

[54] **WEB TENSION CONTROL APPARATUS**

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[58] Field of Search..... **242/75.43, 75.53; 226/38, 44**

[56] **References Cited**

UNITED STATES PATENTS

3,071,300	1/1963	Justus	242/75.43 X
2,343,181	2/1944	Heinz	242/75.43
3,101,915	8/1963	Aaron	242/75.43
3,057,574	10/1962	Justus	242/75.43

3,312,415	4/1967	Jeans	242/75.43
3,164,333	1/1965	Robertson	242/75.43

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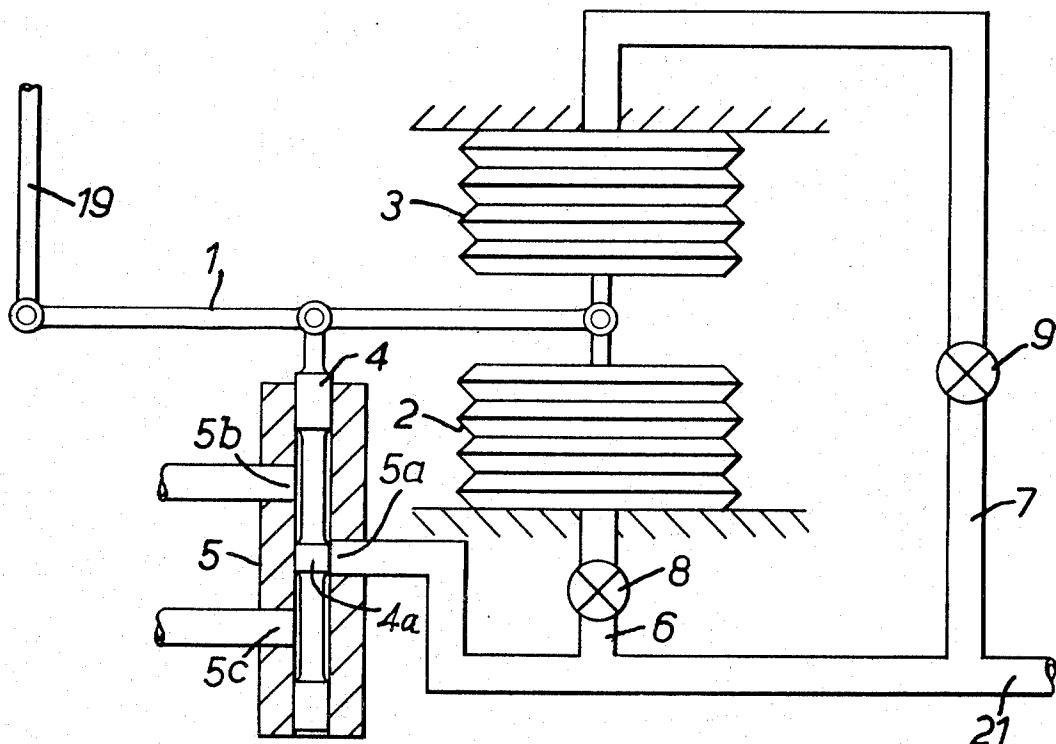
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[57] **ABSTRACT**

A travelling web being wound onto or unwound from a reel has its tension sensed by a dancer roll in a web loop, and the position of the dancer roll is used to control the drive or brake system of the reel to maintain the tension uniform. With pneumatic control, a valve in the air line is actuated by the dancer roll to increase or decrease the supply, and in addition there is feedback from the air line to operate devices which restore the valve to its median position. These devices act in opposition, one acting faster than the other to cause the initial restoration movement. There is a resultant phase difference between dancer roll movement and valve actuation, which can be further modified by interposition of a mechanical phase shifting device between the dancer roll and valve.

16 Claims, 12 Drawing Figures



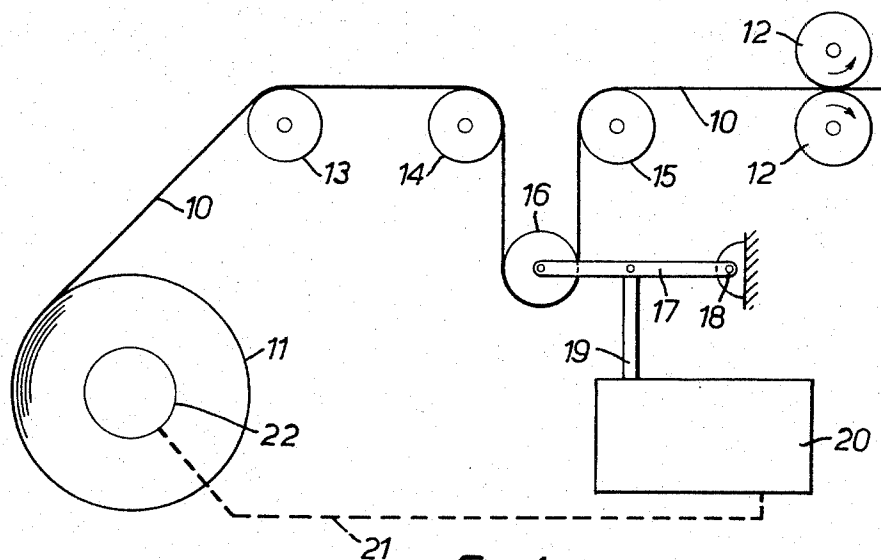


FIG. 1.

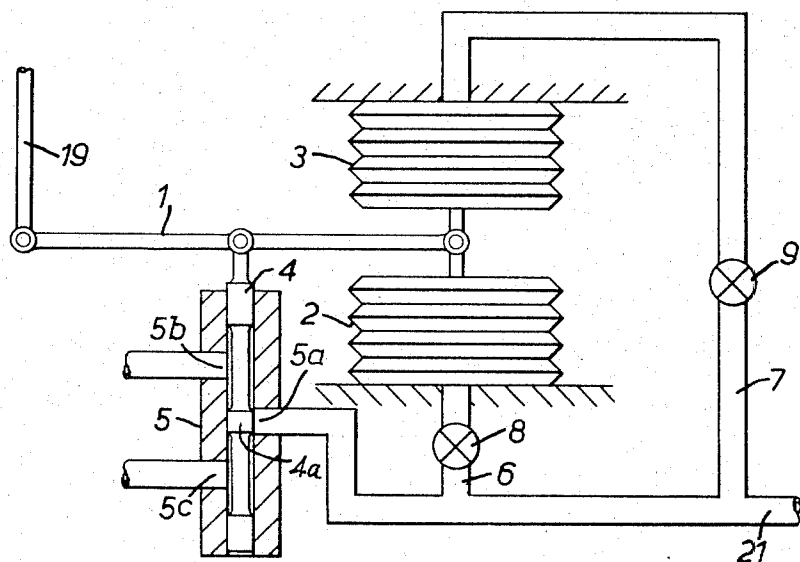
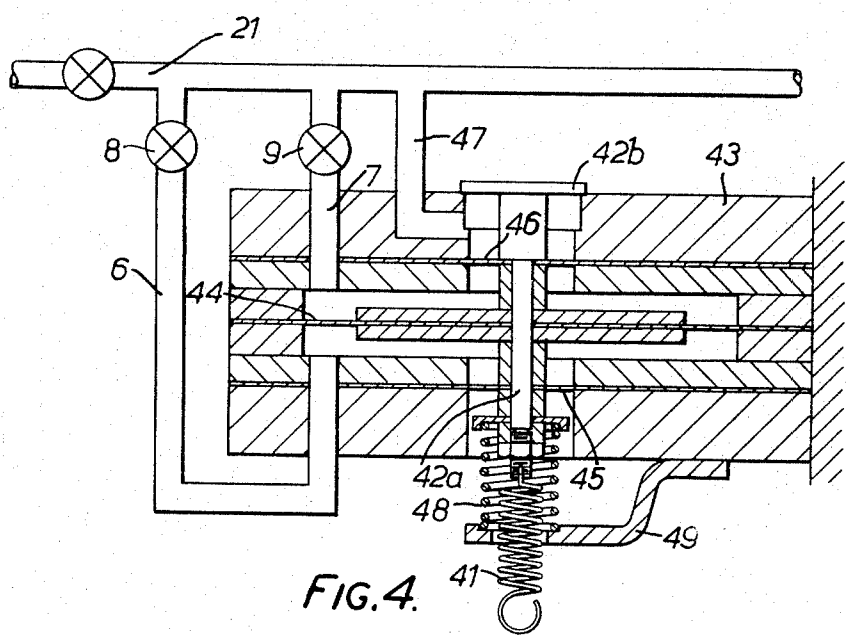
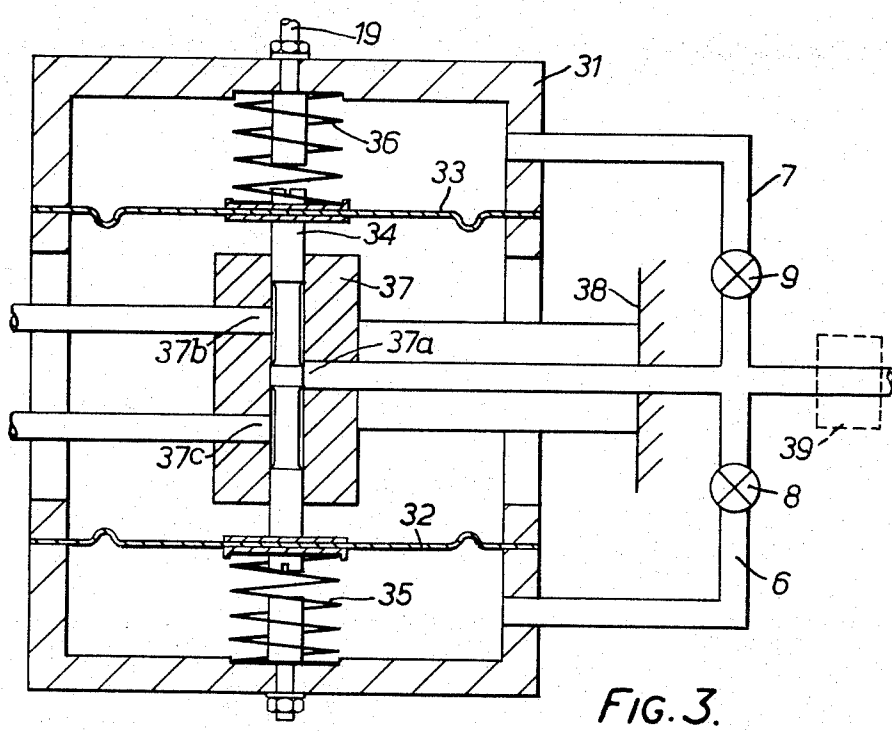
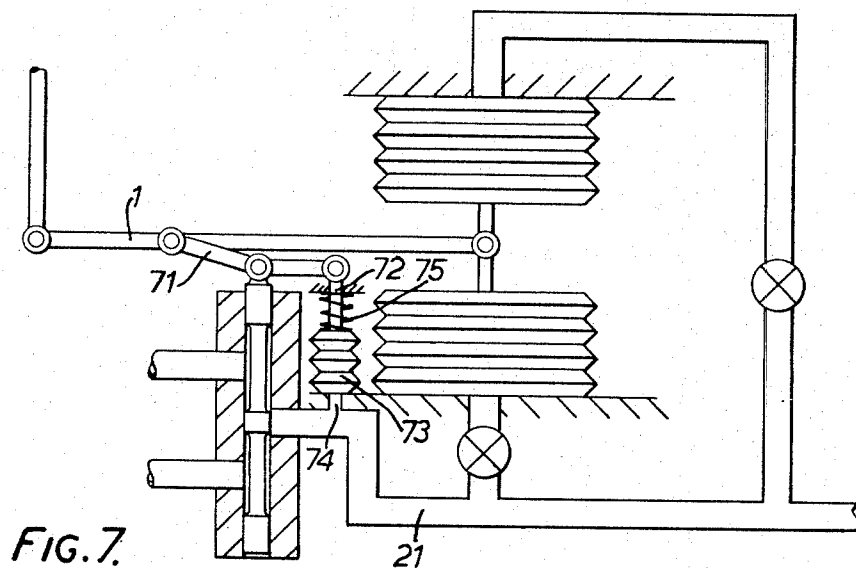
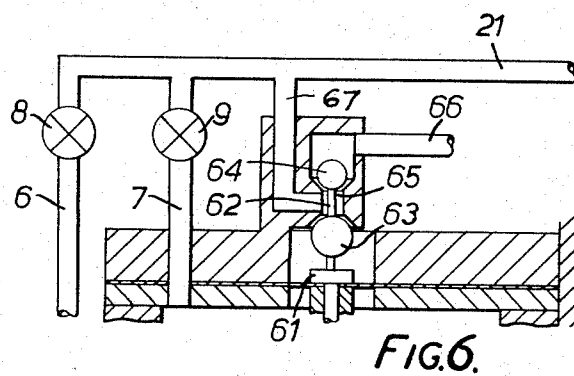
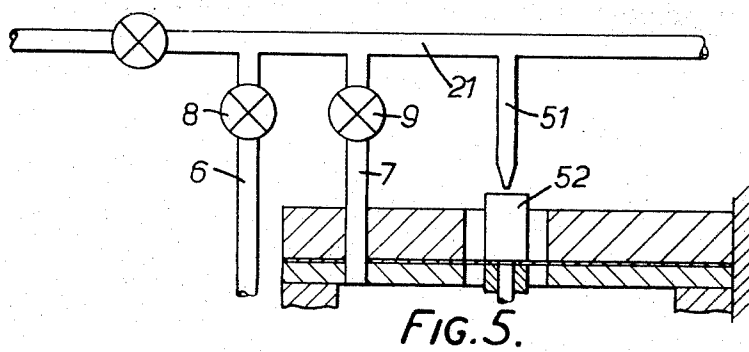
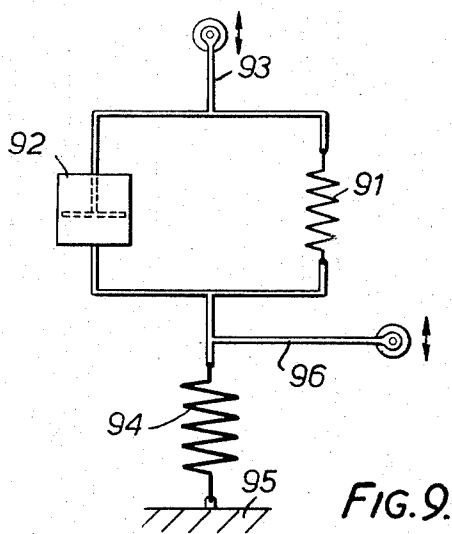
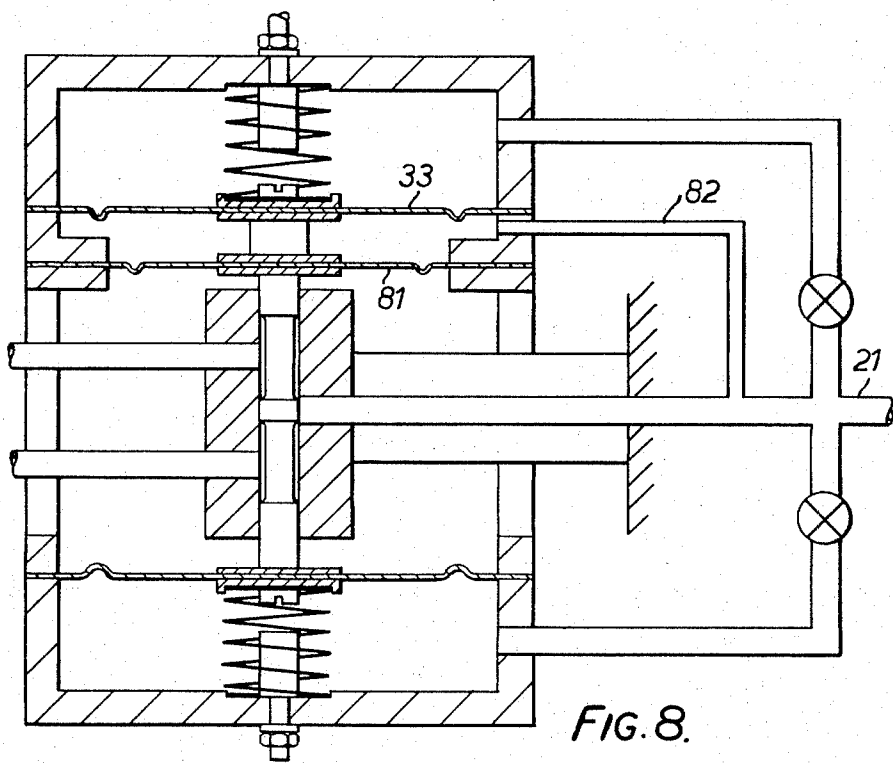


FIG. 2.







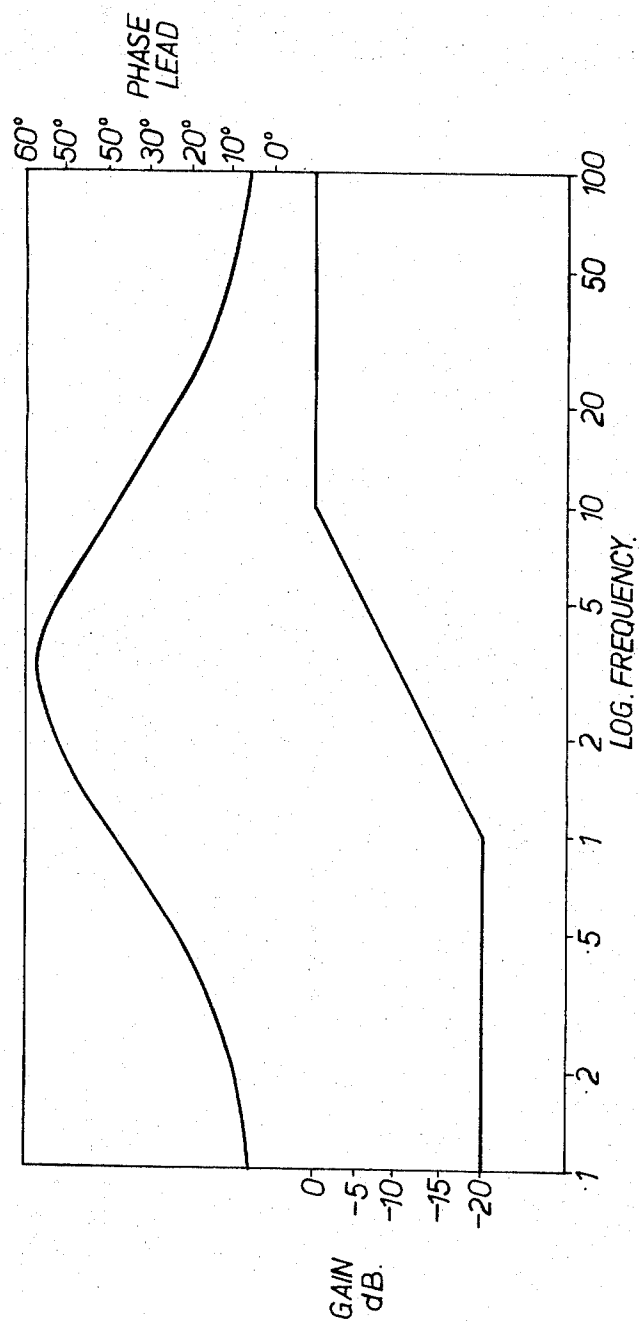
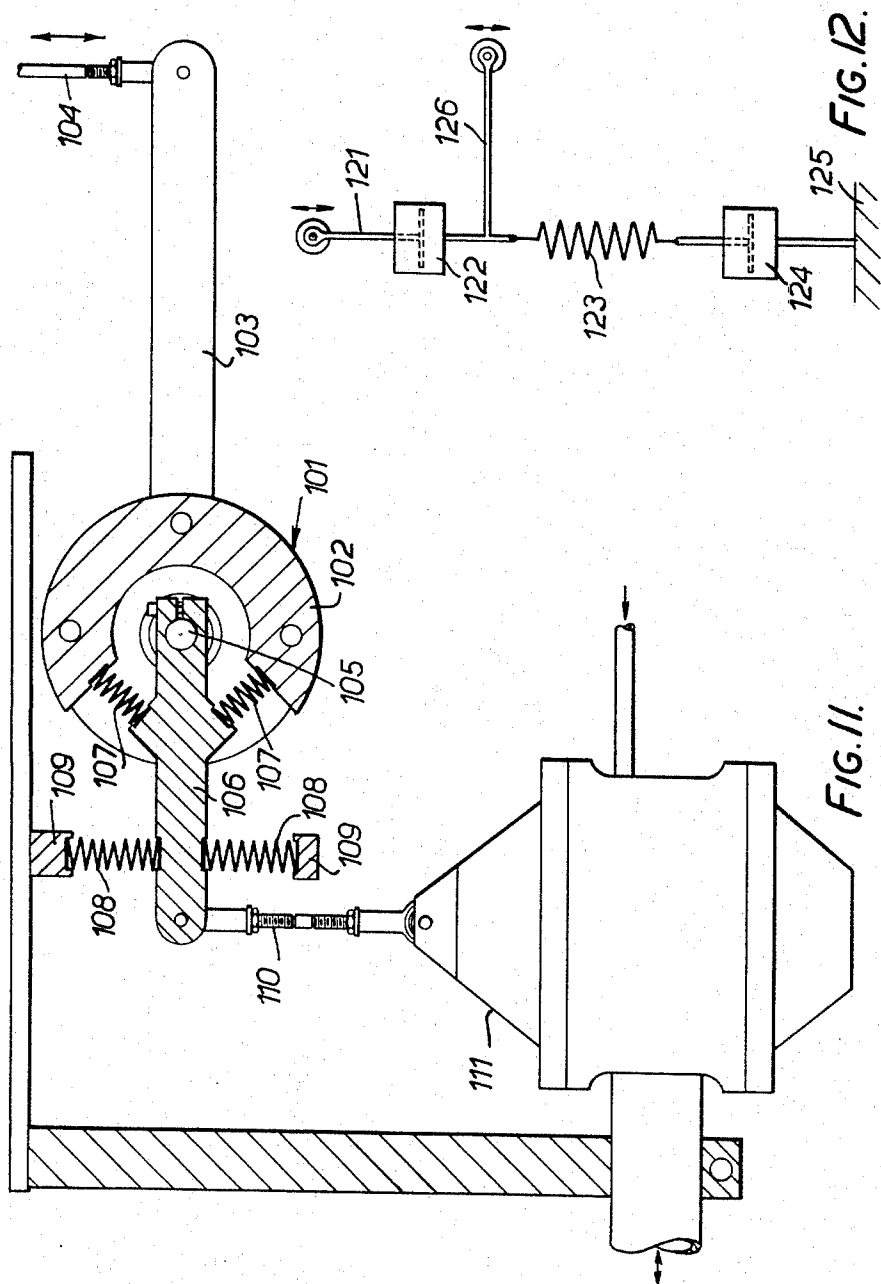


FIG. 10.



WEB TENSION CONTROL APPARATUS

The invention relates to pneumatic or hydraulic web tension control apparatus, for use in machinery, such as paper reel unwinding machinery, in which a travelling web is tensioned by a web drive/brake system.

In such machinery it is common practice to use a sensing roll, known as a dancer roll, which is supported by a downward loop in the web. Increase in tension tends to shorten the loop and raise the dancer roll. A mechanical link to the dancer roll thus provides a position input for a tension control system, which may be loaded by weights, springs or fluid-operated actuating devices. The total effective mass of the dancer roll should be kept to a minimum to minimise inertia effects. The load should be substantially constant, e.g., as provided by a spring or very low rate, and friction should also be kept to a minimum. Damping has generally been required but is undesirable in that it adds a variable element to the load. As is well known, the gain of the tension control apparatus must be kept to a low figure in the range of frequencies around its natural periodic frequency of the system, in order to avoid instability.

Pneumatic web tension control apparatus has previously been proposed in which the gain is reduced from a high value at zero frequency to a lower figure at middle and higher frequencies, by providing the control output with components which are proportional to the signal from the sensing roll and the integral thereof. The integral component provides a resetting action which is valuable when the required web tension is to be adjusted.

According to the present invention there is provided a web tension control apparatus for a travelling web tensioned by a web drive/brake system, the tension control apparatus including a signal generator arranged to produce an input signal in dependence upon the effective web tension, and transmission means arranged to receive said input signal and to supply a control output to the drive/brake system, said transmission means being adapted to introduce a forward phase shift which is frequency-dependent over a middle band of frequencies. In some preferred embodiments said transmission means includes a pneumatic or hydraulic gain and phase-shaping network adapted to reduce the gain of the tension control apparatus in said middle band of frequencies while the forward phase shift increases with frequency.

Preferably this shaping network includes a pair of pneumatic or hydraulic actuating devices arranged in opposite senses and with different time delays, the actuating device with the smaller time delay being arranged to act in the opposite sense to said input signal so as to protrude a component of the control output which is a derivative function of the input signal and thereby introduce the forward phase shift.

The actuating device with the greater time delay, acting in the same sense as an input signal, provides an integral effect to reset the signal generator to a median position. In a preferred form the actuating devices are connected in parallel to receive inputs from a pressure fluid line carrying the control output to the brake system, through restrictors of different throttling characteristics giving the different time delays. The outputs of these devices may be mechanically connected, in opposition to one another, to a control valve controlling the

pressure of fluid in the line, the valve also being connected to receive the input from the signal generator. This connection may be by means of a beam actuated at one end by the signal generator, supported by the actuating device at the other end, with the control valve being connected thereto at an intermediate position.

In this form the actuating devices are preferably pneumatic bellows and they may be sprung, or augmented by spring means, to give stiffness.

In a preferred alternative form, the input from the signal generator is arranged to cause relative movement between the two actuating devices, moving as one, and the control valve, these devices conveniently being formed by diaphragms.

There may be two diaphragms which close off opposed spaces in a common housing and which are mechanically connected to a first operating member of the control valve which is disposed between them. The input from the signal generator is arranged to cause relative movement between the housing and a second operating member of the control valve, one of these operating members being the valve body.

Other embodiments using diaphragms have three in parallel linked by a valve spindle and mounted in a fixed housing to provide two closed spaces at opposite sides of the central diaphragm which communicate, via said restrictors with the pressure fluid line. The diaphragms/spindle assembly is normally balanced between spring means and the pressure of fluid in said line, and said input signal is applied to the valve spindle.

In other preferred apparatus said transmission means includes a mechanical phase shifting system comprising a combination of spring means and viscous damping means arranged so that an input at one point in the system results in a modified output at another point with low gain at low input frequencies and gain increasing with frequency over said middle band of frequencies.

This system may be arranged so that the output phase lead achieves a peak value within said middle band of frequencies, and so that the gain approaches unity at high frequencies.

To achieve this, the system may include an assembly of first spring means and viscous damping means coupled in parallel and in series with second spring means reacting against a fixed structure, the input being to the free end of said assembly and the output from a point effectively intermediate said assembly and said second spring means. In a preferred embodiment said viscous damping means is a rotary dashpot arranged to be partially rotated by said input, the first spring means acting between the dashpot body and an output member and the second spring means acting between a fixed structure and said member. The spring means are conveniently pairs of opposed springs, with the second spring means stiff compared with the first spring means.

In an alternative arrangement, said system includes an assembly of first viscous damping means, spring means and second viscous damping means to one end of which the input is applied and the other end of which reacts against a fixed structure, the output being from a point effectively between the spring means and the viscous damping means at said one end.

The invention may be performed in various ways and some specific embodiments will now be described in more detail by way of example and with reference to the accompanying drawings, in which:

FIG. 1 is a schematic elevation of a paper reel unwinding machine incorporating a web tension control apparatus,

FIGS. 2 to 8 are diagrams of pneumatic web tension control apparatus that can be incorporated in the machine of FIG. 1,

FIG. 9 is a schematic diagram of control mechanism with phase advancing characteristics,

FIG. 10 is a graph illustrating frequency-gain and frequency-phase lead characteristics of the mechanism of FIG. 9,

FIG. 11 is a part-sectional elevation of a practical embodiment of the mechanism of FIG. 9, and

FIG. 12 is a schematic diagram of an alternative phase advance mechanism.

Referring to FIG. 1, a web 10 is unwound from a reel 11 by means of a pair of drawing rollers 12. The web 10 passes over idler rollers 13, 14 and 15 and around a dancer roll 16, forming a downward loop.

The dancer roll 16 is of comparatively light construction and is mounted between the free ends of a pair of pivoted arms 17, only one of which is shown, the other ends of which are pivoted at 18 to a fixed structure. A substantially constant downward force is applied to the dancer roll to tension the web 10. This force may be applied by a spring of very low rate or by a pneumatic or hydraulic actuator with low friction. Use of weights to load the dancer roll 16 is also possible but is undesirable from the point of view of the inertia involved.

The arms 17 are connected by a mechanical link 19 to web tension control apparatus indicated as a block 20 in FIG. 1 and which may take various forms as shown in the remaining Figures. The control apparatus 20 controls the pneumatic or hydraulic pressure in a line 21 to a brake 22 acting on the reel 11, to adjust the tension in the web 10. The link 19 provides a position input from the dancer roll 16. If the tension in the web 10 increases, the effective length of the web between the idler rollers 14 and 15 will decrease and the dancer roll 16 will rise, thus raising the link 19. Conversely a decrease in tension will cause a drop of the dancer roll 16 and link 19.

Referring now to the control apparatus shown in FIG. 2, the lower end of the link 19 is pivotally connected to one end of a substantially horizontal beam 1, the other end of which is supported between a pair of opposed bellows 2, 3 which also act as compression springs, although separate springs may be assembled within the bellows or adjacent them. The beam 1 is also connected near its mid-point to the spool 4 of a control valve 5, controlling the pneumatic pressure in the line 21 to the brake 22. The central land 4a of the spool does not quite cover the outlet port 5a of the valve 5 in the central spool position and consequently the outlet port 5a is then open to both inlet port 5b and exhaust port 5c simultaneously. The pressure established at the outlet port 5a, and hence in the line 21, varies in dependence on the bias of the spool 4 towards either supply or exhaust. This design can give a very high gain and a substantially linear pressure/displacement characteristic over a desired pressure range.

Branches 6, 7 from the line 21 lead in parallel to the interiors of the pair of bellows 2, 3, so that the bellows act on the beam in opposition to one another. Restrictors 8 and 9, are incorporated in the branches 6, 7 respectively, restrictor 8 having a smaller throttling effect

and thus providing a smaller time delay than restrictor 9.

When a position input is received from the dancer roll 16 via the link 19, indicating a change in tension of the web 10, the first effect is to adjust the control valve 5 so as to apply or release the brake 22 to readjust the tension. The change in pressure in the line 21 is transmitted with differing time delays through the restrictors 8 and 9 to the bellows 2 and 3. The bellows 2, being connected to the restrictor 8 giving the lower time delay, reacts more rapidly and adjusts the beam 1 in the opposite sense to the input received through the link 19. As the pressure change is transmitted to the bellows 3, the pressures in the two bellows gradually become equal and the beam is restored towards its mid position. The effect of this is to give a control output to the valve 5 which has one component which is proportional to the input, a second component (produced by bellows 2) which is a derivative function of the input and thereby introduces a forward phase shift which increases with frequency and reduces the gain of the tension control apparatus in a middle band of frequencies, and a third component (produced by bellows 3) which provides an integral effect, resetting the beam to its middle position after adjustment of the web tension.

With this arrangement, the gain at zero frequency is very high. The gain and phase shaping network formed by the bellows 2 and 3 and their connections reduces the gain in a middle range of frequencies to a value at which instability is avoided, this range being arranged to be in the neighbourhood of the normal periodic frequency of the system, but the gain increases again as the frequency is increased, due to the forward phase shift introduced by the bellows 3.

In the control apparatus of FIG. 3, the lower end of the link 19 is connected to a housing 31 which contains a pair of diaphragms 32 and 33 connected to opposite ends of a valve spool 34 and closing off opposed end chambers in the housing. Springs 35 and 36 within those chambers act on the diaphragms to provide the necessary stiffness. The valve spool 34 is free to move axially within the valve body 37 which is fixed with respect to the machine, indicated at 38. The valve is similar to that of FIG. 2 and the position of the valve spool 34 within the valve body 37 determines the pressure supplied to the line 21 as derived from the regulated supply pressure input to the valve at port 37b. The pressure in line 21 is applied to each diaphragm through branch lines 6 and 7 equivalent to those of FIG. 2 and also incorporating respective restrictors 8 and 9, restrictor 8 having a smaller throttling effect and thus providing a smaller time delay than restrictor 9. The lines 6 and 7 supplying the diaphragm chambers are necessarily of flexible construction to enable relative movements between valve body 37 and controller housing 31 to occur.

The pressure in line 21 is also applied to the brake 22. As the control apparatus is essentially a relatively small device, to maintain an adequate frequency response, it may not be capable of supplying the brake 22 and associated lines with the necessary flow. To overcome this a flow booster 39 may be incorporated in the brake line as shown, or a pneumatic to hydraulic converter close to the control apparatus, the brake then being hydraulically operated.

When a position input is received from the dancer roll 16 via the link 19, indicating a potential change in

tension of the web 10, the first effect is to move the spool 34 in unison with controller housing 31. The valve 34,37 is phased so that this action changes the pressure in the line 21 and hence the pressure applied to the brake 22, in the appropriate sense.

The change in pressure in the line 21 is transmitted with differing time delays through the restrictors 8 and 9 to the diaphragms 32 and 33. The diaphragm 32, being connected to the restrictor 8 giving the lower time delay, reacts more rapidly and adjusts the spool 34 within the valve body 37 against the springs 35,36 in the opposite sense to the original input received through the link 19. As the pressure change is transmitted to the diaphragm 33 the pressures acting on diaphragms 32 and 33 gradually become equal and the valve spool 34 is restored towards the position it occupied immediately after the original input from the dancer roll 16.

The effect of this is to give a control output to the valve spool 34 which has one component which is proportional to the input, a second component (produced by diaphragm 32) which is a derivative function of the input and thereby introduces a forward phase shift which increases with frequency and reduces the gain of the tension control apparatus in a middle band of frequencies, and a third component (produced by diaphragm 33) which provides an integral effect, resetting the dancer roll 16 to its middle position after adjustment of the brake.

It will be appreciated that the action described above could equally well be achieved if the link 19 were connected to the valve body 37 and the controller housing 31 were fixed.

In the control apparatus of FIG. 4 the movement of the link 19 is applied to a spring 41 connected to one end of the spindle 42a of a poppet valve whose head 42b seats at the other end on a housing 43. The housing contains a main central diaphragm 44 and two subsidiary diaphragms 45 and 46 which between them define chambers with which the branch lines 6 and 7 are respectively in communication. These lines also contain restrictors 8 and 9 of respectively quick and slow response. The spindle 42a passes centrally through the diaphragms which are clamped thereto. A further branch 47 leads from the main brake pressure line 21 into a space between the head 42b of the poppet valve and the diaphragm 46, the effective area of the underside of the head being slightly larger than the effective area of the diaphragm 46. The output pressure from line 21 is thus applied to both in opposition and the slight inequality means that in the equilibrium position there is a small leakage from the poppet valve. A bias spring 48 acts between a bracket 49 fixed to the housing 43 and the spindle 42 and surrounds the spring 41.

This control apparatus operates as follows. Assuming the dancer roll moves downwards, indicating insufficient brake torque, an increased downward force is applied via the spring 41 to the spindle 42a, causing the poppet valve to seat more closely and to increase the pressure in the brake line 21, which previously had a continuous leakage via the valve. The pressure in the brake line 21 increases by an amount proportional to the dancer roll displacement and this increase is fed via the lines 6 and 7 and their respective restrictors 8 and 9 to opposite sides of the main central diaphragm 44. By virtue of the lesser restriction in the line 6, there is initially a net upward force tending to re-open the pop-

pet valve and reduce the brake line pressure. After an interval, the pressures on opposite sides of the main diaphragm 44 equalize and the balance of the valve is restored.

Corresponding action takes place if the dancer roll moves upwards, tending to relax the spring 41. The spring 41 should not become slack, but this is prevented by the spring 48 which pre-loads it and also produces an upward force that counteracts the pressures on the upper diaphragm 46 from the branch 47.

Referring now to FIG. 5, this shows a modification of FIG. 4 and omits the lower half, which is identical. In this case, a branch 51 from the brake line 21 terminates in an orifice with which the end of a spindle 52 co-operates to control the leakage therefrom. The jet from the orifice exerts a downward force on the spindle. In this case, however, the upward movement of the spindle increases brake line pressure, and so the restrictor 9 is the one with the quick response and the restrictor 8 the one with the slow response.

In the control apparatus of FIG. 6, the lower half is again similar to that of FIG. 4 and is not shown. The spindle 61 operates a "peanut" valve 62 comprising two linked balls 63 and 64 on either side of a throat 65. The pressure supply is through a line 66 and in the intermediate position there is leakage past the balls 64 and 63 to atmosphere while a portion of the supply is taken from the throat 65 via duct 67 to the line 21. The ball 64 on the supply side is smaller than the exhaust side ball 63 and this asymmetry gives a net downward force on the spindle proportional to the brake line pressure and balanced by the compression spring. The branch lines 6 and 7 to opposite sides of the main diaphragm have restrictors 8 and 9, as before, the restrictor 9 being the one with the quicker response.

As with FIG. 5, an upward movement of the spindle 61 is required to increase the brake pressure and it does this by seating the ball 63 against the lower end of the throat 65 to reduce the leakage and unseating the ball 64 to give freer passage to the pressure medium. The increase in pressure is fed more rapidly through the line 7 to urge the main diaphragm back towards the central position. The opposite movement of the spindle 71 releases brake pressure by unseating the ball 63 further and seating the ball 64 more firmly.

In the apparatus shown in FIGS. 4, 5 and 6, the brake line pressure reacts directly on the controller spindle, thereby giving potentially rapid response from the valve, and so the brake line pressure re-adjusts very rapidly to a force change due to dancer roll movement. They have the characteristics of force balance devices whereby any force imbalance between the output, input and two feedback force generators serves to open or close the valve and thus create a force balance. The spool valves shown in FIGS. 2 and 3 will not have such quick response since after the valve has been displaced to a new setting a finite time elapses before sufficient air has been passed into or out of the brake line to re-establish the new pressure, and the larger the volume of the brake line and brake actuator the longer the delay.

FIGS. 7 and 8 show modifications of FIGS. 2 and 3 designed to improve this response.

In FIG. 7, which is in most respects similar to FIG. 2, the spool 4 is connected to the beam 1 by a link 71 which has an extension jointed to a shaft 72 connected to one end of a bellows 73 in direct communication

with the pressure line 21 via a port 74. A spring 75 opposes expansion of the bellows 73.

This arrangement produces an immediate feed back around the spool valve which improves its response. For example, should the dancer roll fall, the increased pressure applied to the line 21 is also practically instantaneously applied to the bellows 73, which expand to raise the link 71 and restore the spool towards its median position.

In the control apparatus of FIG. 8, which is in most respects similar to that of FIG. 3, a secondary diaphragm 81 is introduced between the diaphragm 33 and the valve, and the space between the diaphragms 33 and 81 is in communication via branch 82 with the brake line 21. The diaphragm 81 has an effective area somewhat smaller than the diaphragm 33 and there is therefore a net upwards force proportional to the brake line pressure, which in conjunction with the springs 35 and 36 moves the spool upwards. This is the direction tending to reduce brake line pressure and there is therefore negative feedback around the spool valve.

The finite displacement necessary to cause a change in output pressure in the steady state, which would result in a finite change in the datum position of the dancer roll for different brake torque settings, can be eliminated by designing the controller such that the diaphragm or bellows which provides the positive feedback is slightly larger in effective area than the diaphragm or bellows that provide negative feedback. These differences in area, in conjunction with the rate of springs, can be made just to match the finite gain of the valve so that any change in brake pressure results in a spool movement, in the steady state, which is just sufficient to give that change in pressure.

It would be possible to have hydraulic rather than pneumatic operation of the control apparatus, although this would tend to increase the complexity and cost. It would be necessary to modify the above described apparatus by including compressible elements in the feedback paths, preferably between the restrictors and diaphragms or bellows so that a reasonable volume of fluid has to flow through the restrictors to change the feedback pressures. These compressible elements may take the form of additional bellows free to expand and contract as the pressure changed. Also, it would be necessary to encase the apparatus to collect escaped fluid and return it to a tank.

With the above described tension control apparatus the movement of the dancer roll is transmitted directly to the valve which forms the controller in the pneumatic line. However, it is inevitable that the supply pipe or pipes to the brake or brakes will introduce some phase lag into the system, thereby degenerating the system response and its stability. The longer these pipes are, the greater the lag. The lag due to a length of pipe is basically a combination of the time delay involved for a pressure wave to propagate the pipe length and also resistance/capacitance effects which give increasing signal attenuation as the frequency increases. Careful selection of the bore of the pipe can minimise this delay or lag, but not eliminate it. With pipe lengths of only a few feet the effect is negligible but when pipe runs of tens of feet are necessary it becomes progressively more pronounced. Hydraulic operation would reduce this lag problem, but as mentioned above it is desirable to have compressible elements in a hydraulic system

corresponding to those described, and these would only re-introduce lag.

Mechanism for introducing a compensating phase advance in the control is diagrammatically illustrated in FIG. 9, where a spring 91 and a viscous damping element or dashpot 92 are mechanically connected in parallel. To one end 93 of this assembly 91,92 is applied the movement of a dancer roll and a further spring 94 acts between the other end of the assembly 91,92 and a fixed structure 95. Output movement, which is applied to a controller in the pneumatic brake supply line, is taken at 96 from the junction between the spring 94 and the assembly 91,92. The directions of movement are indicated by arrows. The spring 94 is many times stiffer than the spring 91.

Depending on the relative values of the spring rates and dashpot force/displacement constant the output displacement at 96 is related to the input displacement at 93 depending on the speed, and therefore the frequency, of the input signal. At a very low input speed or frequency the dashpot 92 will present negligible resistance to relative motion between input and output and the ratio of output-input displacements will be close to $K_1/K_1 + K_2$, where K_1 and K_2 are respectively the rates of the springs 91 and 94. In these conditions the output motion will be in phase with the input motion.

As the input frequency is increased the dashpot 92 provides a force opposing relative motion between input and output and the ratio of input-output displacements increases above the minimum $K_1/K_1 + K_2$. This ratio, or gain, is frequency dependent and its logarithm is proportional to the logarithm of frequency, this being shown in the lower half of FIG. 10 where the gain is expressed in dB.

Above a certain frequency the dashpot 92 behaves as a solid link in comparison with the spring 94 and the output approaches the input in magnitude, i.e., the gain approaches unity or 0 dB. The lower half of FIG. 2 shows these three stages as straight line asymptotes, and it can be seen that for a limited frequency range, hereinafter referred to as the operating frequency range, the gain increases at a rate of 6 dB/octave or 20 dB/decade.

FIG. 10 also shows the phase relationship between the output and input. At low and high frequencies there is substantially no phase difference, but particularly over the operating frequency range where the gain is frequency dependent there is a phase lead which achieves a peak value. This, and the frequency at which it occurs, are directly related to the values of the time constants T_1 and T_2 , respectively equal to B/K_1 and $B/K_1 + K_2$, where B is the dashpot constant. The low frequency gain is also related to the time constants, having the numerical value T_2/T_1 . Thus if the spring rates were chosen to give $K_2 = 9K_1$, the low frequency gain would be 0.1 (-20 dB) and the peak phase lead value would be 57°. By suitable choice of spring rates and dashpot constants the variable parts of the gain and phase characteristics can be shifted along the frequency axis to any desired range. For given spring rates, increasing the dashpot stiffness raises both time constants by the same factor and in effect moves the characteristics to the left to a lower frequency range. The operating frequency range can be chosen to give the device the desired effect in compensating for the brake line lag.

Referring now to FIG. 11 a practical embodiment of the mechanism of FIG. 3 employs a rotary dashpot 101 whose hub 102 can be oscillated by an arm 103 fixed thereto and to which the movements of the dancer roll are applied via a rod 104. The dashpot body (not shown) is attached to the hub and partially rotates therewith. The co-axial output shaft 105 of the dashpot has a lever 106 clamped thereto and which is normally centered by a pair of springs 107 reacting against the body 102 and by another pair of springs 108 reacting against a fixed structure 109. The springs 107 have the lower rate and act at the smaller radius and are equivalent to the spring 91. The springs are in opposed pairs so that their preload forces are in balance and to make a compact design. The free end of the lever 106 operates, by means of an adjustable length rod 110, a controller 111 in the pneumatic supply line to the brake. The rod 110 is adjustable to facilitate setting up the system. The springs and the dashpot may also be adjustable to tune to the frequency range desired.

An alternative arrangement is shown diagrammatically in FIG. 12 where the input from a dancer roll is applied at 121 to a dashpot 122, a spring 123 and a further dashpot 124 in series, the dashpot 124 acting against a fixed support 125. The output movement is taken at 126, between the dashpot 122 and the spring 123. Again, by choosing appropriate constants for the spring and the dashpot, characteristics suited to the frequency range desired can be obtained.

These mechanical phase advancing systems are preferably used in conjunction with the hydraulic or pneumatic apparatus described with reference to FIGS. 1 to 8. However, it is possible that they may be used with other controlling devices.

I claim:

1. A web tension control apparatus for a travelling web tensioned by a fluid operated web drive/brake system, the tension control apparatus including a sensor co-operating with the web to produce a physical displacement in either of two opposite directions depending on increase or decrease of tension either side of a norm, a fluid control valve which governs the supply to said drive/brake system and which is responsive to said physical displacement to restore the tension towards said norm, and feedback means to said valve from said supply downstream of the valve to resist the valve movement caused by said sensor and, after a delay, to assist said movement, wherein the improvement comprises introducing delay means into the feedback that resists the valve movement, the assisting feedback being still further delayed, and providing a resilient connection between the sensor and the valve which is sufficiently stiff to transmit said physical displacement for actuating the valve but which substantially eliminates a force reaction from the valve back to the sensor.

2. A web tension control apparatus according to claim 1, wherein said feedback means includes a pair of fluid operated actuating devices arranged to act in opposite senses on the valve and supplied through respective delay means for the supply to the drive/brake system.

3. A web tension control apparatus according to claim 2, wherein a floating lever is connected at one point to said sensor, at another point between said actuating devices and at a third point to the valve, the re-

silience of the connection between the sensor and the valve being provided by said actuating devices.

4. A web tension control apparatus according to claim 3, wherein a link provides the connection between the lever and said valve and means responsive to the outlet pressure of said valve give negative feedback to the sensor through said link.

5. A web tension control apparatus according to claim 2, wherein said actuating devices comprise chambers with diaphragms that are linked to move together by a movable element of the valve, and wherein spring means interconnect said diaphragms and said sensor.

6. A web tension control apparatus according to claim 5, wherein a housing is spanned by said diaphragms, whereby two spaced fluid chambers are formed, and the valve element is disposed within the housing between the diaphragms.

7. A web tension control apparatus according to claim 6, wherein the housing is connected to move with the sensor and the spring means act between the inside of the housing and the diaphragms.

8. A web tension control apparatus according to claim 7, wherein a third diaphragm is provided to close off a space between itself and one of the other diaphragms, this space being in communication with the outlet of the valve to provide a negative feedback to the sensor.

9. A web tension control apparatus according to claim 6, wherein there are three parallel diaphragms, a valve spindle comprising said valve element linking said diaphragms, a fixed housing mounting said diaphragms and providing two closed spaces at opposite sides of the central diaphragm, separate passageways which communicate between each of said two closed spaces and said pressure fluid line, restriction of different throttling characteristics disposed in said separate passageways to give two different time delays, and spring means normally balancing the diaphragm-spindle assembly against the pressure of fluid in said line, said input signal being applied through said resilient connection to said valve spindle.

10. A web tension control apparatus according to claim 9, wherein said valve is a poppet valve and the fluid pressure in said line is applied to a chamber defined by an outer diaphragm, the housing, and the head of the poppet valve which, when balanced, permits leakage of the pressure fluid.

11. A web tension control apparatus according to claim 9, wherein the valve is constituted by an end of said spindle co-operating with an orifice from which the fluid under pressure jets to an extent determined by the spindle position.

12. A web tension control apparatus according to claim 9, wherein the valve has two elements on said spindle at opposite ends of a throat, the fluid under pressure being normally directed past one said element and into said line, which communicates with said throat, and also past said other element as leakage.

13. A web tension control apparatus according to claim 1, wherein said resilient connection includes a mechanical phase shifting system comprising a combination of spring means and viscous damping means arranged so that the input thereto results in a modified output at another point with low gain at low input frequencies and gain increasing with frequency over said middle band of frequencies, the system including an assembly of first spring means and viscous damping

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means coupled in parallel, second spring means in series therewith, and a fixed structure against which said second spring means react, the input being to the free end of said assembly and the output from a point effectively intermediate said assembly and said second spring means. 5

14. A web tension control apparatus according to claim 12, wherein said viscous damping means is a rotary dashpot arranged to be partially rotated by said input, the first spring means acting between the dashpot body and an output member and the second spring means acting between a fixed structure and said member. 10

15. A web tension control apparatus according to claim 13, wherein said second spring means is stiff compared with said first spring means. 15

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16. A web tension control apparatus according to claim 1, wherein said resilient connection includes a mechanical phase shifting system comprising a combination of spring means and viscous damping means arranged so that the input thereto results in a modified output at another point with low gain at low input frequencies and gain increasing with frequency over said middle band of frequencies, the system including an assembly of first viscous damping means, spring means and second viscous damping means in series to one end of which assembly the input is applied and the other end of which reacts against a fixed structure, the output being from a point effectively between the spring means and the viscous damping means at said one end.

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