

[54] CONTROL FOR FLUID MOTOR

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Related U.S. Application Data

[62] Division of Ser. No. 290,531, Sept. 20, 1972, Pat. No. 3,814,532.

[52] U.S. Cl. 418/40, 60/403

[51] Int. Cl. F01c 21/12, F03c 3/00, F04c 15/02

[58] Field of Search 418/40; 60/403, 493; 91/59, 364, 388; 404/103, 117; 74/61

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Primary Examiner—John J. Vrablik

[57] ABSTRACT

A rotatably driven eccentric carrier has its axis parallel to that of a compacting machine drum. A large eccentric mass fixed to one side of the carrier is only partially counterbalanced by a smaller radially movable mass thereon; hence centrifugally responsive movement of the smaller mass decreases vibration amplitude. The radial position of the smaller mass is controlled by a single-acting hydraulic jack on the carrier, connected with an accumulator through a manually operable valve. Speed of carrier rotation is controllable. A pressure responsive cutout switch in the jack-accumulator circuit prevents overspeed.

2 Claims, 6 Drawing Figures

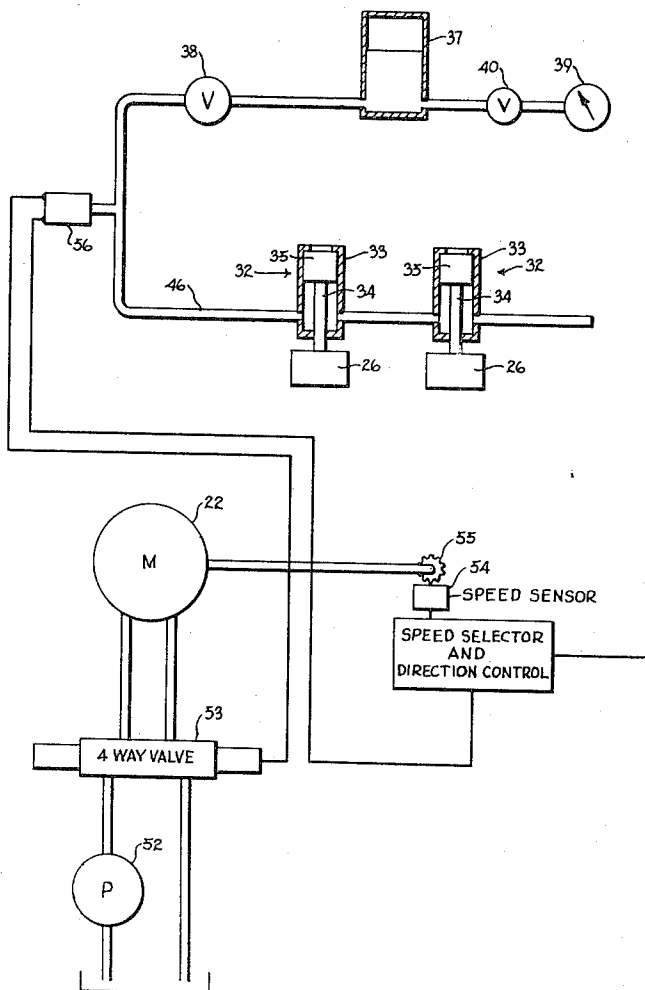


Fig. 1.

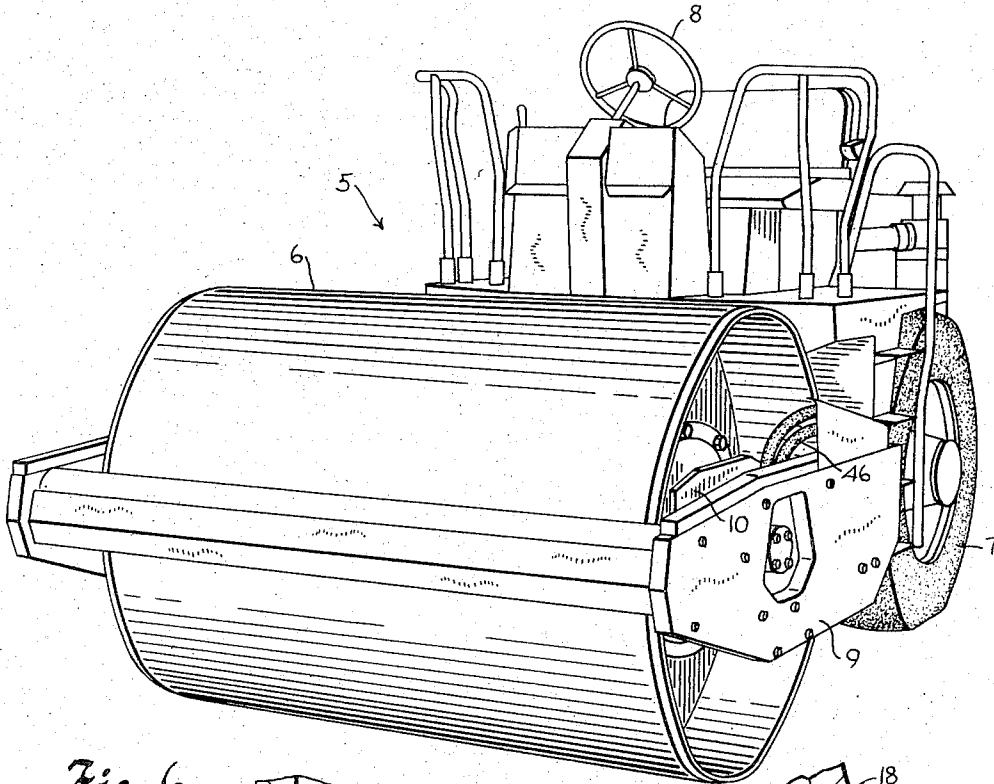


Fig. 6.

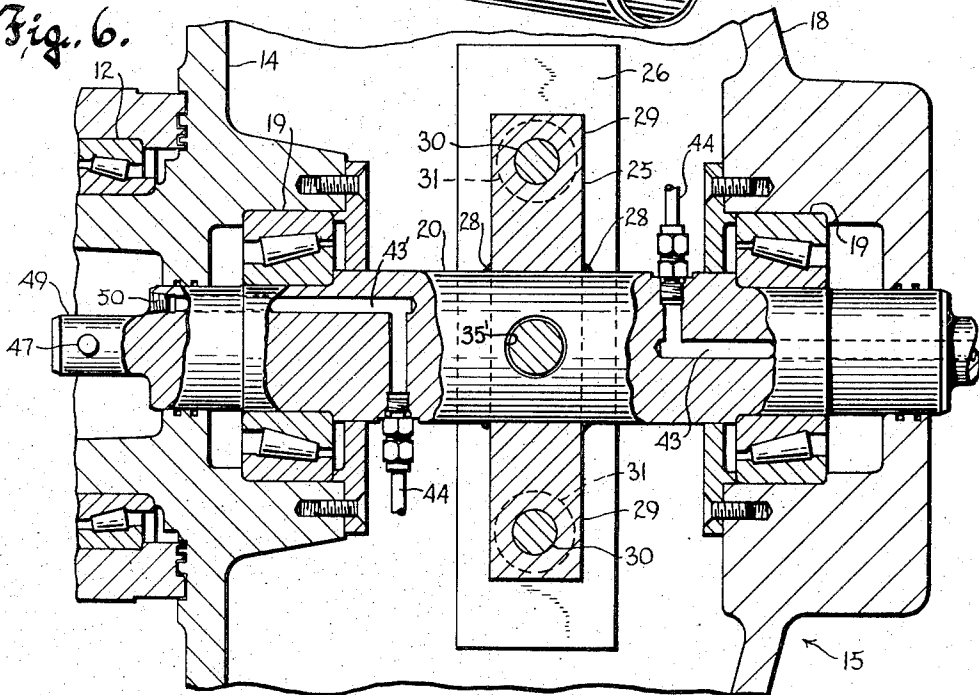


Fig. 2.

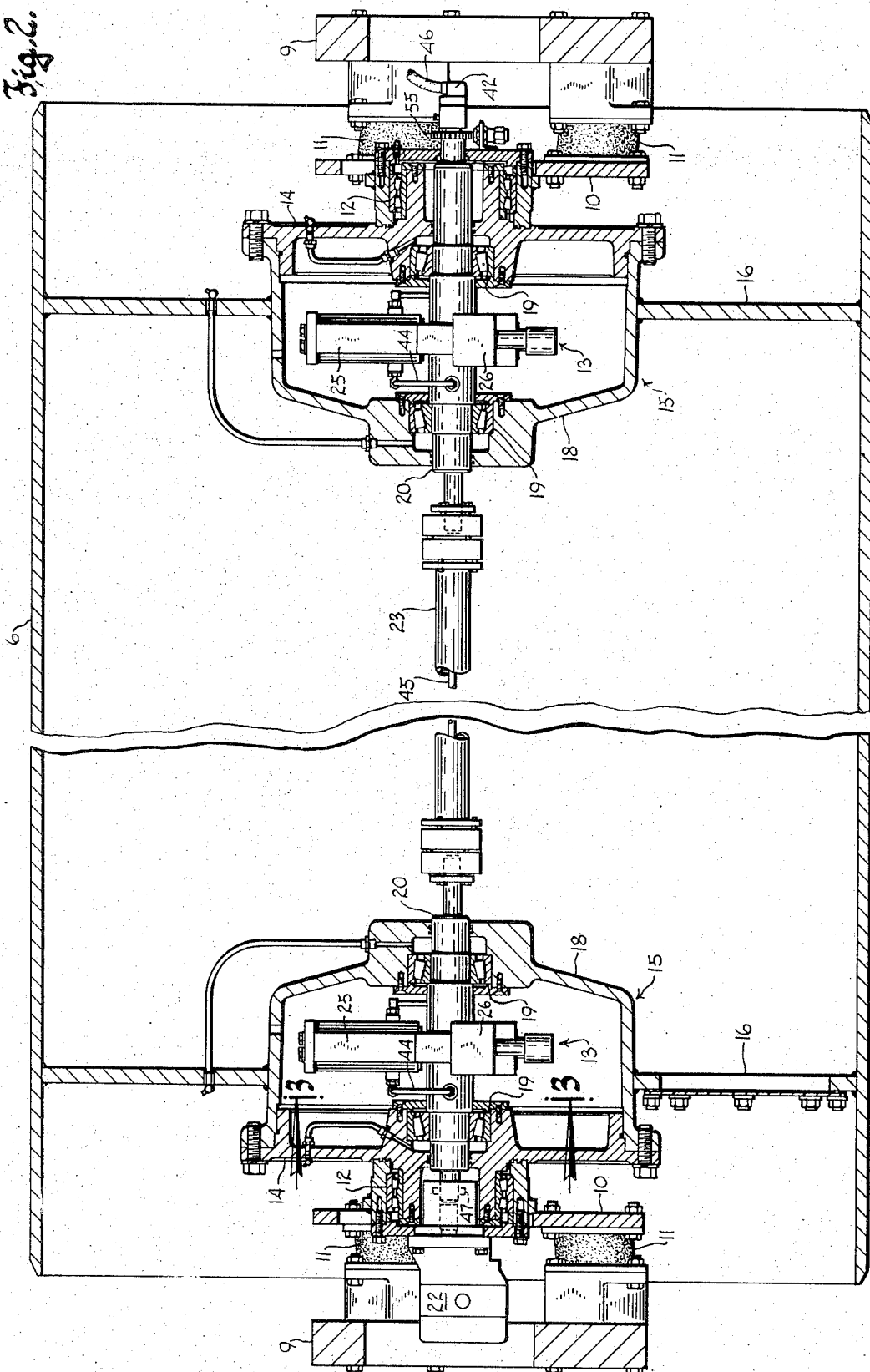


Fig. 4.

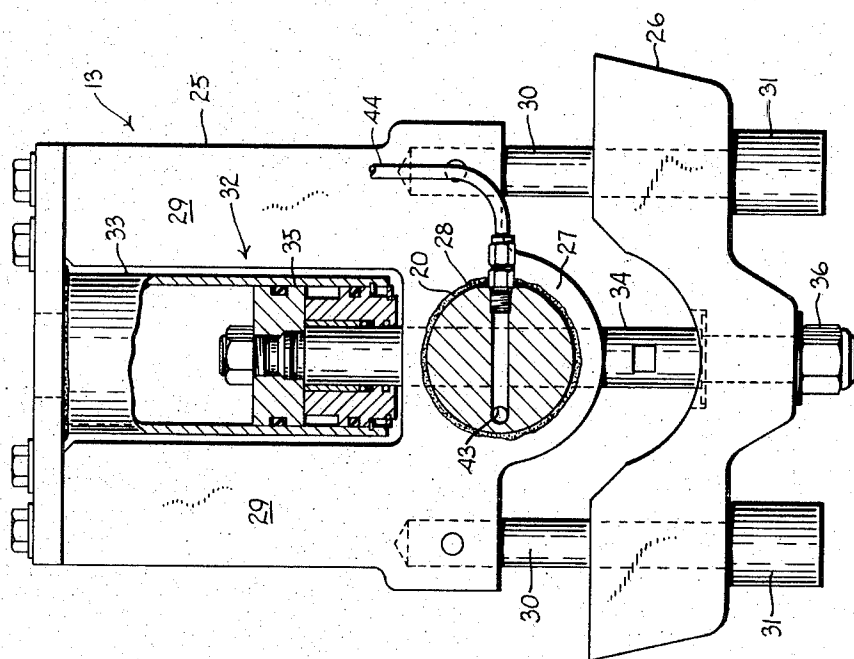


Fig. 3.

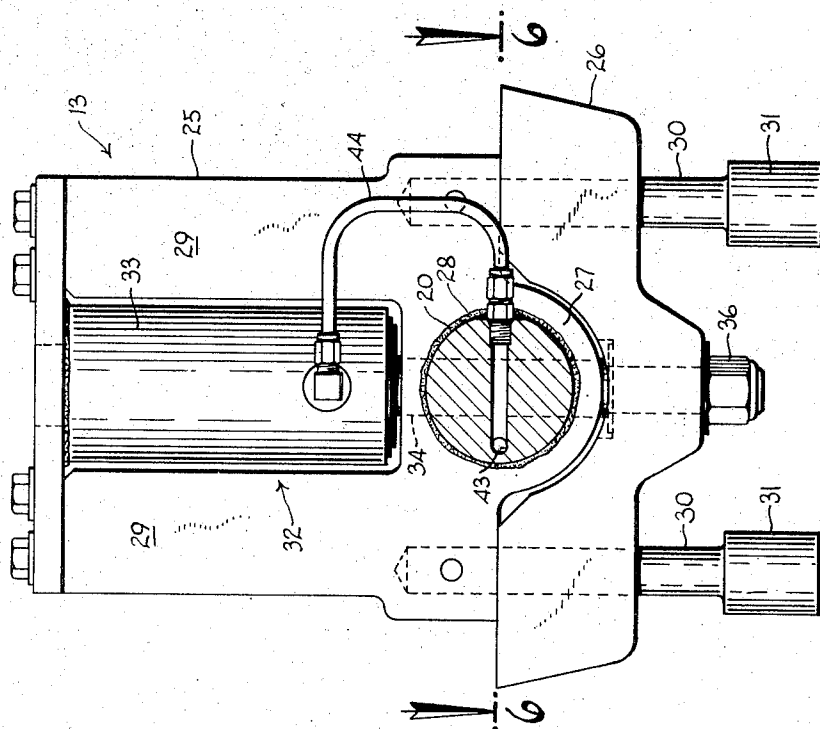
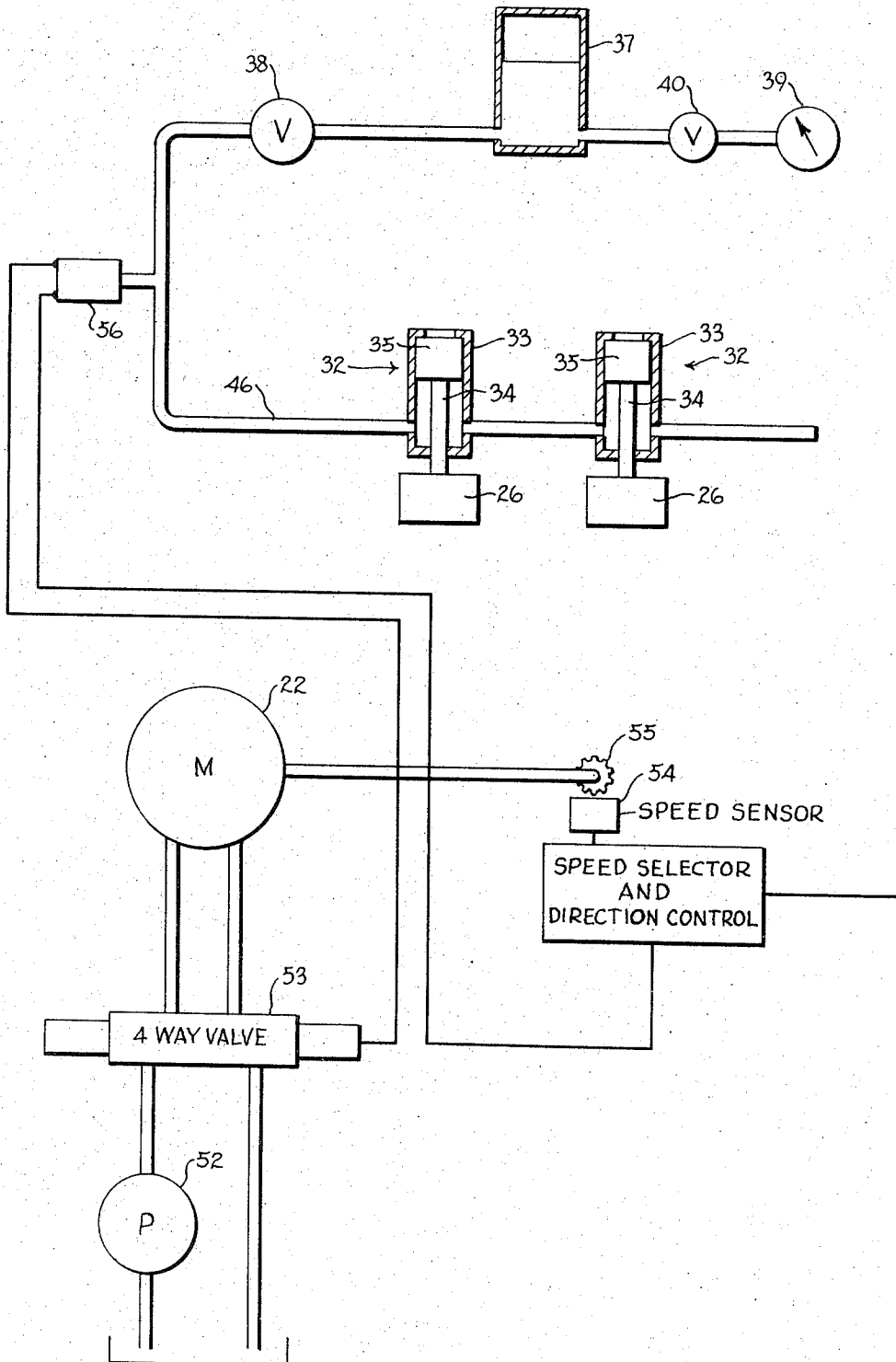


Fig. 5.



CONTROL FOR FLUID MOTOR

This invention, like that of the pending application Ser. No. 290,531, filed Sept. 20, 1972, now U.S. Pat. No. 3,814,532, of which this application is a division, relates generally to machines for compacting freshly laid asphalt paving, soil, and similar materials, in which the desired compaction is produced not only by traverse of a heavy drum or roller over the surface of the material but also by imparting vibration to the drum.

In the machine of the present invention vibration of the drum is produced by eccentric mass means revolving rapidly about an axis parallel to that of the drum as the drum rolls slowly over the surface of material being compacted. Such vibration of the drum is known to improve the ability of the machine to effect a complete and rapid compaction. It is also known that the amplitude of the vibration greatly affects the character of work accomplished by such a machine. High amplitude vibration quickly produces desired densities in the material being compacted but may leave its surface ridged or rough, while low amplitude vibration makes possible the production of exceptionally smooth surfaces. In prior machines, however, the amplitude of vibration was fixed, so that the machine was best suited either for preliminary compaction or for finish work, or — as was most often the case — the machine was designed for a compromise vibration amplitude that did not do either type of work completely satisfactorily.

The object of the invention claimed in the aforesaid patent is to provide a machine of the character described that accomplishes both preliminary and finish compaction with the utmost efficiency by reason of the fact that it can selectively produce either high or low amplitude vibration.

Another object of the invention claimed in the aforesaid patent is to provide a compacting machine in which the operator can cause the machine to operate with a high vibration amplitude during preliminary compacting, and then, as the job nears completion, can manually shift to low amplitude vibration without having to stop the machine or the vibration producing apparatus in order to do so, but which machine fails safe in that it cannot be shifted during operation, nor shift itself, from low amplitude to high amplitude vibration, thus avoiding any possibility of ruining a nearly complete, smoothly finished compacted surface.

In general the objects of the invention disclosed and claimed in the aforesaid U.S. Pat. No. 3,814,532, are achieved by providing a carrier which is driven for rotation about an eccentric axis that is parallel to the drum axis and preferably coincides with the drum axis, and an eccentric mass which is so mounted on the carrier as to be constrained to rotate therewith but to be movable radially toward and from the eccentric axis between defined limits. Centrifugally propelled outward motion of this movable mass is opposed by a releasable locking device, such as a hydraulic jack, which can also be mounted on the carrier for rotation therewith. By closure of a manually controlled valve connected with the jack, fluid can be prevented from escaping therefrom, thus confining the movable mass against centrifugally responsive outward motion. The movable mass is overbalanced by a second, diametrically opposite eccentric mass which is fixed to the carrier and which has a moment larger than the greatest moment exerted by the movable mass; hence vibration amplitude can be

decreased merely by opening the valve to permit the movable mass to move outward in response to centrifugal force. A pressure responsive cutout switch in the hydraulic circuit for the jack prevents overspeeding of the vibration producing means.

The rotatable carrier for the eccentric masses is driven by a fluid pressure motor at a manually selectable speed which enables the operator to set the frequency of the vibration, and it is the purpose and object of the invention claimed herein to provide an improved speed regulating system which not only maintains the speed of the motor and the rotatable carrier driven thereby at the selected magnitude, but also disrupts the connection between the fluid motor and the source of fluid pressure by which it is driven to automatically stop the motor in the event any portion of the speed regulating system fails.

With these observations and objectives in mind, the manner in which the invention achieves its purpose will be appreciated from the following description and the accompanying drawings which exemplify the invention, it being understood that changes may be made in the specific apparatus disclosed herein without departing from the essentials of the invention set forth in the appended claims.

The accompanying drawings illustrate one complete example of an embodiment of the invention constructed according to the best mode so far devised for the practical application of the principles thereof, and in which:

FIG. 1 is a front perspective view of a compacting machine embodying the principles of the present invention;

FIG. 2 is a view generally in longitudinal section through the drum of the compacting machine illustrated in FIG. 1;

FIG. 3 is a fragmentary sectional view on a larger scale, taken on the plane of the line 3—3 in FIG. 2 and showing the vibration producing means in its condition for producing vibration of high amplitude;

FIG. 4 is a view generally similar to FIG. 3, but with portions of the pressure responsive means shown broken away and illustrating the vibration producing means in its low amplitude condition;

FIG. 5 is a schematic diagram of the electrohydraulic control system in the machine of this invention whereby the operator of the machine is enabled to control both the amplitude and the frequency of vibration; and

FIG. 6 is a fragmentary sectional view taken on the plane of the line 6—6 in FIG. 3.

Referring now more particularly to the accompanying drawings, the numeral 5 designates generally a self-propelled compacting machine embodying the principles of this invention, here illustrated for purposes of example as having a single freely rotatable compacting drum 6 and a pair of power driven traction wheels 7 that are part of a tractor which mounts the power plant of the machine. The machine is steerable as it moves over material to be compacted by reason of the fact that the drum assembly and the tractor are articulately connected for relative motion about an upright axis. A steering wheel 8 that is conveniently accessible to the operator of the machine controls the mechanism by which steering is effected. Details of the steering mechanism and of the reversible propulsion system, being conventional and well known, are not shown.

As is also conventional, the drum assembly of the machine comprises a rigid yoke the arms 9 of which embrace the drum and have bearing mounting plates 10 connected thereto by elastic shock mounts 11 (see FIG. 2). The bearing mounting plates 10 are adjacent to the ends of the drum and carry the outer races of bearings 12 by which the drum is freely rotatably connected with the mounting plates, and through the shock mounts 11, with the arms 9 of the yoke. Vibration of the drum is therefore not imparted to the rest of the machine.

The drum is hollow, and in this case the vibration producing means, designated generally by 13, is housed within the drum and rotates about an eccentric axis which coincides with the axis of the drum. As the description proceeds it will be apparent that the invention is also applicable in principle to a machine of the type disclosed in U.S. Pat. No. 3,543,656, wherein the chassis comprises two drums, at least one of which is power driven, and wherein the axes of the vibration producing means are above the drums and parallel to their axes.

In the machine here illustrated the inner race of each drum bearing 12 is mounted on a hub portion of the axially outer wall 14 of a concentric housing unit designated generally by 15. There are two such housing units 15, one fixed to each of the end walls 16 of the drum, and each housing unit encloses one of the vibration producing means 13 of this invention. Each of these housing units also has an inner wall 18, spaced axially inwardly from its outer wall 14. The walls 14 and 18 of each housing unit carry the outer races of bearings 19 by which an eccentric carrier shaft 20 is journaled for rotation on an axis coinciding with that of the drum. Each shaft 20 carries eccentric masses that are described hereinafter, so that as it rotates vibration is imparted to the drum.

Secured to one of the bearing carrying plates 10 is a hydraulic drive motor 22 that is connected with one of the eccentric carrier shafts 20. Rotation is transmitted to the other shaft 20 by means of a concentric torque tube 23 that couples the shafts.

Referring now to FIGS. 3 and 4, the eccentric masses by which rotation of the shafts 20 is translated into vibration of the drum comprise a pair of eccentric mass elements carried by each carrier shaft 20. One of these mass elements, designated by 25, is fixed to the shaft at one side thereof and produces a substantially large eccentric moment; the other mass element 26 is constrained to rotate with the carrier shaft at the side thereof remote from the fixed mass element 25, but is so mounted as to be movable radially relative to the shaft between a high amplitude position in which it is near the eccentric axis (which position is illustrated in FIG. 3) and a low amplitude position (FIG. 4) in which it is at the defined outer limit of its radial relative motion. Even when the movable mass element 26 is at the outer limit of its motion, its eccentric moment is less than that of the fixed mass element 25, which is to say that the fixed mass element always overbalances the movable mass element 26. Consequently centrifugally responsive outward motion of the movable mass element 26 brings the moments exerted by the two mass elements more nearly into balance, decreasing the amplitude of the vibration produced by the apparatus.

In the specific embodiment here shown, the fixed mass element is generally U-shaped, with its bight portion formed as a hub 27 that closely embraces and is

non-rotatably secured to the shaft 20, as by weldments 28. The massive legs 29 of the U are disposed at equal distances to opposite sides of a plane that contains the eccentric axis, and they of course project to one side of the shaft to produce the larger eccentric moment. Projecting from the bight portion of the larger mass element, in the direction opposite to its legs, are a pair of parallel guide posts 30 upon which the smaller mass element 26 is slidable for its motion relative to the shaft. In its radially inner position the smaller mass element lies closely adjacent to the bight portion of the fixed mass element. The radially outermost position of the smaller mass element is defined by enlarged heads 31 on the outer ends of the guide posts 30.

When the vibration producing means is in operation, centrifugally responsive outward movement of the smaller mass element 26 can be prevented, to maintain high vibration amplitude, by means of readily releasable locking means, here shown as a fluid pressure responsive device 32 that comprises a single acting hydraulic jack. The cylinder 33 of the hydraulic locking means is mounted between the legs 29 of the fixed mass element, while the rod 34 of its piston 35 slidably extends through a hole 35' in the shaft 20, (FIG. 6) parallel to and between the guide posts 30. The outer end portion of the piston rod is secured, as by a nut 36, to the movable mass element, through which it also extends.

A hydraulic system that includes the rod end of each cylinder 33 also comprises, as illustrated in FIG. 5, an accumulator 37 and a control valve 38. Essentially, the hydraulic system is a closed duct with the series-connected cylinders 33 at one end thereof, the accumulator 37 at its other end, and the valve 38 arranged to control flow of fluid between the cylinders and the accumulator. The control valve may be directly manually operable, but preferably is a conventional normally open solenoid actuated valve remotely operated by an amplitude selector switch located on the operator's control panel (not shown).

When the valve 38 is closed, fluid is of course locked into the rod end of each cylinder, and if the movable mass elements are in their high amplitude positions, they will thus be confined against centrifugally responsive outward movement. When the amplitude selector switch is in its "high amplitude" position, the solenoid controlling the valve 38 is energized and the valve is closed. Conversely, when the amplitude selector switch is in its "low amplitude" position, the solenoid is de-energized and the valve is open. It follows, therefore, that the only time the valve is not open is when the amplitude selector switch is in its "high amplitude" position. When the vibration device is in operation and the valve 38 is open, centrifugal force on the movable mass elements 26 causes fluid to be forced out of the cylinders and into the accumulator 37. Air pressure in the accumulator, although substantially high, is not high enough to prevent such centrifugally responsive motion. Note that any failure of the hydraulic system or error in manipulation of the valve 38 can only result in low amplitude vibration.

As indicated by FIG. 5, the hydraulic system for the vibration device can also include a pressure gage 39 and a manually operable fill valve 40. The pressure gage can be removably connected to the fluid inlet port of the fill valve.

The accumulator 37 and the amplitude control valve 38 are of course mounted on relatively fixed parts of the machine, whereas the cylinders 33 rotate with the eccentric carrier shaft. Therefore the fluid passage connecting the cylinders with the relatively fixed units is in part defined by portions of each carrier shaft and also comprises a rotation accommodating union 42. Each shaft 20 has two axial bores 43 which define the fluid passage portions therein and which open to the outer ends of the shaft but terminate short of its medial portion through which the rod of the hydraulic jack passes. The inner end of each bore 43 is connected with the rod end of its adjacent cylinder 33 by means of a conduit 44, so that the two bores in each shaft are connected through the cylinder. The mouths of the bores at the axially inner ends of the shafts are communicated with one another through a suitable high pressure conduit or hose 45 inside the torque tube 23. The rotation accommodating union 42 is at the end of the drum that is opposite the hydraulic motor 22, with its stationary component on the adjacent bearing carrying plate 10, and it of course provides for communication between its adjacent shaft bore 43 and a duct 46 that is connected with the valve 38.

The hydraulic drive motor 22 has its stator mounted on the adjacent bearing carrying plate 10. The driving connection between its rotor and the adjacent eccentric carrier shaft 20 comprises a cross pin 47 that extends through a reduced diameter end portion 49 of that shaft. The adjacent axial bore 43' in the shaft is off-center but parallel to the shaft axis, opening to the shoulder defined by the reduced diameter shaft end portion 49, and the outer end of that bore 43' is closed by a bleed plug 50 that can be removed to vent air from the hydraulic system when it is being charged with fluid.

Referring back to FIG. 5, the hydraulic motor 22 is supplied with pressure fluid from a pump 52 under the control of the operator. For sake of simplicity, the control system by which the operator regulates the operation of the hydraulic motor has been illustrated as comprising a solenoid actuated four-way valve 53. The spool of the valve 53 is self centering and unless shifted occupies a neutral position at which the pump is disconnected from the motor. The spool is solenoid actuable to a pair of operating positions in which it provides for selected rotation of the motor in one direction or the other. Those skilled in the art will recognize that in lieu of the motor control system illustrated, a known form of closed loop arrangement may be used, wherein the motor and a swash plate type pump are connected to form a closed loop, and a servo valve controls the swash plate angle of the pump, the servo valve being actuated by a reversible torque motor under control of the operator.

Preferably the direction of rotation of the motor 22 is coordinated with the direction of traverse of the machine, so that the horizontal components of force due to the vibratory means will not oppose motion of the machine. Hence there can be a single instrumentality by which the operator can control both the direction in which the traction wheels are driven and the direction of rotation of the motor 22.

The operator can also have manual means for establishing a desired frequency of vibration, which the of course equal to the rotational speed of the motor 22 and can be controlled by adjustments of the four-way

valve 53 (or the servo valve of the closed loop system if it is used instead of the arrangement illustrated) to move it partway towards or away from one of its fully open operating positions. For the purpose of such frequency control a speed sensor 54 is mounted on the subframe side wall 10 at the side of the drum remote from the hydraulic motor 22, cooperating with a sprocket-like pulse generating device 55 that is secured to the adjacent eccentric carrier shaft 20 to rotate therewith. The speed sensor 54 is essentially a tachometer in which electrical pulses are generated at a rate dependent upon the speed of shaft rotation. Those skilled in the art will recognize that the operator's vibration frequency control adjusts the rate at which pulses are generated by apparatus that produces a reference pulse train, and that the four-way valve solenoids are energized in accordance with outputs based upon a comparison of the reference pulse train with the pulses generated by the tachometer. The apparatus for generating the reference pulse train, for effecting such comparison and for producing such output are all known.

If by any chance the vibration frequency control apparatus should fail, the machine of the present invention has means for preventing an excessively high frequency of vibration, comprising a normally closed pressure responsive switch 56 that is hydraulically connected in the amplitude control circuit comprising the single-acting jacks 32 and the accumulator 37. It will be evident that with increasing speed of the hydraulic motor 22 centrifugal force upon the movable mass elements 26 will cause increasing pressure upon the fluid in the amplitude control hydraulic system. The pressure responsive switch 56 is set to open when pressure in its hydraulic system exceeds a critical value, which critical value somewhat exceeds the highest vibration frequency for which the apparatus is adjustable. Such opening of the pressure responsive switch interrupts the energizing circuit for the solenoids of the four-way valve, permitting the spool of that valve to center itself, thereby preventing fluid from flowing to the motor 22 from the pump 52. The pressure responsive switch 56 thus provides an overspeed safety device for the vibration producing means.

From the foregoing description taken with the accompanying drawings it will be apparent that this invention provides vibration producing means in a compacting machine of the character described wherein both the amplitude and the frequency of vibration can be selected by the operator of the machine. It will also be apparent that the amplitude control apparatus can be shifted from high amplitude to low amplitude vibration while the machine is in operation, but not vice versa, and that such apparatus tends to fail safe because any failure thereof or misuse of the amplitude control will result in low amplitude vibration.

Those skilled in the art will appreciate that the invention can be embodied in forms other than as herein disclosed for purposes of illustration.

The invention is defined by the following claims:

We claim:

1. A control for a fluid motor, comprising:

- A. motor speed selecting means for effecting connection of the motor with a fluid pressure source for operation of the motor at a predetermined speed;
- B. a motor speed sensing device coacting with the motor speed selecting means to regulate the speed

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of the motor and maintain the same at its selected value;

- C. means driven by the motor for producing fluid pressure of a magnitude that varies with the speed of the motor, said means comprising
 - 1. eccentric mass means driven by the motor for rotation about an axis; and
 - 2. a hydraulic system including means to resist centrifugal force acting on said eccentric mass means during rotation thereof, and in which hydraulic pressure is developed by the response of the eccentric mass means to centrifugal force; and
- D. means rendered operative in consequence of rise in said pressure beyond a predetermined magnitude for effecting cessation of fluid flow to the motor so that in the event of failure of any portion of said motor speed regulating means the motor stops.
- 2. A control for a fluid motor, comprising:
- A. motor speed selecting means for effecting connection of the motor with a fluid pressure source for operation of the motor at a predetermined speed;
- B. a motor speed sensing device coacting with the motor speed selecting means to regulate the speed

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of the motor and maintain the same at its selected value;

- C. means driven by the motor for producing fluid pressure of a magnitude which varies with the speed of the motor, said means comprising
 - 1. a motor driven shaft;
 - 2. a weight;
 - 3. means connecting the weight with the shaft for rotation therewith and for limited motion laterally towards and from the shaft so that upon rotation of the shaft centrifugal force tends to displace the weight outwardly away from the shaft;
 - 4. hydraulic cylinder and rod means connected between the shaft and the weight to resist outward displacement of the weight, whereby centrifugal force acting on the weight pressurizes the hydraulic fluid in the cylinder; and
- D. means rendered operative in consequence of rise in pressure in the hydraulic cylinder beyond a predetermined magnitude for effecting cessation of fluid flow to the motor so that in the event of failure of any portion of said motor speed regulating means the motor stops.

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