(12)   EUROPEAN PATENT SPECIFICATION

(54) **Wheel assembly suspending**

Radeinheit-Aufhängung

Suspension d'une structure de roue

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Remarks:
     The file contains technical information submitted
     after the application was filed and not included in
     this specification

EP 0 397 488 B1

Description

[0001] The present invention relates in general to wheel assembly suspending and more particularly concerns novel apparatus and techniques for absorbing energy from a vehicle wheel assembly moving over a rough surface so as to facilitate significantly reducing forces transmitted to the sprung vehicle body supported on the unsprung wheel assembly.

[0002] It is known to use electrically controlled active suspension members, such as a shock absorber containing gas or fluid whose pressure may be electrically controlled to achieve a predetermined characteristic, such as a hard or soft ride, while avoiding bottoming. For example, reference is made to the article in POPULAR SCIENCE for August 1985 pp. 66-69 entitled "Smart suspensions-handling tailored to road and driver." Reference is also made to the article entitled "Optimum Vehicle Suspension With a Damped Absorber" by Ghoneium and Metwalli in Transactions of the ASME for June 1984 p. 148 and the article entitled "On the Application of Optimum Damped Absorber to Vehicle Suspension" by Ghoneium and Cheema in Transactions of the ASME for March 1986 p. 22. Reference is also made to European Application No. 251603 and Australian Application 87174560 believed to have been published on or about January 7, 1988, claiming priority of United Kingdom Application No. 86/15427 filed June 24, 1986. These systems have not been widely implemented on commercial vehicles.

[0003] A paper entitled "Active Suspensions with Vibration Absorbers and Optimal Output Feedback Control" by A. G. Thompson, B. R. Davis and F. J. M. Salzborn copyright 1984 by the Society of Automotive Engineers, Inc. discloses that in active suspensions for road vehicles with full state feedback the performance index may be improved by the use of dynamic vibration absorbers applied to the axles. The paper describes a suspension assumed to be of the electrohydraulic type, and states that the addition of the vibration absorber should result in a significant hydraulic power saving. FIG. 2 of that paper discloses a shock absorber across the active member comprising an actuator. The paper also teaches a ratio of damping mass to unsprung mass should preferably be between 0.2 and 0.5 in the conclusion and states that the improvement of the axle response which can be achieved by increasing the mass ratio is quite small.

[0004] It is an important object of this invention to provide an improved vehicle wheel assembly.

[0005] The literature on vehicle suspension systems sometimes uses different terms to describe similar systems, and sometimes uses the same term to describe different systems. In order to clearly understand this invention and to properly distinguish it from approaches in the prior art, there follows certain definitions.

Definition of a linear system

[0006] Consider an operation T which, when applied to a time function x(t) yields a unique time function T[x(t)]. Then T applied to the time function x_1 yields $T(x_1)$, and T applied to x_2 yields $T(x_2)$. The operation T is linear if and only if

$$T(ax_1 + bx_2) = aT(x_1) + bT(x_2)$$

for all complex time functions x_1 and x_2 and for all complex constants a and b. A linear system is one whose excitation x and response y are related by a linear operation T(x) = y over the usable range of the variable x and y in the system. See, Bose and Stevens, "Introductory Network Theory" Harper and Row (1965).

Definition of a nonlinear system

[0007] A nonlinear system is any system whose excitation and response are not related by a linear operation.

Definition of a passive element

[0008] A passive element is an element for which the energy that can be extracted from its port is less than or equal to the energy that was previously supplied to its port from an external source.

Definition of a passive system

[0009] A passive system is a system composed of all passive elements.

Definition of a controlled element

[0010] A controlled element is an element in which either variable associated with the output port or the relation between these variables can be affected by input signals at the control port or ports.

Definition of a passive controlled element

[0011] A passive controlled element is a controlled element for which the energy that can be extracted from its output port is less than or equal to the energy that was previously supplied to that port from an external source.

Definition of an active controlled system

[0012] An active controlled system is a system that can accept power at some frequency or frequencies at one or more power supply ports and can deliver power at different frequency or frequencies at an output port under the control of signals at one or more input ports called control ports.

[0013] The linearity or nonlinearity of an active controlled system is determined by applying the definition of a linear system to the signals at the input and output ports.

Definition of a quasi-static controlled system

[0014] A quasi-static controlled system is a controlled system in which the bandwidth of the controlling signals is much less than that of the signals at the output port.

Definition of a dynamic controlled system

[0015] A dynamic controlled system is a controlled system in which the bandwidth of the controlling signals is comparable to that of the signals at the output port.

[0016] Prior art suspension systems employing wheel dampers that are described in the literature fall into the following two categories:

    1. Passive linear system exemplified by the system described in the aforesaid article of H. Ghoneiun, S. M. Metualli.
    2. Active linear systems exemplified by the systems described in the aforesaid article of A. G. Thompson, B. R. Davis and F. J. M. Salzborn.

[0017] In both these categories of suspension systems the optimization criterion resulted in significant resistive force (force proportional to the difference in velocities of the sprung and unsprung masses) between the unsprung and sprung masses. This resistive force transmits undesirable force to the sprung mass while providing the benefits of damping to the fundamental resonance of the sprung mass with the suspension spring and partial damping to the tire-wheel resonance of the unsprung mass with the tire compliance. The wheel damper in prior art systems was thus called upon only to supply that portion of the damping to the tire-wheel resonance that was not supplied by the resistive force between the unsprung and sprung masses. The result was that the wheel damper did not produce a significant enough performance improvement to justify its cost, and the concept has not found commercial application in automobiles.

[0018] The present invention deals with the use of a damper in the specific class of nonlinear, active, dynamically controlled systems for vehicle suspension. Only with this class of systems is it possible to have essentially no net resistive force between the unsprung and sprung masses and to simultaneously maintain control of the vehicle for large vertical road excursions. The absence of the net resistive force provides a desired significant reduction of the acceleration of the sprung mass. However, this benefit can only be realised if a wheel damper is employed in order to remove energy from the resonance of the unsprung mass with the tire. Unlike in the prior art systems which were linear, the wheel damper when used in the present system must supply essentially all the damping to the resonance of the tire with the unsprung mass. This requires that the ratio of the mass of the damper to the unsprung mass be larger than in the prior art linear systems in which the wheel damper provided only partial damping. For passive linear systems it has been shown in the aforesaid Ghoneiun et al article that an optimum performance results from a value of this mass ratio of 0.1. And for active linear systems it has been shown in the aforesaid Thompson et al article that the optimum value of this mass ratio is between 0.2 and 0.5. It has been discovered that in the case of active non-linear dynamically controlled systems the optimum value of this mass ratio, considering performance and cost, is substantially 1.0 and the range of practical values of this mass ratio is between 0.5 and 1.5.

[0019] According to the present invention, there is provided suspension apparatus for a vehicle having a sprung mass and an unsprung mass including a wheel and a wheel support member, the suspension apparatus comprising:

    an active dynamically controlled suspension system intercoupling the sprung mass and the unsprung mass;
    a wheel damper comprising a damping mass and a damping device connecting the damping mass to the wheel support member, the damping mass being relatively movable with respect to the wheel support member such that there is no transfer of force between the unsprung mass and the sprung mass through the wheel damper; and wherein the system is free of a damping element in parallel with an active component of the system characterised in that:
    the active dynamically controlled suspension system is a non-linear controlled system; and
    the ratio of damping mass to unsprung mass is within the range of 0.5 to 1.5. Preferably the ratio is substantially unity with the unsprung mass of the wheel assembly, typically comprising tire, wheel and axle.

[0020] Numerous other features, objects and advantages of the invention will become apparent from the following specification when read in connection with the accompanying drawings, in which:

    FIG. 1 is a combined block-diagrammatic representation of a typical prior art automobile wheel suspension;
    FIG. 2 is a schematic circuit diagram of the mechanical system of FIG. 1;
    FIG. 3 is a graphical representation of the response of the prior art mechanical circuit of FIG. 2;
    FIG. 4 is a combined block-diagrammatic representation of a vehicle wheel assembly damped accord-

ing to the invention;

FIG. 5 is a schematic circuit diagram of the mechanical system of FIG. 4;

FIG. 6 is a graphical representation of the response of the mechanical system of FIGS. 4 and 5; and

FIG. 7 is a diagrammatic representation of another embodiment of the invention.

[0021] With reference now to the drawing and more particularly FIG. 1 thereof, there is shown a combined block-diagrammatic representation of a typical prior art wheel suspension. The sprung mass of the vehicle 11, typically comprising about one-fourth the mass of the vehicle including the frame and components supported there-on, is connected to the wheel assembly 12 by spring 13 in parallel with shock absorber 14. Wheel support assembly 12 carries on axle 12A wheel 15 including hub 15B and tire 15C. The wheel, brake assembly, if it moves vertically with the wheel, and wheel support assembly are charac-terized by an unsprung mass M_W. Tire 15C has a compliance C_T. Tire 15C is shown supported on road 16. Spring 13 is characterized by a compliance C_S, and shock absorber 14 is characterized by a mechanical resistance R_{SH}.

[0022] Referring to FIG. 2, there is shown a schematic circuit diagram of the mechanical system of FIG. 1. v_R represents the vertical velocity of the portion of tire 15C in contact with road 16 and is applied to the tire compliance 15C' in series with spring compliance 13' in series with the vehicle sprung mass portion 11'. The spring compliance 13' is shunted by the shock resistance 14'. The series combination of compliance 13' in shunt with the shock resistance 14' and the sprung mass 11' is shunted by the wheel unsprung mass 15'.

[0023] This mechanical circuit is characterized by a resonance where the tire compliance C_T resonates with the unsprung mass M_W and corresponds to a high vertical velocity of wheel 15 at this resonant frequency. Referring to FIG. 3, there is shown a graphical representation of the ratio of the vertical wheel velocity, v_W, to v_R, the vertical velocity of the tire portion contacting the road, as a function of frequency showing the peak at $\omega_1 = 1/(C_T M_W)^{\frac{1}{2}}$.

[0024] A disadvantage of this resonance is that in order to reduce the wheel excursion at this resonance ω_1, shock 14 must transmit considerable forces to sprung mass 11. Furthermore, in an electrically controlled system, this resonance significantly increases the peak energy requirements imposed upon such a system attempting to resist transmission through a controlled suspension member. In a conventional system, this resonance constrains the mechanical impedance presented by spring 13 and shock 14 to be suffiently high to resist transmission of this resonant force component and thereby provide a stiffer ride than would ordinarily be desired.

[0025] Referring to FIG. 4, there is shown a combined block-diagrammatic representation of a system accord-

ing to the invention incorporating a damping assembly that appreciably reduces the resonance effects described above. Sprung mass portion 11 is connected to wheel assembly 12 by suspension member 17, which may be an electrically controlled suspension. A damping assembly comprising damping mass 21 is connected to wheel assembly 12 by damping spring 22 in parallel with damping resistance 23, which may be a conventional shock absorber. The system is free of a damping element in parallel with an active component of the system.

[0026] Referring to FIG. 5, there is shown a schematic circuit diagram of the mechanical system of FIG. 4. The only changes over the circuit of FIG. 2 are the mechanical circuit comprising the series combination of damping spring compliance 22' shunted by damping resistance 23' and damping mass 21' in shunt with unsprung mass 15' and suspension member 17. The result of this arrangement is that the peak shown in FIG. 3 is now significantly damped. If the mass of damping mass 21' is made substantially equal to the unsprung mass M_W, the resulting response is staggered tuned peaks only four db above unity as shown in FIG. 6. FIG. 6 shows a graphical representation of the ratio of vertical wheel velocity v_W to v_R, the vertical velocity of the tire portion contacting the road, as a function of frequency. A typical frequency for the unsprung mass tire compliance resonance is 15 Hz largely established by the tire compliance and unsprung mass. Damping mass 21' absorbs most of the wheel resonant energy. A preferred range for the ratio of damping mass 21' to unsprung mass is 0.5 to 1.5 with a ratio of substantially unity preferred.

[0027] Referring to FIG. 7, there is shown a combined block-diagrammatic representation of another embodiment of the invention in which a sealed cylinder 24 is fastened to the wheel assembly member 12, and the damping mass 21 is suspended from the damping spring 22' inside damping fluid 23' in sealed cylinder 24.

[0028] The invention has been described with respect to a single wheel to avoid obscuring the principles of the invention. The principles are applicable to other wheels on a vehicle. The suspension system may use an electrically dynamically controlled non-linear active system.

Claims

1. Suspension apparatus for a vehicle having a sprung mass (11) and an unsprung mass including a wheel (12) and a wheel support member (12A), the suspension apparatus comprising:

an active dynamically controlled suspension system (17) incercoupling the sprung mass (11) and the unsprung mass (12,12a);
a wheel damper (21,22,23) comprising a damping mass (21) and a damping device (22,23) connecting the damping mass (21) to

the wheel support member (12A), the damping mass (21) being relatively movable with respect to the wheel support member (12A) such that there is no transfer of force between the unsprung mass and the sprung mass through the wheel damper; and wherein the suspension apparatus is free of a damping element in parallel with the active controlled suspension system. characterised in that:

the active dynamically controlled suspension system is a non-linear controlled system; wherein the ratio of the mass of the wheel damper to the unsprung mass is in the range of 0.5 to 1.5

2. Apparatus according to claim 1, wherein the damping device is a damping spring (22) connecting the damping mass to the wheel support member (12A), and the wheel damper further includes:

a mechanical resistance (23) in parallel with the damping spring.

3. Apparatus according to claim 2, wherein the wheel damper comprises a sealed cylinder (24) attached to the wheel support member,

the mechanical resistance comprises damping fluid (23') inside the sealed cylinder, and the damping spring comprises a spring (22') inside the sealed cylinder.

4. Apparatus according to claim 3, wherein the spring (22') is suspended from an end of the sealed cylinder.

5. Apparatus according to claim 3, wherein the damping mass is a mass (21) suspended from the spring inside the sealed cylinder.

6. Apparatus according to any of claims 1 to 5, wherein the ratio of the mass of the wheel damper to the unsprung mass is substantially 1.0.

7. Apparatus according to claim 1, wherein the wheel damper also includes at least a damping spring (22).

8. Apparatus according to claim 7, wherein the wheel damper further includes at least a damping resistance (23).

Patentansprüche

1. Aufhängungsvorrichtung für ein Fahrzeug mit einer gefederten Masse (11) und einer ungefederten Masse, welche ein Rad (12) und ein Radträgerteil (12A) umfaßt` wobei die Aufhängungsvorrichtung

aufweist:

ein aktiv dynamisch gesteuertes Aufhängungssystem (17) , welches die gefederte Masse (11) und die ungefederte Masse (12, 12a) miteinander koppelt; einen Raddämpfer (21, 22, 23) mit einer Dämpfungsmasse (21) und einer Dämpfungseinrichtung (22, 23), welche die Dämpfungsmasse (21) mit dem Radträgerteil (12A) verbindet, wobei die Dämpfungsmasse (21) gegenüber dem Radträgerteil (12A) so beweglich ist, daß über den Raddämpfer keine Kraft zwischen der ungefederten Masse und der gefederten Masse übertragen wird, und wobei die Aufhängungsvorrichtung kein Dämpfungselement parallel zu dem aktiv gesteuerten Aufhängungssystem aufweist; dadurch gekennzeichnet, daß:

das aktiv dynamisch gesteuerte Aufhängungssystem ein nichtlinear gesteuertes System ist; das Verhältnis der Masse des Raddämpfers zur ungefederten Masse im Bereich von 0,5 bis 1,5 liegt.

2. Vorrichtung nach Anspruch 1, bei welcher die Dämpfungseinrichtung eine Dämpfungsfeder (22) ist, die die Dämpfungsmasse mit dem Radträgerteil (12A) verbindet, und der Raddämpfer weiter umfaßt:

einen mechanischen Widerstand (23), welcher parallel zu der Dämpfungsfeder angeordnet ist.

3. Vorrichtung nach Anspruch 2, bei welcher der Raddämpfer einen abgedichteten Zylinder (24) aufweist, der an dem Radträgerteil befestigt ist,

der mechanische Widerstand eine Dämpfungsflüssigkeit (23') innerhalb des abgedichteten Zylinders enthält, und die Dämpfungsfeder eine Feder (22') innerhalb des abgedichteten Zylinders aufweist.

4. Vorrichtung nach Anspruch 3, bei welcher die Feder (22') an einem Ende des abgedichteten Zylinders aufgehängt ist.

5. Vorrichtung nach Anspruch 3, bei welcher die Dämpfungsmasse eine Masse (21) ist, die an der Feder innerhalb des abgedichteten Zylinders aufgehängt ist.

6. Vorrichtung nach einem der Ansprüche 1 bis 5, bei welcher das Verhältnis zwischen der Masse des Raddämpfers zu der ungefederten Masse im wesentlichen 1,0 beträgt.

7. Vorrichtung nach Anspruch 1, bei welcher der Raddämpfer auch wenigstens eine Dämpfungsfeder (22) enthält.

8. Vorrichtung nach Anspruch 7, bei welcher der Raddämpfer ferner wenigstens einen Dämpfungswiderstand (23) enthält.

Revendications

1. Appareil de suspension d'un véhicule ayant une masse suspendue (11) et une masse non suspendue qui comprend une roue (12) et un organe (12A) de support de roue, l'appareil de suspension comprenant :

> un système (17) de suspension commandé dynamiquement de manière active, couplant la masse suspendue (11) à la masse non suspendue (12, 12a),
> un amortisseur (21, 22, 23) de roue comprenant une masse d'amortissement (21) et un dispositif d'amortissement (22, 23) raccordant la masse d'amortissement (21) à l'organe de support de roue (12A), la masse d'amortissement (21) étant mobile par rapport à l'organe de support de roue (12A) afin qu'il n'existe aucun transfert de force entre la masse non suspendue et la masse suspendue par l'intermédiaire de l'amortisseur de roue, et dans lequel l'appareil de suspension est dépourvu d'élément d'amortissement monté en parallèle avec le système de suspension à commande active, caractérisé en ce que :
> le système de suspension à commande dynamiquement active est un système commandé non linéaire,
> dans lequel le rapport de la masse de l'amortisseur de roue à la masse non suspendue est compris entre 0,5 et 1,5.

2. Appareil selon la revendication 1, dans lequel le dispositif d'amortissement est un ressort d'amortissement (22) qui raccorde la masse d'amortissement à l'organe de support de roue (12A), et l'amortisseur de roue comporte en outre :

> une résistance mécanique (23) placée en parallèle avec le ressort d'amortissement.

3. Appareil selon la revendication 2, dans lequel l'amortisseur de roue comprend un cylindre étanche (24) fixé à l'organe de support de roue,

> la résistance mécanique comprend un fluide d'amortissement (23') placé dans le cylindre étanche, et
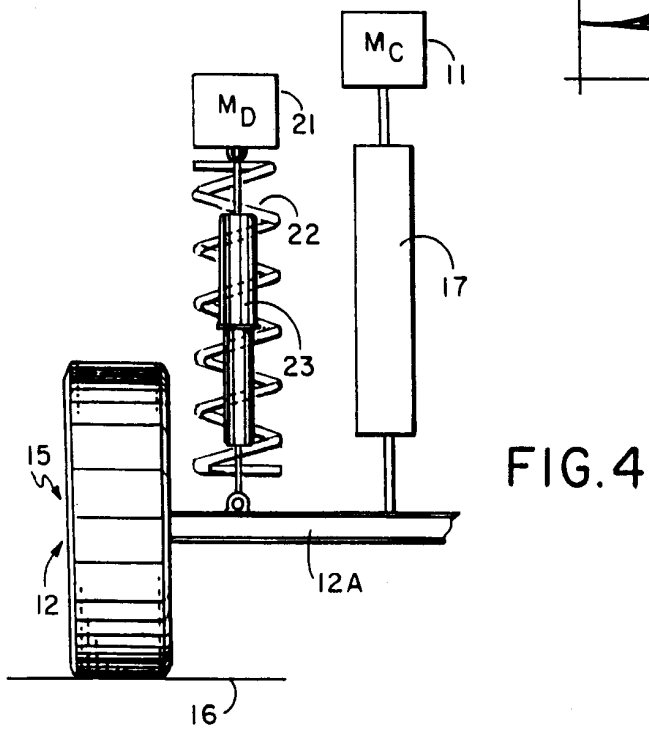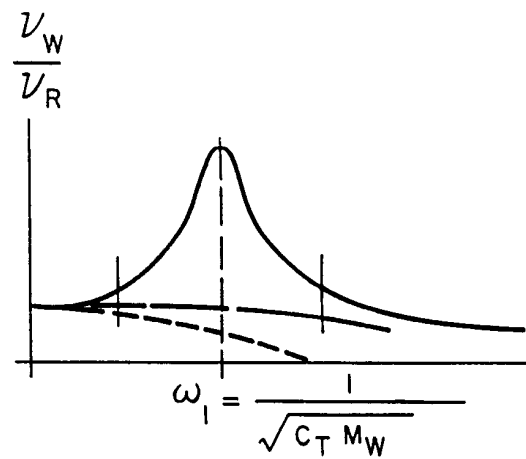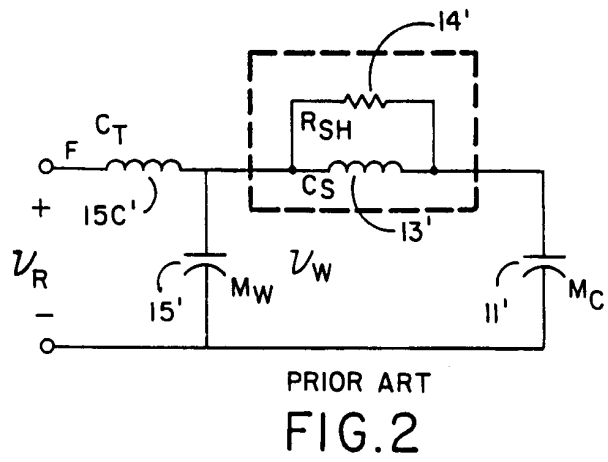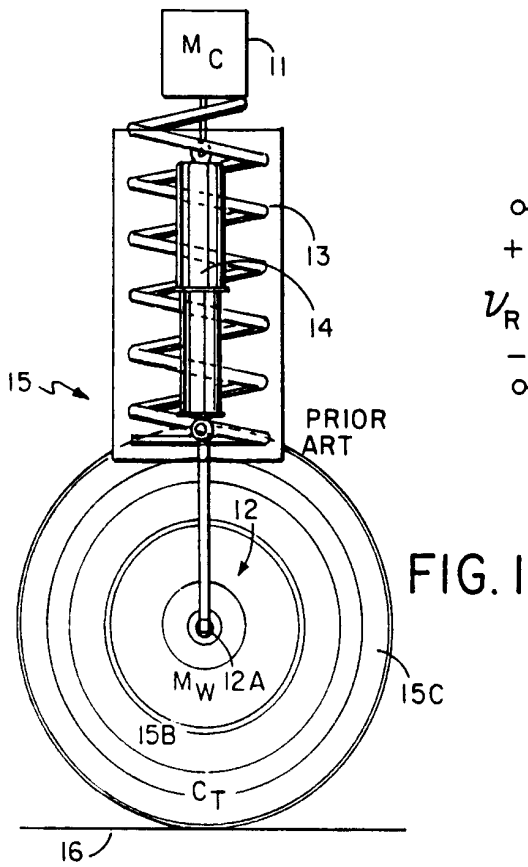> le ressort d'amortissement comprend un ressort (22') placé dans le cylindre étanche.

4. Appareil selon la revendication 3, dans lequel le ressort (22') est suspendu à une extrémité du cylindre étanche.

5. Appareil selon la revendication 3, dans lequel la masse d'amortissement est une masse (21) suspendue au ressort à l'intérieur du cylindre étanche.

6. Appareil selon l'une quelconque des revendications 1 à 5, dans lequel le rapport de la masse de l'amortisseur de roue à la masse non suspendue est pratiquement égal à 1,0.

7. Appareil selon la revendication 1, dans lequel l'amortisseur de roue comporte aussi au moins un ressort d'amortissement (22).

8. Appareil selon la revendication 7, dans lequel l'amortisseur de roue comporte en outre au moins une résistance d'amortissement (23).
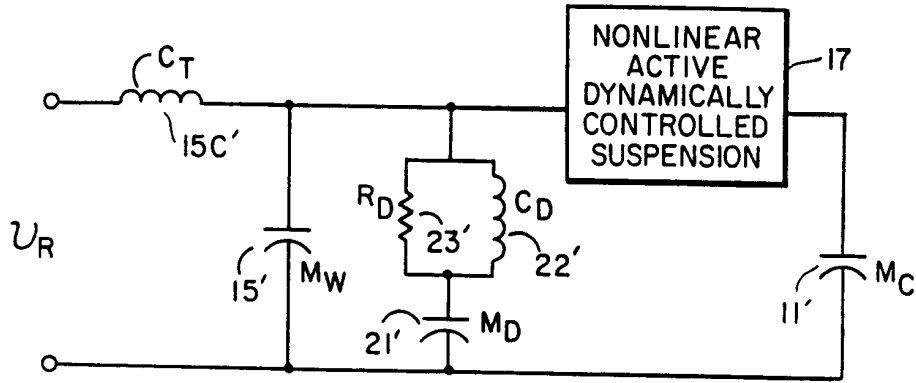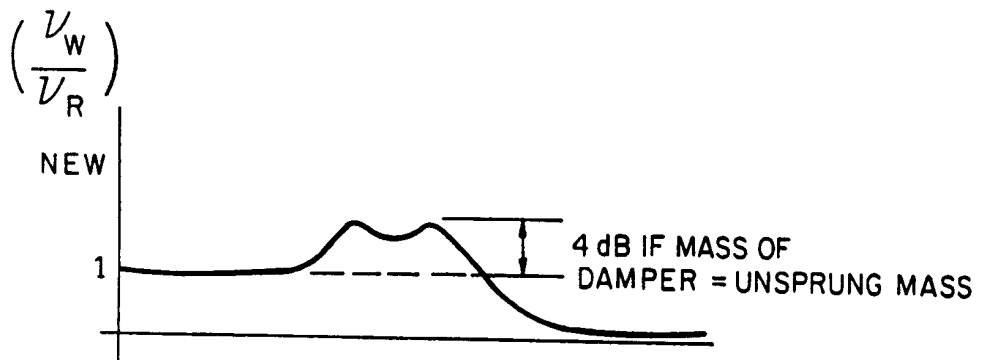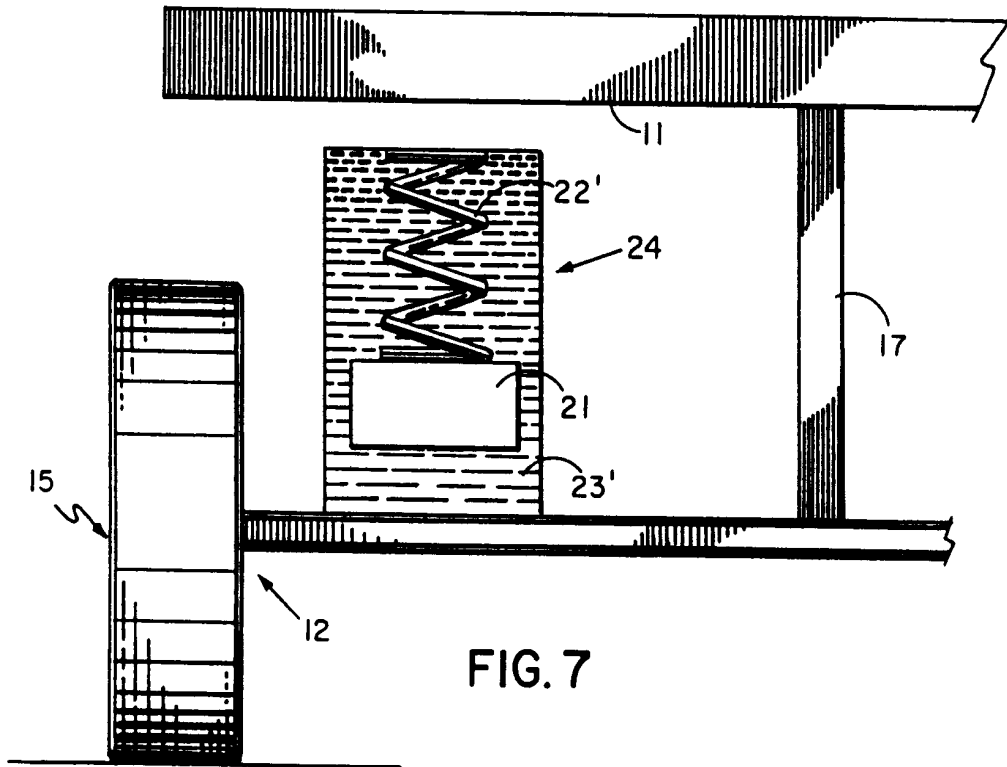
FIG. 1

PRIOR ART

FIG.2

$$\omega_1 = \frac{1}{\sqrt{C_T\, M_W}}$$

FIG. 3

FIG.4

FIG. 5

$\mathcal{U}_R$

C_T

15C'

NONLINEAR
ACTIVE
DYNAMICALLY
CONTROLLED
SUSPENSION

17

R_D 23' C_D 22'

15' M_W

21' M_D

11' M_C

FIG. 5

FIG. 6

$\left(\dfrac{\mathcal{U}_W}{\mathcal{U}_R}\right)$

NEW

1

4 dB IF MASS OF
DAMPER = UNSPRUNG MASS

FIG. 6

FIG. 7

22'

24

21

17

23'

11

15

12

FIG. 7