Nov. 4, 1969 K. M. ALLEN ETAL 3,476,234

SELF-BALANCING VIBRATING STRUCTURES

Filed March 18, 1968 3 Sheets-Sheet 1

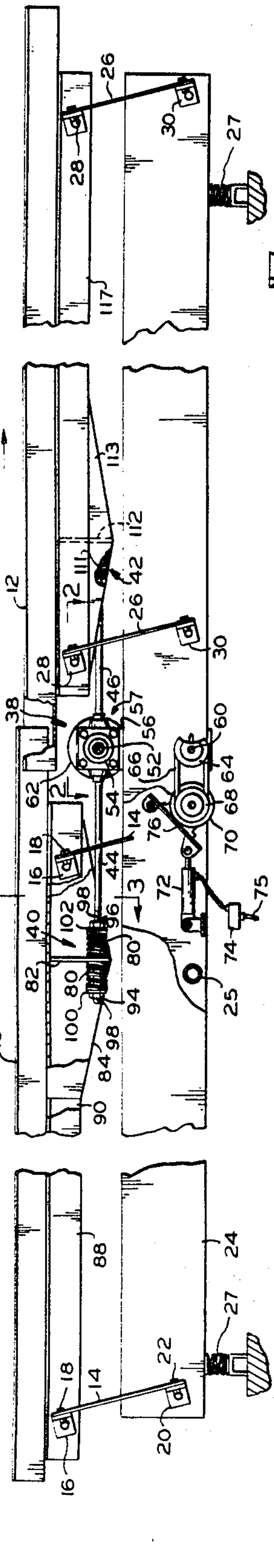

FIG. 1

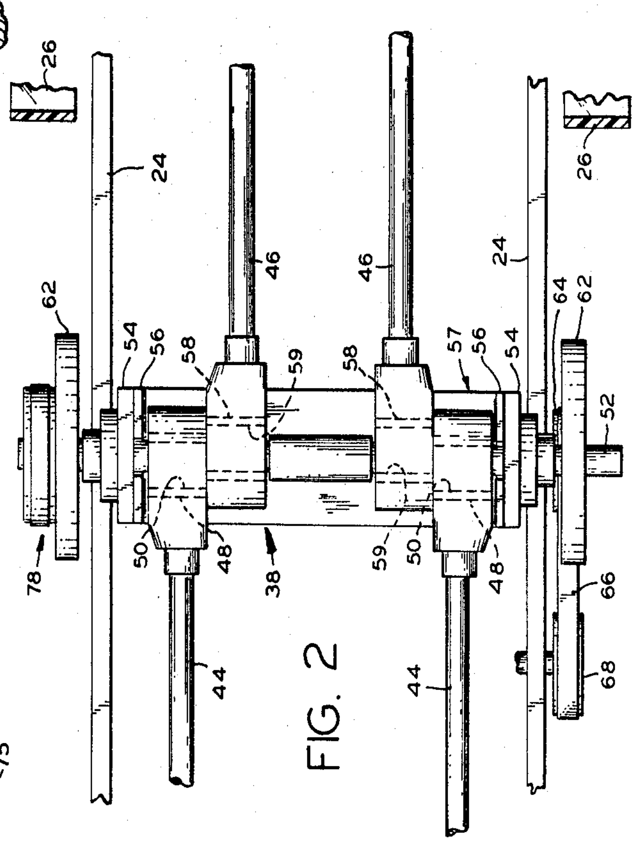

FIG. 2

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FIG. 3

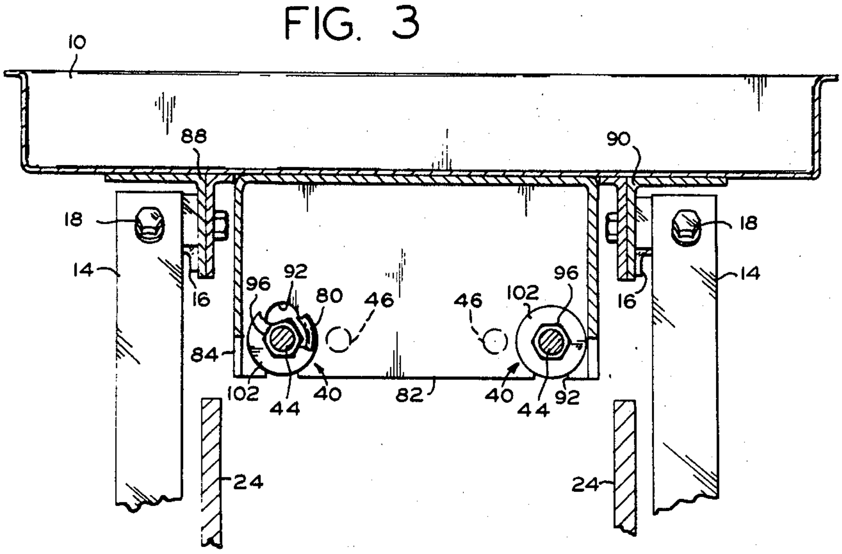

FIG. 4

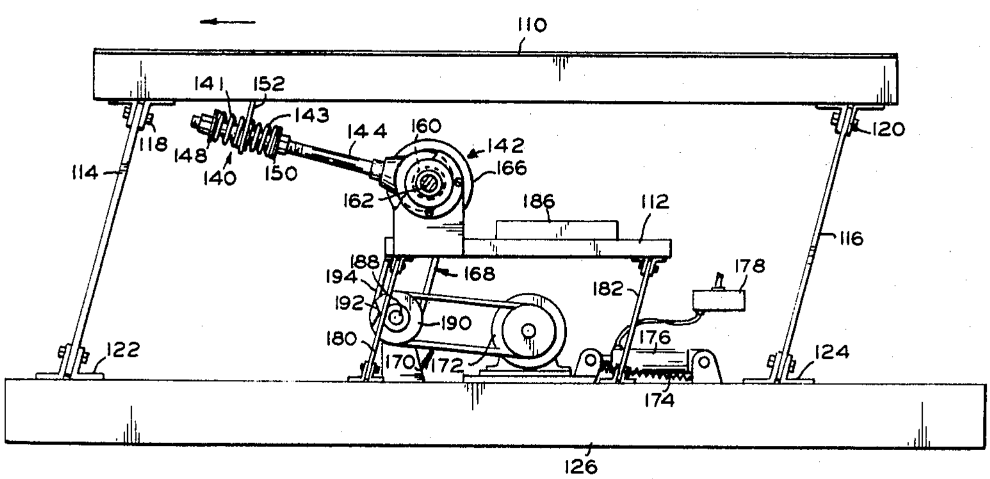

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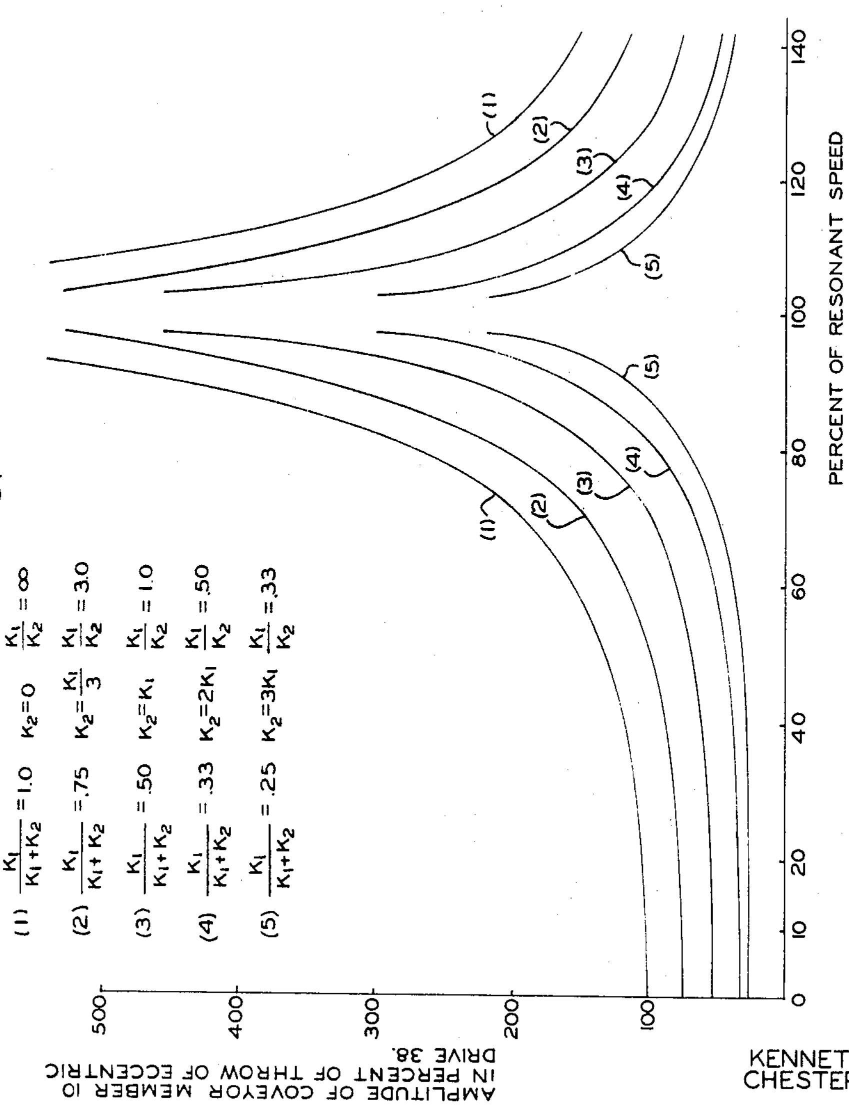
FIG. 5



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SELF-BALANCING VIBRATING STRUCTURES

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Continuation-in-part of application Ser. No. 568,616, July 28, 1966. This application Mar. 18, 1968, Ser. No. 721,539
Int. Cl. B65g *27/00*
U.S. Cl. 198—220 19 Claims

ABSTRACT OF THE DISCLOSURE

A floating eccentric drive (FIGS. 1 to 3) drives connecting rods 180° out of phase to drive two conveyor beds through resilient coupler means permitting lost motion of the conveyor beds longitudinally relative to the connecting rods. The conveyor beds are supported by resilient struts, and the coupler means permit the conveyor beds to balance each other over a wide range of difference in loading of the two conveyor beds. The drive is operated at from 50 to 90% of resonant speed, and the ratio of the coupler spring constant of each coupler means to the total spring constant of the struts is at least 1:3. In FIG. 4, a single conveyor bed mounted on resilient struts is driven through a resilient coupler by a crank drive mounted on a counterweight carried below the conveyor bed by resilient struts.

This is a continuation-in-part of our application Ser. No. 568,616, filed July 28, 1966, now abandoned.

This invention relates to self-balancing vibrating structures, and more particularly to conveyor structures having opposed, resiliently mounted, counter-balancing vibrating mechanisms.

An object of the invention is to provide self-balancing vibrating conveyor structures.

Another object of the invention is to provide conveyor structures having opposed, resiliently mounted, counter-balancing vibrating mechanisms.

A further object of the invention is to provide a self-balancing vibrating structure in which opposed outputs of a vibrating drive are coupled by means including a resilient coupling to a pair of independently and resiliently mounted vibrating systems.

Another object of the invention is to provide a self-balancing conveyor structure in which a pair of conveyor mechanisms are vibrated oppositely by a drive including a resilient drive coupling which permits the conveyor members to shift in amplitude and phase relative to each other to balance each other and loads thereon.

A still further object of the invention is to provide a conveyor structure in which a pair of vibrating conveyor systems balance each other over a wide range of speeds.

Another object of the invention is to provide a conveyor structure in which the throw of an eccentric drive to a conveyor bed is amplified at speeds below resonance by resilient coupling means connecting the drive to the conveyor bed.

The invention provides a vibrating structure in which a conveyor bed supported by resilient support means opposing oscillation thereof is driven by resilient drive means at a speed between 50% and 90% of resonance and the spring constant of the drive means is at least one-third that of the resilient support means. A vibrating structure forming one specific embodiment of the invention includes a pair of independently and resiliently mounted vibrating mechanisms vibrated in opposite directions through drive means including at least one resilient, driven coupling. One or both of the vibrating mechanisms may be a conveyor mechanism. Each vibrating structure includes means for driving at frequencies less than the resonant frequency thereof. A vibrating structure forming one embodiment of the invention includes a pair of conveyor members mounted independently and resiliently and driven by opposed outputs of a drive including a pair of resilient couplings, the resilient couplings having effective spring strengths of the same order of magnitude as those of the resilient mounts of the conveyor members. The couplings permit the conveyor members to shift in phase and amplitude relative to each other to maintain a balance between the conveyor members, and preferably the drive is free to shift to aid in maintaining the balance. Preferably the conveyor members are rigidly connected to flexible glass fiber reinforced plastic struts and are supported by the struts. The speed of the drive preferably is of a type with a wide range of speed adjustment adapted to drive the conveyor members at any selected speed below resonance in a wide range from a low speed at which material on the members is not advanced therealong to a high speed at which the material is advanced rapidly therealong. In a vibrating structure forming an alternate embodiment of the invention, an elongated first conveyor member is mounted resiliently for back and forth movement and an eccentric vibrating drive drives the conveyor member through a resilient coupling and also drives a counterbalancing member oppositely to the drive of the conveyor member, the counter-balancing member being mounted resiliently for back and forth movement opposite to that of the first conveyor member.

A complete understanding of the invention may be obtained from the following detailed description of self-balancing vibrating structures forming specific embodiments thereof, when read in conjunction with the appended drawings, in which:

FIG. 1 is a side elevation view of a self-balancing vibrating structure forming one embodiment of the invention;

FIG. 2 is an enlarged, horizontal sectional view taken substantially along line 2—2 of FIG. 1;

FIG. 3 is an enlarged, vertical sectional view taken substantially along line 3—3 of FIG. 2;

FIG. 4 is a side elevation view of a self-balancing vibrating structure forming an alternate embodiment of the invention; and

FIG. 5 is a graph illustrating conveyor throw for different frequencies of operation.

Referring now in detail to the drawings, there is shown in FIGS. 1 to 3 a self-balancing vibrating structure forming one embodiment of the invention and including a pair of tandem arranged, generally horizontal, trough-like conveyor members **10** and **12**, preferably of substantially equal lengths. The conveyor member **10** is resiliently supported by pairs of parallel, strap-like resilient struts **14**, which are preferably composed of glass reinforced plastic material. The struts **14** slope downwardly and slightly forwardly relative to direction of feed and extend primarily vertically, and the upper end portions thereof are rigidly secured to the conveyor member **12** by brackets **16** and bolts **18**. The lower end portions of the struts **14** are rigidly secured by brackets **20** and bolts **22** to an elongated base or frame **24** supported by cross members **25**, which may be suitably mounted by resilient supports **27** on a floor or suspended from a ceiling. The conveyor member **10** and the struts **14** comprise an independent vibrating mechanism or system.

The conveyor member **12** is resiliently supported, similarly to the supporting of the conveyor member **10**, by pairs of resilient struts **26** identical to and parallel to the struts **14** and connected rigidly to the conveyor member **12** and the base **24** by brackets **28** and **30**. Thus, each of the conveyor members **10** and **12** is resiliently mounted for independent reciprocating movement relative to the base **24**, and each is a part of an independent vibrating mechanism or system. When each of the conveyor members is reciprocated longitudinally, the struts supporting it are flexed so that, as the conveyor member is moved to the left from its normal position, as viewed in FIG. 1, the conveyor member is moved downwardly, and, as the conveyor member is moved to the right from its normal or at rest position, the conveyor member is moved upwardly and feeds material thereon toward the right.

The conveyor member **12** is resiliently supported, similarly to the supporting of the conveyor member **10**, by pairs of resilient struts **26** identical to and parallel to the struts **14** and connected rigidly to the conveyor member **12** and the base **24** by brackets **28** and **30**. Thus, each of the conveyor members **10** and **12** is resiliently mounted for independent reciprocating movement relative to the base **24**, and each is a part of an independent vibrating mechanism or system. When each of the conveyor members is reciprocated longitudinally, the struts supporting it are flexed so that, as the conveyor member is moved to the left from its normal position, as viewed in FIG. 1, the conveyor member is moved downwardly, and, as the conveyor member is moved to the right from its normal or at rest position, the conveyor member is moved upwardly and feeds material thereon toward the right.

A floating drive **38** drives the conveyor members **10** and **12** oppositely and at the same frequency through pairs of resilient drive couplings **40** and **42** receiving outputs of the drive **38** from connecting rods **44** and **46**, which are 180° out of phase relative to each other, and the couplings keep the frequencies of the conveyor members the same while permitting the phase and amplitude of one of the conveyor members to so vary relative to the phase and amplitude of the other conveyor member as to perfectly balance the vibration of the other conveyor member even though the load on one of the conveyor members may be several times greater than the load on the other conveyor member. The couplings provide for substantial lost motion between the drive **38** and the conveyor members.

There are two connecting rods **44** connected by self-aligning bearings **48** to two, in phase, eccentrics or crank portions **50** of a crankshaft **52** mounted rotatably in bearings **54** carried by parallel arms **56** of a floating mounting frame **57** pivotally mounted by a shaft **60** on the base **24**. There are also two of the connecting rods **46** connected by self-aligning bearings **58** to eccentrics or crank portions **59**, which are 180° out of phase from the eccentrics **50**. The crankshaft carries small flywheels **62**. The shaft **60** is driven at any selected rate of speed by a variable pulley **64**, a belt **66** and a variable pulley **68** driven by an electric motor **70**. These elements constitute an adjustable frequency drive. To change the speed, pressure of air in a control cylinder unit **72** is varied by a bleeder valve **74** in an air line **75** to shift a mount **76** to move the motor and pulley **68** farther from or closer to the pulley **64**. This inversely changes the effective diameters of the pulleys **64** and **68** to change the speed of the shaft **60**. A fixed ratio belt and pulley drive **78** drivingly connects the shaft **60** to the crankshaft **52**.

The drive couplings **40** and the drive couplings **42** are substantially identical, and only the drive couplings **40** will be described in detail. Each drive coupling **40** (FIGS. 1 and 3) includes a pair of rather long helical compression springs **80** of substantially equal lengths and of substantially equal spring constants mounted in seats on opposite sides of a vertical abutment plate **82** of a channel-like bracket member **84** fixed to the bottom of the conveyor member **10** and including suitable cross bracing. The conveyor members **10** and **12** have longitudinal stiffeners **88** and **90**, which are substantially T-shaped in transverse cross-section. The connecting rods **44** extend quite loosely through notches or slots **92** in the abutment plate **82** and through the pairs of springs **80**. Nuts **94** and **96** screwed onto threaded end portions **98** of the connecting rods **44** and spring seats **100** and **102** on the rods confine the pairs of springs under some preloading compression therebetween. The combined or effective strength of the springs **80** is substantially of the same order of magnitude as the combined strength of the struts **14**. That is, upon the application of a given static force to the connecting rods to produce a given length of compression of the springs **80**, the upper ends of the struts **14** will be moved a lesser length. For effective operation of the conveyor structure, the spring constant of the coupling should be at least about one-third that of the combined spring constant of the struts, and should not be greater than about three times. The connecting rods **46** are similarly connected to the conveyor member **12** by the drive couplings **42**, which include pairs of compression springs **111** on opposite sides of an abutment plate **113** of a bracket member **115** secured to the conveyor member **12**. Longitudinal stiffeners **117** are secured to the bottom of the conveyor member **12**. The drive couplings **40** and **42** supply energy in generally opposed directions at the same frequency to the two independent vibratory systems comprising the conveyor members **10** and **12** and resilient struts **14** and **26**. However, the phase relationship and amplitudes of reciprocation of the two vibratory systems are permitted by the resilient couplings to change so that the reaction force of each of the conveyor systems balances that of the other. As a result, no vibrations are transmitted to the base, no shocks are imparted to the drive, and very little power is required to drive the conveyor members.

In one constructed conveyor structure forming a specific embodiment of the invention, the throw of each connecting rod was three-eighths of an inch, and, for rotation of the crankshaft **52** at about fifty revolutions per minute, the amplitude or throw of each conveyor member was about one-quarter of an inch. For rotation of the crankshaft at about eight hundred revolutions per minute, the amplitude of each conveyor member was about three-fourths of an inch. The drive was smooth, free of shock and quiet at all speeds throughout this range from fifty to eight hundred revolutions per minute. Each conveyor member **10** and **12** was about thirty feet long and was supported by eight pairs of struts, and, while under load, the power required to drive the conveyor members at eight hundred oscillations per minute was about one-quarter horsepower. The speed at which products were fed was from zero to one-hundred fifty feet per minute, about one and one-half tons being fed per minute.

In the operation of the above-described self-balancing vibrating structure, material is supplied to the lefthand end portion of the conveyor member **10**. The motor **70** drives the crankshaft at a speed below the resonant frequency of the vibrating structure and determined by the selected adjustment of the cylinder unit **72**. The crankshaft reciprocates the connecting rods **44** and **46**, the connecting rods **44** being in phase with each other and the connecting rods **46** also being in phase with each other but 180° out of phase relative to the connecting rods **44**. The resilient drive couplings **40** and **42** impart the opposed forces to the conveyor members **10** and **12** to flex each of the struts **14** and **26** substantially equidistantly on opposite sides of the normal or unflexed position of that strut. At low frequencies of vibration, the material is not moved along the trough. When the frequency is increased, the amplitude or throw of the troughs is increased and the material is advanced by the troughs to the right. When the loads on the two conveyor members are equal, their phase relationships to the drive are substantially the same and their amplitudes are substantially equal. However, when their loads are quite different from each other, the phase relationship of the conveyor members and their amplitudes so shift as to balance their reaction forces on the drive, this being permitted by the resilient drive couplings **40** and **42**.

While the conveyor members **10** and **12** in the disclosed embodiment are mounted by the struts **14** and **26** for feeding material in the same direction, it will be obvious that the struts **26**, instead of being parallel to the struts **14**, may be sloped opposite to the slope of the struts **14** so that the conveyor member **12** advances material oppositely to the direction of advancement by the conveyor member **10**. Thus, the conveyor members can be mounted to both feed to a receiving means at the adjacent ends of the conveyor member. Also, the conveyor members **10** and **12** may be so mounted as to both feed material away from their adjacent ends. The conveyor members **10** and **12**, while being shown as positioned in substantial alignment, may, of course, be offset from each other, and may, if desired, be placed one above the other. However, to achieve the self-balancing condition the conveyor members should be mounted in substantially parallel positions and be driven substantially fully out of phase with one another. If desired, the couplings **42** may be rigid, with the couplings **40** being resilient and providing all the necessary resilient lost motion for balancing.

In FIG. 5 there are shown typical curves of the conveyor member **10** in which the amplitude or throw of the conveyor is plotted against the frequency of the drive **38**. The amplitude is given in percent of the throw of the eccentric drive and the frequency is given in percent of resonant frequency. Curve **1** is that in which the struts **14** supporting the conveyor member **10** are of no spring strength, and, in effect, are freely pivotal arms, with the entire spring structure being in the couplings **40** which have a spring constant such as to give the vibratory system which consists of the member **10**, the struts **14** and the couplings **40** a resonant frequency between 400 and 1500 cycles per minute. A curve **2** shows the amplitude of the conveyor member **10** where the spring constant or strength of the couplings **40** is three times that of the struts **14** with the resonant frequency of the system being between 400 and 1500 cycles per minute. Curve **3** gives the amplitude of movement of the conveyor member **10** where the spring constants of the couplings **40** and the struts **14** are equal with the resonant frequency of the system consisting of the conveyor member **10**, the couplings **40** and the struts **14** being between 400 and 1500 cycles per minute, for example. For the curve **4**, the spring constant of the couplings **40** is one-half that of the struts **14** and the resonant frequency of the system consisting of the member **10**, the struts **14** and the couplings **40** is between 400 and 1500 cycles per minute. The curve **5** represents the conditions where the spring rate or strength of the couplings **40** is one-third that of the struts **14** with the resonant frequency of the vibratory system made up of the member **10**, the struts **14** and the couplings **40** being between 400 and 1500 cycles per minute. In the above examples K_1 represents the spring constant of the couplings **40** and K_2 is the total spring constant of the struts **14**. In each of these examples, the conveyor is best operated below resonant frequency and at a frequency in the range of from 50% to 90% of the resonant frequency and has a wide operating range wherein an appreciable change in amplitude results from a moderate change in frequency and in which there is an amplification of greater than one over a large portion of the operating range. This permits the throw of the eccentric drive **38** to be quite small and still obtain a large throw of the conveyor member **10**, the throw of the eccentric drive being not greater than ¾ of an inch and preferably about ⅜ of an inch.

The ratio of the spring constant of the couplings **40** to the total spring constant of the struts **14** should not be substantially less than 1:3 to provide efficient movement of the conveyor member **10** while operated below resonant frequency with a short throw of the eccentric drive **38** and amplification of the throw of the eccentric drive so that the throw or amplitude of the conveyor member is sufficiently long for excellent movement of material. The ratio of the spring contsant of the coupling **40** to that of the struts **14** may be indefinitely high so long as it is not substantially below the above-mentioned minimum of 1:3 while keeping the total spring constant of the system (the sum of the spring constant of the couplings **40** and that of the struts **14**) at such a value that the resonant frequency of the system is not substantially lower than 400 cycles per minute and is not substantially higher than 1500 cycles per minute. However, it is desirable that the spring constant of the struts **14** be sufficient to enable support of the conveyor member **10** and the material thereon by the struts without excess deflection, though, of course, pivotal arms or other supporting means could be used in place of the struts **14** if the spring constant of the coupling is made sufficiently great. The preferred range of the ratio of the spring constant of the coupling to that of the struts is from 1:2 to 3:1, a ratio of 1:1 being optimum. The conveyor bed should have sufficient weight to cause amplification of the throw of the conveyor over that of the eccentric drive at drive frequencies within the range at which the conveyor efficiently moves material thereon. The preferred weight of the conveyor is from 4% to 10% that of the spring constant of the system with an optimum being about 7 pounds of weight of the bed for a spring constant of 100 pounds per inch. With the conveyor structure within the above-outlined limits and operated at a frequency not substantially lower than 50% and not substantially higher than 90% of the resonant frequency, a short crank throw can be used while imparting an amplified conveyor bed throw. Such amplification will, of course, begin lower in the operating frequency range for a higher ratio of coupler spring constant to the total spring rate of the supporting struts than for a lower ratio. The throw of the eccentric drive may be from ⅛ inch to ¾ inch for most applications.

A very close mathematical approximation of the various values of the conveyor comprising the conveyor member **10**, the struts **14** and the coupling **40** is given by the following formula:

$$D=\frac{dK_1}{2\pi f\sqrt{C^2+\left(\frac{K_1+K_2}{2\pi f}-2\pi fM\right)^2}}$$

where:

D=Throw or distance in inches the conveyor travels between extremes

d=Throw of eccentric in inches

K_1=Static spring constant of coupling springs in pounds per inch

K_2=Total static spring constant or sum of spring constants of struts **14** in pounds per inch

C=Viscous damping factor (resistance to movement of conveyor struts **14**)

f=Frequency of conveyor in cycles per second=r.p.m./60
M=Mass of conveyor=W/386; where W=weight in pounds, and 386=acceleration due to gravity in inches per second per second.

The factor C is sufficiently small that it is negligible within the operating range of frequency of the conveyor so that the above formula reduces to:

$$D=\frac{dK_1}{K_1+K_2-4\pi^2f^2M}$$

So far as the resonant frequency is concerned, little error results from ignoring the factor C so that a formula which gives a close approximation of the resonant frequency of the vibrating system made up of the conveyor member **10**, the struts **14** and the couplings **40** is:

$$f_R=\frac{1}{2\pi}\sqrt{\frac{K_1+K_2}{M}}$$

where:

f_R=Cycles per second at resonance

In a conveyor forming one constructed embodiment of the invention, only the conveyor member **10** with the base **24**, the supporting struts **14**, the resilient couplings **40**, the drive **57**, the motor **70** and the coupling structure between the motor **70** and the drive **57** were used with the drive **57** held against floating. That is, the conveyor was a single conveyor. The member **10** was a channel having a weight about 7% of the spring force of the resilient elements, struts **14** mounted in pairs were used, the spring constant of the couplings **40** was about one-half that of the total spring constant of the struts **14**, and the resonant frequency of the vibrating system thus formed was substantially higher than 700 cycles per minute. At 450 cycles per minute, the throw of the member **10** was ¼ inch; at 500 cycles per minute the throw was slightly less than 5⁄16 inch; at 550 cycles per minute the throw was slightly over 5⁄16 inch; at 600 cycles per minute the throw was ½ inch; at 650 cycles per minute the throw was ⅝ inch; and at 700 cycles per minute the throw was 1¼ inches. In the same constructed single conveyor but with the spring constant of the couplings **40** increased to about one and one-half times the total spring constant of the struts **14** while leaving the spring constant of the struts the same, the resonant frequency of the system increased to substantially higher than 750 cycles per minute, and the throw at 500 cycles per minute was slightly less than ⅜ inch; at 550 cycles per minute was slightly less than ⅜ inch; at 600 cycles per minute was slightly over ⅜ inch; at 650 cycles per minute was ½ inch; at 700 cycles per minute was ⅝ inch; and at 750 cycles per minute was ¾ inch. In another constructed single conveyor embodying the invention the conveyor member **10** weighed 560 pounds, there were 10 struts **14**, which were ¼ inch inch fiber glass arms 2 inches by 8 inches, the ratio of the spring constant of the exciter couplings **40** to the total spring constant of the struts was 1:1, the resonant frequency was substantially above 800 cycles per minute, and the throws of the conveyor member **10** were ¼ inch at 550 cycles per minute, 5⁄16 inch at 600 cycles per minute, ⅜ inch at 650 cycles per minute, 7⁄16 inch at 700 cycles per minute, 9⁄16 inch at 750 cycles per minute and ¾ inch at 800 cycles per minute. With the same conveyor as that just described but with the couplings **40** being weaker to provide a 7:8 ratio of couplings' spring constant to total support struts' spring constant, the throw varied from ¼ inch at 500 cycles per minute to 1⅛ inches at 800 cycles per minute, the resonant frequency being lower than the immediately preceding described conveyor system and being somewhat over 800 cycles per minute.

EMBODIMENT OF FIG. 4

A self-balancing vibrating structure shown in FIG. 4 forming an alternate embodiment of the invention includes a conveyor member **110** and a counterbalancing member **112** forming parts of two independent, opposed vibrating systems or mechanisms. The conveyor member **110** is mounted on a first pair of resilient, slightly inclined struts **114** and a second pair of parallel struts **116**, brackets **118** and **120** connecting the member **110** rigidly to the upper end portions of the struts and brackets **122** and **124** connecting a base **126** rigidly to the lower end portions of the struts. The member **110** is adapted, when moved back and forth, to advance material to the left, as viewed in FIG. 4.

A resilient coupling **140** of substantially the same construction as the previously described couplings **40** serves to resiliently connect the conveyor member **110** to an eccentric type vibrating drive **142**. The coupling **140** includes a pair of elonagted coil springs **141** and **143** on the end portion of a connecting rod **144** and held by spring seats **148** and **150** under preloaded compression abutting opposite sides of a rigid abutment plate **152** fixed rigidly to the bottom of the conveyor member **110**. The coupling **140** provides resilient lost motion between the connecting rod and the conveyor member.

The connecting rod **144** forms a part of the vibrating drive **142**, and is driven by an eccentric or crank portion **160** of a crankshaft **162** journaled in bearings **164** mounted rigidly on counterbalancing member **112**. Flywheels **166** are mounted on the crankshaft **162**, which is driven by a belt and pulley drive **168** driven through a variable speed pulley drive **170** by an electric motor **172** mounted slidably on the base **126** and held in selected position by a spring **174** and a pneumatic cylinder unit **176** controlled by a remotely controlled bleeder valve **178** in a line **179** carrying air under a constant pressure.

The counterbalancing member **112** forms, with resilient pairs of slightly inclined struts **180** and **182** fixed to the member **112** and the base **126**, an independent vibrating system or mechanism which is substantially matched to the vibrating system including the conveyor member **110** and struts **114** and **116**. A counterweight **186** forms an integral part of the counterbalancing member **112**. A shaft **188** journaled on the base is driven by output pulley **190** of the variable speed pulley drive **170**, and has keyed thereto driving pulley **192** of the belt and pulley drive **168**. A belt **194** of the drive **168** extends substantially parallel to the struts **180** and **182**. The drive **142** has two outputs 180° out of phase or directly opposed, one output being nonresiliently connected to the counterbalancing member **112** and the other output being connected for resilient, lost motion to the conveyor member **110**.

When the crankshaft **162** is rotated, it drives the connecting rod **144** and the counterbalancing member **112** oppositely or 180° out of phase. The conveyor member **110** is driven by the connecting rod through the resilient coupling **140**, whose resiliency is greater than the resiliency of either the vibrating system including the conveyor member **110** or the vibrating system including the counterbalancing member **112**. That is, the spring constant of the springs **141** or **143** is substantially less than the total spring constant of the struts **114** and **116** or the total spring constant of the struts **180** and **182**. This permits the member **110** to shift in phase and amplitude or throw relative to the member **112** so that the two vibrating systems are kept well balanced.

The above described conveyor structures have very high capacities, while being very light in weight and requiring very little power. The conveyor structures operate very well with short crank throws and amplified conveyor bed throw, and do so at frequencies below resonance so that it is not necessary to go up to or through resonance in their operation. The above-described conveyor systems balance so well that substantially no vibration is imparted to the bases thereof.

It is to be understood that the above-described arrangements are simply illustrative of the application of the principles of the invention. Numerous other arrangements may be readily devised by those skilled in the art which will embody the principles of the invention and fall within the spirit and scope thereof.

What is claimed is:

1. In combination,
a pair of vibrating members,
base means,
a plurality of resilient struts mounting the vibrating members independently on the base means for independent longitudinal back and forth movements and urging the vibrating members toward predetermined positions,
a drive mechanism having a pair of output drive means driven substantially 180° out of phase with respect to each other,
means mounting the drive mechanism floatingly relative to the base means,
means connecting one of the members to one of the output drive means, and
a resilient coupler connecting the other vibrating member to the other output drive means for relative longitudinal movement between one of the vibrating members and the last-mentioned output drive means.

2. The combination of claim 1
wherein the amplitude of each output drive means is a predetermined distance,
the resilient coupler having a maximum movement at least as great as said predetermined distance.

3. The combination of claim 2
wherein the resilient coupler has a maximum movement substantially greater than said predetermined distance,
the drive means serving to drive the vibrating members at a frequency less than the resonant frequency of the combination to provide a throw greater than said predetermined distance.

4. In combination,
a pair of elongated conveyor members,
a base,
first resilient mounting means mounting one of the conveyor members in a generally horizontal position extending in a predetermined direction for movement back and forth on the base,
second resilient mounting means mounting the other conveyor in a generally horizontal position extending in said direction for movement back and forth on the base,
crank means having a pair of eccentric drive portions 180° out of phase,
a mounting structure mounting the crank means for floating movement in said direction,
a pair of connecting rods driven by the eccentric drive portions,
power means for rotating the crank means, and
means connecting the connecting rods to the conveyor members and including at least one resilient coupler permitting limited longitudinal movement of one of the connecting rods relative to one of the conveyor members.

5. The combination of claim 4 wherein the coupler includes an abutment plate mounted rigidly on one of the conveyor members and having a clearance opening through which one of the connecting rods extends, a pair of compression springs mounted on the last-mentioned connecting rod and a pair of spring seat means on the last-mentioned connecting rod holding the compression springs in engagement with the abutment plate.

6. The combination of claim 4 wherein each of the mounting means includes a plurality of parallel, stiff slats mounting one of the conveyor means on the base.

7. The combination of claim 4 wherein each of the mounting means includes means connecting the end portions of the slats rigidly to said one of the conveyor members and the base.

8. The combination of claim 4 wherein the power means comprises a variable speed drive and means coupling the drive to the crank means.

9. The combination of claim 4 wherein the mounting structure includes a U-shaped frame member pivotally mounted on the base, and a pair of bearing means carried in aligned positions by the arms of the U-shaped frame member and journaling the crank means.

10. In combination,
a pair of elongated conveyor members,
a base,
a plurality of resilient first struts mounting one of the conveyor members in a generally horizontal position extending in a predetermined direction for movement back and forth on the base,
a plurality of resilient second struts mounting the other conveyor in a generally horizontal position extending in said direction for movement back and forth on the base,
floating crank means having a pair of eccentric drive portions 180° out of phase,
a pair of connecting rods driven by the eccentric drive portions,
power means for rotating the crank means, and
a pair of resilient couplers connecting the connecting rods to the conveyor members for longitudinal floating movement therebetween.

11. In a conveyor,
a conveyor member,
resilient means mounting the conveyor member for oscillatory movement and having a predetermined spring strength,
oscillating drive means having a predetermined throw, and
resilient coupling means connecting the drive means to the conveyor member for oscillating the conveyor member, the spring constant of the resilient coupling means being sufficiently high relative to that of the resilient means that the throw of the conveyor member is greater than said throw of the drive means when the drive means is operated at a frequency less than nine-tenths of the resonant frequency of the system including the conveyor member, the resilient means and the coupling means.

12. The conveyor of claim 11 wherein the spring constant of the resilient coupling means is sufficiently high relative to that of the resilient means that the throw of the conveyor member is greater than said throw of the drive means when the drive means is operated at a frequency within a substantial portion of a range of frequency from about one-half to about nine-tenths of the resonant frequency of the system including the conveyor member, the resilient means and the coupling means.

13. The conveyor of claim 11 wherein the spring constant of the coupling means is not substantially less than one-third of that of the resilient means.

14. The conveyor of the claim 11 wherein the spring constant of the coupling means is from one-half to twice that of the resilient means.

15. The conveyor of claim 11 wherein the spring constant of the coupling means is substantially equal to that of the resilient means.

16. In a conveyor,
a conveyor member,
resilient means including support means supporting the conveyor member for oscillatory movement and resilient coupling means for oscillating the conveyor member,
the resilient means and the conveyor member forming a vibrating system having a resonant frequency within a range of from 400 cycles per minute to 1500 cycles per minute,
drive means coupled to the conveyor member by the coupling means serving to oscillate the conveyor member at a frequency of from 50% to 90% of said range,
the resilient means having a predetermined spring constant,

the conveyor member having a weight of from 4% to 10% of said predetermined spring constant,

the spring constant of the resilient coupling means being not substantially less than one-third of said predetermined spring constant.

17. The conveyor of claim **16** wherein the spring constant of the resilient coupling means is from one-half to twice that of the resilient means.

18. The conveyor of claim **16** wherein the spring constant of the resilient coupling means is substantially one-half that of the resilient means.

19. The conveyor of claim **17** wherein the weight of

the conveyor member is about 7% of said predetermined spring constant.

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EDWARD A, SROKA, Primary Examiner

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,476,234 Dated November 4, 1969

Inventor(s) Kenneth M. Allen and Chester H. Harper

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 3, lines 26 to 42, are duplicates of the preceding paragraph and should be omitted.

Column 7, line 55, "inch" should be canceled.

SIGNED AND
SEALED
FEB 17 1970

(SEAL)
Attest:

Edward M. Fletcher, Jr.
Attesting Officer

WILLIAM E. SCHUYLER, JR.
Commissioner of Patents