

May 4, 1954

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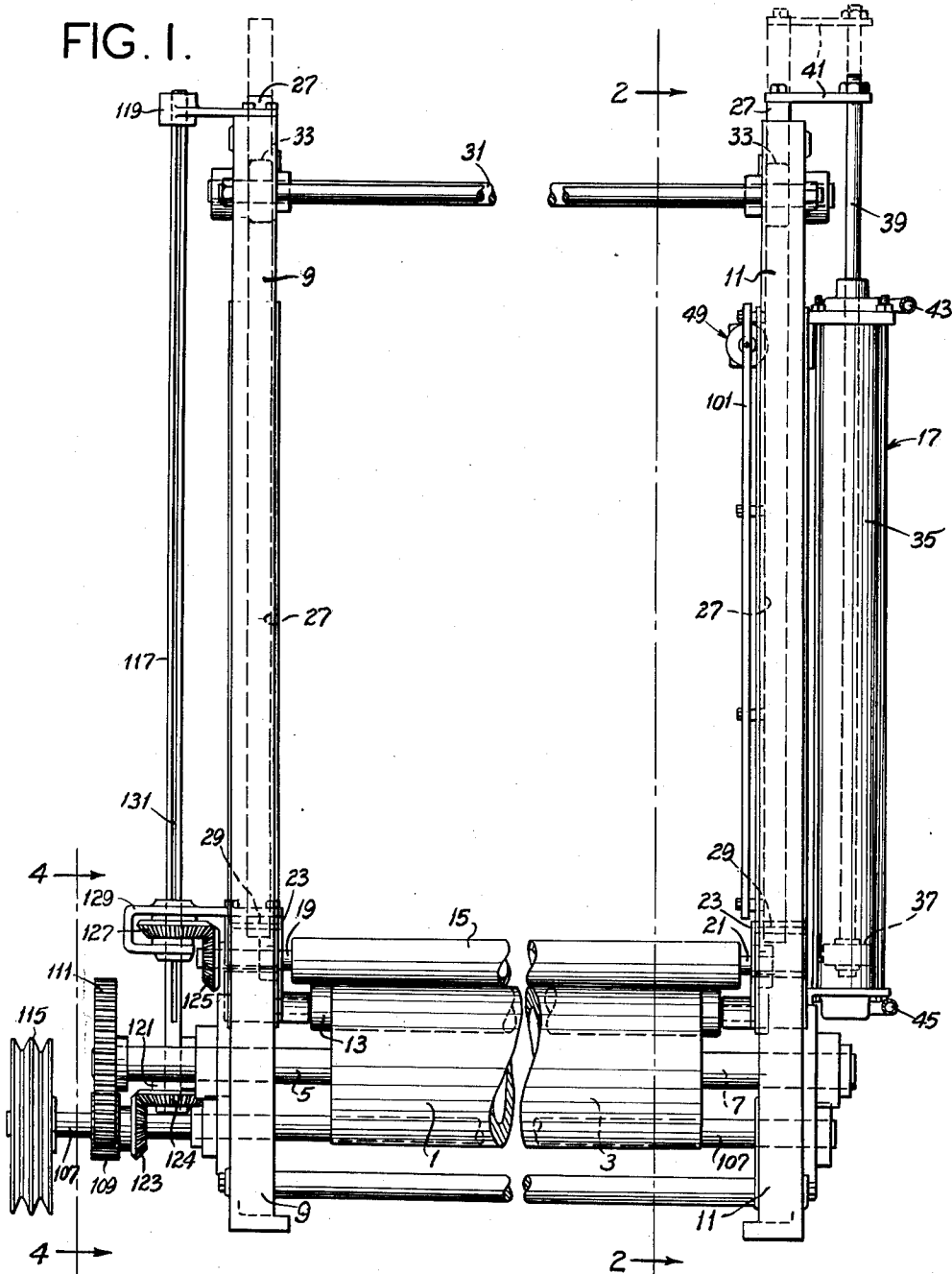
2,677,508

WEB WINDING

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3 Sheets-Sheet 1

FIG. 1.



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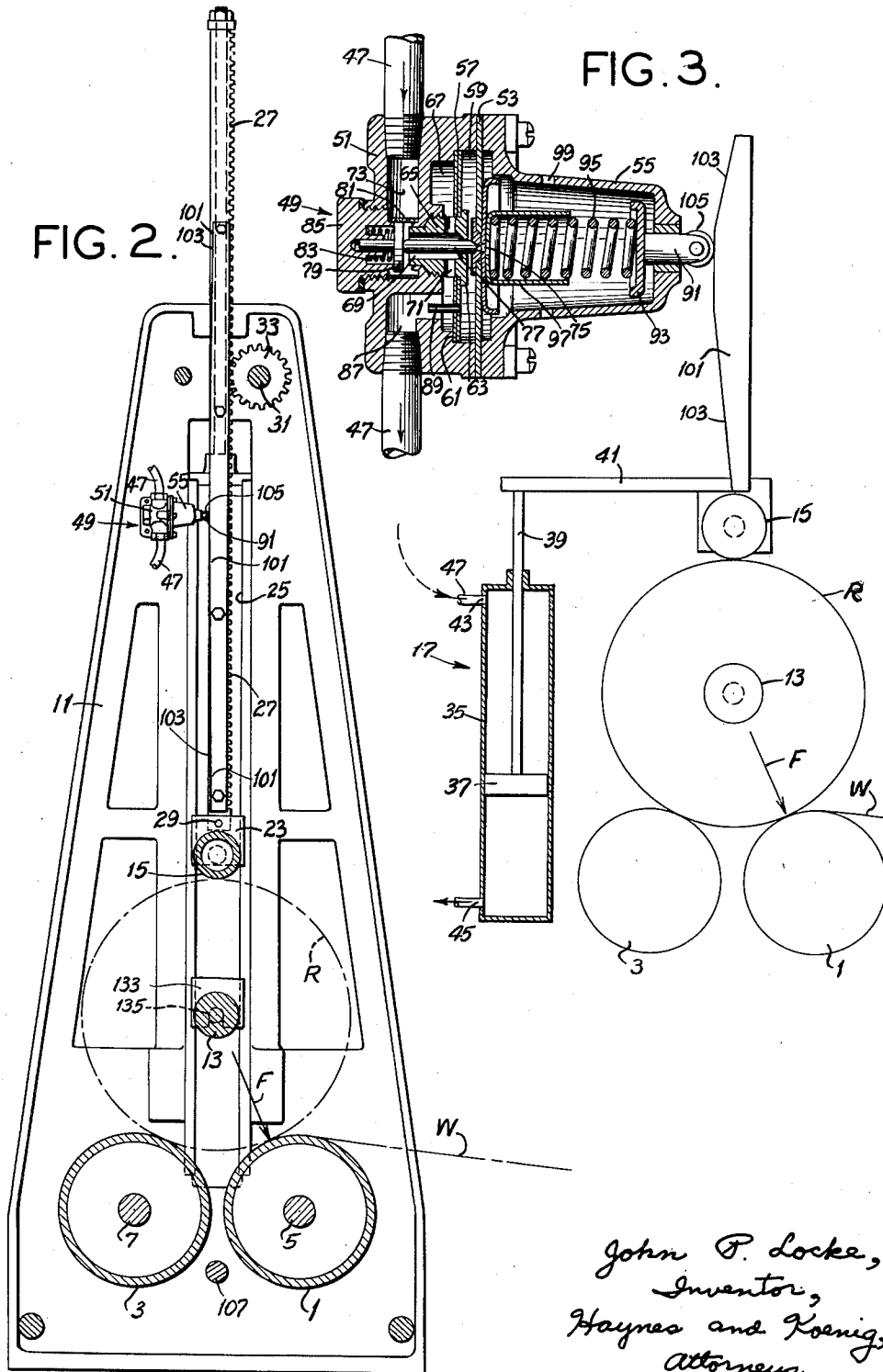
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FIG. 5.

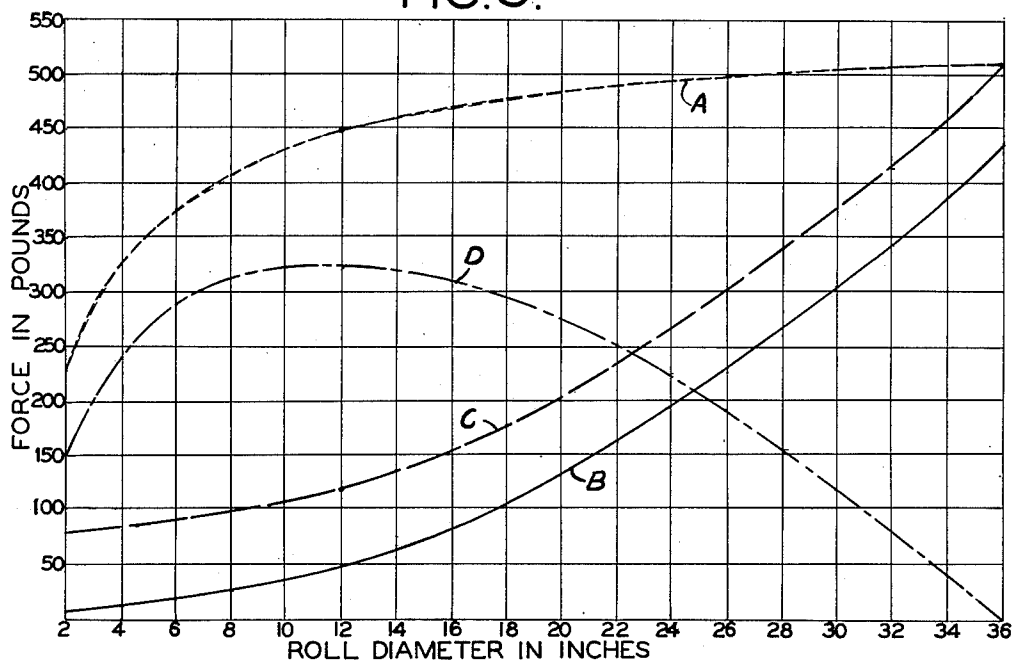
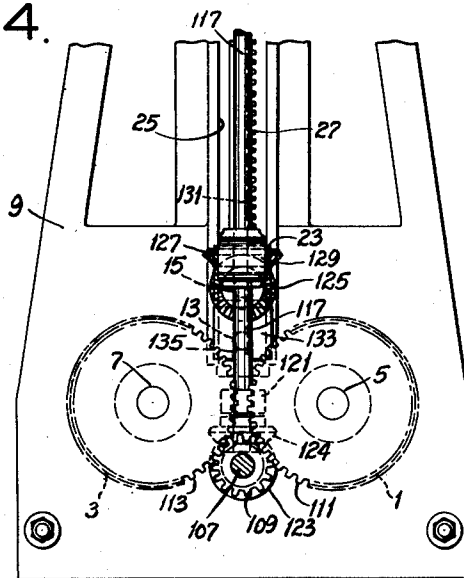


FIG. 4.



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UNITED STATES PATENT OFFICE

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WEB WINDING

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4 Claims. (Cl. 242—66)

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This invention relates to web winding, and more particularly to web winders of the friction drum type.

The object of the invention is the provision of a two-drum friction drum winder for winding a web into a roll with close control over the hardness of the roll, and adapted to wind a roll of substantially uniform hardness throughout its mass from its innermost to its outermost convolution.

According to my observations, the hardness of a roll of web material wound by a friction drum winder is for the most part determined by the compressive force exerted upon the web along a line transverse to the length of the web in the nip of the web roll being wound and the drum of the winder where the web first comes on to the web roll, assuming that the web is being wound under substantially constant tension. This is particularly true of web material that has a relatively low modulus of elasticity in tension. I believe that the hardness of the roll is largely determined in the case of such material by the compression of the web material being wound against the drum of the winder where the material first comes on to the roll. The extent of compression, of course, is dependent upon the compressive force exerted on the web by the web roll. To wind a web roll of uniform hardness, it should be substantially equally compressed throughout its length. For equal compression of the web throughout its length, the compressive force exerted on the web by the web roll should be maintained substantially constant throughout the winding operation.

The invention, in general, involves a friction drum winder having a rider roll bearing on the web roll being wound, and means for applying force to the web roll as the latter is wound in accordance with its increase in diameter in such manner as to maintain the compressive force exerted on the web in the nip of the web roll and the drum of the winder where the web first winds on to the roll substantially constant throughout the winding operation. This produces a web roll of substantially uniform hardness from its innermost to its outermost convolution.

The force is applied to the web roll being wound by fluid pressure means acting on the rider roll for biasing it toward the web roll and supplied with fluid under pressure through a line including a pressure-control valve. This valve is controlled by means movable by the web roll, as it increases in diameter for varying the pressure

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of fluid pressure means as winding progresses. This means may be such as to adjust the valve in such a way as to vary the pressure of fluid in the fluid pressure means (and the resultant force exerted by the web roll against the friction drum) to maintain the compressive force on the web between the web roll and the friction drum substantially constant. This means may be made such as to adjust the valve in other ways where other than uniform hardness of the web roll is desired. The winder also has features of construction which reduce its required headroom to a minimum and allow for the use of a single fluid pressure means while assuring parallelism of the rider roll and friction drums. Other features will be in part apparent and in part pointed out hereinafter.

The invention accordingly comprises the elements and combinations of elements, features of construction and arrangements of parts which will be exemplified in the structures hereinafter described, and the scope of the application of which will be indicated in the following claims.

In the accompanying drawings, in which one of various possible embodiments of the invention is illustrated,

Fig. 1 is a view in elevation of a friction drum winder of this invention, illustrating parts in the position they occupy at the start of winding a roll;

Fig. 2 is a vertical section taken on line 2—2 of Fig. 1, illustrating parts in the position they occupy when a roll is about half wound;

Fig. 3 is a view illustrating a pressure control valve of the winder in cross section and diagrammatically illustrating other elements;

Fig. 4 is a vertical section taken on line 4—4 of Fig. 1; and,

Fig. 5 is a graph illustrating certain force-web roll diameter characteristic curves.

Similar reference characters indicate corresponding parts throughout the several views of the drawings.

Referring to the drawings, a friction drum winder constructed in accordance with this invention is shown to comprise two friction drums 1 and 3 having shafts 5 and 7 journaled at their ends in side frames 9 and 11. The drums, as herein illustrated, are of equal diameter and journaled by their shafts in the side frames near the bottom of the latter with their axes parallel, in the same horizontal plane, and spaced apart a distance somewhat greater than the drum diameter so that there is a space between the drums. The drums 1 and 3 are driven in the same

direction (counterclockwise as viewed in Fig. 2) and are adapted peripherally to support and drive a web roll R (shown about half wound in phantom in Fig. 2) being wound on an arbor 13. The web being wound is designated W and comes on to the web roll in the nip of the web roll and the friction drum 1 of the winder, which is hereinafter referred to as the first drum of the winder, the drum 3 being hereinafter referred to as the second drum of the winder. The web W, coming on to the periphery of the web roll R, is compressed against the first friction drum 1 by the web roll.

The compressive force to which the web is subjected is represented by the force arrow or vector F in Figs. 2 and 3. This force is exerted downward in the direction of the line joining the centers of the web roll and the first drum 1. The magnitude of the vector F depends upon the diameter of the web roll and the downward force applied to the top of the web roll by a rider roll 15, which may be positively biased downward by fluid pressure means generally designated 17. It will be seen that as the diameter of the web roll increases in winding, the weight of the web roll increases, the angle of the direction of vector F with respect to the vertical changes, and the point of application of the vector also changes. The magnitude of vector F is the algebraic sum of the resultant of the weight of the web roll (and arbor) in the direction of the vector and the resultant of the force applied by the rider roll in the direction of the vector.

Referring now to Fig. 5, which is a graph in which the horizontal coordinates represent roll diameter in inches and the vertical coordinates represent force in pounds, the curve A represents the total vertical force which must result from the weight of the web roll (and arbor) and the vertical force due to the rider roll to produce a vector F of constant magnitude in a specific case involving a winder wherein drums 1 and 3 are $10\frac{1}{2}$ inches in diameter with their axes spaced apart $11\frac{1}{4}$ inches, and wherein the web being wound weighs about 0.44 lb. per yard of its length. The curve A is plotted by selecting a desired magnitude for vector F (this being about 260 lbs. for the curve A illustrated) and determining the total vertical force acting directly downward in the line of the center of the web roll necessary to establish this vector magnitude for the various vector angles. For example, the total vertical force acting through the center of the web roll necessary to produce a magnitude for vector F of 260 lbs. when the diameter of the web roll is 2 inches is about 230 lbs. When the diameter of the web roll is 12 inches, the total vertical force required to produce a 260 lb. vector F is about 450 lbs. When the diameter of the web roll is 36 inches (full roll diameter), the total vertical force required to produce a 260 lb. vector F is about 510 lbs.

The curve B represents the dead weight of the web roll (plus the weight of the arbor 13). For example, when the diameter of the web roll is 2 inches, its weight plus the arbor weight is about 10 lbs. When the diameter of the web roll is 12 inches, its weight plus the arbor weight is about 50 lbs. When the diameter of the web roll is 36 inches (full roll), its weight plus the arbor weight is about 440 lbs.

The curved C represents the combined dead weights of the web roll, arbor, and the rider roll (including elements which add dead weight to the rider roll), and thus represents the total dead

weight contributing toward the vector F. This curve is drawn for a rider roll dead weight of about 70 lbs., and is plotted simply by adding 70 lbs. to the vertical coordinates of curve B. Thus, at 2 inch roll diameter, the total dead weight is 80 lbs., at 12 inch roll diameter the total dead weight is 120 lbs., and at 36 inch full roll diameter the total dead weight is 510 lbs.

The curve D represents the extra downward vertical force which must be applied to the rider roll to augment the dead weight of the web roll, arbor, and rider roll to produce the desired magnitude of 260 lbs. for the vector F. This curve is determined by plotting the differences between the vertical coordinates of curves A and C. Thus it will be seen that when the diameter of the web roll is 2 inches, the rider roll must apply an extra force of about 150 lbs. (the difference between the 230 lbs. and 80 lbs. vertical ordinates of curves A and C at 2 inch roll diameter). When the diameter of the web roll is 12 inches, the rider roll must apply an extra force of about 330 lbs. (the difference between the 450 lb. and 120 lb. vertical ordinates of curves A and C at 12 inch roll diameter). When the diameter of the web roll is 36 inches, no extra force is required of the rider roll, other than its own weight. The 260 lb. magnitude for vector F is purposely selected so that at the full roll diameter of 36 inches, no extra force need be applied to the rider roll.

The rider roll 15 bears on the top of the web roll and is adapted to move upward as the web roll increases in diameter while its axis is maintained parallel to the axes of the first and second friction drums and the web roll, and directly above the axis of the web roll. For this purpose, the rider roll is provided at its ends with trunnions 16 and 21 which are journaled in bearings 23 vertically slidable in vertical guideways 25 formed in the side frames 9 and 11. The vertical plane of the guideways is centered with respect to the axes of the drums 1 and 3. Racks 27 extend vertically upward from the bearings 23, being fixed at their lower ends to the bearings as indicated at 29. The teeth of the racks face toward the right as viewed in Fig. 2. A shaft or torque bar 31 is journaled at its ends in the side frames 9 and 11 adjacent the upper ends of the frames and has pinions 33 fixed thereon in mesh with the racks. The axis of the bar 31 parallels the axes of the friction drums 1 and 3. The arrangement is such that the ends of the rider roll are constrained against relative vertical movement, while being free to move upward (or downward), so that the rider roll remains parallel to the friction drums as the web roll R increases in diameter and moves the rider roll upward.

The fluid pressure means 17 comprises an air pressure cylinder 35 mounted in vertical position on the outer side of the side frame 11. A piston 37 slides in the cylinder and a piston rod 39 extends from the piston through the upper head of the cylinder to a connection at 41 with the upper end of the adjacent rack 27. In Fig. 3, the connection 41 is diagrammatically shown as made to the rider roll bearing rather than to the upper end of the rack as in Figs. 1 and 2 for simplification of illustration. The cylinder has a pressure supply port 43 at its upper end and a vent port 45 at its lower end. The arrangement is such that by supplying air under pressure to the cylinder above the piston through port 43, the piston will tend to pull down the adjacent rack 27 and thus tend to bias the rider roll 15 downward against the top of the web roll R. The downward force

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exerted by the piston is equally divided between the two racks due to their being geared together by the torque bar 31 and pinions 33, and thus the downward forces acting on the ends of the rider roll are equalized. Thus, the rider roll is maintained parallel to the friction drums despite the fact that only one air cylinder at one side of the winder is used to exert the downward bias on the rider roll.

The air cylinder 35 is supplied with air under pressure through a line 47 including a relieving type pressure control valve 49 adapted to control the pressure in the cylinder above the piston. The valve 49 herein illustrated is a standard commercially available pressure regulator with some modification to adapt it for use in the winder of this invention. As shown in Fig. 3, the valve 49 in general comprises a generally cup-shaped body 51. The open side of the body is closed by a resilient diaphragm 53, the margin of the diaphragm being compressed against the rim of the body by a bonnet 55 bolted to the body. Within the body, between the diaphragm and a baffle plate 57, is a chamber 59. The baffle plate 57 is held against a shoulder 61 by a head 63 on a fitting 65 which extends through an opening in the center of the baffle plate and which is threaded at its left end as viewed in Fig. 3 in the base of the body 51. Within the body 51, between the baffle plate 57 and the base of the body is another chamber 67. The latter is of smaller diameter than the chamber 59 to provide the shoulder 61.

The fitting 65 has an axial passage 69 which narrows down at its right end as viewed in Fig. 3, and radial ports 71 providing for communication between the passage 69 and the chamber 67. At its left end as viewed in Fig. 3, the fitting opens into a transverse air inlet passage 73 in the base of the body 51. In the center of the diaphragm 53 is a vent 75. This is adapted to be closed by a ball-end valve 77 axially slidable in the narrow right end of the passage 69 through the fitting 65. The valve has a head 79 located in the air inlet passage 73. A strainer screen 81 surrounds the head. A spring 83 reacts against the head from a plug 85 threaded in an opening in the base of the body 51 aligned with the fitting 65 to bias the valve 77 toward seating engagement with the vent 75. The body 51 is formed with an air outlet passage 87 leading from the chamber 67. A siphon tube 89 in the baffle plate 57 provides for communication between chamber 59 and the air outlet passage. A follower rod 91 is slidable in the end of the bonnet 55. At its inner end this rod carries a spring seat 93. A compression spring 95 reacts from the seat 93 against a seat 97 engaging the diaphragm 53 to bias the diaphragm toward the left as viewed in Fig. 3. In the wall of the bonnet are breather holes 99.

The spring 95 normally holds the parts in the position shown in Fig. 3 wherein the ball end of the valve 77 closes the vent 75. The valve 77 is held against the bias of spring 83 in the retracted position illustrated in Fig. 3 wherein the valve head 79 is spaced to the left of the adjacent end of the fitting 65. Under these circumstances, air may flow through the inlet passage 73, through the passage 69 in the fitting 65, through the radial ports 71 in the fitting to the chamber 67 and thence through the outlet passage 87 and line 47 to the cylinder 35, with a pressure drop in flowing through the valve. The chamber 59 is in communication with chamber 67 through the tube 89, so that the diaphragm 53 is subjected

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to the pressure in the outlet 87, which is generally the same as in the cylinder 35. If this pressure should become high enough to overcome the force of spring 95, for a particular setting of this spring, the diaphragm moves away from the ball end of valve 77 and air is vented through the vent 75 into the bonnet and out through the breather holes 99. This venting occurs after the valve 77 has moved to the right in Fig. 3 to a forward position wherein the head 79 on the valve closes the left end of the fitting 65, thereby blocking off the passage 69 in the fitting from the air inlet passage 73. When the pressure has been sufficiently reduced, spring 95 returns the ports to their Fig. 3 position. How much pressure the valve 49 will hold in the cylinder is determined by the compression of the spring 95, and this in turn is determined by the position of the follower rod 91.

The pressure control valve 49 is mounted on the side frame 11 with its follower rod 91 in position to be automatically adjusted by a cam 101 fixed to the adjacent rack 27. The cam 101 moves upward with the rider roll 15 as the diameter of the web roll increases. The cam consists of an elongate flat bar formed with a cam edge 103 engaging a follower roller 105 on the outer end of the follower rod 91. The cam edge, as illustrated, is of such development and so phased as to adjust the control valve to obtain the different air pressures required in the air cylinder 35 at different web roll diameters to supply the extra force which must be applied to the rider roll to maintain the magnitude of force F at least substantially constant throughout the winding. The necessary development of the cam edge may be determined from the curve D and the characteristics of the valve. It will be seen that the development of the cam is such that it first gradually moves the rod 91 inward, thereby gradually to increase the pressure held in the cylinder 35, and ultimately permits the rod 91 to move outward thereby gradually to lower the pressure held in the cylinder.

The friction drums 1 and 3 and the rider roll 15 are all positively driven in counterclockwise direction by gearing illustrated in Figs. 1 and 4. As shown, this gearing comprises a horizontal shaft 107 journaled in the side frames 9 and 11. Fixed on this shaft outward of the side frame 9 is a gear 109 in mesh with gears 111 and 113 on drum shafts 5 and 7, respectively. Shaft 107 has a pulley 115 fixed on its outer end driven through a belt drive from a suitable source of power (not shown). A rider roll drive shaft 117 is mounted in vertical position on the outside of side frame 9 by means of upper and lower bearings 119 and 121. This shaft 117 extends generally the full height of the side frame with its axis intersecting the axis of the rider roll. A bevel gear 123 on the horizontal shaft 107 meshes with a bevel gear 124 on the lower end of the shaft 117 to drive the latter. The trunnion 19 of the rider roll 15 extends out of its bearing 23 and has a bevel gear 125 fixed on its outer end. This meshes with a bevel gear 127 which rotates with the shaft 117 but which is axially slidable thereon. The gear 127 is carried by a yoke 129 fixed to the bearing 23 for the rider roll trunnion 19. It is keyed to rotate with the shaft 117 while being axially slidable thereon by providing it with a key slidable in a spline 131 in the shaft 117. At 133 are shown bearings for engaging the ends of the shaft 135 of the arbor. These

bearings are slidable in the guideways 25 below the bearings 23.

Operation is as follows:

Air is supplied to the air inlet of the pressure control valve 49 at a pressure high enough to accommodate the highest pressure needed in the cylinder 35, taking into account the pressure drop in the valve. At the start of winding, the rider roll 15 is in its lowermost position bearing down on the arbor 13 and wedging it between the first and second friction drums 1 and 3. Racks 27, cam 101 and the piston 37 are in their lowermost position. The cam 101 under these circumstances holds the follower rod 91 in such position that the valve 49 regulates the pressure in cylinder 35 to such value as to exert the necessary extra force on the rider roll 15 to make force F equal the desired value (260 lbs. for example). Referring to Fig. 5, it will be seen that for a 2 inch roll diameter in the particular case illustrated, this extra force would be 150 lbs., and the cam is so developed and phased as, in its position corresponding to 2 inch roll diameter, to set the valve to establish such a value of pressure in the cylinder 35 as to produce 150 lbs. force.

As winding proceeds, the web roll R, increasing in diameter, moves the rider roll 15, racks 27 and cam 101 upward. As the cam moves upward from starting position, it moves the follower rod 91 inward, thus gradually increasing the pressure in the cylinder 35 to increase the extra force exerted by the rider roll on the web roll in accordance with the requirements as dictated by the curve D of Fig. 5. For example, when the diameter of the web roll has increased to 12 inches, the cam 101 has moved the follower rod 91 to a position wherein valve 49 establishes such pressure in the cylinder 35 as to cause an extra force of 330 lbs. to be applied by the rider roll to the web roll. From this point on, however, the follower rod 91 moves outward, thereby gradually decreasing the pressure in the cylinder 35 to decrease the extra force exerted by the rider roll on the web roll in accordance with the requirements dictated by the curve D of Fig. 5. For example, when the diameter of the web roll has increased to 36 inches, the cam meanwhile has permitted the follower to move outward completely to relieve the spring 95, so that the space in cylinder 35 above the piston 37 is fully vented, with the result that no extra force is applied by the rider roll.

It will be understood that instead of maintaining the force F substantially constant to wind rolls of uniform hardness, the force might be varied as desired by changing the development of the cam 101. Or a particular cam might be retained and the entire curve D shifted up or down as by changing the length of the rod 91. Thus, the winder of this invention is useful not only to carry out the method of this invention for producing rolls of uniform hardness but also to control the roll hardness in any manner desired.

The winder construction involving the gearing together of the racks 27 by the torque bar 31 and pinions 33 thereon and the use of only one air cylinder is advantageous in that it assures parallelism of the rider roll 15 to the friction drums 1 and 3 throughout the winding even though only one air cylinder at one side of the winder is used without requiring the use of special anti-backlash gears and racks to eliminate play.

In view of the above, it will be seen that the

several objects of the invention are achieved and other advantageous results attained.

As many changes could be made in the above constructions without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

I claim:

10 1. A friction drum winder comprising two friction drums journaled at their ends in side frames near the bottom of the frames, a rider roll journaled at its ends in bearings which are vertically slidable in vertical guideways in the side frames, the rider roll being adapted to bear on the top of a web roll being wound and to be moved upward by the web roll as the latter increases in diameter, racks extending vertically upward from the rider roll bearings, a torque bar journaled at its ends in the side frames near the top of the frames and carrying pinions in mesh with the racks, a fluid pressure cylinder mounted in vertical position on one of the side frames having an inlet port at its upper end and a vent at its lower end, a piston reciprocable in the cylinder, and a piston rod extending from the piston through the upper end of the cylinder and fixed to the upper end of the adjacent rack.

30 2. A friction drum winder comprising two friction drums journaled at their ends in side frames near the bottom of the frames, a rider roll journaled at its ends in bearings which are vertically slidable in vertical guideways in the side frames, the rider roll being adapted to bear on the top of a web roll being wound and to be moved upward by the web roll as the latter increases in diameter, racks extending vertically upward from the rider roll bearings, a torque bar journaled at its ends in the side frames near the top of the frames and carrying pinions in mesh with the racks, a fluid pressure cylinder mounted in vertical position on one of the side frames having an inlet port at its upper end and a vent at its lower end, a piston reciprocable in the cylinder, a piston rod extending from the piston through the upper end of the cylinder and fixed to the upper end of the adjacent rack, means for supplying fluid under pressure to the inlet port of the cylinder including a pressure control valve for variably controlling the pressure of fluid in the cylinder above the piston, and means movable with the rider roll for controlling the valve.

55 3. A friction drum winder comprising two friction drums journaled at their ends in side frames near the bottom of the frames, a rider roll journaled at its ends in bearings which are vertically slidable in vertical guideways in the side frames, the rider roll being adapted to bear on the top of a web roll being wound and to be moved upward by the web roll as the latter increases in diameter, racks extending vertically upward from the rider roll bearings, a torque bar journaled at its ends in the side frames near the top of the frames and carrying pinions in mesh with the racks, a fluid pressure cylinder mounted in vertical position on one of the side frames having an inlet port at its upper end and a vent at its lower end, a piston reciprocable in the cylinder, and a piston rod extending from the piston through the upper end of the cylinder and fixed to the upper end of the adjacent rack, means for supplying fluid under pressure to the inlet port of the cylinder including a pressure control valve

for variably controlling the pressure of fluid in the cylinder above the piston, said valve being mounted on one of said side frames and having a movable pressure-setting element, and a cam mounted on the adjacent rack and acting on the pressure-setting element of the valve for varying the pressure of fluid in the cylinder above the piston as winding progresses.

4. A friction drum winder as set forth in claim 3 wherein the cam is formed gradually to increase the pressure of fluid in the winder above the

piston from the start of winding to a maximum, and then gradually to decrease the pressure to a minimum.

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