United States Patent [19]

Ward

[11] **3,737,156**

[45] **June 5, 1973**

[54] **V-TYPE TORSION BAR TILLAGE TINES**

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[22] Filed: **Oct. 8, 1971**

[21] Appl. No.: **187,664**

[52] **U.S. Cl.** **267/154,** 267/57

[51] **Int. Cl.** **F16f 1/16**

[58] **Field of Search**..............................267/154, 57

[56] **References Cited**

UNITED STATES PATENTS

3,337,236	8/1967	Peterson	267/57
2,797,434	7/1957	Vigmostad	267/154
2,591,281	4/1952	Musschoot	267/154
3,276,762	10/1966	Thomas	267/154

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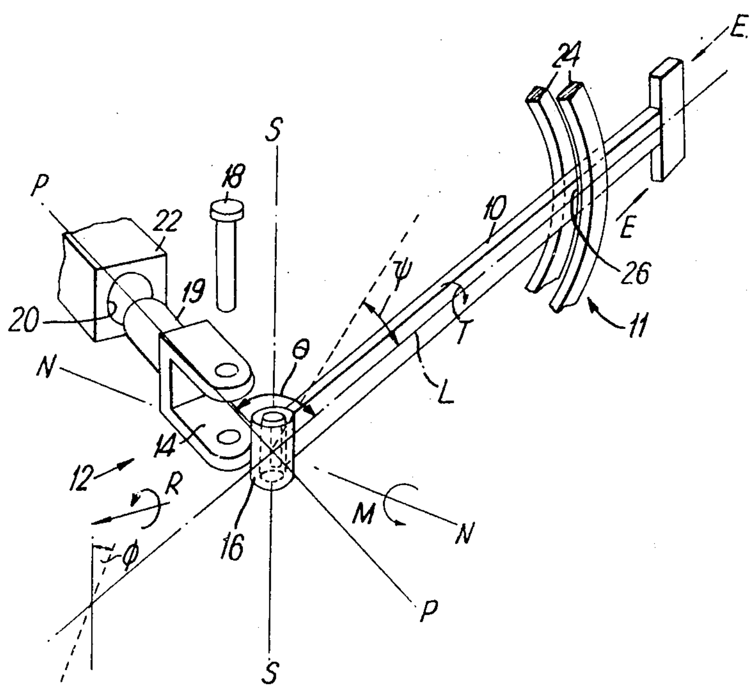
[57] **ABSTRACT**

The invention provides a spring assembly and an energy absorbing linkage wherein end constraints on an elongated resilient member are effective to place the elongated member in torsion, such torsion permitting large deflections of the spring or linkage while maintaining a relatively low stress level in the resilient member. The spring assembly can be conveniently made in the form of an energy linkage unit, such a unit being suitable for building planar or lattice spring structure, or for use in an installation as a means of alternatively storing and releasing energy.

29 Claims, 31 Drawing Figures



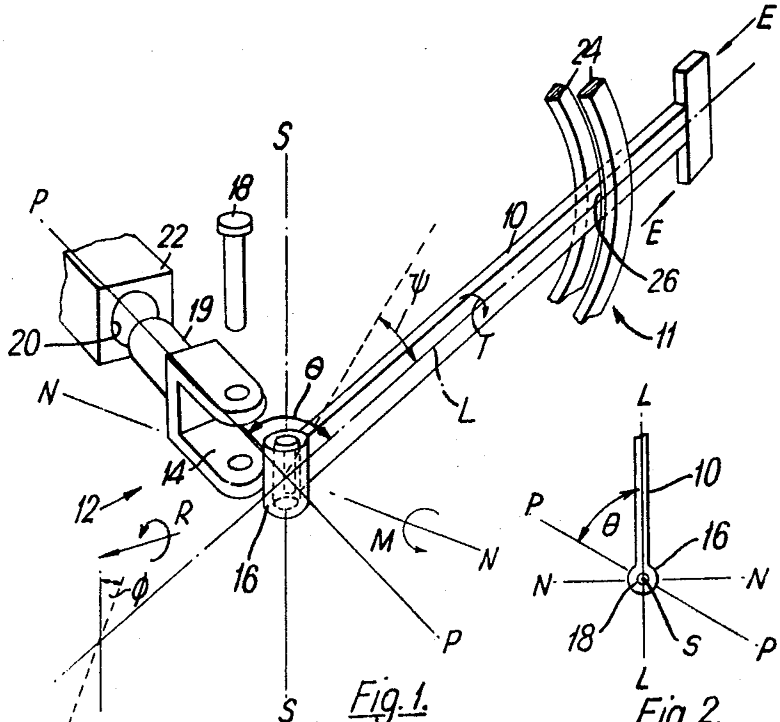


Fig. 1.

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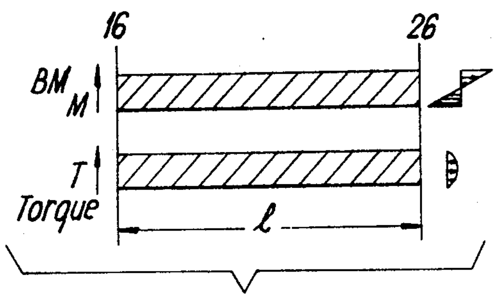

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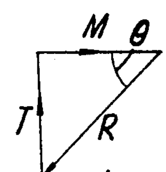

Fig. 3.

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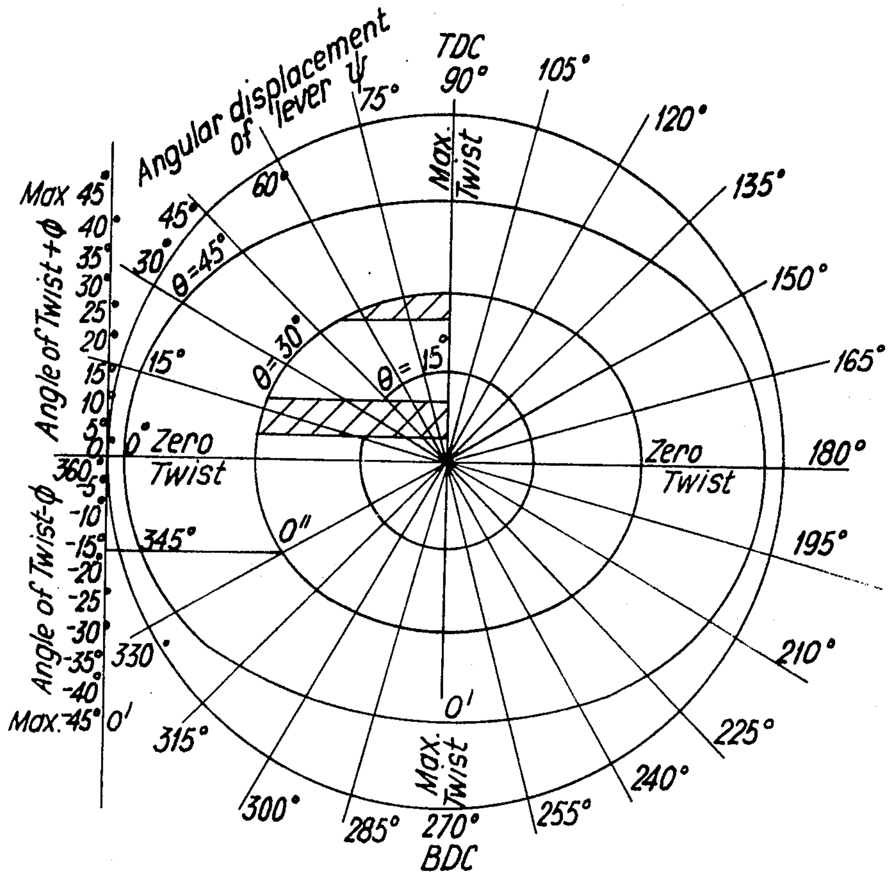

Fig. 5.

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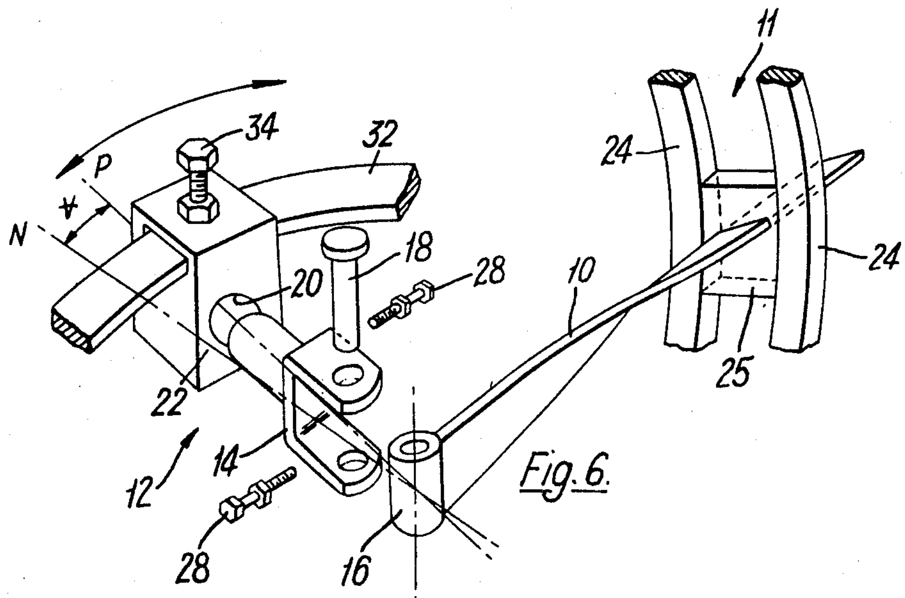

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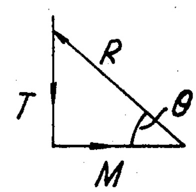

Fig. 7.

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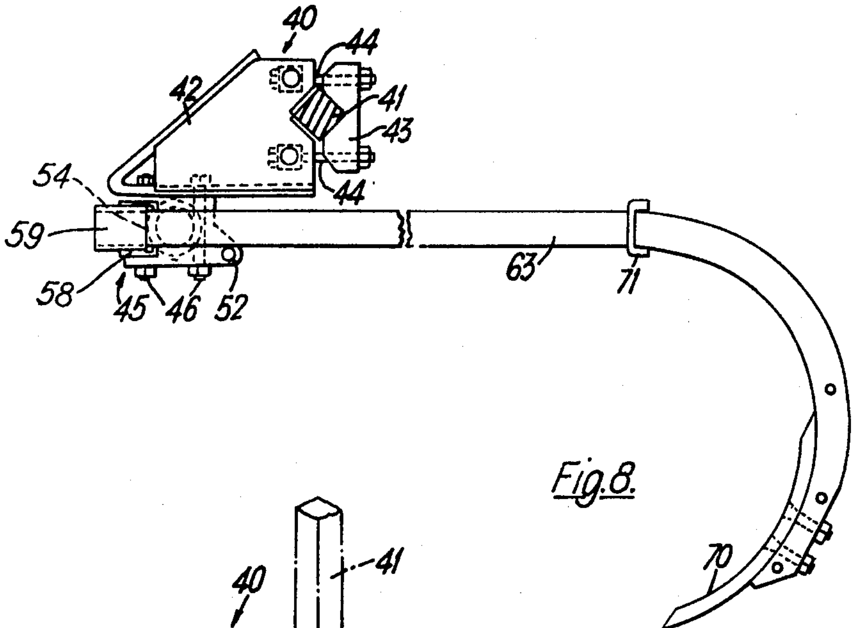

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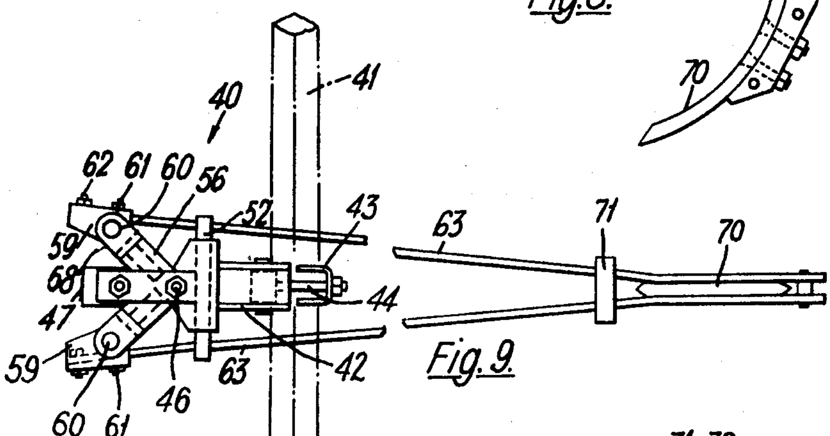

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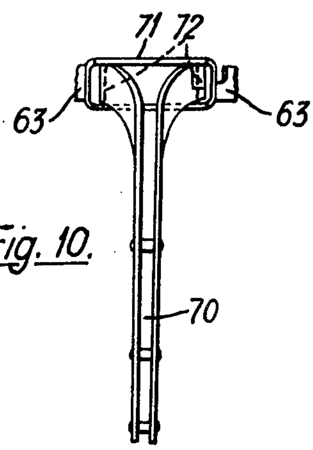

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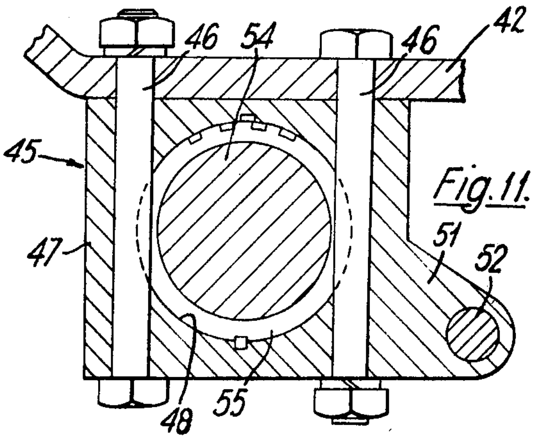

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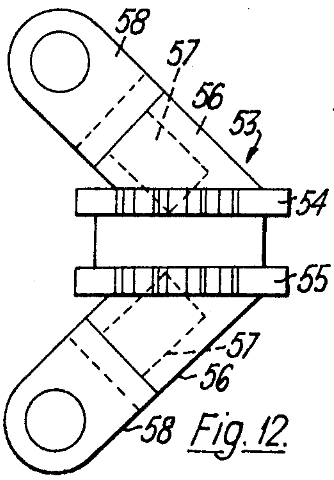

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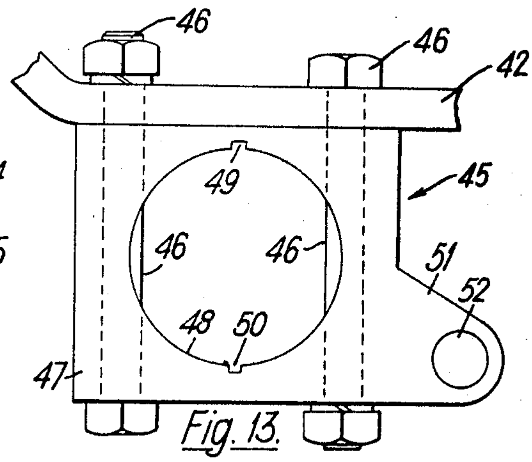

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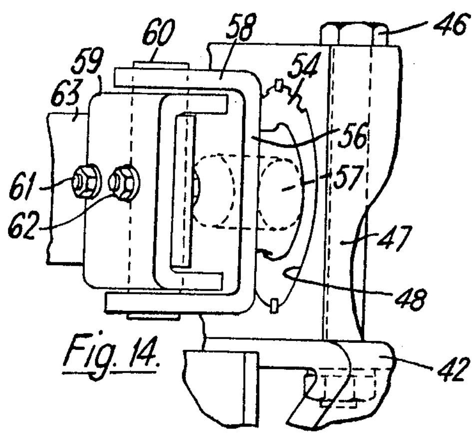

Fig. 14.

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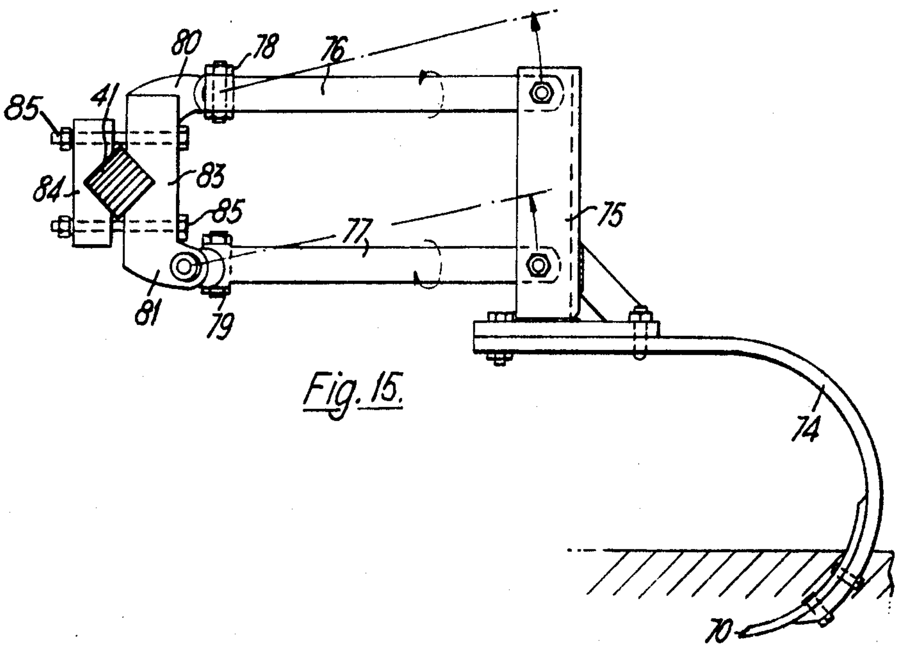

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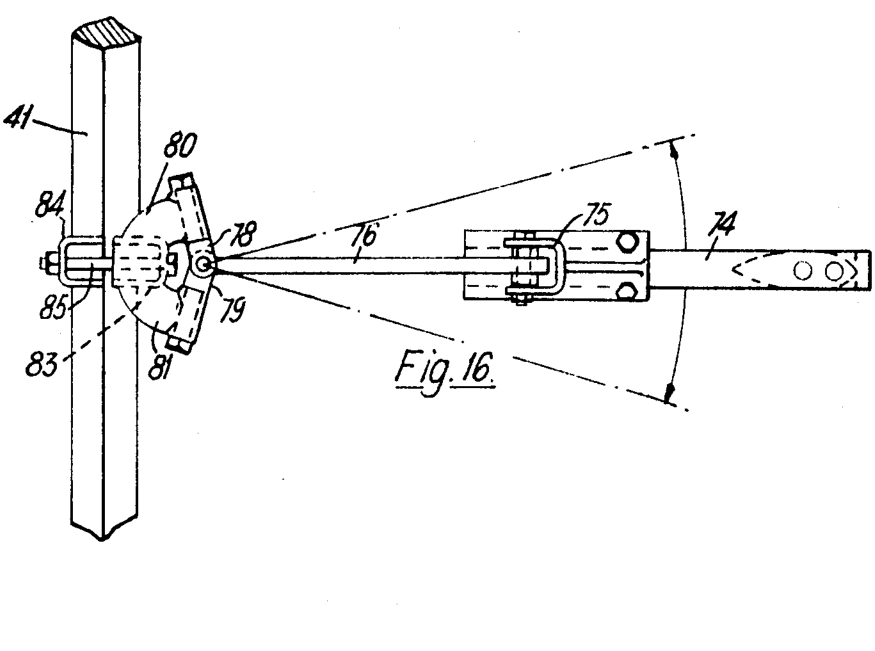

Fig. 16.

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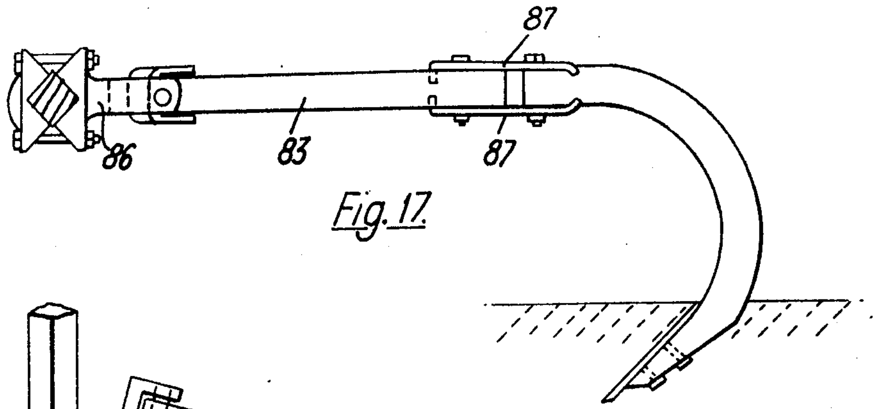

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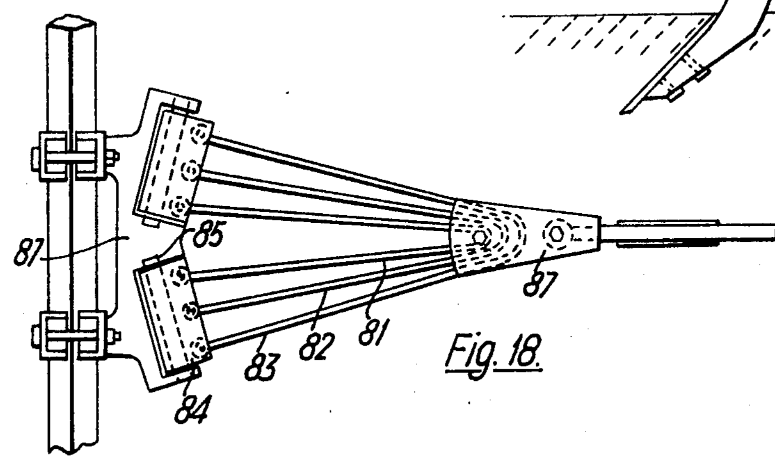

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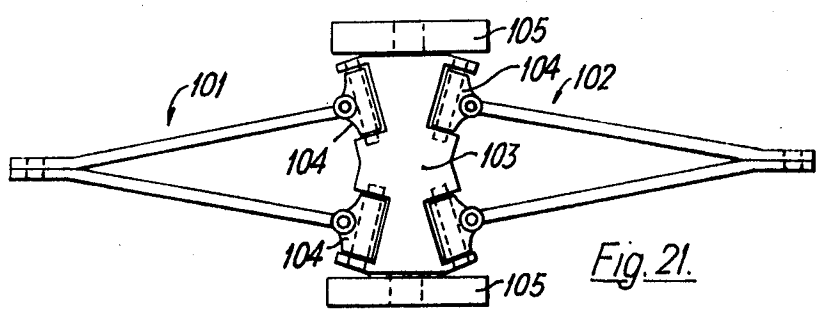

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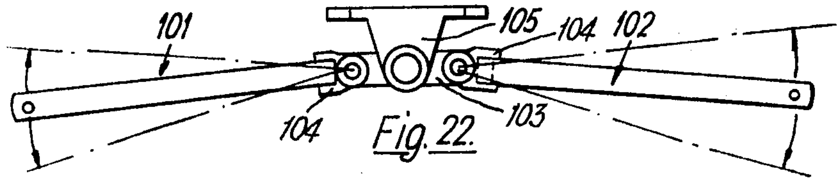

Fig. 22.

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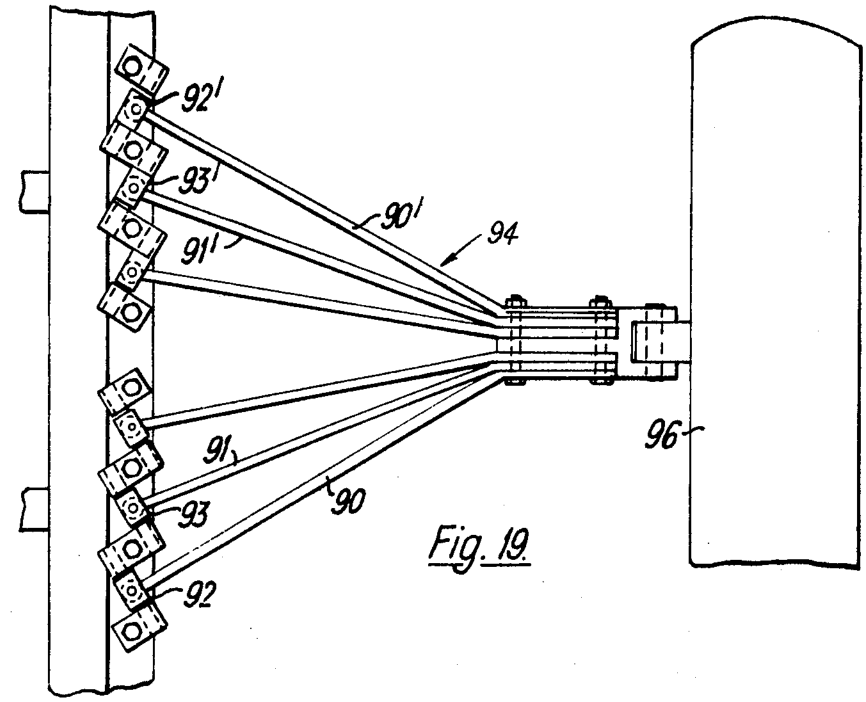

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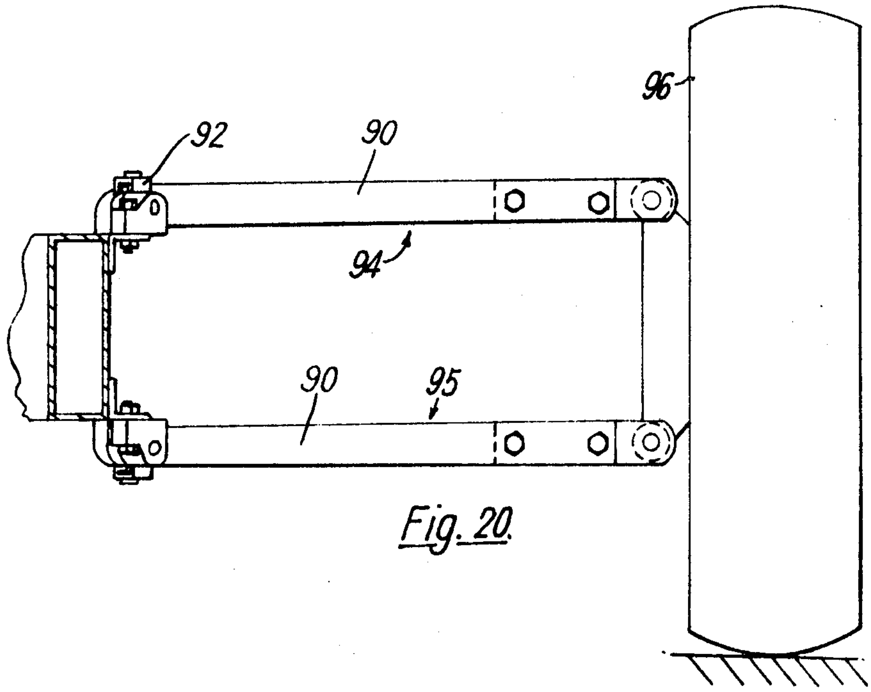

Fig. 20.

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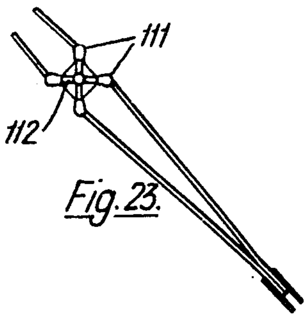

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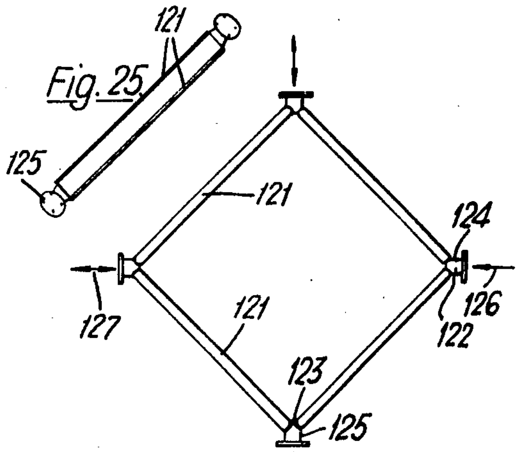

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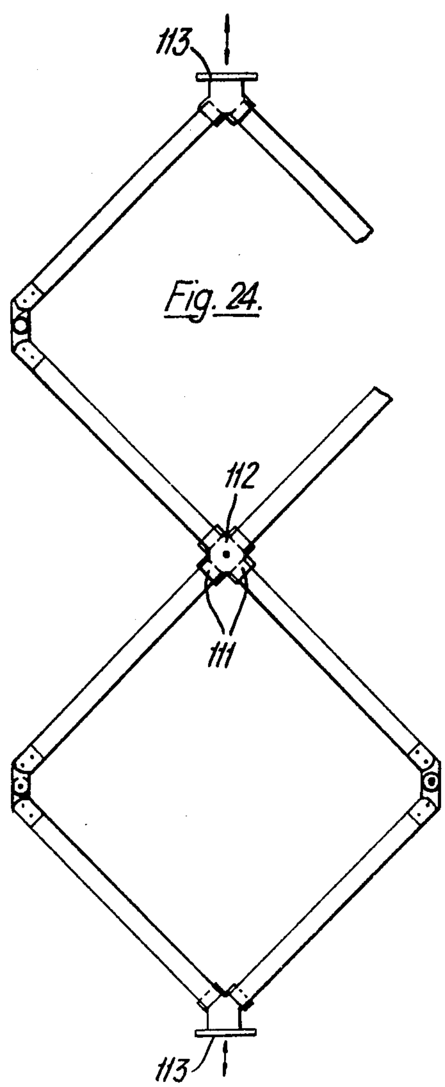

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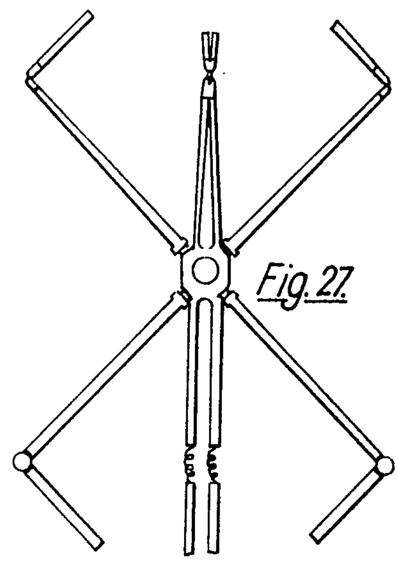

Fig. 27.

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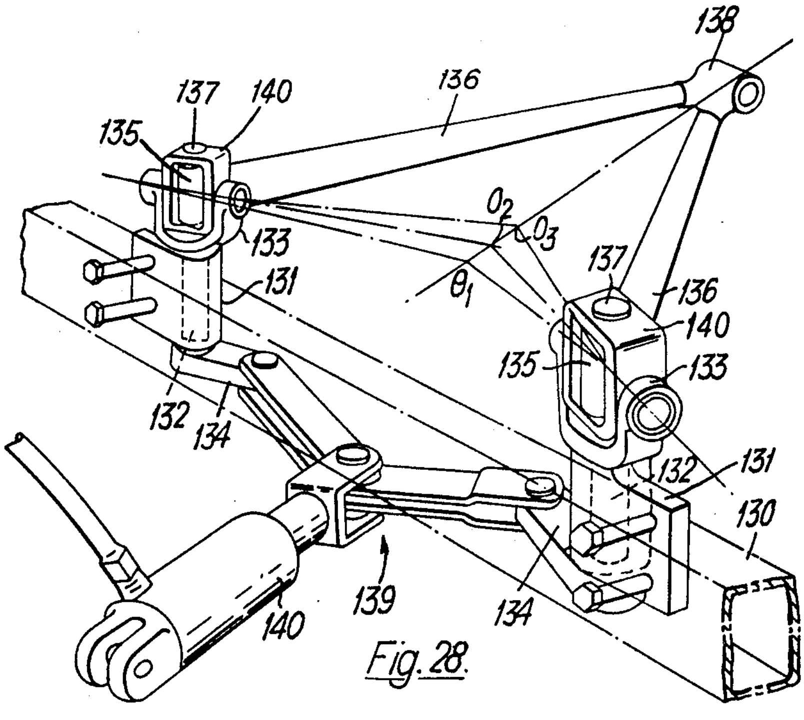

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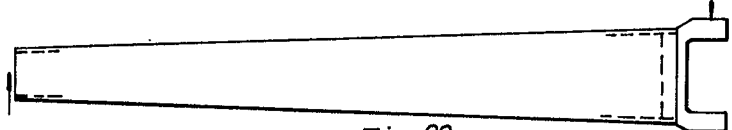

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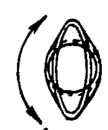

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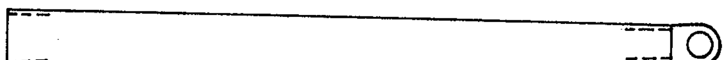

Fig. 30.

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V-TYPE TORSION BAR TILLAGE TINES

The present invention relates to energy storing linkages and especially to spring assemblies therefor.

According to the present invention a spring assembly consists of an elongated resilient member, first and second load attachments positioned on the member at spaced locations, the line extending between them defining a longitudinal axis, said first load attachment being adapted to apply torque to said resilient member, said second load attachment comprising first, second and third journal elements wherein said first journal element and said second journal element are connected by primary pivot means for pivotal movement relative to each other about a primary axis, and the second journal element and third journal element are connected by secondary pivot means for pivotal movement relative to each other about a secondary axis, said third journal element being fixed to or integral with said resilient member.

Preferably the primary pivot axis is inclined to said longitudinal axis and the secondary axis is arranged perpendicular to said primary axis.

According also to the present invention an energy absorbing linkage unit comprises two spring assemblies as defined above wherein the first load attachment is common to both resilient members and transmits equilibrating torque from one resilient member to the other and wherein said first journal element is common to both assemblies.

According also to the present invention an energy absorbing linkage comprises a plurality of energy absorbing units connected together.

Further the invention includes an installation incorporating a linkage unit as referred to wherein angular displacement of the spring through 90° causes an energy input and a further deflection through 90° causes a release of energy. Alternatively the resilient member is pretwisted on assembly so that the first 180° of deflection causes an energy input and a further deflection through 180° causes a release of energy.

The embodiments of the invention will now be described with reference to the accompanying drawings of which:

FIG. **1** is a diagram of a representative spring assembly in accordance with the invention;

FIG. **2** is a detail of a part of the spring assembly in FIG. **1**;

FIG. **3** is a vector diagram of moments as applied to the embodiment in FIG. **1**;

FIG. **4** illustrates a bending moment, torque and stress diagram representative of FIG. **1**;

FIG. **5** is a polar diagram representative of the function of the embodiment in FIG. **1**;

FIG. **6** is a further embodiment of spring assembly according to the present invention;

FIG. **7** is a vector diagram of moments as applied to the embodiment in FIG. **6**;

FIGS. **8** – **14** show various views of an agricultural chisel plow tine embodying the present invention;

FIGS. **15** and **16** show a side elevation and plan view of further chisel plow tine embodiments of the present invention;

FIGS. **17** to **27** illustrate various views of energy storing linkages employed in a variety of possible applications in accordance with the present invention;

FIG. **28** is a view of a spring assembly in which the spring rate can be adjusted in accordance with the present invention; and

FIGS. **29** to **31** are plan, side and end views of a representative embodiment of a resilient linkage member in accordance with the present invention.

FIGS. **1** and **2** show the moments acting on the end of blade **10** at spring attachment **12** which constitutes the second load attachment. The upward or bending moment M is generated by the external forces E applied to the far end of the blade **10** which constitutes the first load attachment. For convenience these are represented by equal and opposite loads. The moment M and the reactive moments R and T are all represented as vectors in the "right hand or clockwise" direction (which could be "counterclockwise," of course, depending on direction of bending moment "M") as shown in FIG **3**. The clevis **14**, which constitutes the second journal element can be connected to the boss **16**, which constitutes the third journal element, by pivot pin **18** and is itself rotatable on a stub axle **19** in a bore **20** in a journal member **22**, which constitutes the first journal element. The longitudinal axis of the resilient member is designated L. The axis of bore **20**, being the primary axis, is designated P. The vertical axis corresponding to the primary axis, is designated S. The axis about which the bending moment M acts is designated N. The clevis **14** cannot generate any moment about axis P because it is free to rotate on stub axle **19**. It can only generate a moment R in the plane of the clevis a component of which will balance M, i.e. $R \cos \theta = M$ where θ is the angle between axes P and L. The other component, $R \sin \theta$, is balanced by moment T which is supplied by a torque arising from twist of the blade. In FIG. **1** the first load attachment **11** is represented by parallel bars **24** between which the blade **10** can slide vertically and which react the torque T. The torque reaction point **26** on blade **10** lies between bars **24**. Thus, the prime purpose of the first load attachment is to react the torque T. By neglecting possible side effects from other forces $T = M \tan \theta$, and whatever the variation in the forces E, there will be a variation in T governed largely by the above relationship.

The deflection of the whole spring assembly is found, at least to a first approximation, by considering the twist in that portion of the blade **10** under torque i.e. between boss **16** and bars **24**. The blade **10** will have inherent characteristics of length, sectional dimension and elastic modulus of material and in general the twist is given by the equation:

$$\phi = T \times c \times l$$

Where ϕ = angle of twist
T = torque
c = torsional rigidity
l = effective length of the blade

In general c is a function of the shear modulus of elasticity, the area of cross section and the moment of area inertia of the cross section. This and the length become an invariable factor for a given blade **10**. The angle of the blade **10** moves up is denoted ψ and in general terms can be regarded as the angular deflection of the torque reaction point **26**. The relationship between the angle θ, ϕ and ψ is:

$$\cos^2 \phi = \frac{1}{\cos^2 U + \frac{\sin^2 U}{\cos^2 \theta}}$$

for a single blade such as **10.** θ being fixed for a particular structure, cos θ will be less than unity and the angular deflection ψ will usually be somewhat larger than the blade twist angle ϕ. The deflection of the blade **10** in bending under moment M will be easily calculated but if the blade has the proportions shown in FIG. **1** it will be much smaller than the deflection due to twisting.

Referring now to FIG. **4**, this represents the bending moment and torsion diagram along the length l of the blade **10** and are not to any particular scale. The bending moment M at the boss **16** generates the stress distribution shown on the right. It will be seen that the extreme fibers of the blade **10** under bending are heavily stressed but that the imposition of a torsion does not add to this stress at all. Similarly, the central region is stressed by the torsion and the imposition of the bending does not contribute any large additional stresses in this region. Thus it affords the possibility of being able to use some material of lower cost than spring steel.

The blades **10** can be designated to resist the bending loads which are to be experienced and can then be converted into springs of suitable characteristics by selection of θ (the angle of inclination) between the primary axis P and the longitudinal axis L. When $\theta = 0°$ there will be no reaction at all from the second attachment and when $\theta = 90°$ there will be no torque applied to the blade and no corresponding torsional deflection.

The broader aspects of the invention as regards deflection and movement will now be described with reference to FIG. **5.** The spring may be mounted so that it can rotate in a vertical plane as seen in FIG. **1** through one or more complete revolutions. The first load attachment would be such as to preserve its torque reaction properties throughout the movement which would be generally about axis N. For this purpose the bars **24** have been shown as parts of spaced circular rails in FIG. **1.** Assume that the blade **10** is shown in its free state in FIG. 1 and that movement occurs in an upward and generally anti-clockwise direction from this position. Assume also that the angle ψ increases from zero during the movement. During the first 90° of movement the spring blade **10** will be deflected and will be absorbing energy until 90° position is reached. Further movement from this point will reduce the deflection progressively and there will be an energy release until when $\psi = 180°$ and the spring will be back in its free state. During the third and fourth quadrants of movement there will occur an energy storage followed by an energy release respectively.

FIG. **5** shows a polar diagram from which the geometry of the spring deflection in terms of the three angles can be studied. This diagram is based on the formula previously given. The radial lines define angular displacement ψ, the contours define the value of θ (the angle between the primary and longitudinal axes) and the angle of twist ϕ is given by a vertical scale alongside of the polar diagram. It has just been described how energy input occurs in the first and third quadrants and energy release occurs in the other two quadrants as the angle ψ increases. However at 90° or T.D.C. (top dead center) the spring goes 'on center' with maximum energy locked in and this energy may be released by either continuing the rotation or by reversing it. Reference is made to the areas shaded at $\phi = 5°$ to 10° and $\phi = 25°$ to 30° for the $\theta = 30°$ contour. The energy, which is proportional to these shaded areas put into the spring for 5° of deflection is much less nearer the T.D.C. position than earlier during the total deflection. This means that the energy storage rate at high deflection is lower than at lower deflection. In the past considerable thought has been devoted to achieving effective spring characteristics of this kind by the use of elaborate linkages. The release of energy is at an increasing rate as the blade approaches zero-torsional deflection.

It is possible to make a storage cycle extend over 180° by employing pretwist of the resilient blade **10.** Such an arrangement is shown in FIG. **6.** The blade **10** is twisted and constrained at the first load attachment by means of a block **25** but is permitted to move in an arc between the bars **24** as before. Stops **28** are introduced to constrain the blade at the second load attachment end. With reference back to FIG. **5**, assuming in a spring where $\theta = 45°$ of pretwist, i.e. $\phi = +45°$, has been introduced, then the scale of twist angles is changed from the range $-45°$ to 0° to $-45°$ to the range of 0° to 45° to 90° and hence the zero for this scale is move to **0'** at B.D.C. (bottom dead center). Consequently, the 180° angular from B.D.C. to T.D.C. will cause 90° of twist in one direction, i.e. a continuous energy input over two quadrants. Movement through the remaining two quadrants (or reverse movement) will release the energy. The amount of pretwist does not have to be as much as **45** degrees and in the case where $\psi = 17°$ and $\theta = 30°$ the base line from which to measure angle ψ would be the 150° – 330° radial line and the T.D.C. or maximum twist point would occur after 120° of angular deflection.

If a stop **28** is arranged, say **5** degrees after T.D.C. point, the spring, having passed over T.D.C., will not return and will stay against the stop thus constituting an over-center device.

FIG. **6** also shows an arrangement in which a spring assembly is provided with a second load attachment having a provision for selectively varying the angle θ. The first journal element **22** is mounted on a rail **32** and can be clamped in position by a screw **34.** It will be seen that when the angle between N and P becomes zero the axis P is coincident with axis N and when θ becomes negative the blade **10** twists in the opposite direction on receiving the same external loads. The corresponding vector diagram when θ becomes negative value is shown in FIG. **7.**

Reverting to a 360° movement with two storage and two release cycles it should be noted that one storage/-release cycle takes place by torsional deflection of the blade **10** in one angular directioni.e. plus ϕ, and during the other cycle in the other direction i.e. $-\phi$. It is possible to restrain the blade **10** against angular deflection only in one of these directions and to let it go free in the other.

In the foregoing description it has been assumed that one end of the assembly is fixed and the other parts of the spring assembly move relative to it. Clearly the other end could be fixed if desired without affecting the fundamental operation.

If the characteristics of the blade and the choice of angle θ is such that a large twist in the order of 90° occurs for a given load then the total bending characteristics will change because of progressive change in the bending section along the blade. Thus the total deflection may have a large proportion of it attributable to bending deflection. From a stress point of view this is not desirable because the stresses become additive but it may be acceptable in certain applications.

One of the difficulties of the spring assembly described in FIGS. **1** and **6** is the nature of the first load attachment which has to provide a torque reaction irrespective of the angular deflections and the provision of a static torque reaction member over a large arc is often inconvenient.

This difficulty is overcome by arranging the spring assemblies in pairs so that the torque from one spring equilibrates that from the other. This can be done in at least two ways exemplified in FIGS. **8** to **14** and FIGS. **15** and **16.**

FIGS. **8** to **10** show in side elevation, lower plan view and partial end elevation, a chisel plow tine assembly **40** mounted on a toolbar **41** which carries a movable bracket **42** clamped to it by a bridge piece **43** and bolts **44.** A spring attachment assembly **45** is bolted to the underside of the bracket **42** by bolts **46.** The support assembly **45** consists of two main parts. These are shown, for the sake of clarity, separated from each other in FIGS. **13** and **14** respectively and in assembled position in FIG. **12.** One part is constituted by a block **47** which has a central circular aperture **48.** The bolts **46** pass partially through this aperture as seen in FIG. **11.** Two keyways **49** and **50** are provided in the aperture **48** of the block at diametrically opposed positions. A projecting boss **51** is bored to accept a long pin **52** shown in FIG. **9.** The other part is a clevis support **53** shown in FIG. **12** and has a cylindrical central portion **54** having an annular groove **55** cut in its periphery. The central portion **54** is adapted to fit snugly into the aperture **48** in block **47** and the bolts **46,** when inserted, partially pass through the groove **55** and hold the central portion **54** in position while enabling it to be rotated. Two bosses **56** each project at equal angles to the central portion and each of these is bored to receive stub shafts **57** which each carry at their outer ends a clevis **58.** FIG. **12** is a plan view of the clevis support assembly.

FIG. **14** shows a detail of one clevis **58** pinned to a blade adaptor plate **59** with a pin **60** which is fast with the latter by having a bolt **61** pass through both. The bolt **61** and a second bolt **62** also pass through the end of a spring blade **63** to hold the blade end fast with the plate **59.**

Thus the construction is such that once the clevis support **53** is fixed in the block **47** by a key in an appropriate keyway **49** or **50,** the clevises **58** are permitted to rotate freely in the bosses **56** subject to the constraints applied by the blade **63.** The blade support plate **59** is also permitted to swivel freely on the pin **60** but is also subject to spring force constraint.

The spring blades **63** on each side side extend to the right, as seen in FIGS. **8** and **9,** to form a mounting for the tine **70.** A torsion lockout yoke member **71** is applied at the end of the straight portions of the blades **63.** This is a bracket having two slots **72** through which the blades **63** pass with small clearance.

The construction of chisel plow tine assembly described in FIGS. **8** to **14** is designed to transmit certain loads to the frame by means of tool bar mounting assembly **40.** These loads are all generated at the tine and in general their resultant is in a rearward direction and displaced below the apex of the two blades which constitute a Vee. These originating forces are carried into the structure and appear as a force and a bending moment at the yoke **71.** The force places both blades **63** in pur tension and this is carried through the clevises **58** and the support assembly **45** to the bracket **42** and tool bar **41.** Negligible deflection occurs during the transmission of this force through the spring assembly. The bending moment present at the apex is divided into equal parts and each passes along a blade **63** to a respective plate **59** as a constant bending moment. This is applied to the respective clevis **58** with the results described previously with reference to FIG. **1** except that the torques carried by the two blades **63** are equal and opposite and are mutually equilibrated by the yoke **71.**

FIGS. **15** and **16** show in side elevation and plan view a chisel plow tine mounted on a parallel arm linkage unit wherein the spring assemblies are mounted one above the other. The tine point **70** is simply mounted on a curved bar **74** which is bolted to the foot of a standard **75.** The standard **75** is carried by two vertically spaced blade spring assemblies constituted by two blades **76** and **77** pivoted on respective clevises **78** and **79** themselves rotatable in clevis supports **80** and **81** which are integral with or fixed to one of a pair of clamp brackets **83** and **84.** Bolts **85** hold the clamp brackets on to a square toolbar **41.** It will be seen that the clevises **78** and **79** are equally and oppositely orientated on each side of a common vertical plane of the blades **76** and **77** and transmit equal and opposite torques arising from vertical loads applied to the standard **75.** It should be recognized that horizontal forces and bending moments arising from ground contacts are likely to be taken as tension and compression loads in the blades and only vertical shear forces causing bending moment at the front ends of the blades **76** and **77** are likely to cause torsional deflection. With particular reference to FIG. **16** it will be seen that in the unloaded state the two clevises form a hinge axis about which the tine can pivot quite freely but as soon as it is loaded in bending and by draft loads it will tend to centralize partially due to the uneven torsion reactions generated. It can be arranged that a tine of this kind be tuned to vibrate from side to side to provide a soil shattering action if such is desired.

Turning now to arrangements involving multiples of paired spring assemblies which can be conveniently termed spring units, FIGS. **17** and **18** show a tine wherein several Vee springs **81, 82, 83** are nested together and each Vee blade provides its own torque equilibration across the apex of its Vee. The opposite front ends of the Vees are mounted in opposite members **84** which constitutes the first journal element referred to. The inner Vee blade **81** only is clamped between the locking plate **87,** the others being free to accommodate differential movement without inhibiting the spring action.

FIGS. **19** and **20** show an arrangement employing several single but paired blades **90—90′, 91—91′,** etc. In this case each blade has its own individual pivot pin **92—92′** etc. which again constitutes the second journal element. The arrangement as a whole consists of similar upper and lower assemblies **94** and **95** suitable for mounting a wheel **96** of a vehicle.

While the arrangements shown in FIGS. **17** to **20** show arrangements of spring units in parallel, the following figures show various forms of end connected spring units.

FIGS. **21** and **22** show in plan and side elevation respectively an equalizer bar wherein two units **101** and **102** are mounted on a single support assembly **103** the latter constituting the first journal element for all four of the second journal members **104** and is itself journalled to rock in a pair of hangers **105.** Loads would be applied or reacted at the apices of the two units which are bolted together and be carried through to the hangers **105.**

FIGS. **23** and **24** show an arrangement of multiple spring units wherein one end of each unit is suspended by a clevis **111** journalled in a common first journal element **112** and the apices are pin connected together. The units, in groups of four are connected in a plane to make a large spring complex capable of large deflections to the extent that the two load pads **113** meet in the middle.

FIGS. **25** and **26** show an arrangement wherein both ends of the blades **121** are mounted on clevises **122, 123** which are journalled in respective first journal elements **124, 125** the multiplicity of springs defining a complex similar to FIGS. **25** and **26.** It should be noted in this case however that load is applied by arrows **126** as a linear force through clevis **124** to the blade. This results in a moment at the other end which generates torque at **124** in one direction. Similarly a torque arises from the reaction forces **127.** It is arranged that the torques cause the same twist in the blade i.e. one in one direction at one end and the other in the other direction at the other end.

FIGS. **27** shows an arrangement wherein blade units having pin connected apices are arranged symmetrically in a three dimensional lattice. Three double units would seem to be the minimum necessary to sustain a complex of this kind. More double springs would be incorporated with an orange-segment effect and would be limited at one level by the width of the spring units. Several levels, columns, and ranks of spring groups could be arranged if desirable.

FIG. **28** shows a practical arrangement of the adjustment of angle θ discussed with reference to FIG. **6.** In this embodiment, which is a wishbone type suitable for a vehicle suspension, the chassis member **130** carries bosses **131** which each rotatably support a shaft **132** having a yoke **133** at one end and a lever arm **134** at the other. The yoke **133** defines the first journal element and an open rectangular bracket **140,** runnioned in the yoke **133,** defines the second journal element. A boss **135** welded on to the inboard end of each spring blade **136** constitutes the third journal element and is secured in the bracket **140** by a pin **137.** The apex ends of the resilient members **136** are fixed in a pierced end fitting **138** capable of taking load. The lever arms **134** are both connected by a Y link **139** to a hydraulic ram which is remotely operated.

Operation of the ram **140** causes rotation of levers **134,** shafts **132** and yoke **133** to change the angle θ. This means that a driver of a truck could adjust the ridge height to a required level as the truck is loaded and also be given an indication, by reading the oil pressure required to achieve ride height, of the load carried.

The resilient members used in the FIG. **28** embodiment could be of the type shown in FIGS. **29, 30, 31** which are a side elevation, plan and end elevation, respectively. The resilient member is a thin walled tube tapering in side elevation and in plan and changing in section from an elipse at the pivot pin end to a circle at the apex end. This enables greater resistance to bending while permitting a fairly uniform torsional stress level to be maintained along the length of the tube.

When the spring unit is in the form of a Vee it will be appreciated that the torque about the longitudinal axis of one arm of the Vee is not in the same plane as the torque in the other arm. Hence a component of the two torques will equilibrate each other but the other components will be additive and will be balanced by part of the applied load. Thus the equation:

$$\cos^2 \phi = \frac{1}{\cos^2 \psi + \frac{\sin^2 \psi}{\cos^2 \theta}}$$

will not give the true deflection and it will be modified by the incorporation of a further term which is a function of B which is the half angle of the Vee. Moreover, the pattern of energy storage in the storage cycle will differ from the pattern of energy release during the release cycle though the total energy stored and given up will be the same.

It should be understood that the primary and secondary axes can be defined by rubber bushes in torsion or like devices wherein sliding of surfaces over one another may not take place.

The main advantages of the present invention are that it affords spring means which can be constructed inexpensively from inexpensive materials and with good scope for unconstrained design.

I claim:

1. A spring assembly consisting of an elongated resilient member, first and second load attachments positioned on the member at spaced locations, the line extending between them defining a longitudinal axis, said first load attachment being adapted to apply a torque load to said resilient member, said second load attachment comprising first, second and third journal elements wherein said first journal element and said second journal element are connected by primary pivot means for pivotal movement relative to each other about a primary axis that is inclined to said longitudinal axis and the second and third journal elements are connected by secondary pivot means for pivotal movement relative to each other about a secondary axis said third journal element being integral with said resilient member.

2. A spring assembly according to claim **1** wherein said secondary axis is arranged perpendicular to said primary axis.

3. A spring assembly according to claim **2** wherein said secondary axis intersects the primary and longitudinal axes at their junction.

4. A spring assembly according to claim **1** wherein said first load attachment is adapted to apply linear loads to said resilient member.

5. A spring assembly according to claim **1** wherein said first load attachment is adapted to apply bending loads to said resilient member.

6. A spring assembly according to claim **1** wherein said first load attachment is similar to said second load attachment.

7. A spring assembly according to claim 1 having means for varying the spring rate thereof comprising a carrier for said first journal element movable to vary the angle between said longitudinal axis and said primary axis.

8. An energy absorbing spring assembly according to claim 1 and comprising two resilient members wherein the first load attachment is common to both resilient members and transmits equilibrating torque from one resilient member to the other and said first journal element is common to both assemblies.

9. A spring assembly according to claim 8 wherein said resilient members are spaced from each other in the direction of the secondary axis and said first load attachment is attached to each said resilient members and transmits equilibrating torque in bending.

10. A spring assembly according to claim 8 having means for varying the spring rate of the unit comprising carrier means for both said first journal elements movable to vary the angles between said longitudinal axes and said primary axes.

11. A spring assembly according to claim 10 wherein said carrier means is constituted by fourth journal elements which are connected to said first journal elements by tertiary pivot means defining respective carrier axes.

12. A spring assembly according to claim 11 wherein said fourth journal elements are movable by remote power means.

13. A spring assembly according to claim 8 having means for holding said resilient members in a preloaded state.

14. A spring assembly according to claim 13 wherein said preloaded state is a torsional deflection of the resilient members and said means for holding said resilient members is a stop which prevents their rotation.

15. A spring assembly according to claim 14 wherein said resilient member is given a pretwist of up to 90° about its longitudinal axis and between the secondary axis and the first load attachment.

16. A spring assembly according to claim 14 wherein said first journal elements are mounted in a rotary block capable of rotation in a housing to impart said torsional deflection.

17. A spring assembly according to claim 15 which includes a plurality of units wherein all of said units have a common first load attachment.

18. A spring assembly according to claim 17 wherein all of said units have a common primary axis.

19. A spring assembly according to claim 17 wherein at least two of said units have separate primary pivot means defining separate primary axes.

20. A spring assembly incorporating a plurality of units according to claim 11 wherein each unit is end-connected to another unit.

21. A spring assembly according to claim 20 which includes at least four units wherein each unit is connected at one end to another through a common first load attachment and at the other through a common first journal element.

22. A spring assembly according to claim 21 wherein said units all lie in a common plane.

23. A spring assembly according to claim 21 wherein at least six units are arranged in three dimensional form.

24. A spring assembly according to claim 20 which includes at least four units wherein each unit is connected at each of its ends to one other through a load attachment constituted by first, second and third journal elements.

25. A spring assembly according to claim 24 wherein at least four units are arranged to lie substantially in a plane.

26. A spring assembly according to claim 24 wherein at least six units are arranged in three dimensional form.

27. A spring assembly incorporating a unit according to claim 11 wherein the angular displacement of the spring through 90° causes an energy input and a further deflection through 90° causes a release of energy.

28. A spring assembly according to claim 27 wherein a preload equivalent to a predetermined angular displacement of the spring is imposed there being provided a limit stop permitting deflection just in excess of 90° to permit the spring to deflect to this limit and stay against it.

29. A spring assembly incorporating a unit according to claim 8 wherein the resilient member is pretwisted on assembly so that the first 180° of deflection causes an energy input and a further deflection through 180° causes a release of energy.

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