG. 1, the vibrating feeder is shown in what may be described as its normal rest position with all of the links 24 disposed in parallel relationship to each other at a position inclined in the same direction approximately 45° to the horizontal. The links 24 constrain movements of the deck or trough 10 to a fixed path relative to the surface 20.

The deck or trough 10 is supported at the rest position of FIG. 1 by the four pairs of air springs 34 which are mounted in compression between each one of the brackets 28 and the trough 10. The longitudinal axes of the springs 34 are all disposed in parallel relationship to each other at an angle approximately 45° from the horizontal and approximately perpendicular to the axes of the links 24 when the trough is at rest. By this arrangement, the springs 34 will be expanded when the trough or deck is moved to pivot the links 24 in a clockwise direction about their lower pivots 26 and compressed when the deck 10 is moved in the opposite direction.

The deck or trough 10 is driven in vibratory movements by a rotating counterweight assembly 36 which is secured to the lower surface of the deck 10 at its longitudinal center. The assembly 36 includes a motor 38 which is fixedly secured to the bottom of the deck 10 and is connected to drive, by means of a belt 40, a first shaft 42 which extends transversely beneath the deck and is journaled for rotation in bearings 44 mounted upon opposite sides of the deck. The shaft 42 projects beyond both sides of the deck. A counterweight 46 is fixed on each projecting end of the shaft 42 to rotate therewith. A second rotating shaft 48, having a counterweight 50 mounted on each end, is journaled beneath the trough 10 and is driven from the shaft 42 by meshing gears illustrated schematically at 52.

The shafts 42 and 48, because of the direct geared connection 52, rotate in opposite directions. The counterweights 46 and 50 are angularly disposed upon their respective shafts, as illustrated in FIG. 1, so that their maximum forces are imparted in the opposite directions, as indicated by the double arrow "D" in this same figure.

In FIGS. 3 and 4 are detailed views of one reactor assembly 22. The fixed parts of this assembly comprise one of the brackets 28 which includes a pair of supporting plates 58 and 60 which lie in planes substantially perpendicular to each other and are inclined at approximately 45° to the horizontal. The plates 58 and 60 are rigidly connected to each other at their upper edges and to the rigid beam 30 at their lower edges, as by welding. A pillow block 62 is mounted upon the plate 58 to rotatably support a shaft 64 which extends entirely across the transverse width of the feeder to a similar pillow block associated with another reactor assembly 22 located on the opposite side. The shaft 64 rotates about an axis which is parallel to the axes of the counterweight shafts

42 and 48. The link 24 is fixedly secured to the shaft 64 immediately inside of the pillow block 62. At its opposite end 32, the link 24 is connected to the lower surface of the deck or trough 10 by means of a stub shaft 66 which is journaled in a pillow block 68. Since the link 24 is rigid, the coupling described above constrains movement of the deck 10 to a curved path having a radius equal to the distance between the axes of the shaft 64 and the stub shaft 68. Since all of the links 24 of the reactor assemblies are of equal length and are disposed in parallel relationship, the motion of the deck or trough 10 is similar to that of a four-bar linkage having parallel rocker arms when the same are caused to oscillate in both directions.

A rigid pedestal 70 is welded to and projects perpendicularly from the plate 60 to support a circular plate 72 in spaced parallel relationship to the plate 60, the spacing between the plates 60 and 72 depending upon the physical dimensions of the air spring 34 to be employed.

The circular plate 72 serves as a seat for one end of the spring 34 which, as best seen in FIG. 3, takes the form of a bellowslike rubber member of hollow, circular cross-section which is open at each of its opposite ends. The rubber body 74 of the spring 34 is molded to a form similar to that shown in FIG. 3, and includes a selected number of corrugations 76 which are integral with each other and usually are provided with steel rings 78 which encircle the spring between adjacent corrugations. At each end, the body 74 is formed with an annular sealing bead 80 which is clamped to a flat surface by means of a ring 82 and a suitable number of bolts 84. The upper end of the spring 34 is seated upon a flat plate 86 mounted upon a support 88 fixed to the lower surface of the deck or trough 10 to dispose the plate 86 in parallel relationship to the plate 72.

The reactor assembly shown in FIGS. 3 and 4 is the one located at the extreme left of FIG. 1 and includes certain limit switches which are employed to control the characteristics of all of the air springs 34, used in the vibratory feeder apparatus, in a manner described more fully below.

A first limit switch 94 is mounted upon an angle member 96 which is welded to the bracket 28 adjacent the pillow block 62. This switch is provided with an actuating arm 98 which is free to pivot in either direction from the normal rest position shown in FIG. 3. The pivotal axis of the arm 98 is located so as to be coincident with the axis of the shaft 64. A bracket 100 is welded upon the coupling link 24 to support a pair of limit switch engaging lugs 102 and 104 which are located to engage the roller of the arm 98 when the coupling link 24 pivots a selected distance from the rest position shown in FIG. 3. Preferably, the lugs 102 and 104 are clamped to the bracket 100 by means of screws 106 which pass through elongated slots 108 to permit the lugs to be adjusted toward or away from each other to vary the amount of deflection of the link 24 which may occur before the arm 98 is engaged by either of the lugs 102 or 104.

Before discussing the pneumatic and electrical control systems which are provided for the air springs 34, it will be helpful to compare the characteristics of the air springs of the reactor assemblies of this invention with the corresponding characteristics of conventional metallic coil springs.

In making this comparison, the metallic coil spring system schematically illustrated in FIG. 5A will be used. This system includes a mass M suspended from a fixed support X by means of a coil spring Y. With such a system, a coil spring under static load conditions and within its elastic limit will always be deflected by an amount that is proportional to the force acting on the spring. This fixed stiffness that is characteristic of a coil spring is normally referred to as its "spring constant" and

is usually defined as the number of pounds of force necessary to deflect the spring one inch.

Considering now the dynamic response characteristics of coil springs and more specifically the response characteristics under harmonically applied load conditions as developed by the vibrator 36 of the feeder shown in this application, the curve C of FIG. 5 is a plot of the amplitude of motion of the mass M in the simple system of FIG. 5A for various frequencies of a harmonically applied driving force with damping. In considering the curve C it will be assumed that the driving force is such that, if statically applied, it would produce a deflection with an amplitude as indicated at S in FIG. 5. It will be further assumed that the force will be exerted in both vertical directions sinusoidally. As shown by the curve C, when the frequency of the driving force is extremely low, the deflection of the mass M from its normal rest position is essentially the same as if the force were statically applied. If, on the other hand, the frequency of the driving force is extremely high, the inertia of the mass M will prevent its responding quickly enough to follow the force and the amplitude of its motion becomes extremely small as indicated at the right-hand end of the curve C. At an intermediate point, however, the curve C rises sharply to a peak, indicating that the amplitude of motion of the vibrating mass M is greatly magnified for certain frequencies of application of the driving force. The frequency of the driving force which produces maximum amplification of movement of the mass M is known as the "resonant frequency" of the spring-mass system.

In the system of FIG. 5A where the spring element Y is a coil spring operating within its elastic limit, the resonant frequency may be easily calculated and is proportional to the $\sqrt{k/M}$ where k is the spring constant discussed above and M is the mass suspended by the spring. It will be readily apparent, therefore, that the resonant frequency of a spring-mass system employing coil springs is dependent on two factors, only one of which, the mass M supported by the spring, is capable of variation.

In comparison to the coil spring, the air springs 34 of the present application differ in several important respects. First, the pressure of the air within the springs will vary in accordance with the load supported thereby. The stiffness of the air springs 34, therefore, is dependent upon the pressure of the air contained within the spring which is in turn dependent upon the load supported by the spring and the higher load and pressure, the stiffer the spring will be.

Further, the deflection of an air spring does not bear a linear relationship to the mass supported by the spring, as was the case with the coil spring. The air spring 34 has a gradually increasing stiffness as the spring is compressed so that if one pound of force will compress an air spring a distance of one inch, a force of three pounds may compress the spring only two and a half inches.

The dynamic response characteristics of air springs also differ from those of coil springs. In FIG. 5, a frequency response curve for an air spring is indicated by the reference character A. Although the curve A indicates a peak or maximum amplitude, it is not possible to speak of a particular frequency as the resonant frequency of an air spring-mass system for the following reasons. If a harmonically applied driving force is applied to an air spring supported mass, the amplitude of the motion of the mass increases as the frequency of the driving force is increased until the point D is reached. A further increase in the frequency of the driving force causes the amplitude of motion of the mass to drop immediately from the magnitude at D to the magnitude indicated at point E. As the frequency of the driving force increases further, the amplitude of motion of the mass follows the curve A on toward the right-hand end.

If the frequency of the driving force decreases from an extremely high frequency, the amplitude of motion

of the air spring supported mass continues to follow the curve A and decreases smoothly along the curve until the point F is reached. At this time, a further decrease in frequency causes the amplitude to immediately increase to that indicated at the point G on curve A. Further decrease in the frequency of driving force causes the amplitude of the mass to follow the curve A back toward the origin.

It will be recalled that the natural frequency of a coil spring-mass system is proportional to $\sqrt{k/m}$ so that a system having a spring with a fixed constant k , has a natural frequency that varies in inverse relationship with the mass M . On the other hand, the frequency response characteristic of an air spring-mass system is substantially unchanged by variations in the mass. If the stiffness of the air spring is increased by an increase in the mass supported by the spring, the value of $\sqrt{k/m}$ remains approximately the same. The natural frequency, therefore, of an air spring-mass system will remain substantially constant despite variations in the mass. In other words, variations in the load handled by an air spring supported device will have a negligible effect on the natural frequency of the device.

The dynamic response characteristics of an air spring may be further varied by connecting the interior of the spring to an enlarged reservoir of fixed volume. Assuming that air may flow in a substantially unrestricted manner between the spring and the reservoir, the effect of the reservoir is to reduce the percentage change of total volume for a given deflection of the air spring. In other words, an increase in the volume of the air in communication with the spring will cause the shape of curve A of FIG. 5 to approach that of the curve C and the larger the reservoir, the more nearly the curve A resembles the curve C.

Both of the curves A and C shown in FIG. 5 represent the response characteristic of the respective springs with a reasonable amount of damping present. To provide a controlled damping characteristic for the air spring, a valve having an adjustable orifice may be connected to regulate the rate of flow of air between the spring and a reservoir. The net effect of increased damping on either of the curves of FIG. 5 is to render the respective peaks of the curves less prominent.

Adjacent the opposite end of the bracket 28, an angle member 110 is rigidly mounted, as by welding, to project upwardly from the base 30 in a direction parallel to the longitudinal axis of the spring 34. A pair of limit switches 112 and 114 are mounted upon the angle member 110 by means of brackets 116 and 118 respectively, which are clamped by bolts 120 which pass through elongated slots 122 to permit adjustment of the position of the limit switches 112 and 114 longitudinally of the member 110. The arms 124 and 128 of the limit switches project into the path of lugs 130 and 132 which are mounted upon an angle member 134 rigidly fixed to the support 88. It is believed apparent that axial deflection of the spring 34 by a sufficient amount in either direction will cause one of the lugs 130 or 132 to engage and actuate the adjacent limit switch arm 124 or 128. The limit switches 112 and 114 are employed in the electrical control system to be described below to adjust the springs 34 to maintain the vibratory motions of the deck 10 within a predetermined portion of its path of movement.

The pneumatic system by means of which flow of air to and from the various air springs 34 may be regulated is shown in FIG. 6. The system includes a motor driven compressor 150 which is connected to supply high pressure air to an outlet conduit 152. In the particular feeder illustrated eight air springs 34 are employed and, for convenience, the high pressure conduit 152 is split into two branches 154 and 156. The conduit 154 is connected to an air filter 158 whose outlet 160 is con-

nected to the inlet of a three-position valve 162 which is provided with three different flow paths 162A, 162B and 162C. As shown in FIG. 6, the flow path 162B is conditioned to block the flow of air from the inlet conduit 160 to the outlet conduit 164. This illustration shows the valve 162 in its neutral position which is its normal condition.

The valve 162 is preferably provided with a pair of solenoids, which will be described in connection with the wiring diagram of FIG. 7, arranged in such a manner that energization of one of the two solenoids will actuate the valve to condition the flow path 162A between the inlet 160 and the outlet 164 so as to conduct high pressure air to such outlet. Energization of the other of the two solenoids places the flow path 162C in condition to connect the inlet 160 with the outlet 164. When the valve is in this condition, the inlet conduit 160 is blocked while the outlet conduit 164 is vented to the atmosphere to exhaust air from the latter. In the event neither of the solenoids is actuated, the valve assumes its neutral position, as shown in FIG. 6. The structural details of the valve form no part of the present invention. For details of a suitable valve, reference is made to the valves manufactured by the Valvair Corporation of Akron, Ohio, as series D-3000, illustrated in Bulletin SK356.

The outlet conduit 164 of the valve 162 is connected, by various branches, to four of the air springs 34. From the description above, it is believed apparent that the flow of air to and from the various springs 34 connected to the conduit 164 may be regulated by suitable actuation of the valve 162. By regulating the flow of air to and from the springs 34 the height of the springs may be maintained substantially constant while the static and dynamic response characteristics of the air springs will automatically vary to compensate for variations in the loads supported thereby. In general it may be said that the greater the air pressure within the springs, the stiffer they become.

Each pair of springs 34 is connected to a reservoir or tank 166 by means of a conduit 168 which includes a controllable orifice valve 170 for regulating the rate of flow of air through the conduit 168. The tank 166, when connected directly to the springs 34, influences the dynamic response characteristics of the springs in a manner depending upon the relative volumes of the spring and tank 166. The valve 170, by controlling the rate of flow of air between a pair of springs 34 and their tank 166, provides a variable damping characteristic for the springs as discussed above.

The pneumatic connections and components included in the branch line 156 are exactly similar to those described above in connection with the branch line 154 and hence will not be described in detail. Briefly, the conduit 156 is connected to an air filter 172 and a three-position valve 174 of a construction identical to that of the valve 162. The outlet conduit 176 of the valve 174 is connected to the remaining four air springs 34, each pair of which is connected to a reservoir or tank 178 by means of a conduit 180 and a controllable orifice valve 182, the construction and functions of which are identical to the corresponding tanks, valves and conduits connected in the outlet line 164 associated with the branch conduit 154.

As shown in FIG. 6, one air spring in each pair is provided with a valve controlled bleed-off 169 and a pressure gauge 171. These same elements are illustrated in FIG. 3.

The electrical circuit for controlling the pneumatic system of FIG. 6 is disclosed in FIG. 7 in which the contacts of the limit switches 112 and 114 of FIG. 3 are shown at 112a and 114a, respectively. The limit switch 94 of FIG. 3 has, as schematically illustrated in FIG. 7, two contacts 94a and 94b. The purpose of the circuit of FIG. 7 is to control the two operating solenoids for each of the valves 162 and 174 of FIG. 6. Therefore, the

control circuit receives its current supply from one of the main lines 95 through the transformer 97 which is located in advance of the main switch 93.

One solenoid with each valve is employed to shift the valve from its neutral position to place the flow paths 162A and 174A, respectively, in condition to supply air under pressure from the compressor 150 to the respective air springs 34 connected to the valves. The solenoids for accomplishing these movements of the valves are designated 162-in and 174-in, respectively. The reference numerals 162-out and 174-out, respectively, designate the solenoids employed on the associated valves for moving the latter from their neutral positions to place the flow paths 162C and 174C, respectively, in their operative conditions to vent the connected air springs to the atmosphere to thereby lower the height of the springs.

The function of the limit switch 94 is to maintain the normal rest position of the deck 10 at a predetermined elevation above the fixed base. Since the vibratory movements of the deck 10, when it is driven by the vibrating mechanism 36, oscillates the deck on either side of its rest position, the contacts 94a and 94b of the limit switch 94 are connected to the energizing circuit of the motor 38 so that actuation of the contacts of the limit switch 94 during vibratory movement of the deck 10 do not affect the operation of the valves 162 and 174. This is accomplished by connecting a time-delay relay TR3 through the transformer TR4 to one of the lines of the energizing circuit of the motor 38 in such a manner that the relay TR3 is active at all times when the motor 38 is energized. The relay TR3 controls a set of normally closed contacts TR3a which are opened immediately upon the energization of the relay TR3 and remain open for a predetermined time interval after the relay TR3 is deenergized by opening the main switch 93 of the energizing circuit to the motor 38.

Assuming that the motor 38 is not energized, the limit switch 94 functions as follows: When the deck 10 is stationary, the addition of a mass of material to the same will cause the springs 34 to be compressed, thereby causing the deck 10 to settle toward the fixed base 20, and the coupling link 24 of FIG. 3 to pivot in a counterclockwise direction. If the weight of the mass of material is great enough, the link continues to pivot in the same direction until the lug 104 mounted on the link 24 contacts and pivots the arm 98 of limit switch 94. As the arm 98 is pivoted, the movable element of the limit switch 94 is moved into engagement with the contact 94a.

Since contacts TR3a are closed at this time, a circuit is completed through a control relay CR3 which, when energized, closes its associated contacts CR3a. These contacts CR3a, when closed, energize the valve controlling solenoids 162-in and 174-in which actuate their respective valves to place the flow paths 162A and 174A in condition to conduct high pressure air from the compressor 150 to the springs 34 connected to the respective valves. Air under pressure is thus forced into all of the springs 34 to increase the height of the springs. As the spring height increases, the deck 10 is moved back toward its normal rest position. Air under pressure is continued to be forced into the springs until the coupling link 24 of FIG. 3 is rotated in a clockwise direction by a sufficient amount so that the lug 104 permits the arm 98 of the limit switch 94 to move far enough to open the circuit through the contact 94a.

Should the mass of material within the stationary deck 10 be decreased for any reason, the deck will rise under the influence of the springs 34, thus pivoting the coupling links 24 in a clockwise direction. This action causes the lug 102 on the coupling link, shown in FIG. 3 to engage the arm 98 of the limit switch 94 and, when the striker 98 has been deflected a sufficient amount a circuit is completed through the contacts 94b to energize the control relay CR4, which when energized, closes the contacts CR4a to energize the valve actuating solenoids 162-out.

and 174-out. This actuation of the valve control solenoids conditions the flow paths 162C and 174C to vent the connected air springs 34 to the atmosphere, thus lowering the height of the springs and permitting the deck 10 to settle back toward its normal position.

It will be noted that when the vibrating motor 38 is energized, the contacts TR3a are open, hence closing of either of the contacts 94a or 94b cannot energize either of the control relays CR3 or CR4. By virtue of the delayed closing characteristics of the contacts TR3a, the circuit containing contacts 94a and 94b cannot be energized until after the deck 10 has stopped its vibratory movements.

The purpose of limit switches 112 and 114 is to control the total stroke or deflection of the springs 34 to maintain this deflection within a selected region relative to the fixed base. The limit switches 114 and 112 thus define a maximum amplitude over which the springs may not be deflected and further limits the displacement of the mid-point of the amplitude. Assuming that the vibrating mechanism 36 is in operation, the deck 10 is vibrated along a path which is determined by the coupling links 24.

Assuming that a mass of material has been deposited on the deck 10 and the motor 38 has been energized to discharge material from the deck 10, the total mass supported upon the springs 34 decreases as the material passes from the feeder. The decrease in mass permits the springs 34 to expand and thus elevate the average level of the deck 10 during vibratory movement. This average level continues to rise as the material is emptied until eventually, during one cycle of vibration, the deck 10 rises to a level at which the lug 130 actuates the arm 124 of the limit switch 112.

The momentary closing of the contacts 112a immediately energizes two relays CR2 and TR2 connected in parallel. The relay TR2 has contacts TR2a which are closed when the relay is not energized and which remain closed for a predetermined time interval after the relay TR2 is energized. Thus, the momentary energization of the relay TR2 by the momentary closing of the contacts 112a initiates a timed delay period during which the relay TR2 holds closed the contacts TR2a to complete an alternate or lock-in circuit through the relays CR2 and TR2 through the contacts TR2a and normally open contacts CR2b, the latter of which are closed by the energization of the relay CR2. The effect of the alternate or lock-in circuit just described is to maintain the relay CR2 energized for a predetermined time interval after its initial energization, the time interval corresponding to the delayed time opening characteristics of the relay TR2. When the relay CR2 is energized, it closes a second set of contacts CR2c which energize the valve control solenoids 162-out and 174-out which in turn actuate their respective valves to move such valves to the position for venting air from the air springs connected to the respective valves.

The valves 162 and 174 remains in their venting positions as long as the solenoids 162-out and 174-out are energized, this time interval again corresponding to the delayed time opening characteristic of the timing relay TR2. After the time interval has elapsed, the contacts TR2a open, thus opening the circuit through control relay CR2 which in turn opens the contacts CR2c to de-energize the valve operating solenoids 162-out and 174-out. The valves 162 and 174 return immediately to their neutral or blocking position when the control solenoids are deenergized.

In the case where, with the vibrating mechanism 36 in operation, an additional mass of material is delivered to the deck 10, the average level of the deck 10 is lower during its vibratory movements. If the average level is lowered beyond a certain point, the lug 132 will engage and actuate the arm 128 of the limit switch 114. Actuation of the arm 128 closes the contacts 114a to mo-

mentarily energize the control relay CR1 and the timing relay TR1. The timing relay TR1 is of precisely similar construction to the timing relay TR2, and as in the previous case, the relays TR1 and CR1 are locked in through the contacts TR2a and contacts CR1a which are energized when relay CR1 is energized.

Energization of the relay CR1 further closes the contacts CR1c to thereby energize the valve operating solenoids 162-in and 174-in which move the respective valves to their opposite positions whereby air under pressure is supplied from the compressor 150 to the various air springs 34. The valves 162 and 174 remain in this position for a time interval corresponding to the delayed opening characteristics of the contacts TR1a. When the contacts TR1a open, the control relay CR1 is deenergized to open the contact CR1c, thereby deenergizing the valve control solenoids 162-in and 174-in to permit the respective valves to return to their blocking positions.

It will be noted that the energizing circuit for the relays TR1 and CR1 is exactly similar to the energizing circuit for the relays TR2 and CR2 with the exception of the inclusion of the normally closed contacts CR2a in the energizing circuit for the relays TR1 and CR1. The purpose of the normally closed contacts CR2a is to eliminate the possibility of having the valve control solenoids 162-in and 162-out and 174-in and 174-out energized simultaneously. This could occur during starting or stopping when the feeder is vibrating at a frequency which is close to the natural frequency of the air spring system.

In the case where the feeder is operated at a frequency well above its natural frequency, each time the feeder is started the motor 38 must come up to its driving speed from a dead stop. Thus, each time the motor is started the frequency of vibrations applied to the feeder builds up from zero to the operating frequency and must, of necessity, at some time pass through the natural frequency of the spring system. As the natural frequency of the spring system is approached, the amplitude of the feeder builds up and, for the sake of discussion, will be assumed to build up to a point where the amplitude of the stroke of the feeder is greater than the distance between the actuated positions of limit switches 112 and 114. As the stroke builds up, one of the limit switches 112 and 114 will necessarily be actuated before the other.

Assuming that the limit switch 114 is actuated at one end of the stroke and the limit switch 112 is actuated at the opposite end of the same stroke, actuation of the limit switch 112 energizes the relays CR2 and TR2 which would condition the control system to vent air from the air springs. Energization of the relay CR2 immediately opens the contacts CR2a, thus immediately deenergizing the relay CR1 and hence, by means of the contacts CR1c deenergizing the valve operating solenoids 162-in and 174-in to permit the valves to move to their venting positions immediately since energization of the relay CR2 simultaneously closes the contacts CR2c to energize the solenoids 162-out and 174-out. If, on the subsequent stroke of the feeder the limit switch 114 is again actuated, no reversal of the valve can occur since energization of the relay CR1 by the closing of the contacts 114a occurs only during the momentary interval during which the contacts 114a are held closed. As soon as the contacts 114a are opened as the feeder moves away from this limit of its stroke, the relay CR1 is deenergized since the contacts CR2a are open by virtue of the energization of the relay CR2.

Returning now to a further consideration of the dynamic response curve A of FIG. 5, an examination of the curve will reveal that the feeder of this invention may operate in either of two ways. From an examination of the curve A it is seen that if the frequency of the driving force is selected to be some frequency less than that corresponding to the point R on the ordinate, the amplitude of movement of the feeder will be magni-

fied. If the frequency of the driving force is greater than the frequency corresponding to the point N on the ordinate of the curve, the amplitude of movement of the feeder is substantially reduced. Feeders of the type disclosed are generally operated in this latter manner—that is, the frequency of the driving force is selected to be such as to be greater than a frequency corresponding to the ordinate N. The control system described above is especially adapted for this type of operation.

In spite of the greatly reduced amplitude of motion inherent in this last mentioned type, its operation at a relatively high driving frequency affords two important advantages. First, the air spring system acts as a vibration isolator to prevent the transmission of vibration to surrounding structures. Second, because of the relatively flat configuration of the curve in the higher frequency range, slight changes in the driving frequency or characteristic of the air spring system do not affect the amplitude of motion to any substantial degree, thus achieving greater uniformity of feed.

However, in certain instances it may be desirable to operate a device at driving frequencies less than the frequency R on the FIG. 5 plot. The primary advantage achieved in this type of operation is that a much smaller driving force is required to produce a given amplitude. However, the increase in amplitude is due to the fact that the air spring force is used more effectively upon the vibrating element and this additional force, being the result of the reaction between the air spring and the foundation, requires a much more rigid and massive foundation. It is believed apparent that the control system shown could easily be modified to adapt the system for this latter type operation.

Therefore, while only one structural embodiment of the invention has been described, it will be apparent to those skilled in the art that the described embodiment may be modified. Therefore, the foregoing description is to be considered exemplary rather than limiting and the true scope of the invention is that defined in the following claims.

Having thus described the invention, I claim:

1. A vibratory feeder or the like comprising a base, a deck for supporting and feeding materials, means supporting said deck upon said base for translational movement including coupling means between said deck and said base for constraining movement of said deck to a predetermined path relative to said base, and spring means mounted in compression between said deck and said base to resiliently support said deck at a rest position located centrally of said path and at a selected elevation above said base, vibrating means operable to impart an alternating driving force to said deck which exists independently of the deck movement to vibrate said deck upon said support means, and means for adjusting said spring means while said vibrating means is in operation to confine the vibratory movement of said deck to a selected portion of said path and to maintain the midpoint of said selected portion substantially at said selected elevation irrespective of variations in the weight of the material on said deck.

2. A vibratory feeder or the like comprising a base, a deck for supporting and feeding materials, means for supporting said deck upon said base for translational movement including coupling means between said deck and said base for constraining movement of said deck to a predetermined path relative to said base, and spring means mounted in compression between said deck and said base to resiliently support said deck at a rest position located centrally of said path and at a selected elevation above said base, vibrating means operable to impart an alternating driving force to said deck which exists independently of the deck movement to vibrate said deck upon said support means, and means responsive to the mass of material supported by said deck for adjusting said spring means to maintain said rest posi-

tion of said deck at a selected central location on said path.

3. A vibratory feeder or the like comprising a base, a deck for supporting and feeding materials, means for supporting said deck upon said base including coupling means between said deck and said base for constraining movement of said deck to a predetermined path relative to said base, and spring means mounted in compression between said deck and said base to resiliently support said deck at a rest position located centrally of said path, means responsive to the mass of materials supported by said deck for adjusting said spring means to maintain said rest position of said deck at a selected central location on said path, vibrating means operable to vibrate said deck upon said support means, means for rendering said responsive means inoperable when said vibrating means is in operation, and means for adjusting said spring means while said vibrating means is in operation to confine the vibratory movement of said deck to a selected portion of said path.

4. A vibratory feeder or the like comprising a base, a deck for supporting and feeding materials, means for supporting said deck on said base including coupling means between said deck and said base for constraining movement of said deck to a predetermined path relative to said base, and spring means mounted in compression between said deck and said base to resiliently support said deck at a rest position located centrally of said path, vibrating means operable to vibrate said deck on said support means, means for adjusting said spring means while said vibrating means is in operation to confine the vibratory movement of said deck to a selected portion of said path, and damping means coupled to said spring means for varying the response characteristic of said spring means independently of said adjusting means.

5. In a vibratory feeder or the like having a base, a deck for supporting and feeding materials, a plurality of symmetrically located reactor assemblies supporting said deck upon said base for translational vibratory movement relative thereto, and vibrating means operable to impart an alternating driving force to said deck which exists independently of the deck movement to vibrate said deck upon said reactor assemblies; the improvement wherein each of said reactor assemblies comprises a coupling assembly connected between said base and said deck to constrain vibratory movement of said deck to a predetermined path relative to said base, an air spring mounted in compression between said deck and said base to be extended and compressed by changes in the weight of the material supported on said deck and by vibratory movement of said deck along said path, and means operable during operation of said vibrating means to selectively admit air to and release air from said spring to maintain the vibratory movement of said deck within a predetermined portion of said path irrespective of variations in the weight of the material supported on said deck.

6. In a vibratory feeder or the like comprising a base, a deck for supporting and feeding materials, a plurality of symmetrically located reactor assemblies supporting said deck upon said base for translational vibratory movement relative thereto, and vibrating means operable to impart an alternating driving force to said deck which exists independently of the deck movement to vibrate said deck upon said reactor assemblies; the improvement wherein each of said reactor assemblies comprises a coupling assembly connected between said base and said

deck to constrain vibratory movement of said deck to a predetermined path relative to said base, an air spring mounted in compression between said deck and said base to be extended and compressed by changes in the weight of the material supported on said deck and by vibratory movement of said deck along said path, means connected to said air spring for selectively admitting air to and releasing air from said spring to compensate for changes in the weight of the material on said deck, a reservoir, and means for connecting said reservoir to said air spring to vary the response characteristic of said spring independently of said means for selectively admitting air to and releasing air from said spring.

7. In a vibratory feeder or the like as defined in claim 6, a valve mounted in said means connecting said reservoir to said spring to adjustably control the rate of flow of air between said spring and said reservoir to thereby provide said spring with a variable damping characteristic.

8. In a vibratory feeder or the like comprising a base, a deck for supporting and feeding materials, a plurality of symmetrically located reactor assemblies supporting said deck upon said base for constrained vibratory movement relative thereto, and vibrating means for vibrating said deck upon said reactor assemblies; the improvement wherein each of said reactor assemblies comprises a coupling member connected between said base and said deck to constrain movement of said deck to a predetermined path relative to said base, an air spring mounted in compression between said deck and said base to be extended and compressed by vibratory movement of said deck along said path, a source of air under pressure, conduit means for connecting said source to all of said air springs, valve means connected in said conduit means operable in a first position to prevent the flow of air to or from said springs, said valve means being selectively movable from said first position to a second position or to a third position and operable in said second position to connect said springs to said source to admit air to said springs and operable in said third position to vent air from said springs, first control means responsive to movement of said deck beyond a lower limit on said path for moving said valve means to said second position, and second control means responsive to the movement of said deck beyond an upper limit of said path for moving said valve means to said third position.

9. The combination defined in claim 8 wherein each of said first and said second control means includes means for maintaining said valve means in the position to which it is moved by the respective control means for a predetermined time interval upon each actuation of the control means.

10. The combination defined in claim 8 wherein said lower and said upper limits are spaced apart upon said path by a distance greater than the normal range of vibratory movement of said feeder along said path, said second control means including means for rendering said first control means ineffective while said second control means is energized.

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