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VIBRATORY DRIVE MECHANISM

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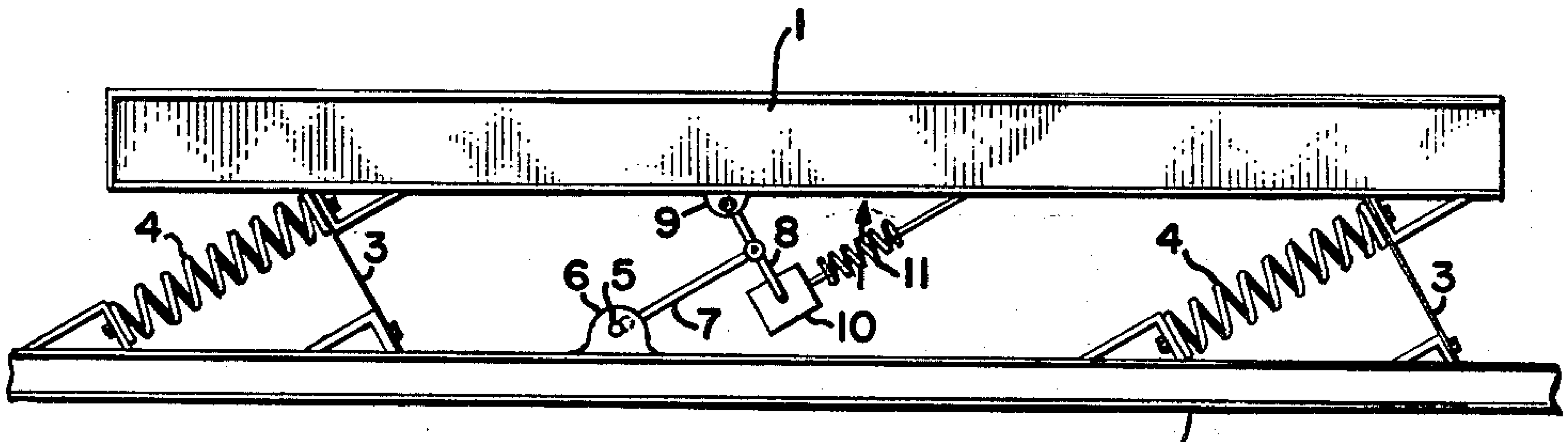

Fig-1

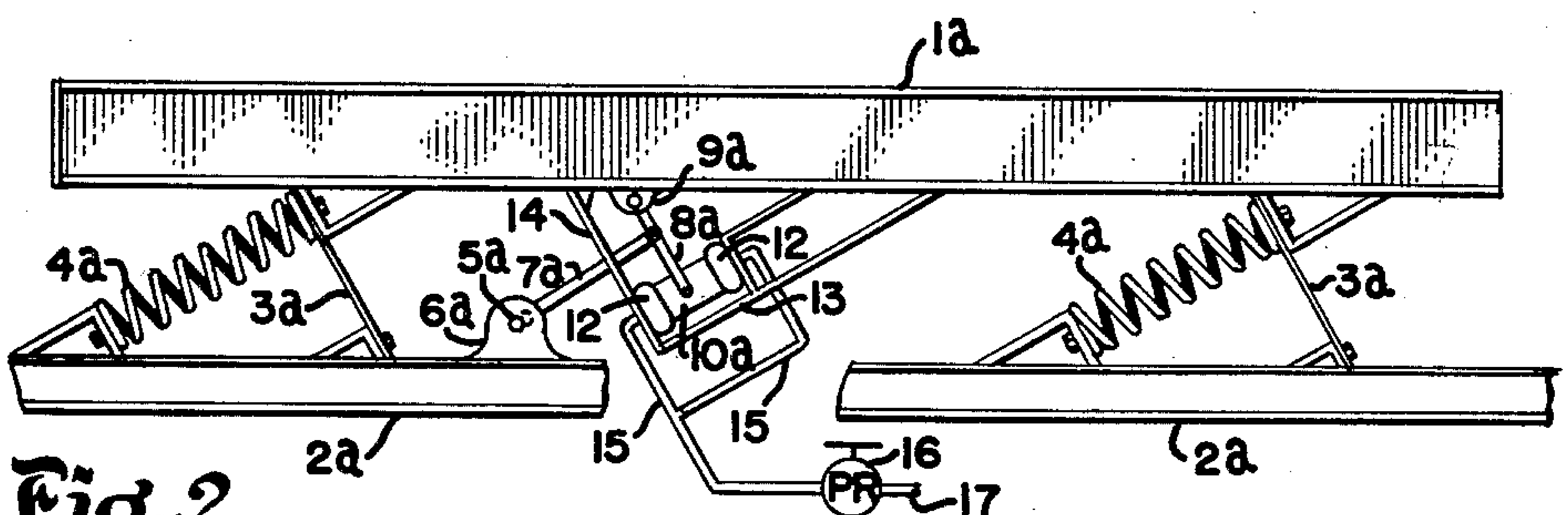

Fig-2

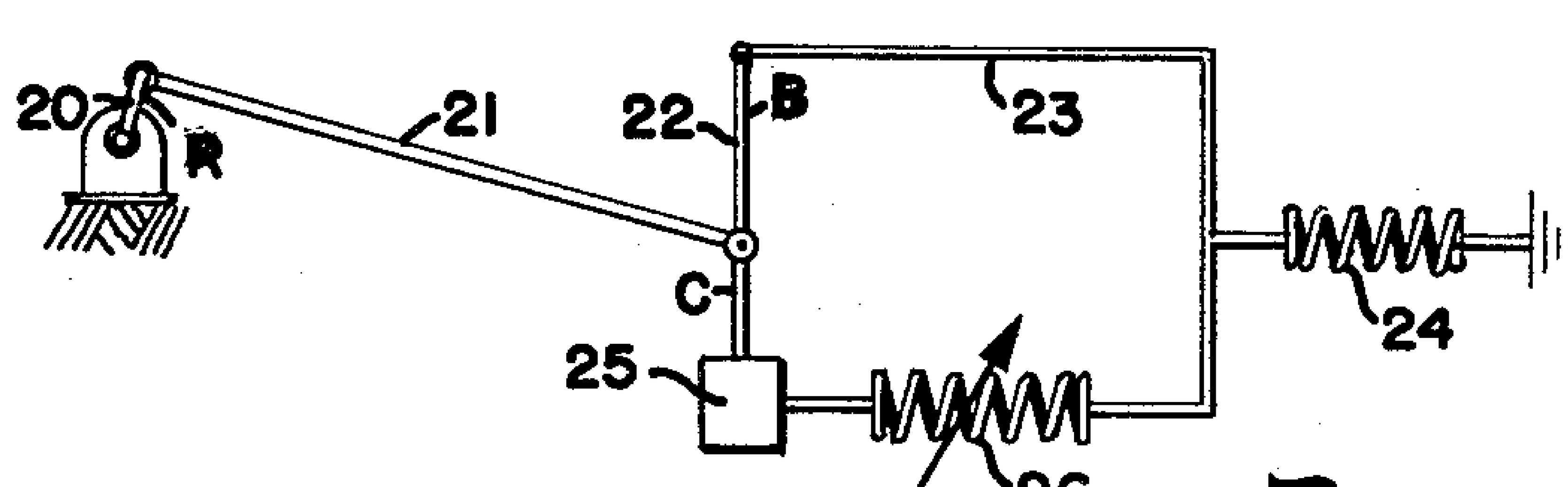

Fig-3

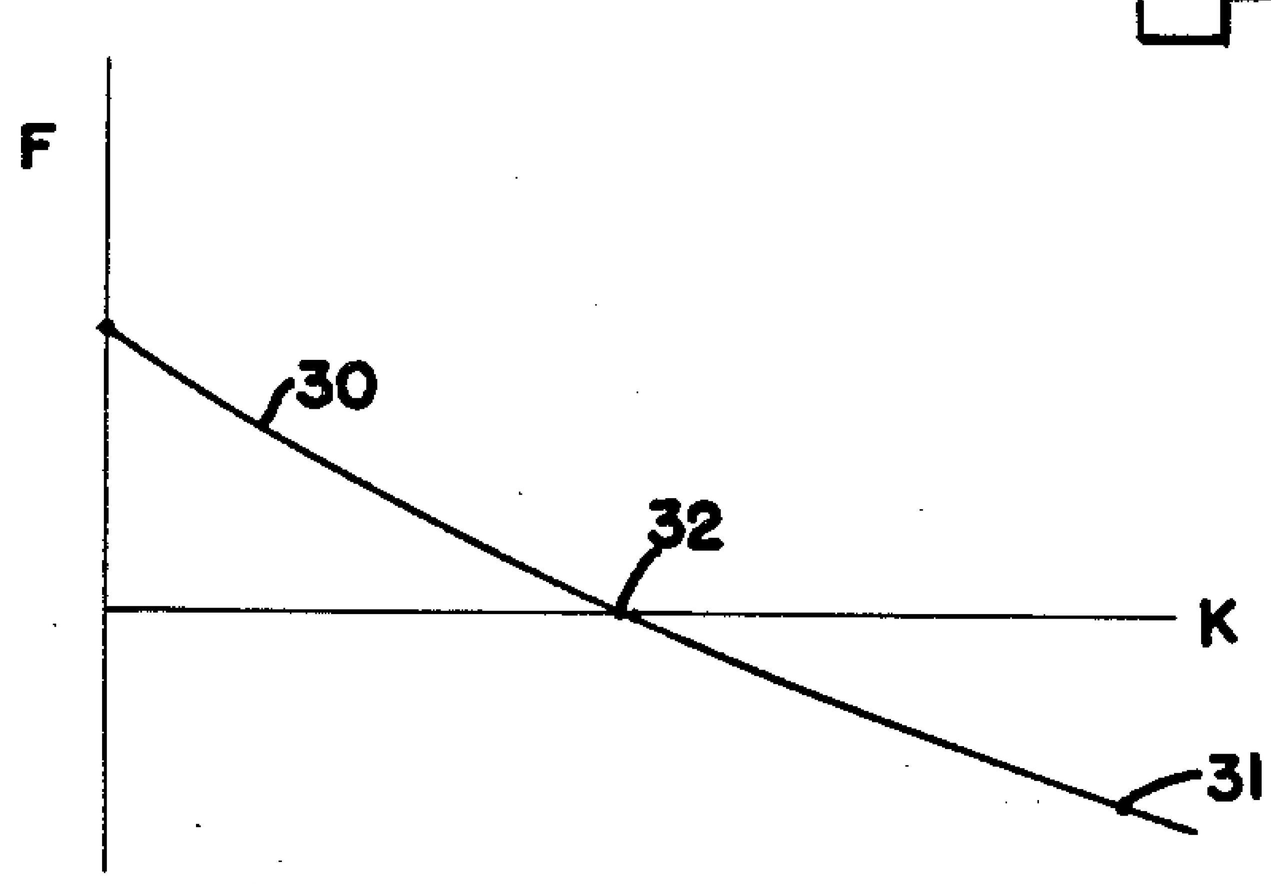

Fig-4

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VIBRATORY DRIVE MECHANISM

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This invention relates to a vibratory drive mechanism and in particular to means for adjustably controlling the stroke of vibratory apparatus driven by a constant stroke crank and connecting rod mechanism.

Many types of vibratory apparatus such as conveyors, feeders, vibratory screens, earth tampers, and similar apparatus are driven by crank and connecting rod mechanisms that provide a substantially constant stroke regardless of the variations in load or forces being overcome by the vibratory members. These drives are quite satisfactory as long as it is not necessary to vary the amplitude of the vibratory movement of the vibrating members. However these drives cannot be readily adapted for variable amplitude operation.

The principal object of this invention is to provide means for varying the amplitude of vibration of a vibratory member driven by a constant stroke crank and connecting rod mechanism.

Another object of the invention is to provide means that are readily adaptable for remote control for controlling the amplitude of vibration of a vibratory member.

A still further object of the invention is to provide tunable coupling means between a crank and connecting rod mechanism and a tuned vibratory member arranged so that at one condition of tuning there is no force transmitted from the crank and connecting rod mechanism to the vibratory member.

A still further object of the invention is to provide means for adjustably controlling the amplitude of vibration of a vibratory member regardless of the vibratory force requirements of the vibatory member.

A still further object of the invention is to provide tunable driving means for driving a tuned vibratory system from a constant stroke drive mechanism arranged so that the tuning for zero force transmission from the drive mechanism to the tuned vibratory member occurs with a substantial spring rate in the tunable drive means.

More specific objects and advantages are apparent from the following description of a preferred form of the invention.

According to the invention, the drive force from a crank and connecting rod mechanism is divided into two parts by differential mechanism with one part being applied directly to the work member and the other part being applied to a mass that is coupled to the work member through an adjustable rate spring. The range of adjustment of the rate of the coupling spring coupling the mass to the work member is such that at one extreme the mass and spring form a resonant system having an extremely low resonant frequency that is much lower than the operating speed and at another condition of tuning or adjustment of the rate of the spring they form a resonant system having a resonant frequency higher than the operating speed. Ordinarily, the full range of tuning is not employed but rather either one of the two ranges of tuning is used, one employing spring rates equal or greater than that providing zero force transmission and the other providing spring rates equal or less than providing zero force. The upper range of tuning is preferable for control purposes because of the improved regulation of the system with respect to changes in load when operating at full stroke.

A preferred form of the invention is illustrated in the accompanying drawings.

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In the drawings:

FIG. 1 is a schematic side elevation of a vibratory conveyor or feeder embodying a drive constructed according to the invention.

FIG. 2 is a schematic side elevation of a vibratory conveyor or feeder employing a preferred form of the invention employing pneumatic springs as the adjustable resilient elements.

FIG. 3 is a schematic diagram illustrating the general cooperation of the elements including the division of drive force between the two paths.

FIG. 4 is a graph illustrating, in general, the variation in transmitted force with changes in tuning of the coupling spring.

These specific figures and the accompanying description are intended merely to illustrate the invention and not to impose limitations on the claims.

A vibratory system that includes the improved drive mechanism may comprise a conveyor trough or feeder trough **1** that is supported from base **2** by means of inclined struts or cantilever leaf springs **3** and coil springs **4**. The springs cooperate with the trough **1** to form a vibratory system that is preferably tuned to be resonant near the operating speed. Vibratory energy to drive the trough **1** is provided by a rotating crank or eccentric shaft **5** that is journaled in a standard **6** erected from the base **2**. The crankshaft **5** is connected through a connecting rod **7** to a lever **8** that is pivotally connected to a bracket **9** depending from the conveyor **1**. The lever **8** carries a mass **10** on its lower end, the mass being connected through an adjustable spring **11** to the conveyor **1**. The lever **8** is a form of differential mechanism to distribute the force from the connecting rod.

The crankshaft **5**, which may also be considered as an eccentric shaft because of its relatively short throw, is preferably driven at a constant speed by means not shown. This constant speed should, for best results under load, be approximately 5 percent less than the natural or resonant frequency of the conveyor trough **1** on the springs **3** and **4**. While coil springs **4** are shown as assisting the cantilever springs **3** in providing the resilient force for the conveyor trough **1**, other types of springs may be substituted or the cantilever springs **3** may be made stiff enough to provide all of the force themselves. The system is tuned in this manner so that the connecting rod **7** and crankshaft **5** need not supply the inertia forces required to vibrate the deck or trough **1**.

Forces from the connecting rods **7**, applied to the lever **8**, are divided with one component being applied to the bracket **9** at the upper end of the lever and thus applied directly to the conveyor trough **1** whereas the other component of the force is applied to the mass **10**. Any movement of the mass **10** transmits a force through the adjustable spring **11** to the conveyor trough **1**. By adjustment of the adjustable spring **11** the magnitudes of the forces applied to the conveyor trough **1** may be varied over a wide range and thereby vary or control the amplitude of vibration of the trough **1** while the stroke of the eccentric or crankshaft **5** and connecting rod **7** remains constant. When the adjustable spring **11** is adjusted for a very low spring rate the mass **10** tends to form a fulcrum for the lever **8** and the force transmitted through the lever bracket **9** to the conveyor deck varies according to the weight and acceleration of the mass **10**. This represents one extreme condition of tuning.

As the spring rate of the adjustable spring **11** is increased, a force is transmitted through the spring **11** to the conveyor deck **1** in accordance with the spring rate and the deflection of the mass **10** from its neutral position. This spring force acts in opposition to the force transmitted through the lever and mounting bracket 9 to the conveyor to thus reduce the net force applied to the conveyor trough 1.

If the spring rate of the adjustable spring 11 is increased until the spring becomes very stiff, the mass 10 moves in phase with the conveyor deck 1 and both the lever fulcrum 9 and the spring 10 apply force in phase to the conveyor deck 1. At an intermediate adjustment of the spring rate of the adjustable spring 11, the force transmitted through the adjustable spring 11 exactly counterbalances or is equal and opposite to the force transmitted through the lever bracket 9 so that no force is transmitted from the connecting rod 7 to the conveyor trough 1. In this condition of tuning the lever mass 10 oscillates through an amplitude of motion which is greater than the motion of the connecting rod in accordance with the ratio of length of the arms of the lever 8 and the force transmitted through the connecting rod equals the inertia force of the mass 10. In order that the amplitude of motion of the conveyor trough 1 may be reduced practically to zero when the force transmission through the lever and adjustable spring system is zero or approximately zero the system comprising the conveyor trough 1 and the springs 3 and 4 must not be operated exactly at resonance. Preferably the system is tuned to a frequency slightly higher than the operating speed.

FIG. 2 shows a similar arrangement in which pneumatic springs, arranged to be variably inflated, serve as the adjustable rate spring 11. In this particular arrangement a conveyor trough *1a* is supported from a base *2a* by means of cantilever leaf springs or struts *3a* assisted by coil springs *4a* to form a tuned vibratory system having a natural frequency substantially at the operating speed.

This system is driven by a crank or eccentric shaft *5a* mounted in a drive housing *6a* and connected through a connecting rod *7a* to a lever *8a*. The lever *8a* is connected to the work member or trough *1a* through a lever fulcrum bracket *9a*. The other end of the lever *8a* carries a mass *10a* that is sandwiched between a pair of air springs 12 held in a bracket 13 connected to the work member or trough *1a*. The lower end of the bracket 13 is braced by a strut 14 to provide rigidity. The air springs 12 are pneumatically connected through flexible connection lines 15 and an adjustable pressure regulator 16 to an air pressure supply pipe 17. It is a characteristic of the air springs, which are inflated pneumatic chambers having flexible but non-stretchable side walls, such as those sometimes used as automobile suspension springs, that the spring rate of an individual spring is not linear with displacement but rather increases nonlinearly as the height of the spring decreases. However, when two air springs are used as a pair in opposition, as illustrated in the figure the nonlinear characteristics of one substantially cancels the nonlinear characteristics of the other so that the combination presents a spring rate that is substantially constant with displacement and which varies with the inflation pressure.

A single air spring can be used in this system provided that the other air spring is replaced with a spring having a spring rate generally equal to the spring rate of the air spring when inflated to approximately half its maximum working pressure. Such a substituted system performs in approximately the same manner except that the neutral or rest position of the mass *10a* varies with the inflation pressure. However, the range of spring rates available in this arrangement are quite small and the use of a pair of opposed air springs is preferred.

FIGS. 3 and 4 are included to more clearly illustrate the dynamic conditions that exist in the drive constructed according to the invention. As illustrated in FIG. 3 a crank 20 having a radius R drives a connecting rod 21 that is connected to an intermediate pivot point of a lever 22, the overall length of which is equal to $B+C$ units. The upper end of the lever 22, the end of the arm B, is pivotally connected to a member 23 which corresponds to the work member 1 of FIG. 1. This member is connected to ground through the spring 24 having a spring rate K_0. The spring 24 is not equivalent to the springs 3 and 4 but rather is an equivalent spring that represents the tuned system comprising the work member 1 together with the springs 3 and 4 when operated at the operating speed of the crankshaft 5 and tuned to a frequency higher than the operating speed. The spring 24 provides force which is equal to the difference between the spring force of the actual springs and the inertia force of the conveyor deck at the operating frequency.

The second arm of the lever 22 having the length C is connected to a mass 25 which in turn is connected through an adjustable spring 26 to the member 23. In the event the conveyor is tuned to have a resonant frequency below the operating speed the spring 24 of FIG. 3 would be replaced by an equivalent mass.

As indicated above, when the adjustable spring such as the spring 26, is adjusted for low or zero spring rate the lever 22 is free to rotate except as it is restrained by the inertia force of the mass 25. Therefore, with the condition of tuning illustrated, i.e., the resonant frequency of the conveyor being above the operating speed, the force applied to the conveyor member or member 23 of FIG. 3 is out of phase with the motion of the connecting rod 21. Thus as the connecting rod 21 approaches the right hand end of its stroke with the mass 25 deflected to the right from the position shown, it is being decelerated and the resulting force applied through the lever to the member 23 acts toward the left. As the spring rate of the coupling spring is increased from the zero condition, the spring applies a force to the member 23 corresponding to the relative deflection of the mass 25. Therefore, when the mass 25 is at the right hand end of its stroke corresponding to the connecting rod 21 also being at the right hand end of its stroke, the spring 26 applies a force acting toward the right to the member 23 and thus in opposition to the force applied through the lever and acting to the left. This condition is indicated in FIG. 4 which represents the net force applied to the work member or to the spring 24 of FIG. 3 for various conditions of tuning of the adjustable spring 26 corresponding to the adjustable spring 11 or the air springs 12. It may be noted that in this condition of tuning the work member 1 or the member 23 of FIG. 3 moves counter to the motion of the connecting rod 21 but at a lesser stroke. Thus the actual deflection of the adjustable spring 26 is equal to the sum of the motions of the member 23 and the mass 25. This condition is illustrated by the branch 30 of the curve shown in FIG. 4.

At the other extreme of tuning, with the spring rate of the adjustable spring set for a very large value corresponding to a very stiff spring approaching a solid connection, there is very little if any relative movement between the mass 25 and the member 23 and therefore there is no rotation of the lever 22 and both ends of the lever therefore apply force to the member 23 in phase, the amount of the force developed depending upon the spring rate and deflection of the spring 24 representing the tuned system. This condition is indicated by the right hand end 31 of the curve shown in FIG. 4.

At an intermediate adjustment of the spring rate of the adjustable spring, a point 32 is reached at which there is no net force applied from the connecting rod to the work member. If the spring 26 were connected between the mass 25 and a fixed point, the condition of tuning to accomplish the zero force transmission would occur when the spring 26 and the mass 25 were in resonance with the speed of operation of the crank 20. If in the illustrated system the mass 25 and spring 26 are resonant at the operating speed and are driven to a substantial amplitude of vibration the force in the connecting rod 21 is quite small but the spring force of the spring 26 is applied to the member 23. This is a force in phase with the motion of the mass 25 and hence in phase with the motion of the connecting rod 21 and would be a force represented by a point on the branch of the curve between the point 32 and the point 31. The point 32 is reached, from the resonant condition, when the spring rate of the adjustable spring 26 is reduced in the proportion of the length C of the lever arm to the total length of the lever from the spring rate at which the mass 25 is resonant on the spring. This is the condition that the force transmitted through the lever directly to the member 23 is equal and opposite to the force transmitted through the lever and mass 25 and adjustable spring 26 to the member 23.

In a practical use of this arrangement the mass 25 or the mass 10*a* sandwiched between the pneumatic springs is made quite small so that the quasi-resonant condition, the condition of no force transmission, occurs at a relative low spring rate of the adjustable spring K. The operation is then confined to that branch of the curve in FIG. 4 lying between the points 32 and 31, the full amplitude of vibratory motion occurring near the point 31 which represents a practically direct connection between the connecting rod 7 and the work member 1.

The advantages of this type of system over the use of the air springs directly in series with the connecting rod resides in the ability of this arrangement to reduce the transmitted force to zero with a non-zero spring rate as well as the provision of a substantially direct connection to the work member at the higher spring rates.

Various modifications of the arrangement shown may be made such as varying the order of the pivots in the lever 8 without departing from the spirit and scope of the invention.

Having described the invention, I claim:

1. A vibratory drive system that comprises, in combination, a base a work member, resilient means supporting the work member from the base, a differential mechanism having a first point connected to the work member, a mass connected to another point of the mechanism, tunable resilient means connecting said mass to the work member, a crank shaft that is journaled on the base, said crankshaft being operated at a substantially constant speed and being connected to the differential mechanism, and means for tuning said tunable resilient means.

2. A vibratory drive system that comprises, in combination, a base, a work member, resilient means supporting the work member from the base and cooperating with the work member to form a resonant system, a lever having a first point connected to the work member, a mass carried on the lever remote from the first point, tunable resilient means connecting the mass to the work member, a crankshaft journaled on the base that is rotated at a speed that is near but not equal to the resonant frequency of the work member on its support means, a connecting rod connecting the crankshaft to said lever at a point separated from said mass and said first point, and means for tuning the tunable resilient means.

3. A vibratory drive system that comprises, in combination, a base, a work member, resilient support means that support the work member from the base for vibration at a resonant frequency, a lever having a first point pivotally connected to the work member, a mass on the lever remote from said first point, tunable resilient means connecting that portion of the lever carrying said mass to the work member, a crankshaft journaled on the base that is driven at a speed in the order of but not equal to the resonant frequency of the work member on the support means, a connecting rod connecting the crankshaft to a point of the lever intermediate the mass and the first point, and means for tuning the tunable resilient means.

4. A vibratory drive system that comprises, in combination, a base, a work member, resilient support means that support the work member from the base for vibration at a resonant frequency, a lever having a pair of spaced apart pivot points and a mass remote from said points, one of said points being connected to the work member, tunable resilient means connecting the mass to the work member, a crankshaft journaled on the base that is driven at a speed near but not equal to said resonant frequency, a connecting rod connected from said crankshaft to the other of said pair of points, and means for tuning said tunable resilient means.

5. A vibratory drive system that comprises, in combination, a base, a vibratory work member, means for supporting the work member from the base for movement along a work path, a lever having a pair of spaced apart pivot points and a mass spaced from said points, one of said points being connected to the work member, means fixed relative to the base for applying vibratory force to the other of said points, tunable resilient means connecting the mass to the work member, and means for tuning the tunable means.

6. A vibratory drive system that comprises, in combination, a base, a vibratory work member, means supporting the work member from the base for movement along a work path, a lever having a mass carried on one end, tunable resilient means connecting the mass to the work member, pivot means connecting a point of the lever remote from the mass to the work member, a crankshaft journaled on the base that is driven at a substantially constant speed, a connecting rod connected between the crankshaft and a point of the lever intermediate the mass and the first point, and means for tuning the tunable resilient means.

References Cited by the Examiner

UNITED STATES PATENTS

1,722,767	7/29	Schieferstein	74—26
2,947,181	8/60	Carrier et al.	74—26
2,947,410	8/60	Carrier	74—26
2,984,339	5/61	Musschoot	74—87 X
3,019,889	2/62	Carrier.	
3,024,663	3/62	Carrier	74—26

FOREIGN PATENTS

515,267	1/31	Germany.
495,498	11/38	Great Britain.

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