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Van der Burgt et al.
[11] 3,767,181
[45] Oct. 23, 1973

[54] SPRING SYSTEM

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[21] Appl. No.: **160,332**

[30] **Foreign Application Priority Data**

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July 7, 1970 Netherlands 7009994

[52] **U.S. Cl.** **267/136,** 267/152
[51] **Int. Cl.** **F16f 7/00**
[58] **Field of Search** 267/136, 152, 175, 267/176, 177, 166

[56] **References Cited**

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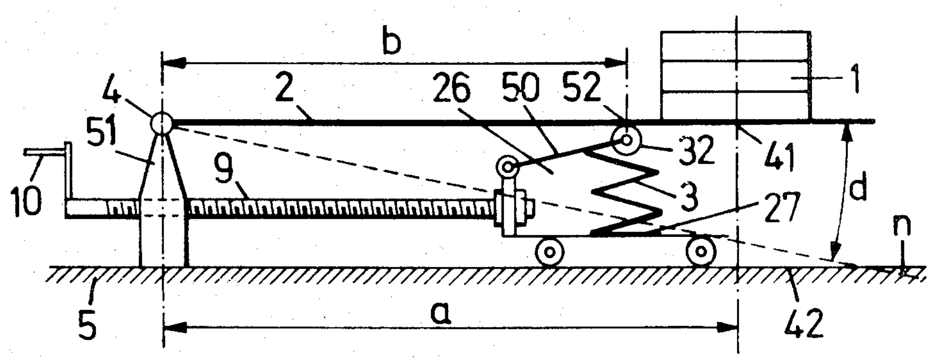
Primary Examiner—James B. Marbert
Attorney—Cushman et al.

[57] **ABSTRACT**

A spring system is proposed which comprises a lever mechanism with a first point of application of a load and a second point of application of a spring force, the lever mechanism bringing about an effective transmission ratio between the displacement of the first point and that of the second point, which ratio is adjustable so that the compression of the spring system and hence distance of the load from a reference plane can be maintained constant. In a preferred embodiment the system comprises a bell crank lever which cooperates with guide means to perform a translatory and a rotary movement in response to a displacement of said first point, the action of said guide means being adjustable.

15 Claims, 18 Drawing Figures



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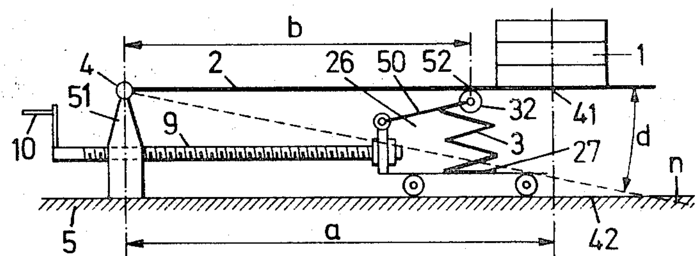


FIG. 1

FIG. 1a

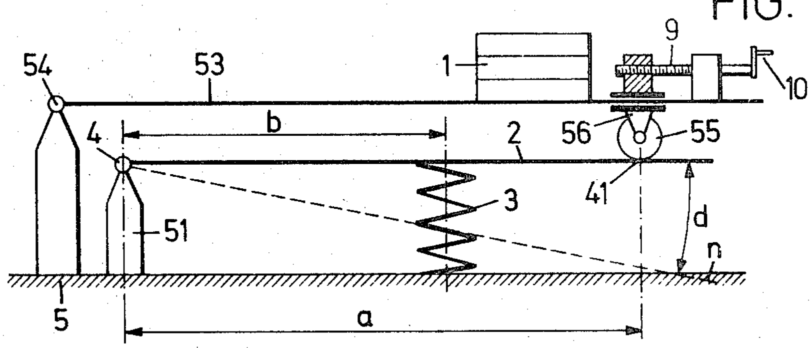


FIG. 1b

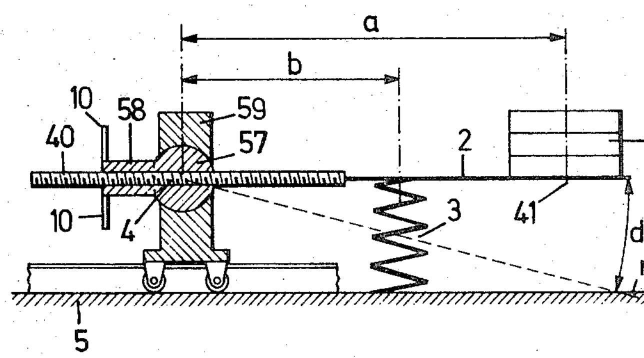


FIG. 2

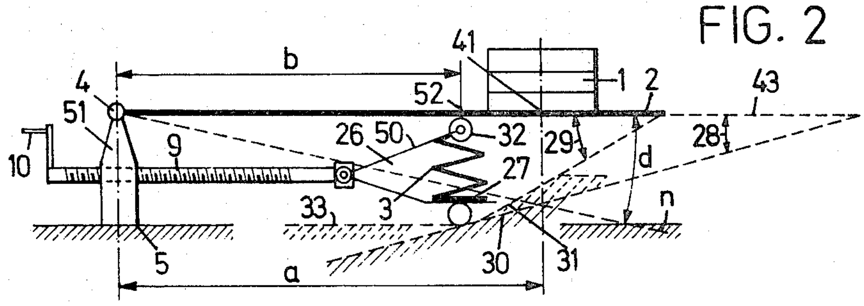


FIG. 3

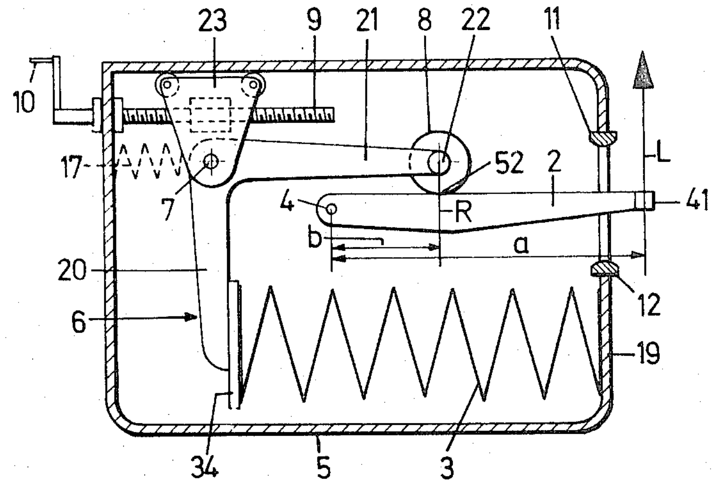
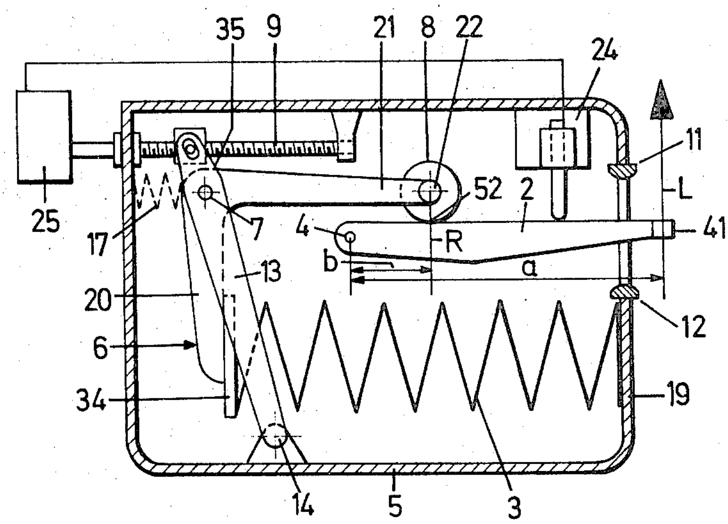

23
9
21
8
22
11
10
17
7
52
2
L
4
R
41
b
a
20
12
6
19
34
5
3

FIG. 3a


35
9
21
8
22
24
11
25
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2
L
17
7
4
R
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12
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34
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14
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3

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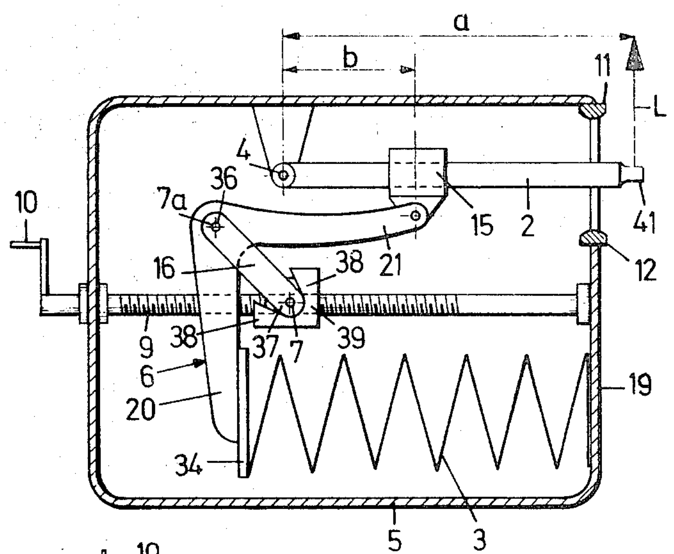

FIG. 4

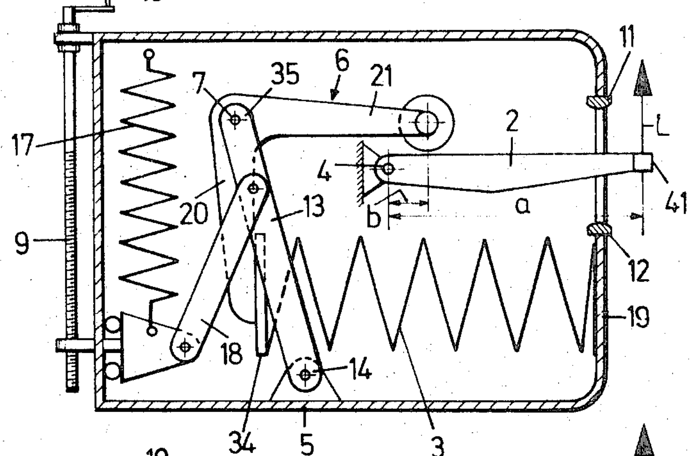

FIG. 5

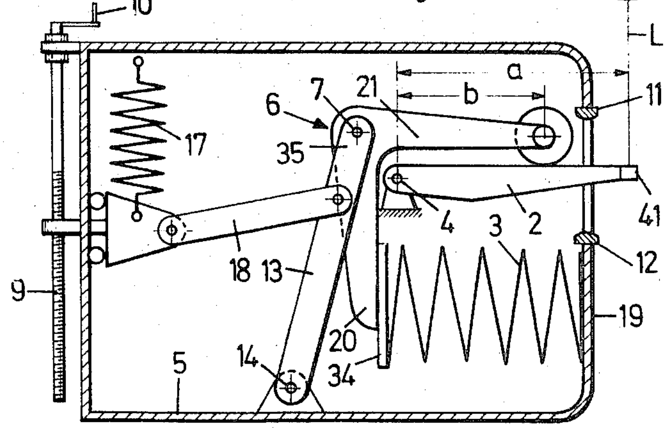

FIG. 6

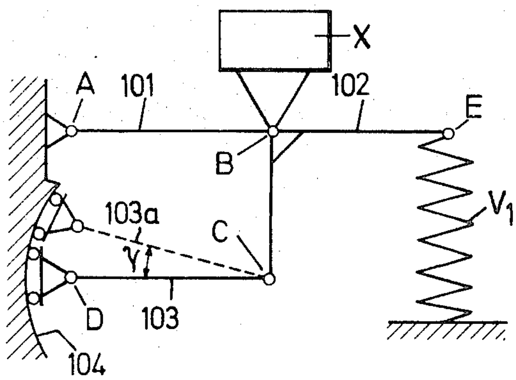

FIG. 7

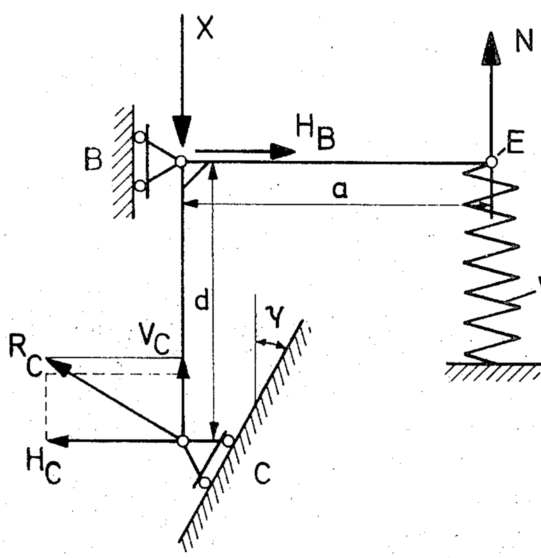

FIG. 8

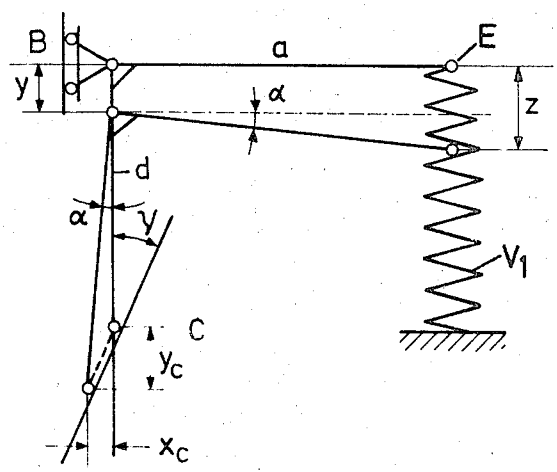

FIG. 9

FIG.10

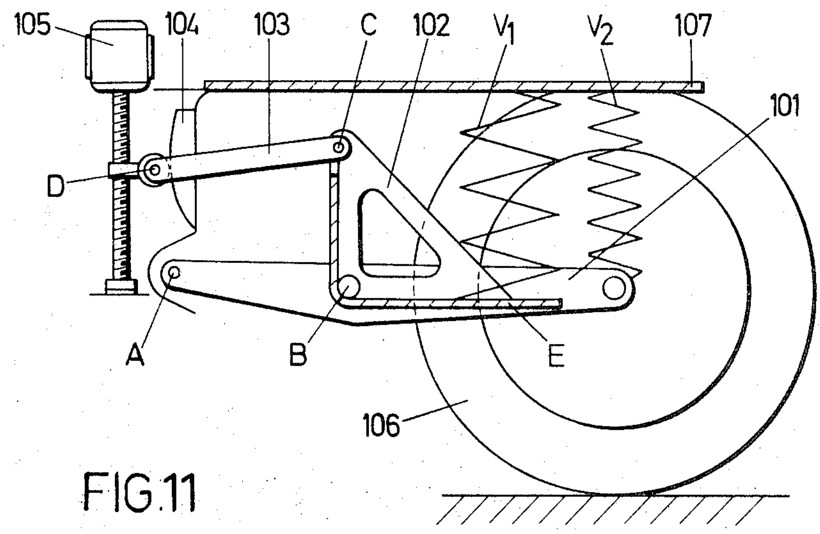
105
104
103
C
102
V1
V2
107
101
D
A
B
E
106

FIG.11

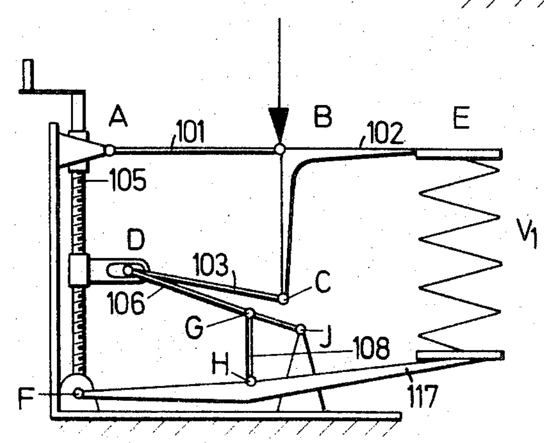
A
101
B
102
E
105
V1
D
103
106
C
G
J
108
H
117
F

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FIG. 12

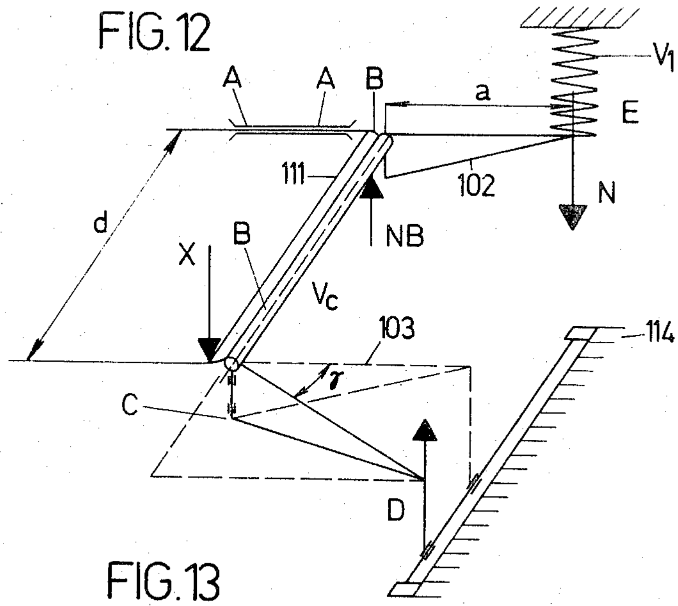


FIG. 13

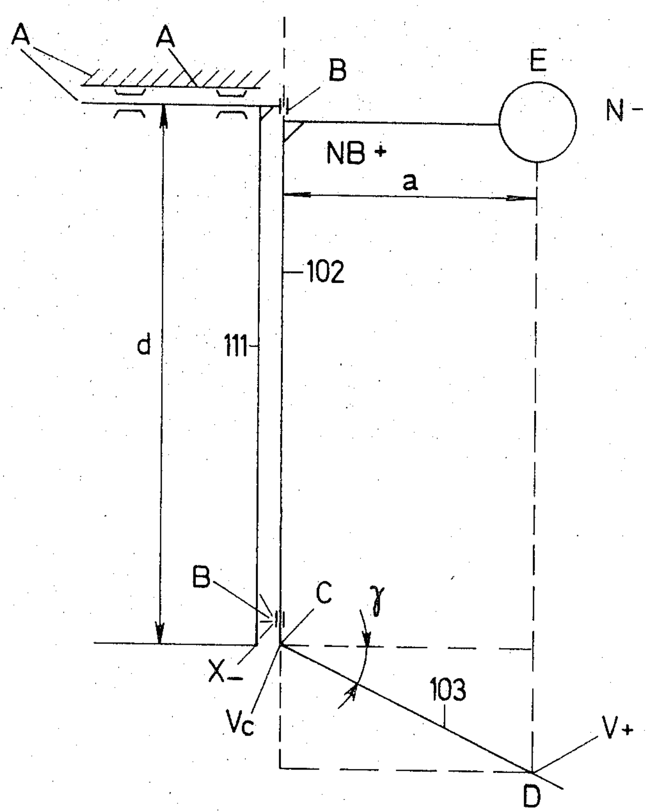


FIG. 14

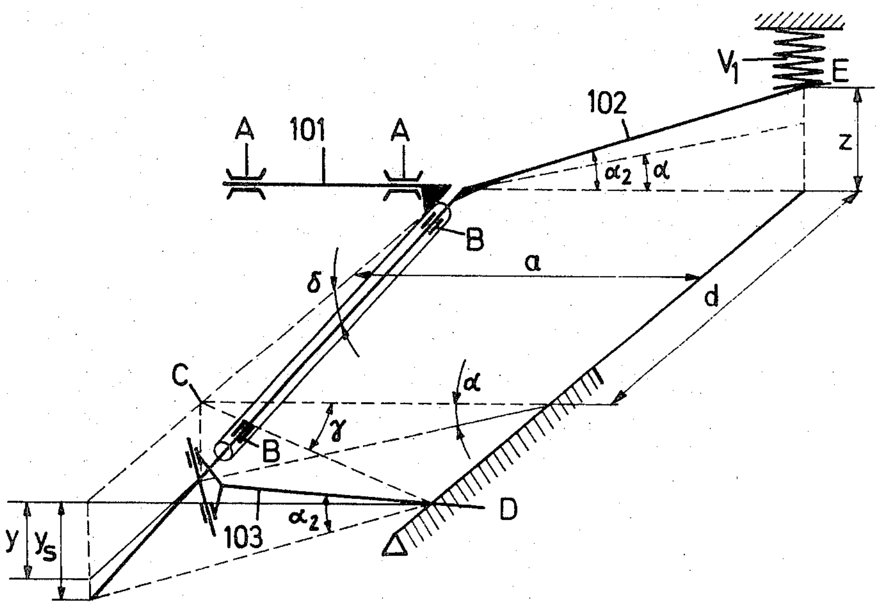
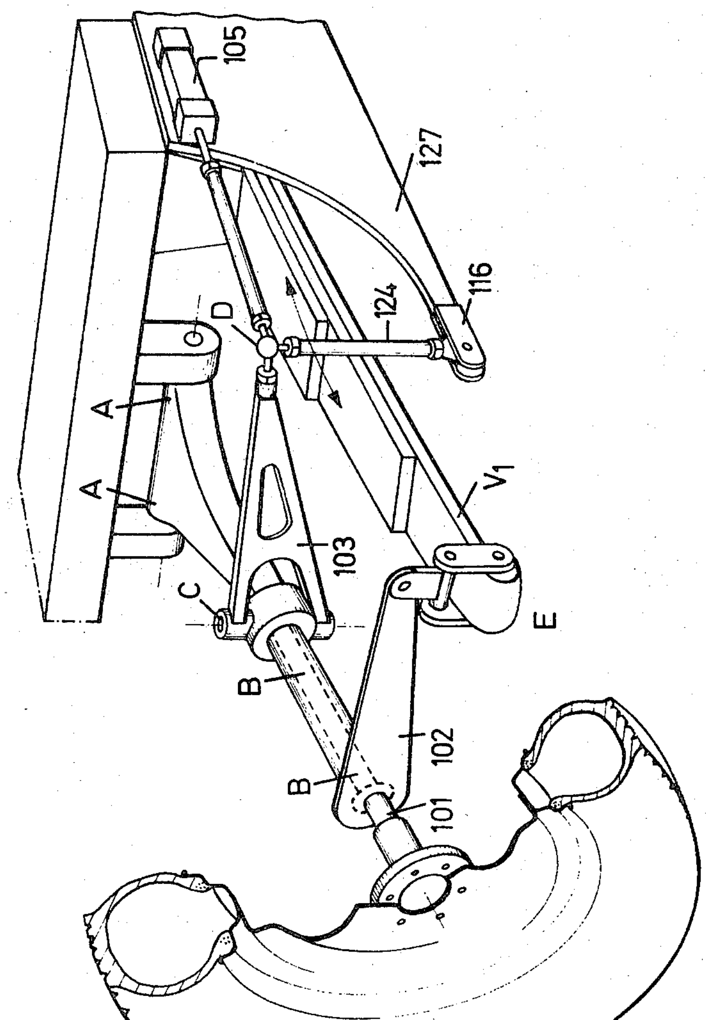

A
101
A
102
V_1
E
z
α_2
α
B
a
δ
d
C
α
γ
B
D
103
α_2
y
y_S

FIG. 15



SPRING SYSTEM

This invention relates to an adjustable spring system comprising a lever mechanism with a point of application of a load and a point of application of a spring force, which mechanism brings about an effective transmission ratio between the displacement of the point of application of the load and the displacement of the point of application of the spring force, and in which spring means acting on the lever counteract the displacement of the point of application of the load in the direction in which the load exerts its force.

Known adjustable spring systems are generally based on a displacement of the point of application of spring force in the case of an increasing or decreasing load such that the effective torque of the spring, and hence the reaction force at the point of application of the load, is increased or decreased. This, however, makes possible a limited adjustment only.

It is an object of the present invention to provide a spring system of the above kind, in which the rigidity of the spring can be adapted to varying load values, and which can be manufactured at a fair cost price.

It is a further object of the invention to provide a spring system for arrangements in which a load supported by the spring system is subject to relatively great variation, for example, between 100 percent and more than 200 percent. This is in particular the case with the main springs of many road vehicles, but also occurs with separately sprung driver's seats in agricultural and industrial tractors, stretchers in ambulances, etc.

The invention accordingly also extends to vehicles and other equipment including at least one spring system according to the invention.

It will be clear to those skilled in the art that the term "spring" as used in this specification and in the appended claims should be interpreted in a broad sense, and includes, for example, pneumatic and hydraulic devices.

According to the invention, there is provided an adjustable spring system comprising a lever mechanism with a point of application of a load and a point of application of spring force, which mechanism brings about an effective transmission ratio between the displacement of the point of application of the load and the point of application of the spring force, and in which the spring which acts on the lever counteracts the displacement of the point of application of the load in the direction in which such load exerts its force, characterized in that the effective transmission ratio is adjustable.

Owing to the adjustability of the effective transmission ratio of the lever mechanism, there is created a mass-spring system, the natural frequency of which can be varied in a given, desirable ratio to the load of the spring system by manual or mechanical adjustment, by virtue of which adjustment the compression of the spring system and hence the distance of the load from a reference plane can be constant.

A preferred embodiment of the invention is characterized by a bell crank lever which co-operates with guide means to perform a translatory and a rotary movement in response to a displacement of the point of application of the load, the action of said guide means being adjustable.

As shown by theoretical studies and confirmed by practice, the vibration insulation of a spring system depends on the ratio between the exciting frequency and the natural frequency of the system, and is better according as this ratio is higher than $\sqrt{2}$. This applies to simple harmonic vibrations, but has been found to be true of vehicles as well. In the case of vehicles, in which the weight of the cargo may vary greatly, a linear spring between the wheels and the vehicle body results in great fluctuations in natural frequency, in the sense that when the vehicle is empty the shock insulation is greatly inferior to that in the fully loaded condition.

If the behaviour of the entire vehicle (without damping) may be represented by the following simplified formula:

$$f = (1/2\pi)\ \sqrt{(c/m)} = (1/2\pi)\ \sqrt{(cg/G)}\ \text{Hz}$$

in which G is the weight in kgf, c the rigidity of the spring in kgf/cm, and $g = 980$ cm/cm² is the gravitational acceleration, the frequency in Hz is proportional to $f :: \sqrt{(1/G)}$.

The above and other features of the invention will be elucidated in the following description with reference to the accompanying drawings.

In said drawings, which are diagrammatical,

FIG. 1 shows an adjustable spring system in a known embodiment;

FIG. 1*a* and 1*b* each represent a spring system according to the present invention;

FIGS. 2, 3, 3*a*, 4 and 5 each show a further developed spring system according to the present invention;

FIG. 6 shows the spring system of FIG. 5 in a different position;

FIG. 7 shows a different spring system according to the invention;

FIGS. 8 and 9 each show a variant of FIG. 1;

FIG. 10 shows the embodiment of FIG. 7 in a practical implementation;

FIG. 11 shows a further development of a spring system according to the invention;

FIGS. 12 and 13 respectively show a perspective view and a plan view of still another embodiment of the spring system according to the invention;

FIG. 14 shows a further development of the spring system shown in FIGS. 12 and 13; and

FIG. 15 shows a vehicle with a spring system according to the invention reduced to practice.

The principle of known embodiments of an adjustable spring system is shown diagrammaticaly in FIG. 1. Resting on the cantilever end of a lever 2 mounted in a frame 5 so as to be turnable about an axis 4, at a point of application 41 is a variable weight 1. There is a constant first distance a between the point of application 41 and axis 4.

A spring 3 is mounted between two support members 27 and 32, one 27 of which is a carriage bearing on the base 42 of frame 5, and the other 32 is a roller engaging lever 2. Support member 32 is the free end of a pivoting arm 50 which like carriage 27 constitutes part of a holder 26, displaceable longitudinally of lever 2 and mounted on adjusting means constituted by a threaded spindle 9 with a handle 10, screwed into a bracket 51. The point 52 where spring 3 acts on lever 2 is at a variable second distance b from axis 4. In the nominal condition of the spring system, in which the amplitude of the vibration is nil, weight 1 can be maintained at a constant level above base 42, by virtue of the fact that a helical spring 3, which in the nominal condition is compressed by the same length and so exerts a constant force, is displaceable longitudinally of lever **2** and, depending on the magnitude of weight **1** is set at such a distance *b* from axis **4** that the nominal angular position *d* of lever **2** relative to reference angular position *n* is the same. When spring **3** is in its ideal position weight **1** can exercise vibrations on spring **3,** the natural frequency of which in the case of a variable weight is proportional to the square root of the leverage *b/a*, that is, decreasing with a decreasing weight **1** by which the displaceable spring **3** is loaded.

In the embodiment of an adjustable spring system according to the invention as shown diagrammatically in FIG. **1***a*, it is not the distance *b* which is adjustable, but the distance *a*. To this effect weight **1** is supported on an auxiliary lever **53** which is mounted in frame **5** for pivoting movement about axis **54,** while lever **2** is turnable about point **4** spaced from axis **54.** The point **41** at which the load acts on lever **2** is displaceable along the lever by virtue of the fact that a supporting roller **55** of a slide piece **56** is adjustable by means of a threaded spindle **9.**

In the embodiment according to the invention shown in FIG. **1***b*the ratio *b/a* is adjustable owing to the fact that axis **4,** about which lever **2** is turnable in frame **5,** is displaceable longitudinally of lever **2.** To this effect the threaded end **40** is received in an adjusting nut **58,** which is also formed as a ball **57** of a swivel bearing **59,** which in turn is guided relatively to frame **5** longitudinally of lever **2.**

In the spring system shown in FIG. **2,** which is a further development according to the invention, the natural frequency of weight **1** on lever **2** can be varied by adjusting in addition to the second distance *b* the precompression force of spring **3.** For this purpose supporting member **27** is supported on a curved surface **30** or **31** enclosing an acute angle **28, 29** with lever **2.**

This manner of adjustment, in which the precompression force is linear with distance *b* and is zero when *b* is zero, is productive of a constant natural frequency with varying weight loads and with a constant angular position *d*.

The construction of the adjustable spring systems shown in FIGS. **1 – 2** is less attractive than the further developed spring systems described hereinafter, especially when long helical springs **3** are used.

Accordingly, according to the invention, the lever mechanism is preferably formed in accordance with FIG. 3, in which holder **26** is replaced by a bell-crank lever **6.** In this spring system, the load acts on lever **2** at a point **41** at the free end thereof, the lever being pivotable between two stops **11** and **12,** and the other end of the lever being pivoted to frame **5** about axis **4.**

Frame **5** also provides a support member, namely supporting surface **19,** for spring **3,** shown as a helical spring, but optionally taking any other form. The other end of the helical spring bears on support member **34** carried by arm **20** of bell crank **6.** Bell crank **6** is in its centre journalled in a fulcrum **7** displaceable along frame **5** longitudinally of lever **2,** the other arm **21** of bell crank **6** carrying a roller **8** rotatable about a shaft **2,** and bearing on lever **2** at point **52.** A threaded spindle **9,** capable of being rotated with a handle **10** is shown diagrammatically in FIG. **3** for the adjustment of bearing block **23** of fulcrum **7.**

The operation of the spring system of FIG. **3** is as follows.

Lit is be supposed that fulcrum **7** is in the extreme left-hand position. The tension or bias of spring **3** is then minimal, for the spring has its maximum length. The force of spring **3** is transmitted to lever **2** through roller roller **8** according to the ratio of the lengths of arms **20** and **21** of bell crank **6.**

In this position roller **8** is at the shortest distance from axis **4,** so that the ratio *a/b* has the lowest value. Let it be supposed that in this case the product $a \times L$ of the tensile force *L* of the load and the arm *a* is considerably higher that the product $b \times R$ of the roller pressure R and arm *b*. Lever **2** is then displaced upwards until it touches stop **11.** Spindle **9** is now rotated to displace fulcrum **7** of bell crank **6** to the right, so that the arm length *b* of roller **8** relative to lever **2** is increased, but in addition spring **3** is compressed to a greater extent and thus acquires a higher bias. Bearing block **23** is adjusted until the position of fulcrum **7** is such that the end of lever **2** is centrally between stops **11** and **12.** If, in a given position of fulcrum **7** lever **2** comes to rest against stop **12,** fulcrum **7** is displaced in the opposite sense, that is, to the left in FIG. **3.**

The spring system illustrated in FIG. **3***a* corresponds with that of FIG. **3,** except that a senser **24** is provided for the adjustment of lever **2** to a central position between stops **11** and **12,** which senser **24** operates a motor **25** for adjusting bearing block **23,** and that fulcrum **7** is guided in a different manner. Fulcrum **7** is secured to a guide arm **13,** pivoting about a fixed point **14** in frame **5.** Pivot **14** has such a position that the fulcrum **7** of bell crank **6** describes a path substantially longitudinal of lever **2.**

FIG. **4** shows a different mechanical construction of the spring system according to the invention, in which the above roller **8** has been replaced by a sliding sleeve **15** mounted on bell crank **6.** In order to prevent sleeve **15** to slide along lever **2** with all springing movements of lever **2,** bell crank **6** is not directly connected with fulcrum **7** which is displaceable relatively to frame **5,** but a rocker **16** has been added between fulcrums **7** and **7***a*, so that sleeve **15** only has to slide on lever **2** when the spring system is adjusted. In the case of springing movements rocker **16** makes small angular movements about fulcrums **7** and **7***a*. These angular movements are bounded by stops **38,** so that when fulcrum **7** is adjusted sleeve **15** is forced to slide on lever **2** after rocker **16** has come to rest against one of the two stops **38.** The correct position of sleeve **15** for springing movements without the latter sliding on lever **2** is assumed after some strokes of the spring of sufficient magnitude.

The spring system shown in FIGS. **5** and **6** is provided with a compensatory spring **17,** which substantially reduces the drive forces required for overcoming the bias of spring **3** in moving adjusting means **9, 10.** Also, the energy to be supplied by the adjusting mechanism for biasing spring spring **3** can be materially reduced by it. Very little external energy is required for the adjusting work of adjusting means **9, 10** when a toggle **18** is used or any other known mechanism for converting the force of counter-spring **17** to such a magnitude that there is virtual equilibrium with the force of spring **3** at fulcrum **7** throughout a considerable part of the latter's stroke.

It should be noted that a compensatory spring **17** can also be used in the spring systems of FIGS. **3** and **4,** in which such spring is shown in dash lines.

FIG. 10 shows an embodiment of the invention applied to an independent wheel suspension of a motorcar, comprising a longitudinal arm, or an axle suspension with a rigid axle guided by reaction arms.

As shown in the figure, two linear spring are used, one V_2 of which is placed directly on the longitudinal arm in known manner, while spring V_1 constitutes part of the variable spring system according to the invention.

In FIG. 15 the invention is shown applied to a half-axle suspension, the spring action being provided for by one leaf spring V_1 constituting the spring element in the linkage according to the invention. In FIGS. 10 and 15 the adjustable support point in the system is consistently designated by D.

As regards the structural elements used, reference is made to FIGS. 7 and 10. FIG. 7 only shows the essential building elements of the spring system in diagrammatic form, while FIG. 10 shows the application of these elements to a wheel arm. Because the direction of loading in FIG. 7 is opposite to that in FIG. 10, the arrangement has been symmetrically reversed about the longitudinal axis.

FIGS. 7 and 10 show a bell crank 102, which is engaged by load X in fulcrum B, while spring V_1 exerts its force at the end of one arm of lever 102. Bell crank 102 is guided by rod 101, which connects fulcrum B with fixed pivot A in the frame. Pivot C at the end of the arm of bell crank 102 parallel to the direction of deformation of spring V_1 is guided by a rod 103, the other end of which is connected to a pivot D, which pivot D can be displaced relatively to the frame by suitable adjusting means (shown in FIG. 10 only) in a given track, owing to which the characteristics of the spring system can be altered. It will be seen in FIG. 10 that arm 101 with the fixed pivot A is extended and carries at its end the axle for wheel 106. Pivot A is mounted on the car body, the bottom of which is designated by 107. The above-mentioned adjusting device 105 regulates the place of pivot D when as a result of a change in the weight of the load this point has to assume a different position on track 104 to re-adjust wheel 106 to its central position relative to body 107.

FIG. 11 shows an addition to the spring system of FIG. 7. In this arrangement pivot D is guided in a slot for the pivot to be displaced in an arcuate path described by rod 106 about pivot J fixedly mounted in the frame. A second fixed pivot F is mounted in the frame, to which lever 117 is secured, which at its end carries a dish supporting spring V_1. Rod 106 and lever 117 are coupled together by means of connecting rod 108 between pivots G and H. This extension results in a spring system in which both the length of the tensioned spring and the location of point D are together adjusted by one mechanism 105, in this case a threaded spindle with a handle.

In FIG. 12, the principles of the present invention are shown in perspective view. There is shown a spring system having identical properties and possibilities as that described with reference to FIG. 7. The arm 101, which carries load X on one end, pivots about axis AA. Bell crank 102 pivots about axis BB and about one member of arm 101, which is L-shaped in this instance. The other member extends in the direction AA perpendicular to axis BB. The spring V_1 acts at the end of this arm in point E. At the end of the other arm of bell crank 102 is an arm 103 rotatable in the plane of bell crank 102, and the other end D of which can be displaced along a ruler 114 secured to the frame to vary an angle between the perpendicular to the part BB of bell crank 102 and arm 103.

FIGS. 13 and 15 show the application of this principle, using a modification for the displacement of end D, to a half axle. Bell crank 102, in this case a torsion tube with a perpendicular arm welded to it, is mounted around axle body 111 (this is the suspension BB of FIG. 12). A leaf spring V_1 acts on the end E via a pair of links. At the other end of the tube of bell crank 102 two small shafts extending at right angles to the tube provide an axis C about which arm 103 can pivot. The end of arm 103, that is, support point D, is connected to a fixed point 116 of chassis 127 by a rod with ball joints 124. The position of point D is controlled by an adjusting device 105.

That the system according to the invention makes it possible to adjust the spring suspension, regulating both the natural frequency of the spring system and the level of the load relative to a base can be shown by means of FIGS. 8, 12 and 13. The essence of the system according to the invention is that the spring force N and the oppositely directed loading force X are out of alignment with each other, and apply to a bell crank lever having a fulcrum, so that the two forces act on the bell crank at different distances from the fulcrum. The resulting torque N.a on the bell crank about fulcrum BB is taken up in two points of the frame which may or may not be connected to the bell crank by links, the relative positions and tracks of these points being controlled to vary the magnitude of force X with N being constant.

FIG. 8 shows the equilibrium of forces acting on bell crank 102. The bell crank is guided by two rectilinear guideways in B and C, which situation is an approximation of the diagram of FIG. 7, the curves described by arms 101 and 103 being equalized to the chords of these arcs. The guideway of C makes an angle γ with the vertical and the direction of the guideway of B. If springing movements are first disregarded, the spring force N in E can be supposed to be constant. The external force on bell crank 102 acts in B and is designated X. The equilibrium satisfies the following formulas, using the notation and the direction of the arrows for the forces of FIG. 8:

The sum of the horizontal forces $= 0 \quad H_B - H_C = 0 \quad (1)$

The sum of the vertical forces $= 0 \quad X - N - V_C = 0 \quad (2)$

The sum of the moments about

$$B = 0 \quad Na - H_C d = 0 \tag{3}$$

It further follows that the direction of the force in C is perpendicular to the direction of movements, so that

$$V_C = H_C \, tg\gamma \tag{4}$$

H_C is found from (3) and substituted in (4) to find V_C the substitution of the result for V_C in (2) gives

$$X = N\,(1 + (a/d)\, tg\gamma) \tag{5}$$

It follows that magnitude X is varied by the variation of γ, that is the position of lever 103 or pivot D, so that the spring maintains a constant force N, which is possible by "circling" lever 103 about pivot C. The position of B is not changed if equilibrium between X and N, according to (5), is reached in the central position. The variation of the effective spring rigidity at pivot B can be shown by means of FIG. 9. Let it be supposed that B is displaced downwards over distance y. C is then displaced over distance y_C in a vertical direction and $x_C =$

$y_C tg\gamma$ in a horizontal direction. Neglecting the effect of angle $\alpha = (x/d)$ on the length of arm d of bell crank 102, $y_C = y$.

The vertical displacement at E then becomes

$$z = y + \alpha\, a = y\ (1 + (a/d)\ tg\gamma) \tag{6}$$

With a spring rigidity c of spring V_1 the change in force of the spring $\Delta N = c\cdot z$

or

$$\Delta N = c\cdot y\ (1 + (a/d)\text{tg}\gamma) \tag{7}$$

The force ΔN of the spring results in the change of force of X, ΔX, as per (5), so that

$$\Delta X = c\cdot y\ (1 + (a/d)tg\gamma)^2 \tag{8}$$

The effective spring rigidity in B becomes with $c_{eff} = (\Delta X/y)$

$$c_{eff} = c\ (1 + (a/d)tg\gamma)^2 \tag{9}$$

The variation in the position of pivot D changes the effective spring rigidity of the system as per equation (9).

If now the force X represents the weight G of a mass acting on the spring system, the angle γ to be adjusted complies with the following formula:

$$G = N\ (1 + (a/d)tg\gamma) \tag{10}$$

N and G in kgf; then we have $m = (G/g)\cdot g = 980$ cm/sec^2.

The undamped natural frequency complies with the following formula:

$$f=\frac{1}{2\pi}\sqrt{\frac{c_{eff}}{m}}=\frac{1}{2\pi}\sqrt{\frac{c_{eff}\cdot g}{G}} \tag{10a}$$

by the substitution of (9) and (10), this becomes:

$$f=\frac{1}{2\pi}\sqrt{\frac{cg}{NH}}\sqrt{G} \tag{10b}$$

or:

$$f::\sqrt{G} \tag{11}$$

This means that the natural frequency of the spring system according to the invention is proportional to the square root of the weight acting on the system. Consequently, with a decreasing load there is a lower natural frequency and a better shock insulation. However, when such spring systems, which becomes less rigid with an increasing load are used in motorcars, there may be drawbacks from the point of view of road performance owing to rolling movements when the vehicle traverses a bend or excessive pitch upon braking if the differences in load become substantial. This can be prevented by affecting in the adjustment of the spring system not only the angle γ, but also the spring force N, such that $(\sqrt{G/N})$ is constant. Such an adjustment can be effected in three ways, namely,

1. by raising B, so that E follows;
2. by keeping B in a fixed position and moving D in FIG. 7 more to the right as corresponds with an arc about C;
3. by lowering the base of spring V_1 when D is raised as shown in FIG. 11.

Naturally the three methods can be applied together.

FIG. 12 and 13 show a spring system which can also be adjusted by displacing a support point, and the construction of which is three-dimensional rather that two-dimensional as is the case in FIG. 7. FIG. 12 shows a perspective view of the system and FIG. 13 a plan view, the forces acting perpendicular to the plane of drawing being indicated by a "minus" if they act downwards, and by a "plus" if they act upwards. Considering the equilibrium of bell crank 102 with supporting arm 103, the following formulas should apply:

sum of the vertical forces

$$= 0\ N_B + V_D - N - V_C = 0 \tag{12}$$

sum of the moments about axis AA $V_D\cdot(d + a\ tg\gamma) - V_C d = 0$ (13)

sum of the moments about axis BB $V_D a - Na = 0$ (14)

Substitution of (14) in (13) results in

$$V_C = N\ (1 + (a/d)\ tg\gamma) \tag{15}$$

From the equilibrium of arm 101 follows that $V_C = X$, so that

$$X = N\ (1 + (a/d)\ tg\gamma). \tag{16}$$

In order to determine the effective spring rigidity in C, FIG. 14 is given by way of illustration. If arm 101 turns through angle δ, and the amount by which point C is lowered is y, then $\delta = (y/d)$.

If it is assumed that bell crank 102 is not deformed under torsion and bending then, if lever 102 does not turn in bearing BB, the amount by which point D of arm 103, which encloses the angle γ with the perpendicular to arm 102, y_s, is determined by the following equation:

$$y_s = \delta\ (d + a\ tg\gamma) = y\ (1 + (a/d)\ tg\gamma) \tag{17}$$

If now the point D of arm 103 is guided by a ruler extending parallel to the initial position of arm 101, bell crank 102 must rotate through angle $\alpha_2 = (a/y_s)$.

FIG. 14 shows the angle α as an example when $tg\gamma = 0$. As a result the fixed part of the arm lying in the plane of AA is raised at its end by an amount $z = y_s$, so that spring V_1 is compressed by an amount z. This results in a change in spring force $\Delta N = cz$. Identical to (16), this gives $\Delta X = \Delta N\ (1 + (a/d)\ tg\gamma)$ (18)

and for the effective spring constant at X

$$c_{eff} = c\ (1 + (a/d)\ tg\gamma)^2 \tag{19}$$

The formulas derived for the bell crank subject to torsion with regard to carrying capacity and effective spring rigidity (16) and (19) are identical to (5) and 9) for the bell crank subjected to bending. The considerations concerning the variation of the natural frequency with the load and the compensation of this variation by adapting the spring force apply likewise. In the derivation given for this, it has been supposed for the sake of convenience that bell crank 102 and arm 103 are undeformable. Naturally, in this system the spring V_1 may be omitted and parts of bell crank 102 may be formed as torsion or bending springs instead. Also, it is not of essential importance whether the spring is applied at the fixed pivot of arm 101 or changes places with the support point D of adjustable support arm 103.

The use of an adjustable spring is then useful if it is desired to have a constant low natural frequency less than about 1 Hz for the suspension of a motorcar with great differences in load. This is almost always accompanied by the need to be able to adjust the ground clearance for it to be virtually constant with such differences in load. As described before, in the system according to the invention a constant natural frequency can be achieved by changing the deformation of the spring in a suitable manner. In a low rigidity spring system this may involve a rather considerable difference in length of the spring in various positions, for which the required space may sometimes be lacking. By the application of the present invention, it is possible, by using for the spring suspension a normal spring parallel to adjustable spring systems as described hereinbefore, to realize a fair approximation of a system having a constant frequency throughout the range of load possibilities without there being a need to change the length of the springs (in the middle position between the extreme spring amplitudes). It follows from the formula for the carrying capacity of the spring system according to the invention that the carrying capacity can be made zero by selecting $tg\delta = -(d/a)$ at the tensional force N of the spring. When a normal spring and an adjustable spring system are used, there is the possibility of adjusting the adjustable spring system to zero when the vehicle is empty and having the empty vehicle body — having a weight G_0 — carried by the normal spring, which is then selected for a given natural frequency, while the variable load is carried by the adjustable spring system. The rigidity of the spring V_1 in this adjustable spring system is so selected that if the full load is carried solely by the adjustable spring system, the natural frequency is substantially equal to that of the empty vehicle.

In FIG. **10** this principle has been adopted: when the vehicle is empty, its weight is carried solely by spring V_2, owing to spring V_1 included in the spring system according to the invention being adjusted to zero carrying capacity. According as the vehicle is loaded to a greater extent the spring system is adjusted to a higher carrying force, always such that the vehicle has a constant road clearance, and spring V_2 as well as spring V_1 have the same length and the concomitant spring force. That such an arrangement gives a fair approximation of a constant natural frequency can be shown as follows: When the vehicle is empty the weight G_0 acting on the spring is c_2; this gives the frequency

$$f_o = (1/2\pi)\sqrt{g}\ \sqrt{(c_2/G_o)} \quad (20)$$

c_2 in kg/cm, G_0 in kgf, $g = 980$ cm/sec^2.
The load has the weight G. At full load G_v, the frequency becomes:

$$f_v = (1/2\pi)\sqrt{g}\ \sqrt{(c_1/G_v} \quad (21)$$

We now make $f_0 = f_v$ or $(c_2/G_o) = (c_1/G_v)$ (22)
According to the properties of the spring system (cf (5) or (16)), it applies that $X = N\ (a + (a/d)\ tg\delta)$. (22a)
Since X is proportional to G, we can write:

$$G = N\ y \quad (23)$$

in which $y = 1 + (a/d)\ tg\delta$. (23a)

At a given load G, with a constant spring force N, y acquires a certain magnitude, from which the effective spring rigidity follows as per equation (9) or (19):

$$c_{eff} = c_1\ (1 + (a/d)\ tg\delta)^2 \quad (23b)$$

which here becomes:

$$c_{eff} = c_1 y^2 \quad (24)$$

Now the dimensioning is supposed to be so selected that when $y = 1$ the full load force G_v is attained. Then we have

$$G = y\ G_v \quad 25)$$

in which $0<y<1$. In the case of the partly loaded vehicle the load $G_0 + G$ acts on both springs c_2 and c_{eff} together, so that the natural frequency becomes:

$$f = (\sqrt{g/2\pi})\ \sqrt{c_2 + c_{eff}/G_o + G)} \quad (26)$$

The substitution of (24) and (25) in (26) produces:

$$f = (\sqrt{g/2\pi})\ \sqrt{c_2 + y^2 c_1/G_o + yG_v} \quad (27)$$

By using equation (22), this can be developed to

$$f = (\sqrt{g/2\pi})\ \sqrt{c_2/G_o}\ \ \sqrt{(1 + y^2\ G_v/G_o/1 + y\ G_v/G_o)} \quad (28)$$

By introducing $\beta = G_v/G_0$, the nett load/tare ratio, we get

$$f = f_o\ \sqrt{(1 + y^2\beta/1 + y\ \beta)} \quad (29)$$

To the adjustable spring system with a constant spring force applied

$$f :: \sqrt{G_{total}} \quad (11)$$

For a practical value of G_v/G_0 of 3, the following differences in frequencies of the two systems are found:

	1 adjustable spring	1 zero load spring 1 adjustable spring
zero load	0.5	1
⅓ load	0.71	0.82
⅔ load	0.87	0.89
full load	1	1

which shows a fair approximation of the constant natural frequency.

We claim

1. An adjustable spring system, comprising a frame, a level mechanism pivotably mounted in said frame, in which said lever mechanism is adapted for supporting a load, spring means acting between said lever mechanism and said frame to counteract the displacement of the load, further comprising adjusting means for changing the effective transmission ratio between the displacement of the load and the displacement of the spring means, characterized in that said adjusting means are arranged in such a way that with variable loads the height of the load and at same time the natural frequency of vibration of the load and spring means are adjusted for maintaining a predetermined height of the load with a predetermined natural frequency of vibration.

2. A spring system according to claim **1**, characterized by two separate levers pivotably mounted in the frame, and extending substantially parallel to each other, in which said levers have spaced pivots and in which the adjusting means comprise a pivoting connection between said levers and adjustable along said levers.

3. A spring system according to claim **1** in which the lever mechanism comprises one single lever on which the load and the spring means act with a constant distance between them and in which the adjusting means are adapted to displace the pivot point between the lever and the frame along the lever.

4. A spring system according to claim **1** in which the pivot point of the lever on which the load acts is mounted fixedly in the frame, and in which the spring means is mounted in a holder adjustable substantially longitudinally of the lever and in which the supporting surface in the frame for the spring means makes a variable angle with the lever.

5. A spring system according to claim **12**, in which the spring means acts in the plane of the bell-crank lever and the reaction means is constituted by rollers in slots or by arms acting between the frame and the lever and pivoting relatively to these two parts, characterized in that the angle of the paths described by the guide points of the lever is adjustable by turning the position of the guide slot or by changing the place of a pivot of the reaction means, mounted in the frame.

6. A spring system according to claim **12**, characterized in that the means for adjusting the reaction means are coupled with means for adjusting the length of the spring means.

7. A spring system according to claim **6**, characterized in that the path of adjustment of the reaction means is such that the change in place of the reaction means is accompanied with a change in length of the spring means upon a change of the angular position of the bell-crank lever.

8. A spring system according to claim **6**, wherein the spring force acting between the bell crank lever and the frame acts on a support point movable in the frame in the direction of the spring force, which point is displaced together with the displacement of the reaction means owing to a mechanical coupling.

9. A spring system according to claim **1**, further comprising a spring directly interposed between the frame and the load, characterized in that the deformation of said directly inter-posed spring can be kept at a given value with changing magnitudes of the load by adjusting the effective spring tension of a spring system according to any one of the preceding claims.

10. A spring system according to claim **4**, characterized in that the adjusting means are coupled with the adjustable holder by means of a pivoting rod.

11. A spring system according to claim **1** in which the force of the load acts on the center of a bell-crank lever, substantially parallel to the direction of the force of the spring means acting on one end of the bell-crank lever, and in which reaction means are active between the other end of the bell-crank lever and the frame and in which the reaction means are adjustable to change the direction of the reaction force between the bell-crank lever and the frame and in which the lever cooperates with guiding means to effect a translatory and a rotary movement as a result of a displacement of the load.

12. A spring system according to claim **11** in which the spring means disposed between the frame and the bell-crank lever act on a supporting point mounted for displacement in the frame in the direction of the force of the spring means, comprising a second lever mechanism connected with the adjusting means for the reaction means.

13. A spring system according to claim **11**, wherein the load acts on an arm, pivotable about an axis perpendicular to the arm and to the direction in which the load acts, and in a point remote from the point where the load acts on the lever, and in which the bell-crank lever is mounted for pivoting movement about an axis parallel to one arm of the bell-crank lever and parallel to the arm on which the load acts and in which the spring means act on the end of the bell-crank lever and substantially parallel to the direction in which the load acts, and in which the reaction force between the frame and the bell-crank lever acts on a reaction lever pivotable about an axis parallel to the direction in which the load acts.

14. A spring system according to claim **1** in a vehicle in which a wheel is mounted on a wheel guiding arm, pivotably mounted in the vehicle frame, and in which there is disposed a spring means between the vehicle frame and the wheel guiding arm for supporting the load of an empty vehicle, further comprising an adjustable spring system according to claim **23** between the wheel guiding arm and the vehicle frame for supporting any excess load in the vehicle.

15. A spring system comprising a frame, a bell-crank lever, bearing means for pivotably supporting the bell-crank lever in the frame, spring means between a supporting surface in the frame and a point at one end of the bell-crank lever, transmission means for transmitting to the bell-crank lever the force exercised by the load reaction means between the bell-crank lever and the frame for supporting the reacting force which results from the force of the load and the action of the spring means, further comprising adjusting means for displacing the reaction means between the bell-crank lever and the frame.

16. A spring system according to claim **15** in which the reaction means between the bell-crank lever and the frame act on the center of the bell-crank and in which the other end of the bell-crank lever is movably connected with a further lever pivotably mounted in the frame and on which end the load acts and in which this further lever is mounted substantially in the plane of the bell-crank lever and in which the adjust-ing means are active to displace the pivot point between the center of the bell-crank lever and the frame in the direction of the lever on which the load acts.

17. A spring system according to claim **15**, characterized in that there is disposed a lever between the parts of the adjusting means which are connected with the frame and the pivot point in the center of the bell-crank lever.

18. A spring system according to claim **15**, characterized by further spring means disposed between the bell-crank lever and the frame for compensating the action of the spring means when the spring system is adjusted.

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