July 3, 1962 S. WILDER, JR 3,042,333

TENSION MAINTAINING WIND OR UNWIND CONTROL

Filed June 5, 1961 3 Sheets-Sheet 1

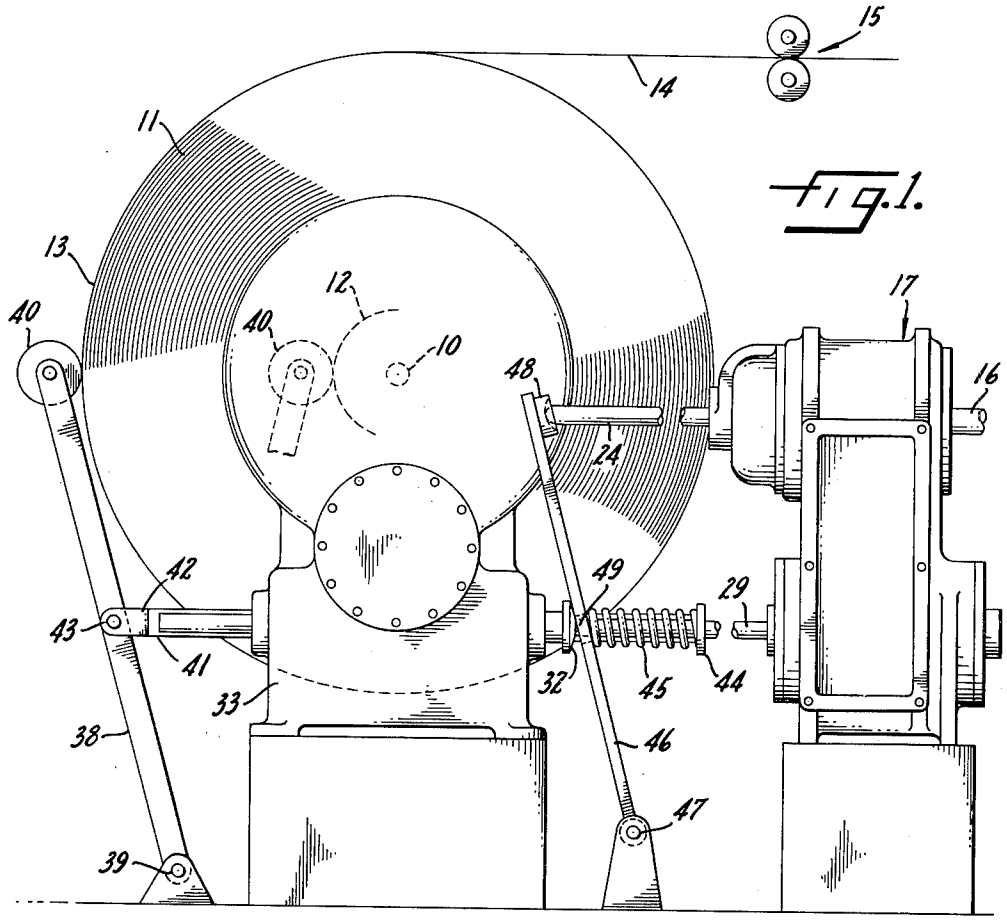


Fig. 1.

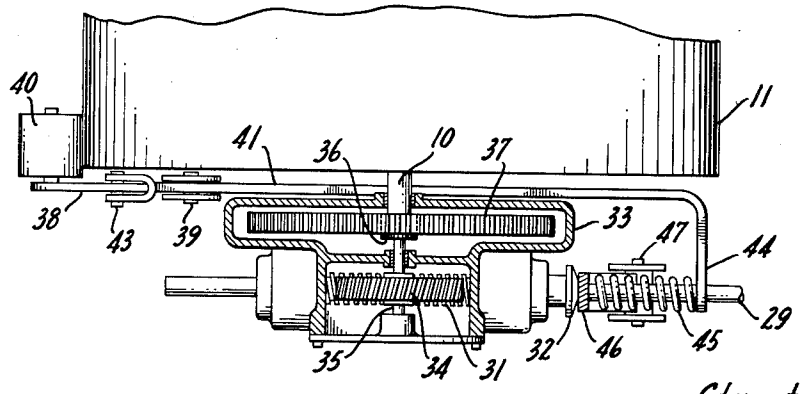


Fig. 2.

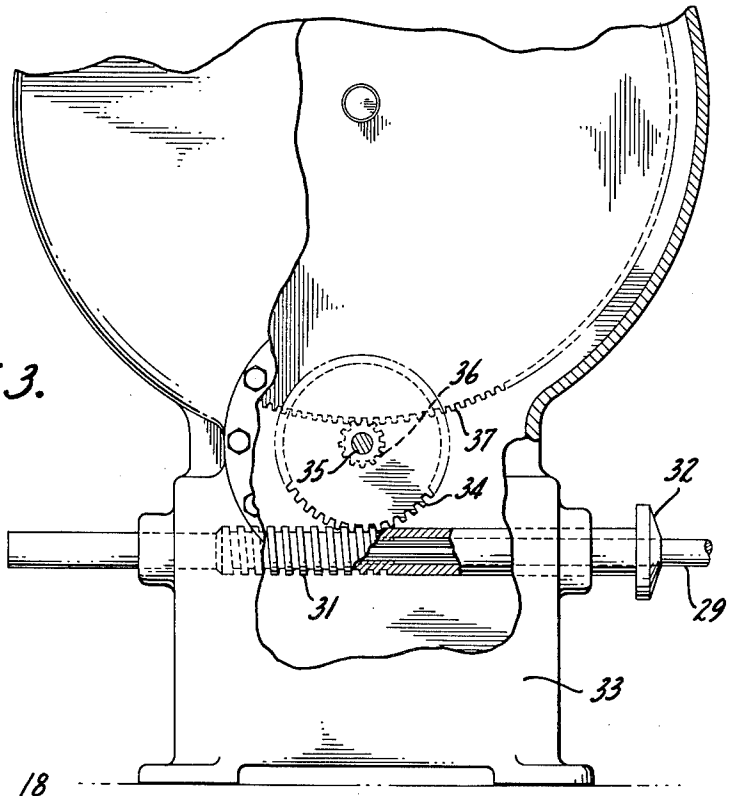
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TENSION MAINTAINING WIND OR UNWIND CONTROL

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Fig. 3.



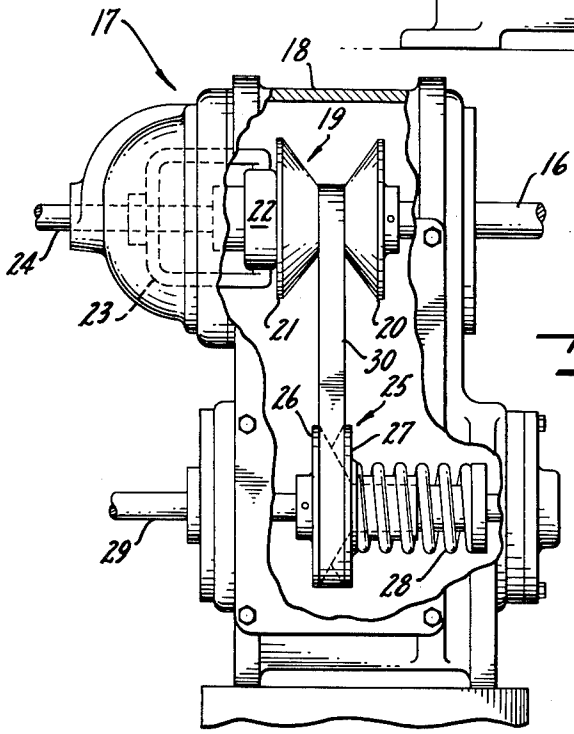


Fig. 4.

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TENSION MAINTAINING WIND OR UNWIND CONTROL

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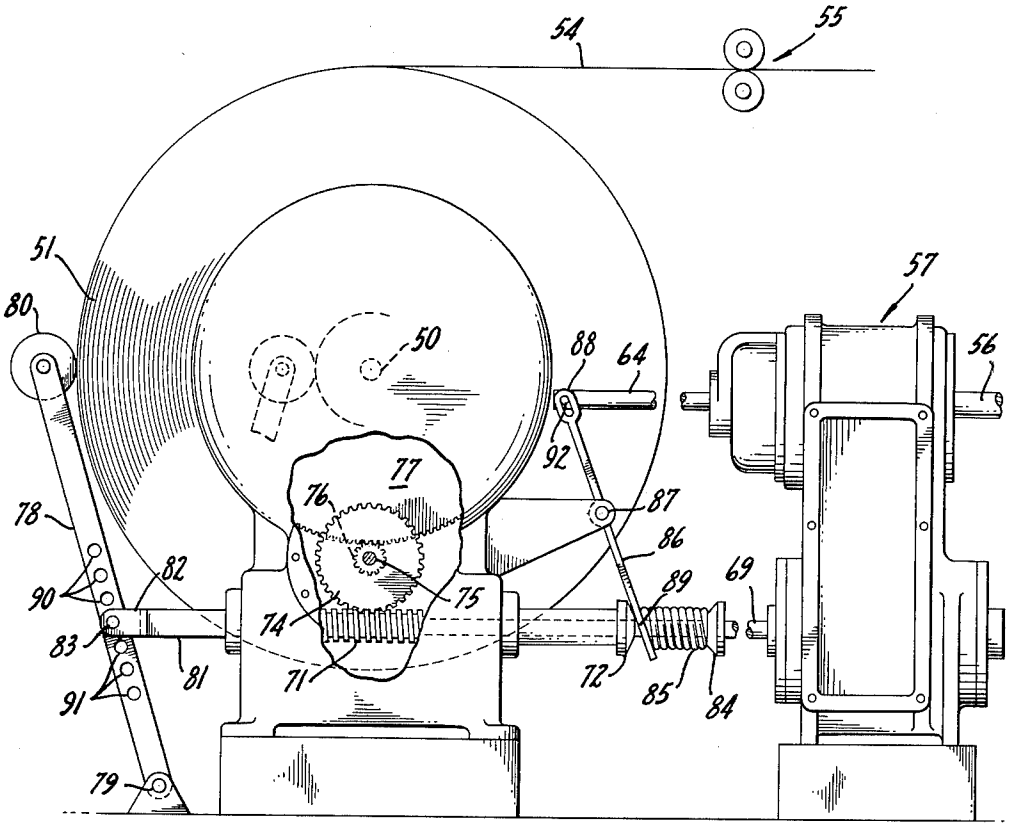


Fig. 5.

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United States Patent Office 3,042,333 Patented July 3, 1962

3,042,333
TENSION MAINTAINING WIND OR UNWIND CONTROL

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Filed June 5, 1961, Ser. No. 115,002
9 Claims. (Cl. 242—75.5)

The present invention relates to a control for maintaining desired tension in a strand, web or sheet of flexible material as that material is wound upon, or unwound from, a storage roll. In many operations in industry, such material is delivered to, or carried away from, such a roll at a controlled linear velocity; and it will be obvious that, because of the changing effective radius of material wound upon such a roll, the angular velocity of the roll must be continually reduced during a winding operation or continually increased during an unwinding operation.

In many applications of either winding or unwinding, it is desirable to maintain constant tension in the length of material which is travelling linearly relative to the roll; but in some applications, particularly in connection with winding operations, a so-called "tapered tension" is desired whereby the outer layers of the roll are more tightly wound than the inner layers, or vice versa. It is an object of the present invention to provide control means whereby constant tension may be maintained or whereby, alternatively, tapered tension of either type may be achieved.

In previously-available devices for controlling the tension in systems of the character here under consideration, a dance roll or a whip roll supported upon the material at a point removed from the effective peripheral surface of the material on the roll has been used, or, through one expedient or another, it has been attempted to sense torque and control tension without contact with the material. A dance or whip roll has been found to be impractical for very wide webs or groups of serried strands, because of sagging or deflection which occurs in a whip roll shaft when it must cover a wide expanse. The devices which seek to sense tension without direct contact with the material are generally complex, expensive and difficult to maintain.

A further object of the invention, then, is to provide a mechanically-simple control of the character under consideration in which the element bearing upon the material to be wound or unwound senses merely the effective radius of the material currently on the roll and therefore is indifferent to the effective width of the material, since the information to which it is responsive may be taken from a very narrow portion of the material selected at any point in the width of the material.

A further object of the invention is to provide, in a control of the character under consideration, an element which is movable in response to the current torque required at the storage roll, which will always seek a position within its path of movement corresponding to the current torque required at the roll and which is resiliently restrained against movement beyond such a corresponding position by a force which is variable in accordance with the current value of the effective radius of the material on the storage roll.

A still further object of the invention is to provide, in a control of the character described, means whereby constant tension in the material may be maintained during an unwinding or winding operation or, by adjustment, tapering tension in the material may be achieved during a winding or unwinding cycle.

Further objects of the invention will appear as the description proceeds.

To the accomplishment of the above and related objects, my invention may be embodied in the forms illustrated in the accompanying drawings, attention being called to the fact, however, that the drawings are illustrative only, and that change may be made in the specific constructions illustrated and described, so long as the scope of the appended claims is not violated.

FIG. 1 is a somewhat diagrammatic end elevation of a control constructed in accordance with the present invention and illustrated in operative association with an unwinding storage roll;

FIG. 2 is a horizontal section through parts of the control, a fragment of the storage roll with material wound thereon being shown in plan;

FIG. 3 is an enlarged, fragmental elevation of parts of the control mechanism, with portions of the casing broken away for clarity of illustration;

FIG. 4 is an elevation of one form of speed varying transmission which may advantageously be used in the control of the present disclosure; and

FIG. 5 is a view similar to FIG. 1 but illustrating a modified form of control associated with a winding storage roll and provided with adjusting means whereby tapering tension may be achieved.

Referring more particularly to FIGS. 1 to 4, it will be seen that I have illustrated a storage roll **10** upon which flexible material **11** is adapted to be wound. The reference numeral **12** indicates the minimum effective radius of material on the roll and the reference numeral **13** indicates the maximum effective radius of material on the roll. The reference numeral **14** indicates a length of material moving linearly away from the roll under the influence of forwarding means, such as the nip rolls **15**, driven from an external source (not shown). Usually, but not necessarily, the forwarding means **15** will be so driven as to move the length of material **14** at a constant linear velocity.

The external power source is suitably connected to drive a rotary means **16** at a constant ratio with respect to the forwarding means **15**. In the illustrated embodiment of the invention, the rotary means **16** may be the input shaft of a speed varying transmission indicated generally by the reference numeral **17**. One suitable form of speed varying transmission is illustrated in FIG. 4 and comprises a housing **18** in which the shaft **16** is journalled. An expansible V-pulley indicated generally by the reference numeral **19** comprises a coned disc **20** fixed to the shaft **16** and a mating coned disc **21** axially movable relative to the disc **20**. A thrust bearing **22** is operatively connected to shift the disc **21** and has associated therewith a yoke **23** from which extends an axially shiftable operating member **24**. A second expansible V-pulley indicated generally by the reference numeral **25** comprises a coned disc **26** fixed to the output shaft **29** of the transmission **17** and a mating coned disc **27** axially shiftable relative to the disc **26** and urged toward said disc **26** by a spring **28**. A V-belt **30** provides a driving connection between the pulleys **19** and **25**; and it will be clear that, as the disc **21** is moved toward the disc **20**, the belt **30** will be crowded outwardly between said discs and will be pulled more deeply between the disc **26** and **27**, thereby forcing the disc **27** away from the disc **26** against the tendency of the spring **28**. Thus, the effective diameter of the pulley **19** will be increased and the effective diameter of the pulley **25** will be decreased to increase the speed of the shaft **29** relative to the speed of the shaft **16**. Conversely, when the disc **21** is moved away from the disc **20**, the belt **30** may move more deeply between the discs **20** and **21** as the spring **28** crowds the disc **27** toward the disc **26** to force the belt outwardly relative to said discs **26** and **27**, thereby decreasing the speed of the shaft **29**.

A worm **31** is mounted for rotation with the shaft **29**

and for limited reciprocation relative thereto in suitable bearings in a housing **33**, one end of said worm projecting from the housing and carrying, at its extremity, a head **32**. Within the housing **33**, the worm **31** meshes with a worm wheel **34** fixed to a shaft **35** which carries, as well, a pinion **36** meshing with a bull gear **37** fixed to rotate with the storage roll **10**.

A first lever **38** is pivoted at its lower end upon a fixed fulcrum **39** near the housing **33** and at its distal end, said lever carries a follower roller **40**. A link **41** terminates at one end in a clevis **42** which is pivoted, as at **43**, to the lever **38** intermediate the ends thereof. At its opposite end, the link **41** terminates in a toe **44** which straddles, and is guided upon, the shaft **29**. A coiled spring **45** is sleeved on said shaft and is confined between the link toe **44** and the head **32** of the worm **31**.

A second lever **46** has one end pivotally mounted on a fixed fulcrum **47** near the housing **33** and at its opposite end carries a pad **48** cooperatively engaging the distal end of the shiftable element **24**. An intermediate portion of the lever **46** straddles the shaft **29** and is engaged by that end of the spring **45** which is adjacent the worm head **32**.

When a loaded storage roll **10** is mounted as shown, the lever **38** will be in its solid line position, the lever **46** will be in the position of FIG. 1 and the transmission **17** will be in the low-output-speed condition as shown. The free end of the material will be drawn off the roll and threaded between the nip rolls **15** and will be led thence to a point of use (not shown). Now, when the external power source is energized, the forwarding means **15** and the rotary means **16** will be driven at a constant ratio.

As material is drawn off the storage roll by the forwarding means **15**, force is applied thereby to tend to rotate the roll **10** in a clockwise direction. Through the gearing **37**, **36**, **34**, that turning force is converted into a force tending to shift the worm **31** to the right, as viewed in FIG. 3. Such movement of the worm is resiliently resisted by the spring **45** to a degree dependent upon the current setting of the abutment toe **44**; and it will be apparent that the current position of said toe is in turn dependent upon the position of the roller **40** which measures the current effective radius of the material **11** on the storage roll.

The parts are so proportioned and designed that the external power source drives the worm **31** in a direction corresponding to the counter-clockwise rotation of the worm wheel **34**, but at a velocity such as to resist such worm wheel rotation. Thus, clockwise rotation of the roll **10** is resisted to a degree dependent upon the current force of the spring **45** and upon the current speed of rotation of the worm **31**. It follows that the tension in the run **14** of material is a function of current effective radius of the material on the roll **10** and current spring resistance to axial movement of the worm. If tension in the run **14** should increase beyond a predetermined value, the tendency of the worm to move to the right will be correspondingly increased to compress the spring **45** and swing the lever **46** in a clockwise direction, thus shifting the element **24** and the disc **21** to the right to increase the speed of the shaft **29** to arrest rightward movement of the worm. Similarly, if tension in the run **14** should decrease below the predetermined value, the rightward tendency of the worm will be correspondingly decreased, the spring **45** will shift the worm to the left, the lever **46** will be moved in a counter-clockwise direction and the speed of the shaft **29** will be correspondingly decreased.

As material is progressively drawn off the roll **10**, the effective radius of material on the roll is correspondingly decreased and tension in the run **14** will tend to rise, but the reduction in effective roll radius will tend to decrease the torque applied to the roll. The above-described tendency of the worm to compress the spring **45** will arise; but because of the above-described torque-reduction effect, a proper balance would not be achieved if the abutment **44** were stationary or even if it moved at a rate equal to the rate of worm axial movement. To compensate for this difference, the current position of said abutment **44** is placed under the control of the current effective radius of material on the roll **10**.

It will be seen that the spring **45** acts on the toe **44** of the link **41** to hold the roller **40** yieldably in contact with the perimetral surface of material on the roll **10**, thus measuring the current effective radius of such material. The parts are so proportioned and designed that, as the roller **40** follows progressive reduction of such effective radius, the rate of travel of the abutment **44** will exceed the rate of rightward movement of the worm **31** to such a degree as to compensate for the torque reduction resulting from reduction in effective radius, to maintain constant tension in the run **14** of material.

The necessary proportions may be defined as follows:

$$l' = \frac{w.l''}{s}$$

Where

l' is the distance from fulcrum **47** to the point **49** at which worm head **32** engages lever **46**;
w is the total range of axial movement of worm **31**;
l'' is the distance from fulcrum **47** to pad **48**; and
s is the total range of travel of shiftable element **24**.

$$L' = \frac{(w+F)L''}{(r'-r'')}$$

Where

L′ is the distance from fulcrum **39** to point **43**;
L″ is the distance from fulcrum **39** to roller **40**;
r' is maximum roll effective radius **13**;
r'' is minimum roll effective radius **12**; and
F is spring length change or the difference between the axial force of the worm at the beginning of an unwind cycle and the axial force of the worm at the end of such a cycle, divided by the spring rate in pounds per inch.

The axial force of the worm at the beginning of a cycle is

$$\frac{2r'T}{2R.p}$$

Where

T is desired tension in material run **14**;
R is the pitch ratio of gear **37** to pinion **36**; and
p is the pitch radius of worm wheel **34**.

The axial force of the worm at the end of a cycle is

$$\frac{r''T}{Rp}$$

Now, if it be assumed that:

$r'=12.5''$
$r''=2.5''$
$w=4''$
$p=2.5''$
Spring rate=400 pounds/inch
$R=20$ to 1
$T=2000\#$
$s=8''$

then

Torque at storage roll at start of unwind cycle
$=12.5\times2000=25{,}000$ inch-pounds

Torque at storage roll at end of cycle
$=2.5\times2000=5000$ inch-pounds

Torque at worm wheel at start of cycle

$$=\frac{25000}{20}=1250 \text{ inch-pounds}$$

Torque at worm wheel end of cycle

$$=\frac{5000}{20}=250 \text{ inch-pounds}$$

Axial force at worm at start of cycle

$$=\frac{1250}{p}=\frac{1250}{2.5}=500\#$$

Axial force at worm at end of cycle $=\frac{250}{p}=\frac{250}{2.5}=100\#$

Difference in necessary spring force from start to end of cycle $=500-100=400\#$

Required length increase of spring

$$=\frac{\text{Required spring force change}}{\text{spring rate}}=\frac{400}{400}=1 \text{ inch}$$

Since $w=4''$ (assumed), t must be $4''+1''=5''$, where t is total range of travel of abutment toe **44**.

Since

$$l'=\frac{w.l''}{s}=\frac{4l''}{8}=\frac{l''}{2}$$

point **49** must be midway between fulcrum **47** and pad **48**.

Since toe **44** must travel 5" while roller **40** moves from r' to r'' ($12.5''-2.5''=10''$), point **43** must be midway between fulcrum **39** and roller **40**.

Thus it will be seen that, as the unwinding cycle of the disclosed assembly progresses, the rate of travel of the abutment toe **44** exceeds the rate of axial movement of the worm **31** in the same direction by an amount which is constantly proportional to the rate at which the effective radius of the material wound on the storage roll **10** decreases. It will be seen, further, that the force with which the spring **45** resists axial movement of the worm **31** under the influence of the driving reaction of said worm upon the worm wheel **34**, is progressively reduced at a rate equivalent to the rate of reduction of the effective radius of material wound on the roll **10**.

In FIG. 5, I have illustrated a similar mechanism designed, however, for carrying out a winding cycle. I have shown a storage roll **50** on which is to be wound a length of flexible material **51**, and I have illustrated a run **54** of such material being advanced toward said roll by forwarding means **55** driven from an external power source (not shown). Rotary means **56** is driven from said external power source at a constant ratio relative to the forwarding means **55**; and as shown said rotary means may be the input shaft of a speed varying transmission **57** which may be structurally identical to the transmission **17** of FIGS. 1–4. A shiftable element **64** is movable oppositely to vary oppositely the speed of the output shaft **69** which is connected to drive the worm **71**.

The worm **71**, like the worm **31** above described, is mounted to rotate with the shaft **69**, but is axially movable relative thereto and is provided at one end with an exposed head **72** for a purpose which will become apparent. The worm **71** meshes with a worm wheel **74** fixed to a shaft **75** which carries also a pinion **76** meshing with a bull gear **77** fixed relative to the storage roll **50**.

A first lever **78** has one end mounted on a fixed fulcrum **79** near the worm **71**, and a follower roller **80**, adapted to bear on the effective peripheral surface of material wound on the roll **50**, is carried at the opposite end of said lever **78**. A link **81** carries at one end a clevis **82** which is pivotally connected to the lever **78** at a suitable point intermediate the ends of said lever; and at its opposite end the link is provided with an abutment toe **84** which straddles, and is guided on, the shaft **69** at a point spaced from the worm head **72**. A coiled spring **85** is sleeved on the shaft **69** and is confined between the worm head **72** and said abutment toe **84**. Obviously, the spring **85** acts to press the follower roller **80** resiliently against the effective periphery of the roll **51** of material wound on the storage roll **50**, and yieldably to resist movement of the worm **71** axially to the right. As obviously, the degree of force with which said spring resists such worm movement will be a function of the current spacing between the worm head **72** and the abutment **84**, and of the spring rate of such spring.

A second lever **86** is pivotally mounted at a suitable point intermediate its ends on a fixed fulcrum **87**, and a yoke **88** at one end of said lever **86** engages a cross pin **92** to provide a two-way operative connection between said lever and the shiftable element **64** of the transmission **57**, while the other end **89** of said lever is operatively engaged by said worm head **72** and is resiliently held thereagainst by the spring **85**.

In this instance, the external power source acts affirmatively to drive the roll **50** instead of serving as a brake or drag. Thus, the rotary element **56** is driven at a velocity which is constantly proportional to the velocity of the forwarding means **55** and the ratio of the transmission **57** is set at such a value that the shaft **69** is driven at a speed to drive the roll **50**, through the worm **71**, worm wheel **74**, pinion **76** and gear **77** at a velocity such as to exert upon the run **54** of advancing material a predetermined tension. The tension will be, in effect, measured by the spring-affected worm **71**. That is, since the reaction effect upon the worm **71** of its effort to drive the worm wheel **74** in a clockwise direction tends to move the worm toward the right, only so much torque can be exerted upon the worm wheel **74** as is proportionate to the force with which the spring **85** currently resists rightward movement of the worm **71**. As material is wound upon the roll **50**, the effective radius of the roll increases, thereby tending to increase the tension in the material run **54** which is advancing at constant velocity, not only by reason of the increase in effective peripheral dimension of the roll but also by reason of the increase in the radial arm through which the rotating roll is acting on the run **54**.

As tension in the run **54** tends to increase, however, the axial force acting on the worm **71** will overcome the current force of the spring **85** and the worm **71** will move toward the right to swing the lever **86** in a counter-clockwise direction whereby the shiftable element **64** will be affirmatively moved toward the left to reduce the speed of the transmission output shaft **69**. At the same time, the increase in the effective radius of the material on the roll **50** will move the roller **80** to the left to shift the abutment **84** proportionately toward the left. Thus, worm movement and abutment movement additively compress the spring **85** to increase the force with which the spring **85** resists further worm movement toward the right.

The factors which, in such a system, coact to determine the value of the tension in the run **54** may be selected to hold that value substantially constant throughout a windup cycle.

Again, let

l'=distance from fulcrum **87** to point **89**
l''=distance from fulcrum **87** to yoke **88**
w=total range of axial movement of worm **71**
s=total range of travel of shiftable element **64**
L'=distance from fulcrum **79** to point **83**
L''=distance from fulcrum **79** to roller **80**
r'=maximum storage roll effective radius
r''=minimum storage roll effective radius
F=spring length change or the difference between the axial force of the worm at the end of a windup cycle and the axial force of the worm at the beginning of such a cycle, divided by the spring rate in pounds per inch
T=the desired tension in material run **54**
R=the pitch ratio of gear **77** to pinion **76**
p=the pitch radius of worm wheel **74**
t=total range of travel of abutment toe **84**

Then assume

$w=4''$
$s=4''$
$r'=12.5''$
$r''=2.5''$
$T=2000\#$
$R=20$ to 1
$p=2.5''$
Spring rate=50#/inch

Then

Torque at storage roll at start of windup cycle
$=2.5\times2000=5000$ inch-pounds

Torque at storage roll at end of windup cycle
$=12.5\times2000=25000$ inch-pounds

Torque at worm wheel at start of windup cycle

$$=\frac{5000}{20}=250 \text{ inch-pounds}$$

Torque at worm wheel at end of windup cycle

$$=\frac{25000}{20}=1250 \text{ inch-pounds}$$

Axial force at worm at start of windup cycle $=\frac{250}{2.5}=100\#$

Axial force at worm at end of windup cycle $=\frac{1250}{2.5}=500\#$

Difference in necessary spring force from start to end of cycle=500−100=400#

Required length reduction of spring

$$=\frac{\text{required spring force change}}{\text{spring rate}}=\frac{400}{50}=8''$$

Since it has been assumed above that $w=4''$ and that $s=4''$, l' and l'' must be equal.

Since **84** and **72** travel in opposite directions, $w=4''$ and spring length reduction must be $8''$, t must be $8''-4''=4''$.

Since follower roll must travel $(r'-r'')=10''$,

$$\frac{L'}{L''}=\frac{4}{10}$$

and point **83** must be located 0.4 of the distance from follower **79** to roller **80** if constant tension is to be maintained in the material run **54**.

If, however, it is desired to control the system in such a way as to provide "tapered tension" in which the tension in the run **54** decreases linearly from a value of, for instance, 2000 pounds at the start of a windup cycle to a value of, for instance, 1000 pounds at the end of the cycle, one way to accomplish that result is to select a spring **85** having a rate different from that assumed above. Thus

Torque at storage roll at start of windup cycle=$2.5\times2000=5000$ inch-pounds

Torque at storage roll at end of windup cycle=$12.5\times1000=12500$ inch-pounds

Torque at worm wheel at start of windup cycle

$$=\frac{5000}{20}=250 \text{ inch-pounds}$$

Torque at worm wheel at end of windup cycle

$$=\frac{12500}{20}=625 \text{ inch-pounds}$$

Axial force at worm at start of windup cycle $=\frac{250}{2.5}=100\#$

Axial force at worm at end of windup cycle $=\frac{625}{2.5}=250\#$

Difference in necessary spring force from start to end of cycle=250−100=150#

Length reduction of spring during cycle=8''

$$\text{Spring rate}=\frac{\text{required spring force change}}{\text{spring length reduction}}$$

$$=\frac{150}{8}=18.75\#/\text{inch}$$

Alternatively, such tapered tension may be achieved by shifting the point at which the link **81** is pivotally connected to the lever **78**; and to that end, I have shown a series of spaced openings **90** above the point **83** and a series of spaced openings **91** below that point whereby adjustments may be selectively made in the point of connection between the link and the lever. Thus, if it is desired to use the spring **85** having a spring rate of 50#/inch and to achieve a linear reduction in tension from 2000 pounds to 1500 pounds during the cycle, a spring length reduction of 5.5 inches must occur during the cyle. Since $w=4$, t must be $5.5-4=1.5''$. Since roller **80** must move $10''$ as above, the ratio of L' to L'' must be

$$\frac{1.5}{10}$$

and if L'' is, for instance, 48'', the link **81** must be pivoted to lever **78** at a point 7.2'' from the fulcrum **79**.

Or, if it is desired to increase the tension from, for instance, a starting value of 1000 pounds to a value of 2000 pounds at the end of the cycle, the reduction in spring length must be 9 inches, the travel of the abutment **84** must be $5''$ and the point of connection of link **81** to lever **78** must be midway between the fulcrum **79** and the roller **80**. If a starting tension of 2000 pounds is to be increased to 2500 pounds at the end of the cycle, abutment **84** must travel 6.5'' while worm **71** travels $4''$ in the opposite direction, and the point of pivotal connection between link **81** and lever **78** must be 0.65 of the length of the lever or 31.2'' from the fulcrum if roller **80** is carried $48''$ from the fulcrum as above.

It will thus be seen that, in the windup system, the current tension in the run of material **54** is determined, when the control mechanism includes a spring having a predetermined spring rate, by the ratio between the rate of travel of the abutment **84** and the rate of axial movement of the worm **71**; and that, for any selected rate of axial movement of the worm, if the ratio between the force with which the spring resists axial movement of the worm and the product of tension by the effective radius of material wound on the storage roll is held constant, the tension in the run **54** will remain constant.

It will also be seen that, if the rate of travel of the abutment **84** is so modified as to change the force with which the spring **85** resists such worm movement at a rate different from the rate of change of the effective radius of material on the storage roll, tension in the run **54** will be progressively changed during a windup cycle to produce a "tapered tension" effect. Expressed otherwise, if the ratio between the force with which the spring resists axial movement of the worm and the product of tension by the current effective radius of material wound on the storage roll is progressively varied during the cycle, the tension in the run **54** will be correspondingly progressively varied.

The above calculations ignore the factors of frictional resistance to movement of the control parts and any net holding force which may be required to maintain a particular setting of any specific form of speed-varying transmission used in the system. These are factors which are familiar to those who are skilled in the art to which the present invention pertains, and compensation therefor may be achieved, in accordance with conventional practice, by suitable linkage changes and/or by interposing a servo-motor device between the shiftable element which controls the transmission ratio and the equivalent of the disc **21**. Also it will be obvious to those skilled in the art that suitable dimensions for the

follower roller and its mounting lever, relative to the force of the spring and the character of the material being wound or unwound, will be selected so that the follower roller will not exert an undue crushing effect upon the material wound on the storage roll.

While the external power source for the system will ordinarily be operated at constant speed so that the material will be forwarded toward or away from the storage roll at constant linear velocity, it will be appreciated that, since the means **15** or **55** and the rotary element **16** or **56** are so connected to the power source as to be driven at a constant ratio, the disclosed control mechanisms will perform their intended functions without significant loss of effectiveness even if the power source is operated at varying speeds.

While a push-pull connection between the lever **86** and the shiftable element **64** has been shown in FIG. 5 and is believed to be preferable in most windup systems, it will be clear that in some such installations a one-way connection like that shown in FIG. 1 may be used. It will also be understood that it may sometimes be desirable to use a two-way connection between the elements **46** and **24** in a system of the type shown in FIG. 1.

I claim as my invention:

1. Means for automatically controlling the rate of rotation of a storage roll for flexible material comprising, in combination with such a roll, a length of flexible material wound on said roll, means for forwarding said material relative to said roll, rotary means driven at a constant ratio with respect to said forwarding means, a speed-varying transmission including an input shaft connected to be driven from said rotary means, an output shaft, and an element shiftable oppositely to vary oppositely the speed of said output shaft relative to the speed of said input shaft, a worm wheel having a constant-ratio driving connection with said roll, a worm having a constant-ratio driving connection with said output shaft, mounted for axial reciprocation, and meshing with said worm wheel, spring means yieldably resisting axial movement of said worm under the influence of the driving reaction of said worm on said worm wheel, means operatively connecting said worm to shift said shiftable element oppositely in response to opposite axial movement of said worm, a movable backing abutment for said spring means, and means variably positioned in accordance with variations in the effective radius of flexible material wound on said roll and operatively connected to move said abutment.

2. Means for automatically controlling the rate of rotation of a storage roll for flexible material comprising, in combination with such a roll, a length of flexible material wound on said roll, means for forwarding said material relative to said roll, rotary means driven at a constant ratio with respect to said forwarding means, a speed-varying transmission including an input shaft connected to be driven from said rotary means, an output shaft, and an element shiftable oppositely to vary oppositely the speed of said output shaft relative to the speed of said input shaft, a worm wheel having a constant-ratio driving connection with said roll, a worm having a constant-ratio driving connection with said output shaft, mounted for axial reciprocation, and meshing with said worm wheel, follower means arranged to bear on the effective periphery of material on said roll, spring means cooperatively associated with said follower means and with said worm to urge said follower means yieldably toward the axis of said roll and yieldably to resist movement of said worm axially in one direction, and means providing an operative connection between said worm and said shiftable element so constructed and arranged that axial movement of said worm results in corresponding movement of said shiftable element.

3. Means for automatically controlling the rate of rotation of a storage roll for flexible material comprising, in combination with such a roll, a length of flexible material wound on said roll, means for forwarding said material relative to said roll, rotary means driven at a constant ratio with respect to said forwarding means, a speed-varying transmission including an input shaft connected to be driven from said rotary means, an output shaft, and an element shiftable oppositely to vary oppositely the speed of said output shaft relative to the speed of said input shaft, a worm wheel having a constant ratio driving connection with said roll, a worm axially shiftably mounted on said output shaft but rotationally fixed thereto and meshing with said worm wheel, a lever fulcrumed adjacent said worm and carrying a follower arranged to bear on the effective periphery of material on said roll, abutment means spaced from one end of said worm and arranged for travel along the axis of said output shaft, a coiled spring sleeved on said output shaft and confined between said abutment means and said one end of said worm, link means providing an operative connection between said abutment means and said lever, said spring acting through said abutment means, said link means and said lever to urge said follower yieldably toward the axis of said roll, and means providing an operative connection between said worm and said shiftable element so constructed and arranged that axial movement of said worm results in corresponding movement of said shiftable element.

4. Means for automatically controlling the rate of rotation of a storage roll for flexible material comprising, in combination with such a roll, a length of flexible material wound on said roll, means for forwarding said material relative to said roll, rotary means driven at a constant ratio with respect to said forwarding means, a speed-varying transmission including an input shaft connected to be driven from said rotary means, an output shaft, and an element shiftable oppositely to vary oppositely the speed of said output shaft relative to the speed of said input shaft, a worm wheel having a constant-ratio driving connection with said roll, a worm axially-shiftably mounted on said output shaft but rotationally fixed thereto and meshing with said worm wheel, a first lever fulcrumed at one end near said worm and carrying a follower at its other end arranged to bear on the effective periphery of material on said roll, abutment means spaced from one end of said worm and arranged for travel along the axis of said output shaft, a coiled spring sleeved on said output shaft and confined between said abutment means and said one end of said worm, link means secured at one end to said abutment means and pivotally connected at its opposite end to said first lever between its fulcrum and said follower, said spring acting through said abutment means, said link means and said lever to urge said follower yieldably toward the axis of said roll, and a second lever fulcrumed at one end near said worm and, at its opposite end, operatively engaging said shiftable element, said one end of said worm operatively engaging said second lever intermediate the ends thereof.

5. Means for automatically controlling the rate of rotation of a storage roll for flexible material comprising, in combination with such a roll, a length of flexible material wound on said roll, means for forwarding said material relative to said roll, rotary means driven at a constant ratio with respect to said forwarding means, a speed-varying transmission including an input shaft connected to be driven from said rotary means, an output shaft, and an element shiftable oppositely to vary oppositely the speed of said output shaft relative to the speed of said input shaft, a worm wheel having a constant-ratio driving connection with said roll, a worm axially-shiftably mounted on said output shaft but rotationally fixed thereto and meshing with said worm wheel, a first lever fulcrumed at one end near said worm and carrying a follower at its other end arranged to bear

on the effective periphery of material on said roll, abutment means spaced from one end of said worm and arranged for travel along the axis of said output shaft, a coiled spring sleeved on said output shaft and confined between said abutment means and said one end of said worm, link means secured at one end to said abutment means and pivotally connected at its opposite end to said first lever between its fulcrum and said follower, said spring acting through said abutment means, said link means and said lever to urge said follower yieldably toward the axis of said roll, and a second lever fulcrumed intermediate its ends near said worm, one end of said second lever operatively engaging said shiftable element and the other end of said second lever being operatively engaged by said one end of said worm.

6. A constant-tension unwind control according to claim 1 wherein the rate of travel of said abutment exceeds the rate of axial movement of said worm in the same direction by an amount constantly proportional to the rate at which the effective radius of the material wound on said storage roll decreases.

7. A constant-tension unwind control according to claim 4 wherein the effective lengths of said levers and the points at which said link means is pivotally connected to said first lever and said worm operatively engages said second lever are so chosen that the force with which said spring resists axial movement of said worm is reduced at a rate equivalent to the rate of reduction of the effective radius of material on said roll.

8. A constant-tension windup control according to claim 1 wherein the rate of travel of said abutment bears a predetermined ratio to the rate of axial movement of said worm to maintain substantially constant the ratio between the force with which said spring means resists axial movement of said worm and the product of desired tension by the effective radius of material on said roll.

9. A controlled-tension windup control according to claim 1 wherein the rate of travel of said abutment relative to the rate of axial movement of said worm modifies the force with which said spring means resists such worm movement at a rate different from the rate of change of the effective radius of material on said roll.

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