Dec. 27, 1960 W. J. CASEY III 2,966,124

OSCILLATION CONTROL ARRANGEMENT

Filed Aug. 1, 1955 2 Sheets-Sheet 1

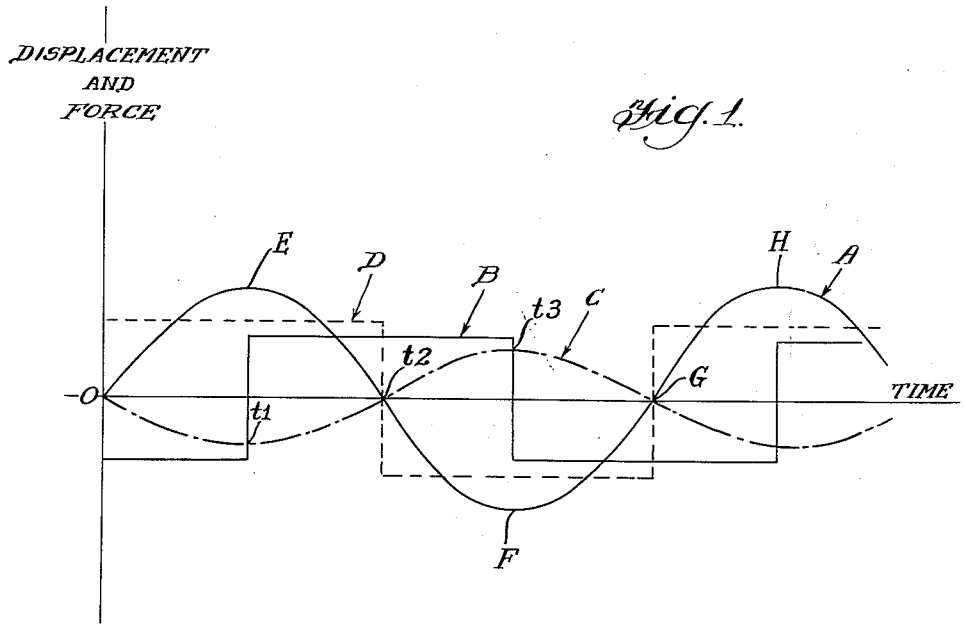
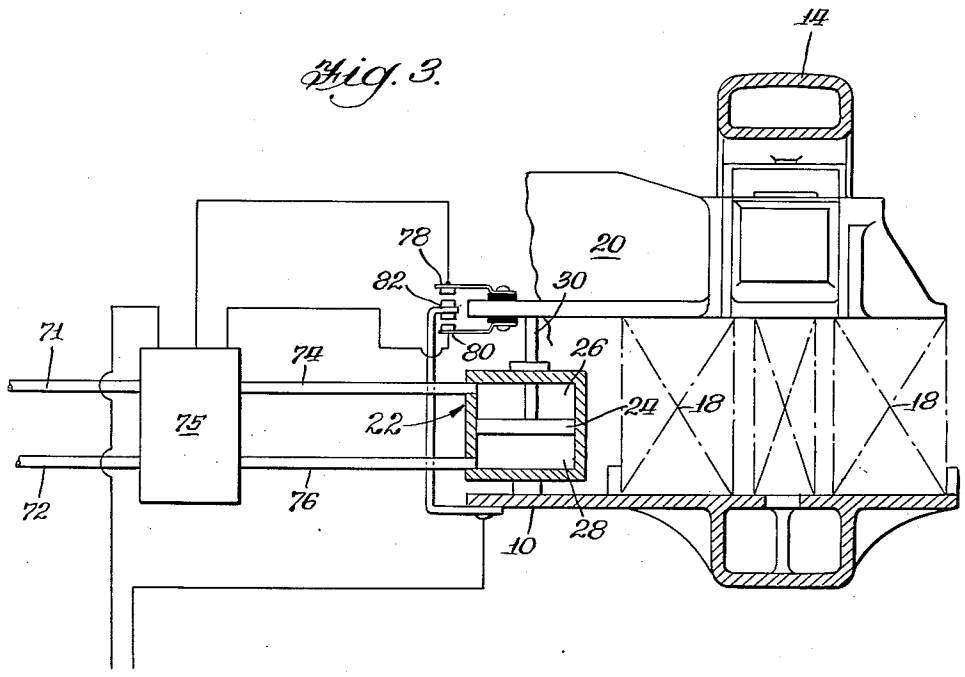


Fig. 1.

Fig. 3.



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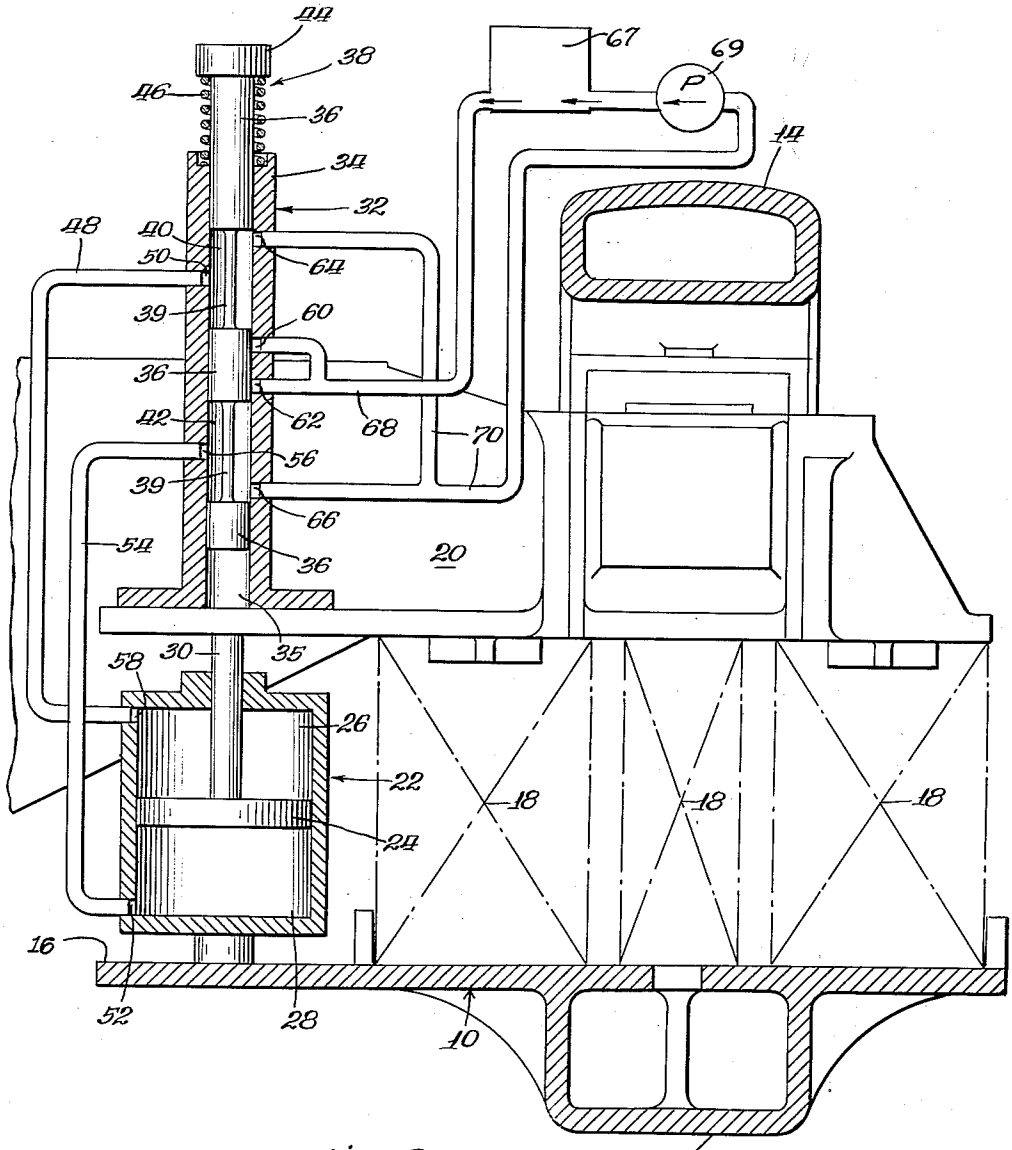


Fig. 2.

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OSCILLATION CONTROL ARRANGEMENT

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This invention relates to railway car trucks and more particularly to means for controlling oscillation of a sprung member relative to an unsprung member of a vehicle such as a railway car truck assembly.

As is well known in the railway truck art, the bolster which supports the car body extends transversely thereof, and the ends of the bolster are spring supported by spaced side frames, the ends of which provide journal bearings for wheel and axle assemblies. Thus the car body or vehicle is resiliently supported by the side frames which function as equalizers to transmit the car body load to the wheel and axle assemblies.

The carrying qualities of such a resiliently mounted vehicle are a function of both the spring rate and the damping rate. To provide good carrying qualities, it is necessary to control both the frequency of oscillation and the effects of impact on the sprung member. It is well known to those skilled in the art that softer long-travel springs generally afford better carrying qualities. This, however, is offset by violent oscillations which occur at certain speeds when resonant frequency occurs. In order that these violent oscillations may be controlled, it has been prior art practice to introduce a vibration damper into the system which offers a resistive force opposed to the movement of the sprung member. The larger the damping resistance, the fewer the number of oscillations traversed before the member comes to rest from an initial displacement. However, increasing damping resistance has the adverse effect of lessening the ability to properly cushion impact forces. Thus it has been necessary in practice to effect a compromise between oscillation control and control of impact forces. Practically, it has been found that there is a particular degree of damping which will produce the best ride qualities for a given oscillating mass and spring rate system. It should be noted that a conventional vibration damper or snubber has the effect of decreasing the magnitude of the oscillations while having little, if any, effect on the frequency of oscillation.

Considering the relationship between spring mass, spring rate, and carrying qualities, exclusive of damping, it is apparent that the largest possible ratio of mass to spring constant is the most desirable condition from a standpoint of both oscillation control and impact cushioning. This may be emphasized by considering a particular spring in combination with masses of different sizes. If a very small mass is used with this spring, displaced some small amount and allowed to vibrate freely, the rate of oscillation or frequency will be very high since the accelerations required in combination with the very small mass must be high in order to produce a force equal to that developed by the spring at any time. On the other hand, a relatively large mass supported by the same spring would oscillate more slowly because the accelerations required to balance the spring force would be less. From a standpoint of uncontrolled oscillations therefore, a system employing a larger mass would experience smaller accelerations and better ride quality.

Again from the standpoint of impact cushioning, a large mass has an advantage since a unit spring displacement over a very short interval of time will cause more flexing in the spring and less displacement of the mass.

Ideally then, a large mass to spring rate constant is most desirable for oscillation control and impact cushioning. Furthermore, the magnitude of the oscillations should be materially decreased and frequency of oscillation should be as low as possible. There are practical limitations, however, in any particular design both as to the mass which may be employed and also to the design of the particular spring. Therefore, the solution of the present invention lies in lowering the natural frequency of the system. This can be accomplished by applying control forces to the mass, or sprung member, in such a way that its effective inertia will be artifically increased. According to the invention, it has been discovered that these control forces should be opposed to acceleration at all times, and ideally, that the forces be proportional to the magnitude of acceleration.

Accordingly, a primary object of the present invention is to devise a means for improving the carrying qualities of a mass-supporting spring system without frictionally damping oscillation of either the mass or its supporting springs.

A more specific object of the invention is to control oscillation of such a mass by opposing acceleration thereof at all times, regardless of the direction of movement of the mass.

Another object is to provide means to apply a force to an oscillating mass to artificially increase its effective inertia.

It is also an object of this invention to increase the effective inertia of a resiliently supported mass by exerting a force on the mass proportional to its acceleration and in a direction opposite thereto.

Another object is to provide means to detect the acceleration of an oscillating mass and to apply a force to the mass opposed to the acceleration.

Another object is to increase the effective inertia of a resiliently supported mass by exerting forces thereon which are always opposed to the displacement of the mass from a mean position.

These and other objects and advantages inherent in the invention will become apparent from the following specification and the accompanying drawings, in which:

Figure 1 is a graph illustrating diagrammatically the relationship between the action of a conventional snubber and that of the invention;

Figure 2 is a cross-sectional view of a railway freight car side frame with the bolster fragmentarily shown in elevation and with a preferred form of the novel control means schematically illustrated, and

Figure 3 is a schematic drawing of another embodiment of the invention.

Referring to the drawings and particularly to Figure 1, it will be noted that time is measured along the horizontal axis and both displacement and force are measured along the vertical axis. No attempt has been made to calibrate the reference axes inasmuch as quantitative relationships would vary considerably depending on particular conditions. The purpose of the graph of Figure 1 is merely to draw a comparison between the invention and prior art. The body supporting bolster of a moving freight car oscillates upwardly and downwardly from a mean or neutral position relative to the side frame and wheels, and the time-displacement curve of the bolster or spring member approximates a sine curve depicted at A in Figure 1. The resistive force exerted in a conventional friction snubber is depicted by the line B. It will be noted that the force represented by line B is upward during any portion of the downward travel of the

bolster and is downward during any portion of the upward travel of the bolster. The resistive snubbing force always acts oppositely of the direction of motion, regardless of whether the displacement of the bolster is above or below the mean position.

The dot dash line C in Figure 1 represents the directional sense and qualitative magnitude of the vertical acceleration of the bolster. It will be noted that the maximum downward acceleration occurs at point *t*1, decreases to point *t*2, and at that instant becomes deceleration, indicated in Figure 1 as acceleration in an upward direction, and once again reaches its greatest magnitude at point *t*3. Thus acceleration is always directionally opposed to the displacement. It should be understood that throughout the disclosure, the word acceleration is used in its technical sense, meaning the rate of change in speed.

If a force is exerted on the sprung member, which is at all times opposite in sense to its acceleration, the effective inertia of the sprung member will be thereby increased. The direction of the control force applied according to the invention is designated in Figure 1 by the dotted line D. It will be seen from the graph that the force line D is out of phase with the force line B by one-half cycle and, therefore, during certain portions of the cycle acts in the same direction as the resistive snubbing force B and during other portions of the cycle acts in a direction opposite to the resistive snubbing force B. It is also demonstrated in Figure 1 that the force represented by line D is always opposed to the acceleration and, therefore, acts at all times in the same direction as the displacement of the sprung member, regardless of whether the sprung member is displaced upwardly or downwardly from its mean position. Furthermore, according to the invention, the applied force represented by line D may be of any desired magnitude which will preferably approximately maintain the amplitude of uncontrolled bolster displacement but will greatly decrease the frequency of oscillation, resulting in greatly improved carrying qualities as described heretofore. The conventional snubber, on the other hand, tends to decrease the amplitude of displacement but has little, if any, effect on the frequency of oscillation and, therefore, is limited in its ability to provide excellent carrying qualities.

Figure 2 shows one embodiment of the invention as applied to a freight car truck of conventional design. A side frame generally designated **10**, supported in the usual manner on a wheel and axle assembly (not shown), is comprised of spaced tension and compression members **12** and **14** shown in cross-section. The member **12** has an increased width portion **16** on its upper side which serves as a spring seat for supporting springs **18** on which is resiliently mounted one end of a conventional bolster **20** which in turn carries a freight car body (not shown). It will be understood that the opposite end of the bolster (not shown) is similarly supported on another side frame (not shown) in the usual manner and is preferably controlled in the same manner as that hereinafter described in connection with the bolster end shown in Figure 2. The above truck structure may be of any conventional design and is not described in detail inasmuch as it forms no part of the present invention.

Interposed between the sprung bolster member **20** and the unsprung side frame member **10** is a cylinder **22**. A piston **24** divides the cylinder cavity into an upper chamber **26** and a lower chamber **28** and is attached to the sprung member **20** for movement therewith by a piston rod **30**. In actual practice the connection of piston rod **30** to bolster **20** would be flexible to accommodate horizontal bolster oscillation laterally and longitudinally of the bolster. In Figure 2 this connection is schematically illustrated to simplify disclosure of the invention. Mounted on and movable with the member **20** is a sensing valve generally indicated at **32**. The valve comprises a valve body **34** secured to member **20** and having an opening or bore **35** which slidably receives enlarged portions forming lands or spools **36** of a valve stem generally indicated at **38**. The stem **38** is also provided with small diameter portions **39** which define an upper cavity or chamber **40** and a lower valve cavity or chamber **42**. The valve stem **38** is provided at its upper end with a shoulder portion **44** and is resiliently balanced in a neutral position by means of a spring **46** which is disposed between the upper surface of the valve body **34** and the shoulder of the portion **44**. A port **50**, communicating with cavity **40**, is interconnected with a port **58** communicating with chamber **26** by means of a flexible line **48**. Communication is also afforded between the cavity **42** and chamber **28** by means of ports **56** and **52** and flexible line **54**. A flexible line **68** is connected to a source of pressure fluid such as an accumulator **67** connected to the high pressure side of a pump **69**. The line **68** is connected to the valve opening **35** through high pressure ports **60** and **62**. A flexible exhaust line **70** connected to low pressure ports **64** and **66** exhausts these ports as, for example, to a reservoir of fluid which supplies the pump. The pump and associated apparatus may be supported from any convenient part of the vehicle. It should be noted that in neutral position of the bolster member **20**, which is illustrated in Figure 1 and at which time there is no relative acceleration between the member **20** and the member **10**, each of the low pressure ports communicates through associated lines with a chamber of the cylinder **22** and both of the high pressure ports from the pump are blocked off by the intermediate spool **36** of the stem **38**. At this time the control system will accommodate slow upward or downward movement of the sprung member **20** relative to the unsprung member **10** without developing pressure in either chamber of cylinder **22**.

Line A of Figure 1 schematically illustrates typical bolster oscillation in railway service when the railway car is in motion. For the purpose of the following discussion such oscillation is assumed to be vertical; however, it will be readily understood by one skilled in the art that the invention is equally applicable to horizontal oscillations of the bolster laterally as well as lengthwise thereof.

Assuming that the bolster is moving downwardly from point E in the oscillation cycle depicted by line A of Figure 1, it will be understood that the speed of downward bolster movement increases toward a mean or neutral bolster position at point *t*2, although downward bolster acceleration decreases toward *t*2 as depicted by line C of Figure 1, until such acceleration is zero at the instant the bolster reaches *t*2. During this phase of the cycle, the stem **38** lags behind the bolster causing high pressure fluid to be delivered from port **62** through line **54** to chamber **28** thereby developing an upwardly acting control force on the bolster which continues to act thereon as shown by dotted line D of Figure 1, until the bolster reaches mean or neutral position at *t*2. As the bolster moves downwardly from *t*2 toward point F of Figure 1, downward bolster movement accelerates upwardly (increasing) as depicted by line C, until the bolster reaches its lowermost position at point F whereupon maximum acceleration occurs as depicted at *t*3. Upward acceleration of bolster movement from point *t*2 causes the stem **38** to move downwardly relative to the bolster due to inertia of the stem thereby accommodating delivery of pressure fluid from port **60** through line **48** to chamber **26** exerting downward control force on the bolster as depicted by line D of Figure 1.

As the bolster moves upwardly from point F toward point G of Figure 1, upward bolster acceleration gradually decreases as depicted by line C until the bolster reaches mean or neutral position at G. During this phase of the cycle stem **38** lags behind the upwardly moving bolster so that downward control force on the bolster continues until it reaches point G whereupon upward bol-

ster movement accelerates downwardly (increasing) until the bolster reaches its uppermost position depicted at H in Figure 1. This acceleration causes valve stem **38** to move by its inertia upwardly relative to the bolster delivering high pressure fluid through port **62** to chamber **28** causing an upward control force to be exerted on the bolster. As previously described this upward control force against the bolster continues until it again reaches mean or neutral position on its downward stroke whereupon downward control force is developed against the bolster as previously described.

It will be understood that the sensing valve **32** is schematically shown in the drawings, and the spring **46** is a very delicate balancing spring which does not interfere with the above described action of the stem **38** caused by inertia forces thereon as the result of bolster oscillation in service as depicted by line A of Figure 1. Ordinarily, a commercial sensing valve such as that schematically illustrated at **32** would be provided with means (not shown) to damp oscillation of a balancing spring such as that depicted at **46**, to eliminate undesirable resonance in such oscillation. It should be noted that the directional sense of the control force is always opposite to the directional sense of the acceleration, regardless of the direction of the motion (Figure 1). A control force is exerted on the member **20** which is always directly opposed to its acceleration. The effect of this force will be to urge the member **20** to a comparatively large magnitude of oscillation against the counteracting forces of the springs **18** and to oppose the efforts of the springs to return the vehicle to the mean position. The net effect will be to greatly retard the frequency of oscillation and to provide thereby greatly improved carrying qualities as heretofore explained.

For purposes of illustration, the valve means **32** illustrated in Figure 2 will provide a constant upward or downward force in opposition to the acceleration, but it will be apparent to those skilled in the art that a commercial valve could be utilized which would regulate the forces to be directly proportional to the magnitude of the acceleration.

Figure 3 diagrammatically illustrates another embodiment of the invention. Referring to Figure 3, the sprung member **20** is, in the same manner as illustrated in Figure 2, resiliently supported by means of springs **18** on the unsprung member **10**. A cylinder **22** is mounted on member **10** and is divided into upper and lower chambers by the piston **24** which is mounted on and movable with the member **20**. A four-way electrically operated control valve **75** affords communication with chamber **26** by means of line **74** and with chamber **28** by means of line **76**. The valve **75** is a conventional type, such as an electro-pneumatic operating valve, Catalogue No. 711M427, Ross Operating Valve Company, Chicago, Illinois. The valve **75** is also connected by means of line **71** to a source of high pressure fluid and by means of line **72** to exhaust. An electrical contact **82** which is movable with the member **10** is disposed, when the member **20** is in neutral or mean position, between a pair of contacts **78** and **80** which are mounted on and movable with the member **20**. Relative downward displacement of the member **20** from mean position will cause the contacts **82** and **78** to complete the circuit to the valve **75** and high pressure fluid will be directed through line **74** into chamber **26**, and simultaneously, fluid will be exhausted from the chamber **28** through line **76** and low pressure line **72**. Relative upward displacement of the member **20** will similarly cause high pressure fluid to be directed into the chamber **28**. It is apparent then that in this embodiment, a force will be exerted on the piston **24** and thereby on the member **20** which will always be in the same direction as the displacement of the member **20** from its mean position. This force can also be represented by the line D in Figure 1. It is apparent that the manner of operation of this embodiment is exactly the same as the operation which is attained by the embodiment of Figure 2.

It will be understood that the control force applied to the bolster must be less than the gravitational force acting thereon at the top of its stroke and less than the force developed against the bolster by springs **18** at the bottom of the bolster stroke. Within this limitation, the force may be adjusted in any conventional manner to any desired valve or valves, preferably proportional to bolster acceleration.

Although the invention has been illustrated and described with regard to a freight car truck structure, it is readily apparent that the invention will be applicable to many types of machines or mechanisms which comprise a sprung and an unsprung member, such as automotive vehicles, railway draft gear, gun recoil mechanisms, and the like. It is also readily apparent that many modifications could be made to either of the embodiments shown in Figure 2 and Figure 3 without departing from the spirit and scope of the invention, and the inventor does not wish to be limited to the means illustrated and described in this application. For example, one such obvious modification would be to attach the piston **24** and piston rod **30** to the unsprung member **10** and to secure the cylinder **22** to the sprung member **20** for movement therewith.

I claim:

1. In a device for controlling oscillating movement of a resiliently supported mass by increasing the effective inertia thereof, a fluid responsive cylinder, a piston having a rod operatively associated with said mass for movement therewith, said piston dividing said cylinder into an upper and a lower chamber, pump means for supplying fluid to and exhausting fluid from said cylinder, valve means for directing said fluid selectively to one of said chambers and exhausting the other of said chambers, said valve means comprising a valve body secured to said mass for movement therewith, a port affording communication between said valve and said upper chamber, another port affording communication between said valve and said lower chamber, a pair of low pressure lines connecting said valve and said pump means, a pair of high pressure lines connecting said valve and said pump means, resiliently supported spool means in said valve disposed, when said mass is at rest, to close said high pressure ports, said spool means, when said mass accelerates downwardly, being movable to connect a high pressure port with said first mentioned port to accommodate fluid flow into said lower chamber and out of said upper chamber, said spool means, when said mass accelerates upwardly, being movable to connect a high pressure port with said other port to accommodate fluid flow into said upper chamber and out of said lower chamber, whereby force will be applied to said mass in opposition to the directional sense of the acceleration.

2. In a device for controlling oscillating movement of a resiliently supported mass by increasing the effective inertia thereof, a fluid responsive cylinder, a piston having a rod operatively associated with said mass for movement therewith, said piston dividing said cylinder into an upper and a lower chamber, pump means for supplying fluid to and exhausting fluid from said cylinder, valve means for directing said fluid selectively to one of said chambers and exhausting the other of said chambers, said valve means comprising a valve body secured to said mass for movement therewith, a first port affording communication between said valve and said upper chamber, a second port affording communication between said valve and said lower chamber, a high pressure line and a low pressure line from said pump means selectively communicable with said first port, a high pressure and a low pressure line from said pump means selectively communicable with said second port, spool means in said valve, said spool means being movable to establish, when said mass accelerates downwardly, communication between a

high pressure line and said second port, said spool means being movable to establish, when said mass accelerates upwardly, communication between a low pressure line and said second port and communication between a high pressure line and said first port.

3. In an oscillation control arrangement for a railway car truck, the combination of: spaced side frames; a bolster having its ends spring supported by the side frames; and means for developing force against the bolster during oscillation thereof opposed at all times to the directional sense of bolster acceleration.

4. In an oscillation control arrangement, the combination of: an oscillating member; spring means supporting said member; and means for developing control force against said member during oscillation thereof opposed at all times to the directional sense of the acceleration of said member.

5. In a device for controlling the acceleration of an oscillating resiliently supported mass, the combination of: means to apply an opposing force to said mass, said means comprising a cylinder operatively associated with said mass, said cylinder having a piston separating said cylinder into an upper and a lower chamber; and other means responsive to said acceleration to control the direction of said opposing force, said responsive means comprising a valve movable with said mass, said valve having ports affording communication between said valve and said chambers, respectively, said valve having high pressure and low pressure lines communicable therewith, and a freely suspended spool disposed within said valve to selectively open and close said lines.

6. In a device for controlling the movement of an oscillating resiliently supported mass, the combination of: a fluid responsive cylinder having a piston operatively associated with said mass, means responsive to downward acceleration of said mass to urge said piston in an upward direction, said means being responsive to upward acceleration of said mass to urge said piston in a downward direction.

7. A device according to claim 6, wherein said means comprises a supply of pressure fluid and valve means comprising a body secured to said mass, a plurality of lines affording communication between said valve and said cylinder, and a freely floating valve spool which upon acceleration of said mass and said valve body lags behind said body thereby controlling the fluid flow through said lines.

8. In a device for controlling the movement, relative to a mean position, of a resilient supported mass, a fluid responsive cylinder having a piston dividing the cylinder into upper and lower chambers, a piston rod operatively associated with said piston and said mass, means responsive to downward acceleration of said mass to cause fluid to flow into said lower chamber and simultaneously to exhaust the fluid from said upper chamber, said means being responsive to upward acceleration of said mass to cause fluid to flow into said upper chamber and simultaneously to exhaust the fluid from said lower chamber.

9. A device according to claim 8, wherein said means comprises a valve having a body secured to said mass and movable therewith, a valve spool slidably received in said body and resiliently supported in a neutral position therein, said spool having port opening and port closing means thereon operable when there is movement of said spool relative to said body, said relative movement resulting from the inertia time lag in the movement of said spool in following the movement of said mass.

10. A device according to claim 9, wherein said opening and said closing means comprise spaced portions on said spool of smaller diameter than other portions of said spool, said smaller diameter portions accommodating flow of fluid through an associated port when aligned therewith, and said larger diameter portions preventing flow of fluid through an associated port when aligned therewith.

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