

May 13, 1958

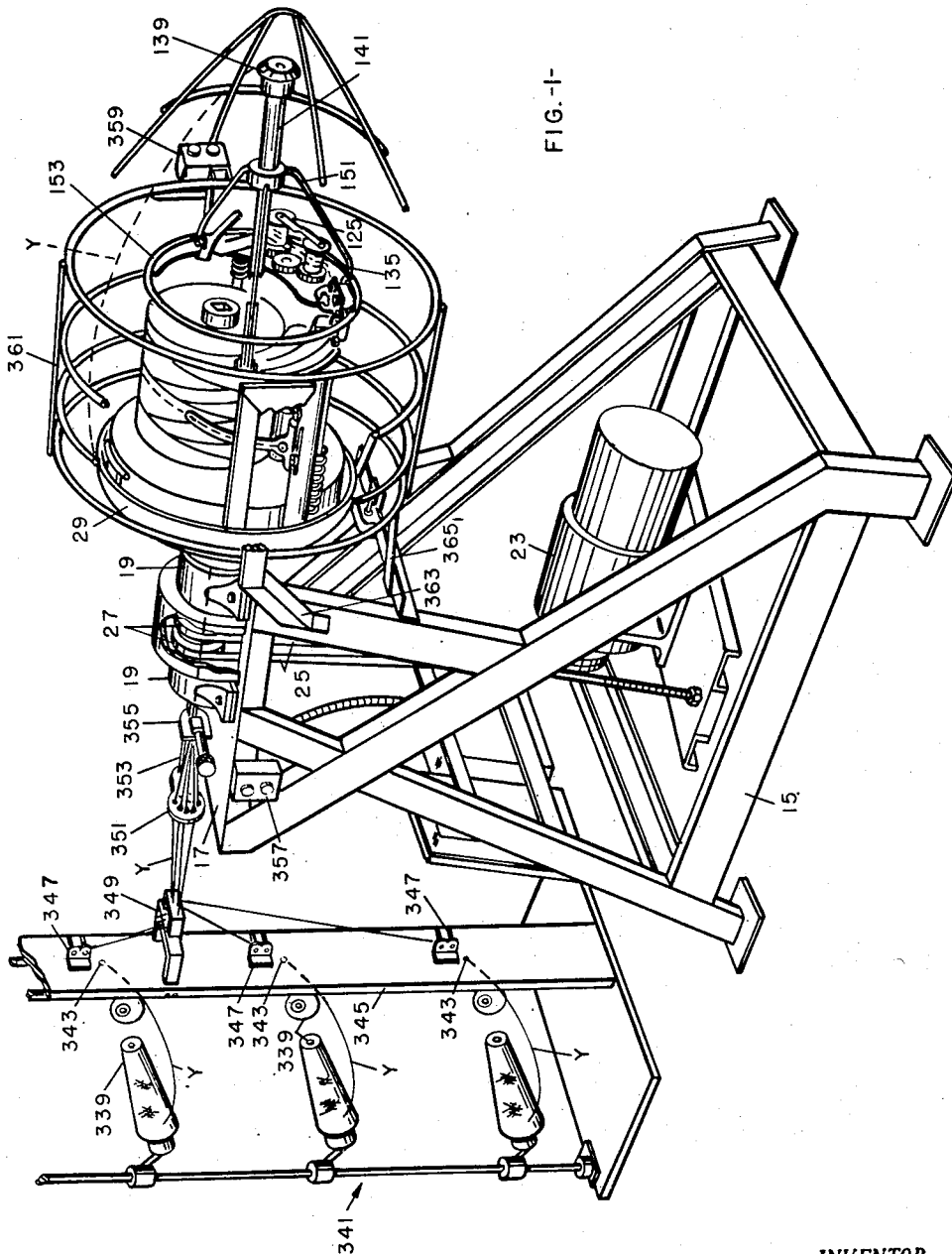
N. E. KLEIN

2,834,178

STRAND HANDLING MACHINE

Filed April 20, 1954

7 Sheets-Sheet 1



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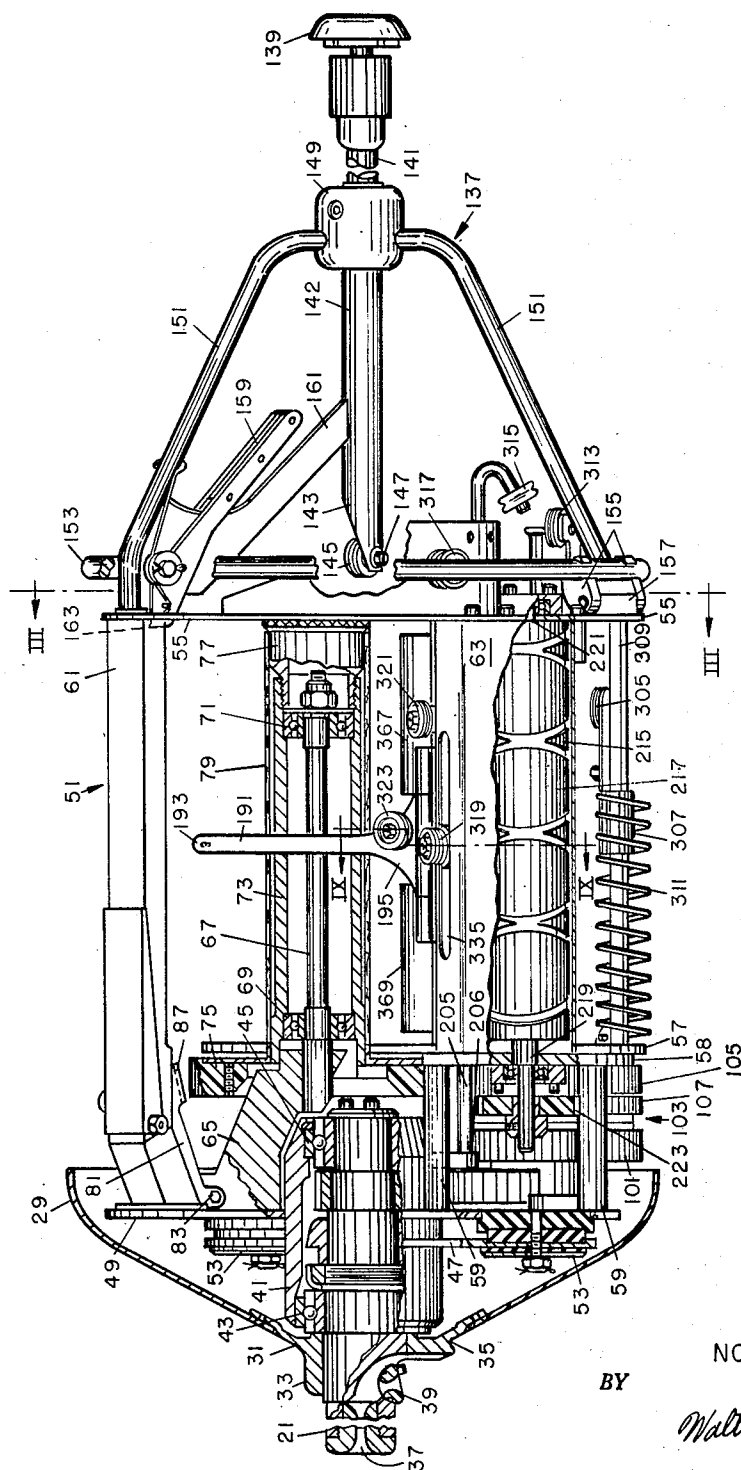
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N. E. KLEIN
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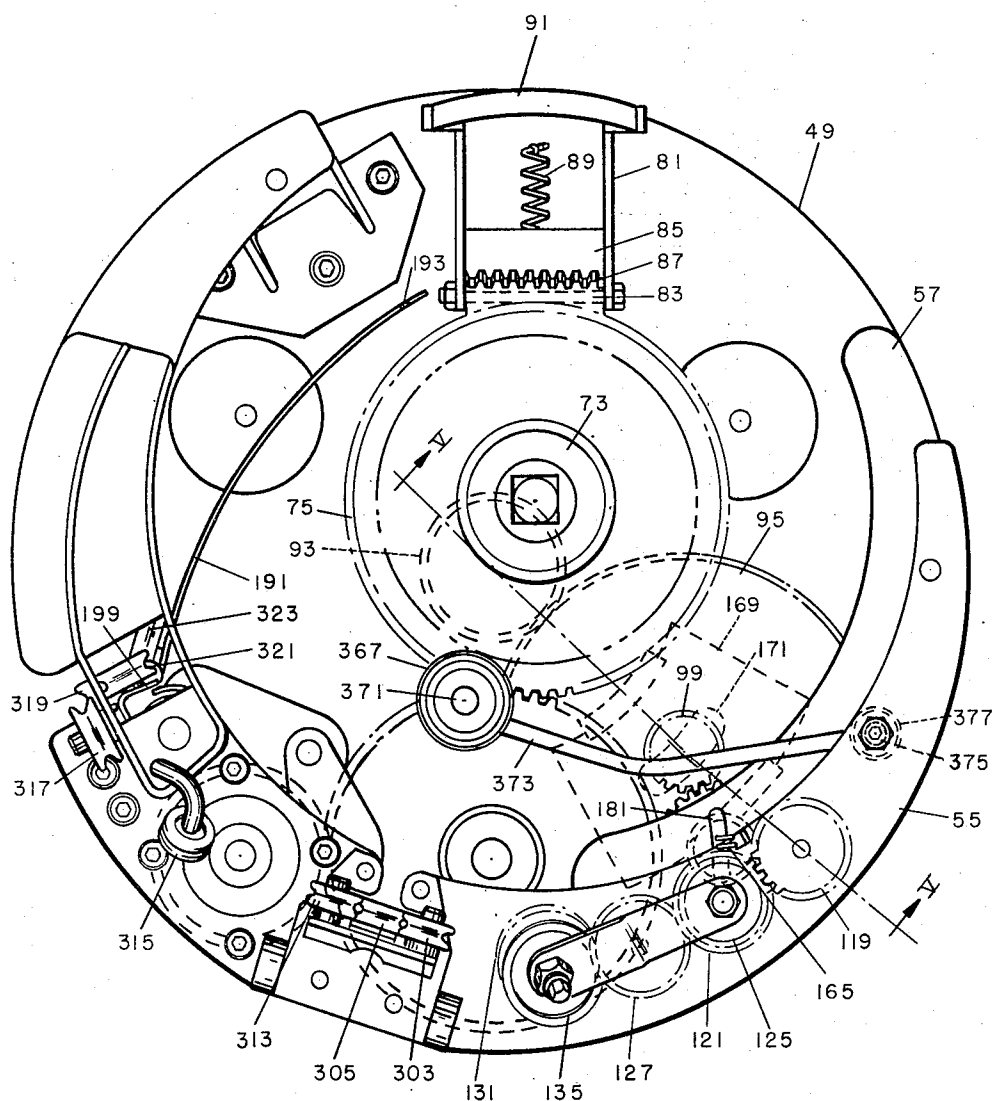


FIG. -3-

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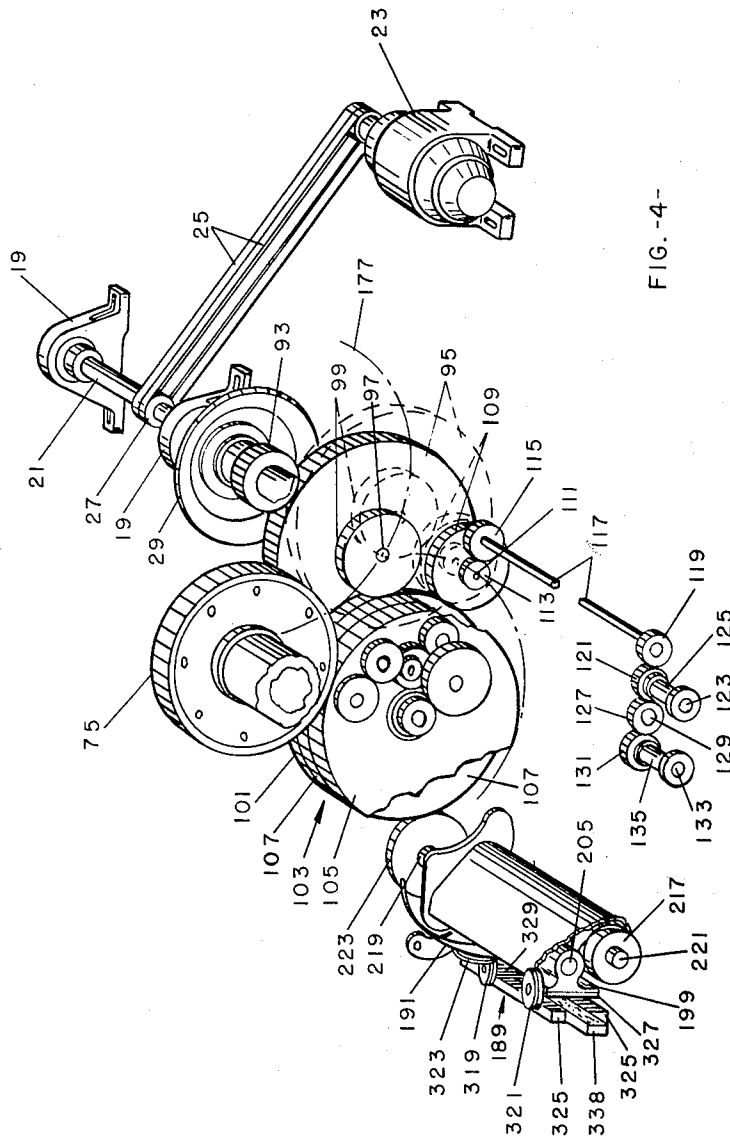
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STRAND HANDLING MACHINE

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7 Sheets-Sheet 4



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STRAND HANDLING MACHINE

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7 Sheets-Sheet 5

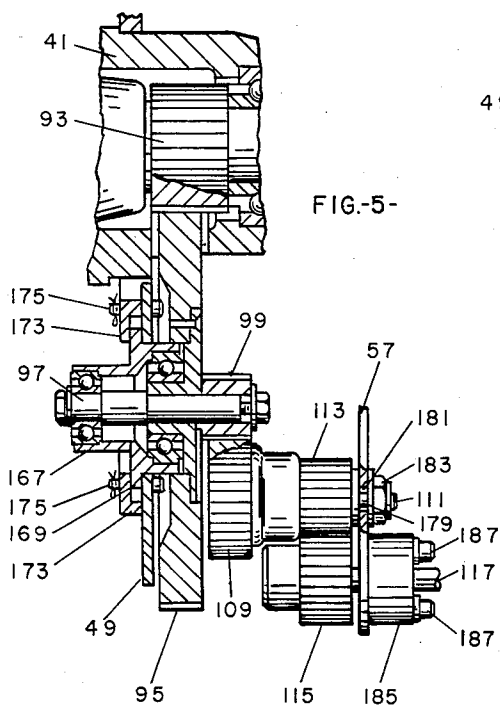


FIG. 5-

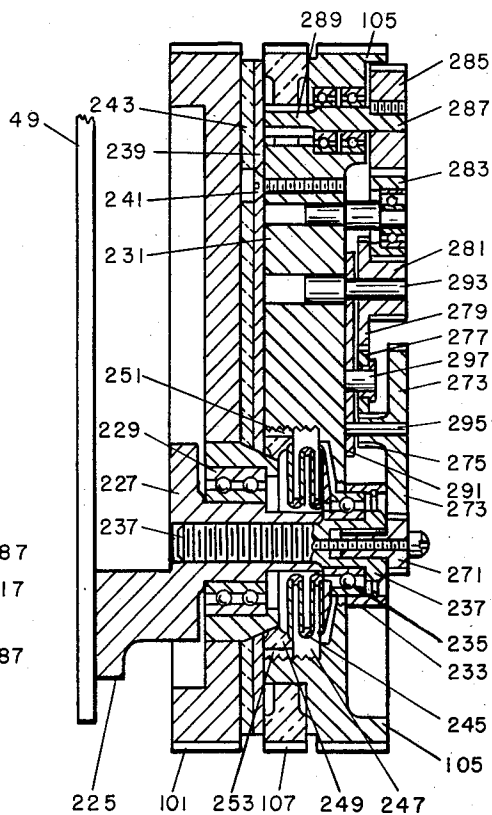


FIG. 7-

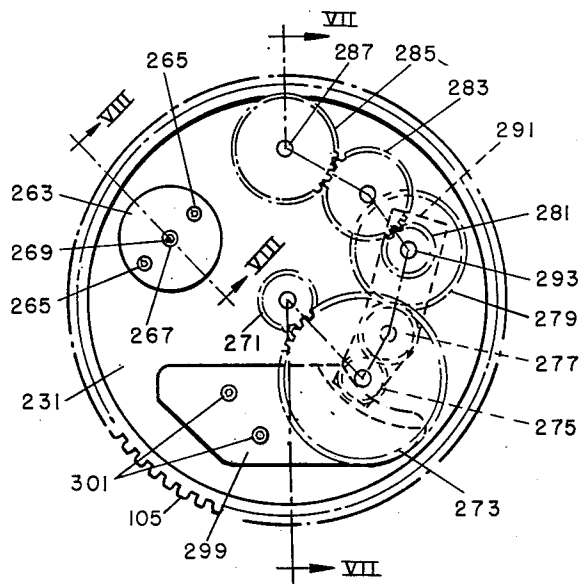


FIG. 6-

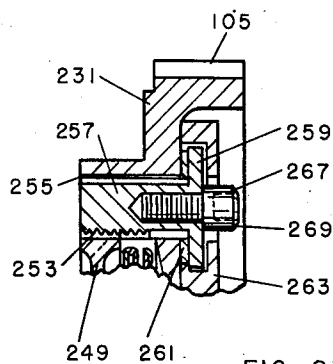


FIG. 8-

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STRAND HANDLING MACHINE

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7 Sheets-Sheet 6

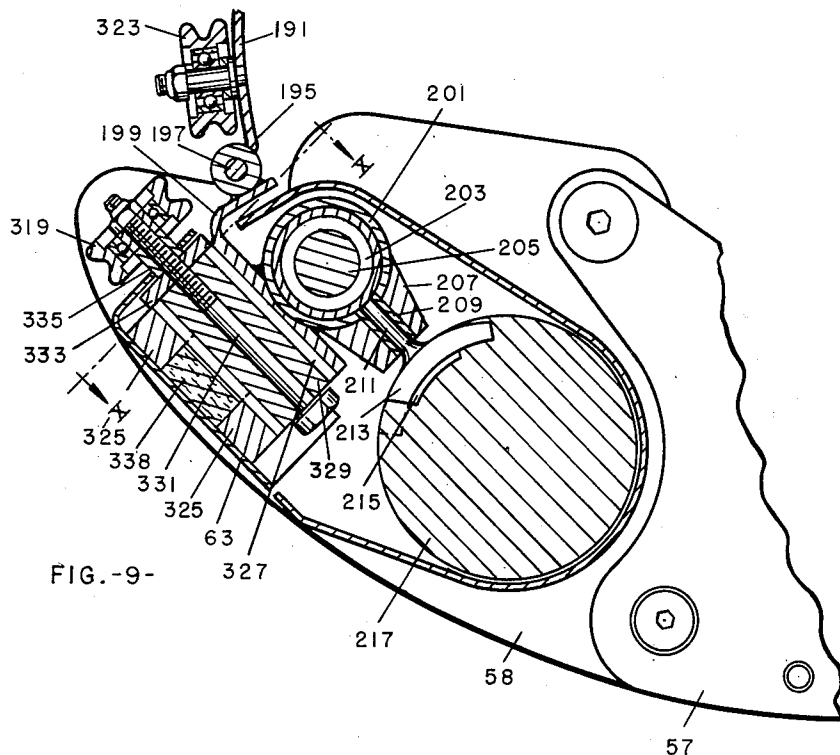
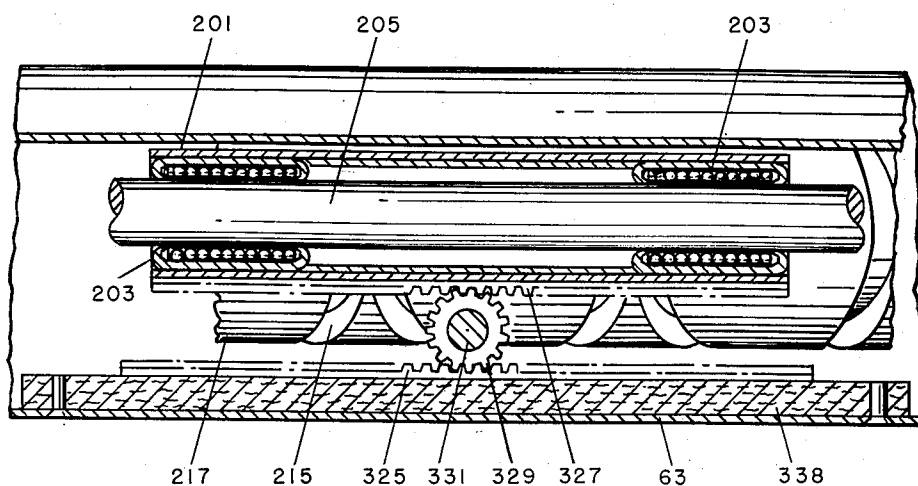


FIG. -10-

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7 Sheets-Sheet 7

FIG. -12-

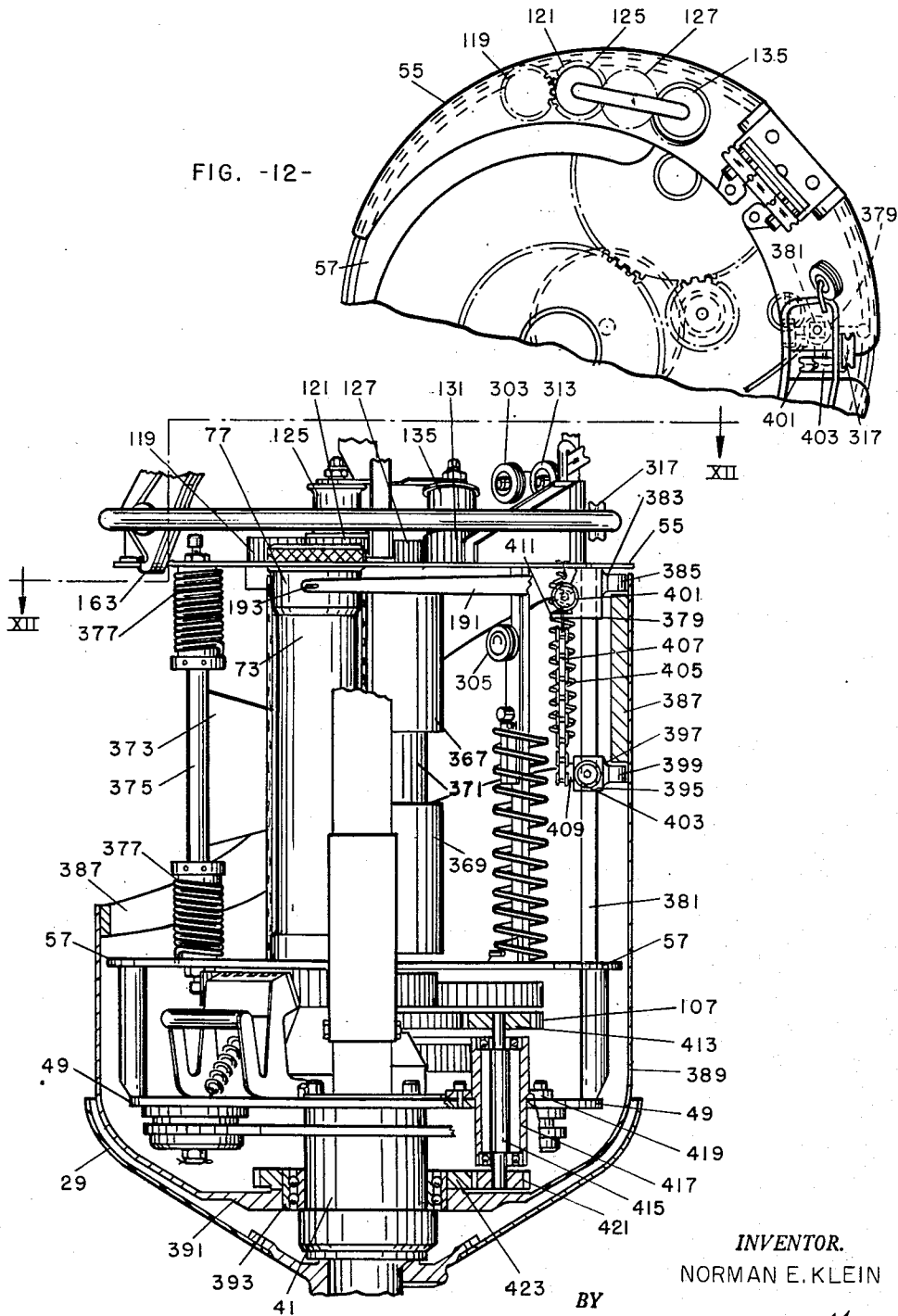


FIG. -II-

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2,834,178

STRAND HANDLING MACHINE

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Application April 20, 1954, Serial No. 424,461

29 Claims. (Cl. 57—58.52)

This invention relates to a strand handling machine and more particularly to a cabling machine for simultaneously plying a plurality of strands together while twist is being inserted therein. The machine of this invention is generally of the type shown and described in my co-pending application Serial Number 417,619, filed March 22, 1954, and is an improvement thereover.

The combined twisting and plying machine of this invention may be further characterized as being of the inflow type capable of receiving a large number of strands from individual packages carried by a creel and which are drawn in combined form axially into the twister balloon and then fed axially into the interior of the balloon and wound on suitable takeup means to form a commercially acceptable precision wound package of high density.

More specifically, although not particularly limited thereto, the machine of my invention is especially adapted to produce a twisted multiple end strand or cord suitable for a variety of uses such as in the manufacture of belts of all kinds, fire hose, nets and seines, parachute cords, tires, chenille, fringes, tufted rugs and carpets. The machine by virtue of the combination of a number of improved elements is capable of handling flexible strands or ends of a variety of materials such as cotton, rayon, nylon or other synthetic materials and combinations of any of these materials to produce a continuous cord without knots in the form of a package that is precision wound, of high density and of a better and more uniform lay of ply than has been heretofore achieved.

The improvements in ply-twisting machines, constituting the instant invention and as will be more specifically described hereinafter, also facilitate the economic production of a wide denier range of finished cord which may run from 2,000 to 50,000 denier.

In general, the machine of this invention includes a twister spindle of tubular construction having a flyer disclosed adjacent one end thereof and a tension control device in the form of a double-step eccentric strand wrap-around storage surface associated therewith. A plurality of strands or ends of yarn supplied from a magazine creel are fed axially into the tubular spindle shaft, which emerge through an orifice in the spindle wall adjacent the strand storage surface. The combined strands make contact with the surface of the yarn storage device in wrap-around fashion and then proceed along the surface of the flyer and, when the spindle shaft is rotated, form a balloon about a support floatingly carried by the end of the spindle shaft. This balloon extends between the flyer and an apex guide mounted on the floating frame support and positioned on the extended axis of the spindle.

The combined strands or cord, in which a double twist is being inserted for each revolution of the spindle, are drawn inwardly from the apex guide by a capstan arrangement carried by the end of the floating frame and driven at constant speed from the rotating spindle. The floating frame also carries a takeup package support in the form

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of a rotatable mandrel or sleeve mounted with its axis of rotation parallel to the axis of rotation of the spindle and slidably receives a winding core. The twisted strands or cord are advanced from the capstan through a precision wind traverse mechanism onto the winding core. The traverse mechanism is driven in timed relation with respect to the winding core and arranged to produce a headless package having a close precision wind. The winding core is driven from the spindle through a slip coupling which delivers a constant torque so that, as the winding builds up on the form, the tension in the twisted strands or cord is gradually decreased. The close precision wind together with the controlled winding tension produces a much higher density and better shaped package than those produced on prior art machines of this type.

The instant invention more specifically provides an improved drive for the rotatable strand package support and the strand traverse mechanism which is simpler in design, of compact form but yet of more rugged construction than prior devices of this kind.

The details of my improved combined ply-twisting and winding machine whereby the foregoing advantages are achieved will now be described in connection with the accompanying drawings in which:

Figure 1 is a perspective view of the strand handling machine of my invention with certain parts thereof broken away to bring other parts into view;

Figure 2 is a fragmentary side elevation view partially in section of the machine of my invention;

Figure 3 is a sectional view of the machine of my invention taken along line III—III of Figure 2;

Figure 4 is a schematic perspective view showing the relationship of the several driving and driven elements of the machine;

Figure 5 is a sectional view of a portion of the main drive gearing taken along the line V—V of Figure 3;

Figure 6 is a plan view of the gear assembly constituting a part of the drive for the rotary package support and the strand traverse mechanism as viewed from the gainer gear assembly side;

Figure 7 is an extended sectional view of the gear assembly of Figure 6, with parts omitted for clarity, taken along line VII—VII of that figure;

Figure 8 is a fragmentary sectional view taken along line VIII—VIII of Figure 6 showing details of the driving clutch tension adjusting means;

Figure 9 is a fragmentary sectional view of the strand traverse mechanism taken along line IX—IX of Figure 2;

Figure 10 is a fragmentary sectional view of the traverse mechanism taken along the line X—X of Figure 9;

Figure 11 is a fragmentary side elevation view, partially in section, showing an alternate strand traverse arrangement for the machine of my invention; and

Figure 12 is a sectional view of the alternate form of my invention shown in Figure 11 and taken along line XII—XII of that figure.

In Figure 1 of the drawings the numeral 15 designates a frame for supporting the combined plying, twisting and winding machine having a support platform 17 on which is mounted in spaced relationship a pair of pillow blocks 19. Journaled within the pillow blocks 19 by means of suitable low friction bearings is a tubular spindle shaft 21 as more clearly shown in Figure 2.

A motor 23 mounted on the frame 10 is coupled in driving relation through a pair of V belts 25 to a pair of split pulleys 27 adjustably mounted on the shaft 21 between the pillow blocks 19. The motor 23 is preferably of a constant speed type and the desired spindle shaft speed is obtained by adjusting the spacing of the split pulleys 27 as is well known in the art.

Disposed intermediate the ends of the spindle 21 is a flyer 29 carried by a hub 31. The flyer hub 31 as shown in Figure 2 has two merging cylindrical surfaces 33 and 35 that are respectively eccentrically disposed with respect to the axis of the spindle shaft 21 so as to provide a two-step wrap-around cord storage device for controlling the tension in the cord balloon in a manner as more fully covered in my co-pending applications Serial Number 244,812 filed September 1, 1951, now Patent No. 2,811,013 and Serial Number 417,619, filed March 22, 1954. The spindle shaft 21 is provided with an axial passage 37 extending inwardly from the left-hand end thereof as viewed in Figure 2 and communicates with an outwardly flared orifice 39 in the wall of the spindle shaft at the surface 33.

The right-hand end of the spindle shaft 21 as viewed in Figure 2 and which is disposed within the cup-shaped flyer 29, has a sleeve 41 mounted thereon by means of low friction bearings 43 and 45. The sleeve 41 has a three-lobed flange 47 thereon disposed intermediate its ends and to which is secured an annular support plate 49 which forms the inner end support for a floating support frame generally indicated at 51. The support plate 49 is preferably attached to each lobe of the flange 47 by resilient mounting means generally indicated at 53 of conventional construction. The support frame 51 includes an outer plate 55 generally of C shape as more clearly shown in Figure 3. The frame 51 also includes a second C-shaped support plate 57 having a lateral offset extension 58, disposed intermediate the plates 49 and 55. The plates 57 and 58 are held in fixed spaced relationship with respect to the plate 49 by a plurality of tie rods 59, two of which are shown in Figure 2. The outer plate 55 is supported with respect to the inner plate 49 by means of a channel-shaped member 61 joining these two plates and a hollow casing member 63 extending between the plates 55 and 58.

An offset bracket 65 suitably mounted on the inner plate 49 serves as a fixed mount for an axle 67 extending in a direction generally parallel to the axis of rotation of the spindle 21. Axle 67 carries a pair of low friction bearings 69 and 71 on which a cylindrical rotatable mandrel or sleeve 73 is mounted. The left-hand end of mandrel 73 as viewed in Figure 2 carries in fixed relation thereto a drive gear 75 whereas the right-hand end of the mandrel is provided with a flanged annular cap 77. The cap 77 is attached to the mandrel 73 by means of screw threads so as to be readily removable and replaceable upon the end of the mandrel. When the cap 77 is removed, the mandrel 73 is adapted to slidably receive a winding core 79 which upon replacement of the cap 77 is held in clamping engagement between the flange of the cap and the hub of the drive gear 75 so as to prevent its turning with respect to the mandrel during a winding operation as will be more fully explained hereinafter.

The mandrel 73 may be held against rotation during the removal and replacement of the flange cap 77 by a suitable locking device in the form of a bell crank member 81 mounted for rocking movement about a bolt 83 passing through the upper portion of the offset bracket 65. The horizontal arm portion of the member 81 as viewed in Figure 2 terminates in a block portion 85 having a plurality of downwardly projecting teeth 87. A coil tension spring 89, shown in Figure 3, connected between the horizontal arm portion of the member 81 and the back plate 49 normally maintains teeth 87 out of registration with the teeth of the drive gear 75 secured to the mandrel 73. A handle 91 projecting from the upper end of the member 81 provides a ready means for rocking the member about its pivot bolt 83 in a clockwise direction as viewed in Figure 2 to bring the teeth 87 into engagement with the teeth of the gear 75 thereby locking the gear and consequently the mandrel 73 against rotation.

The mandrel 73 supporting the winding core 79 is adapted to be driven from the spindle shaft 21 through a

gear train and slip coupling mounted on the back support plate 49 of the floating frame 51 and disposed between the plates 49 and 57 generally below the axis of rotation of the spindle shaft 21 as more clearly shown in Figures 2 and 3. The coaxing relationship between the several gears comprising this gear train is more clearly illustrated in Figure 4. The spindle shaft 21 has fixed thereto a pinion gear 93 which meshes with a larger gear 95 fixed on a shaft 97 mounted on the back support plate 49 in a manner which will be more fully described hereinafter. The shaft 97 carries a twist gear 99 which in turn meshes with a gear 101 which is one of three coaxially disposed gears forming a slip coupling gainer gear assembly generally indicated at 103. A second gear 105 of the gear assembly 103 is arranged to be driven through a slip clutch from the gear 101 whereas the final gear 107 of the assembly 103 constitutes a gainer gear driven through mechanism from gear 105 all of which will be more fully described hereinafter. Gear 105 is disposed in driving relation with gear 75 fixed to the rotatable mandrel 73.

The gear 101 also serves as the input drive gear for a second gear train for driving the mechanism for advancing the twisted cord into the machine. As shown in Figure 4, a capstan drive gear 109 mounted on a stub shaft 111 meshes with gear 101. Fixed to the capstan gear 109 is a smaller gear 113 also adapted to rotate about shaft 111. The gear 113 drives a counter shaft 117 through engagement with a gear 115 on the end of the shaft. The counter shaft 115 is journaled at each end respectively in plates 55 and 57 of the floating frame 51. The outer end of shaft 117 carries a gear 119 in mesh with a gear 121 journaled for rotation on a shaft 123 carried by the outer support plate 55. Also journaled about the shaft 123 and secured in fixed relation to the gear 121 is a capstan 125. An idler gear 127 is disposed for rotation on a stub shaft 129 extending from the support plate 55 and engages in meshing relationship the gear 121 and in turn drives a gear 131 fixedly associated with a second capstan 135 journaled upon a stub shaft 133 also carried by the outer plate 55. It will thus be seen that rotation of the counter shaft 117 will rotate both capstan members 125 and 135 in the same direction. The diameters of the gears 121 and 131 are equal as well as the diameters of the capstan members 125 and 135 so that a number of wraps of cord may be placed about the capstan members and the cord advanced by the rotation thereof.

The combined yarn ends being twisted into cord are directed from the balloon area through an apex guide array 137 hingedly mounted on the end plate 55 of the floating frame 51 as more clearly illustrated in Figures 1 and 2. This guide array includes a trumpet guide 139 rotatably mounted on one end of a tubular support 141. The tubular support 141 is arranged to fit telescopically within a tubular member 142 the left-hand end of which as viewed in Figure 2 terminates in a diagonally cutoff portion 143. The diagonal portion 143 carries a stub shaft 147 on which is rotatably mounted a guide pulley 145. The tubular member 142 extends through and is supported by a collar 148 which is carried on the ends of three support arms 151 diverging with respect to the collar and the support member. The other ends of the support arms 151 are rigidly secured in equally spaced circumferential relation to a support ring 153. A pair of hinge arms 155 rigidly mounted to the support ring 153 coact with a hinge block 157 which in turn may be bolted or otherwise secured to the end plate 55 thereby providing a hinged mount for the apex guide array. In order to provide a quick release latch means for holding the apex guide array 137 in operative position, two pivotally mounted spring bias latch levers 159 only one of which is shown in Figure 2, are mounted on respective latch support arms 161, fixed to and extending in diverging relationship from

the support member 142. Each of the latch levers 159 have a latch portion 163 positioned to be pressed into engagement with the back side of the front support plate 155.

The apex guide array 137 thus provides means for directing the cord being twisted from the cord balloon axially inwardly toward the cord take-up package. The cord undergoing twist and delivered through the apex guide 138 and the tubular support 141 is directed over the guide pulley 145, through a pigtail guide 165, shown in Figure 3, carried by the front plate 55 and then wrapped about the two capstan members 125 and 135. It will be noted that stub shaft 133 carrying the capstan member 135 is slightly skewed out of parallelism with respect to the shaft 123 carrying the capstan member 125 as more clearly shown in Figures 3 and 4 to provide a slight misalignment of the respective capstan surfaces thereby causing a separation of the several reaches of the cord passing between and about the capstans 125 and 135. In order to accommodate the misalignment of the gears 127 and 131 the teeth on gear 131 are cut at an angle with respect to its axis of rotation so that it may properly mesh with the idler gear 127. Since the spindle shaft 21 is driven at a substantially constant rate and capstan members are positively driven through a gear train from the spindle shaft, the cord will be advanced by the capstan members at a substantially constant rate.

From the foregoing description it is apparent that for a given spindle speed the degree of twist imparted to the cord will depend upon the rate at which the cord is advanced by the capstan members 125 and 135. Thus the speed ratio of the gear train between the spindle shaft 21 and the capstan gear 121 will determine the amount of twist being imparted to the cord. Changes in twist can thus be achieved by making appropriate changes in the gear ratio. In practice I have found it desirable to obtain these changes by the interchange of only one gear of the train and prefer gear 99 as the change gear on account of its accessibility. To avoid making changes in the diameter of the mating gear 101 with changes in size of the twist gear 99, I prefer to mount the gears 99 and 101 in the manner more clearly shown in Figures 3, 4 and 5.

As more particularly shown in Figure 5 the stub shaft 97, carrying the reducing gear 95 and the change gear 99, is journaled in a cylindrical bearing housing 167 having a laterally extending flange 169 thereabout in the form of a support plate. The housing 167 projects through an arcuate opening 171 in the back plate 49 and is adjustably held in the desired position by means of two elongated brackets 173 of L-shaped cross sections which respectively engage a marginal portion of the flange 169 on opposite sides of the housing. Suitable fastening means, such as captive cap screws 175, passing through the back plate 49 and making threaded engagement with the brackets 173 serve to clamp the flange into contact with the back plate.

The center of curvature of the arcuate opening 171 coincides with the axis of rotation of the driving pinion 93 on the spindle 21 so that the housing 167 carrying shaft 97 may be swung about such axis within the limits determined by the length of the opening 171, while maintaining the pinion 93 in driving engagement with reducing gear 95. The position of this swing path is more clearly designated in Figure 4 by the broken arcuate line 177. The swing path 177 is in a direction that permits positioning of the shaft 97 at different distances from the gear 101. Consequently, twist gears 99 of different diameter may be interchangeably mounted on the shaft 97 and brought into meshing engagement with gear 101 by adjusting the position of the housing 167 in which the shaft 97 is journaled. Once the proper adjustment for a given size gear 99 is made, the housing

is securely held in position by tightening the clamping screws 175.

The gear arrangement shown schematically in full line position in Figure 4 applies to a rotation of the spindle shaft 21 and the flyer 29 in a direction to produce a Z twist in the cord. Cord having an S twist is also readily produced with the gear train elements herein shown by making two minor adjustments of gear positions and driving the spindle shaft 21 in the opposite direction. Since it is advantageous to arrange the cord advancing capstan and winding mechanism for uni-directional operation, a directional change in the gear train ahead of the gear 101 must be made when changing from Z to S twist or vice-versa. To provide the requisite condition for a change from Z to S twist the clamping screws 173 holding the bearing housing 167 in position are loosened so that the housing supporting the gears 95 and 99 may be swung to the right as viewed in Figure 4 until gear 99 is out of meshing engagement with gear 101. As more clearly shown in Figure 5, the right-hand end portion of stub shaft 111 is of reduced diameter and projects through an arcuate slot 181 in the front support plate 57. The end portion 179 is threaded to receive a clamping nut and washer 183 so as to permit the shaft 111 carrying the gears 113 and 109 to be moved within the limits of the arcuate slot 181. The arcuate slot 181 as more clearly shown in Figure 3 and Figure 4 is disposed with its center of curvature coincident with that of the axis of rotation of gear 101. It will thus be seen that gear 109 will always remain in mesh with gear 101 for all positions of the stub shaft 111 in the slot 181. Stub shaft 111 can thus be adjusted with respect to the arcuate slot 181 until gear 109 engages the change gear 99. Gear 109 thus becomes an idler between gear 99 and gear 101 thereby accomplishing the desired gear change to accommodate the change in direction of the spindle. Movement of the stub shaft 111 will of course impair the proper meshing relation between the gears 113 and 115. To correct this condition I provide a self-aligning bearing 185 for the counter shaft 117 adjustably mounted on plate 57 by cap screws 187 as shown. By adjusting the position of the bearing 185, gear 115 is again brought into proper meshing relation with the gear 113 with but a misalignment between the shafts 111 and 117 so slight that good operating conditions are maintained.

The twisted cord advanced by the capstan drive members 125 and 135 is directed by suitable guide means to be hereinafter more specifically described to a traverse mechanism generally indicated at 189 in Figure 4. The traverse mechanism 189 serves the purpose of producing a lay of cord on the winding core 79 of predetermined pattern as will be more fully described hereinafter. The several components comprising the traverse mechanism are more clearly shown in Figures 2, 9 and 10. The numeral 191 designates a traverse arm having a guide eye 193 at its upper end and through which the cord to be laid on the winding core 79 passes. The lower end of the traverse arm 191 as viewed in Figure 2 terminates in a laterally flared portion 195 and is hinged about a pin 197 carried by an elongated bracket 199 of angular cross section. As more clearly shown in Figure 9 the bracket 199 is securely attached to a sleeve 201. The sleeve 201 is mounted on a pair of ball bushings 203 slidably disposed about a guide rod 205 the ends of which are respectively supported by the plate 55 and a bracket 206 extending from the plate 58. The sleeve 201 also is provided intermediate its ends with a downwardly projecting boss 207 having an axial bore 209 therein for receiving the cylindrical shank 211 of a crescent shaped cam follower 213. The cam follower 213 is adapted to ride in a continuous reversing cam groove 215 provided in a barrel cam 217. Suitable means are provided for rotatably mounting the barrel cam 217 as for example shaft ends 219 and 221 which are respectively journaled

in end plates 58 and 55. It will thus be seen that as cam 217 is rotated causing the cam follower 213 to be driven throughout the length of the cam groove 215 the sleeve 201 and the traverse arm 191 will sweep back and forth along the guide rod 205.

The cam 217 is arranged to be driven in timed relation with respect to the winding core 79 and to this end a gear 223 is disposed on the shaft end 219 of the barrel cam 217 and meshes with the gainer gear 107 of the gainer gear drive assembly 103. A gainer gear drive for the cam 217 is utilized to provide a small continuous rate of advance of the cam and consequently the traverse arm 191 so as to obtain an accurate processional displacement of adjacent turns of the cord on the package. This is of particular importance in the building of a precision wind headless package wherein high density is desired. In this particular instance I prefer to use a rapid traverse producing approximately two turns per length of package or what is known in the trade as a "two wind." Thus unless the rate of traverse is increased by a given increment during each excursion of the traverse arm 191, the cord will hill or pile up. The size of the increment will, of course, depend upon the size of cord and the closeness of the wind desired. If a close wind of maximum density is desired, the small increase in the rate of traverse will be just enough to constantly advance the traverse arm by an amount slightly greater than the diameter of the cord being wound on the winding core.

More specifically details of the mandrel drive and gainer gear assembly 103 are shown in Figures 6, 7 and 8. As illustrated in Figure 7 the assembly is supported upon an offset bracket 225 mounted upon the back plate 49. The bracket 225 carries a stub axle 227 having a low friction bearing 229 on which the drive gear 101 is mounted. Gear 105 of the assembly is provided with a relatively thick web or body portion 231 having a central sleeve like hub 233. The hub 233 is mounted for limited axially slidable movement upon a bearing 235 carried by a shoulder bolt 237 threaded into axle 227 of bracket 225.

Gear 105 is adapted to be driven by gear 101 through a slip clutch arrangement wherein the right-hand face of gear 101 as viewed in Figure 7 constitutes the driving surface. A driven clutch plate 239 is secured to the left-hand face of the body portion 231 of gear 105 by means of a plurality of screws 241 only one of which is shown. The clutch plate 239 is preferably provided with a facing or lining 243 secured thereto as by rivets or any other conventional manner. Gear 105 is resiliently biased toward gear 101 by a compound annular metal disk spring 245 surrounding the shaft 227 and disposed within a cylindrical cavity 247 provided in the body portion 231 of the gear 105 so as to bring the clutch facing 243 into driving relation with the driving face of gear 101. Tension of the spring 245 and consequently the torque delivered by the slip clutch is controlled by an annular tension adjusting ring 249 having its outer circumferential surface in threading engagement with the cylindrical surface of the cavity 247. It will thus be seen that by rotating the annular ring 249 in the appropriate direction the tension of spring 245 may be either increased or decreased. In order to provide readily accessible means for rotating the tension adjusting ring 249 the outer circumferential surface is also provided with radially cut spur gear teeth as shown at 253. A bore 255 is provided through the body portion 231 of the gear 105 as more clearly shown in Figure 8 in registration with the outer circumference of the ring 249 and of diameter equal to that of a pinion 257 for rotatably receiving such pinion which is adapted to mesh with the teeth 253. The pinion 257 is preferably provided with a flange 259 at one end thereof which flange coacts with a flanged collar 263 in turn secured to the body portion 231 of the gear 105 by means of clamping screws 265

as shown in Figure 6. A spring washer 261 disposed between the flange 259 and the body portion 231 of the gear 105 provides frictional resistance to prevent the free rotation of the pinion 257. A cap screw 267 firmly threaded into the end of the pinion 257 having a socket 269 therein serves as a means for the insertion of an appropriate tool for rotating the pinion. It will thus be seen that by rotating the pinion through the use of such tool the annular adjusting ring will be advanced axially of the shaft 237 so as to increase or decrease the tension of the spring 245 as is required to cause the slip clutch to deliver the desired amount of torque. The manner in which the slip coupling functions to control the tension of the cord being wound on the winding form will be more fully explained hereinafter.

The gainer gear 107 which drives the barrel cam 217 through gear 223 is operably connected to and driven by the gear 105 through a planetary gear cluster as more clearly shown in Figures 6 and 7. The gear teeth of gear 105 are offset to the right as viewed in Figure 7 so as to provide a circumferential channel between these gear teeth and the clutch plate 239. This channel provides a bearing about which the gainer gear 107 may rotate. In this instance the gainer gear is a composite internal-external gear annular in form. The planetary gear train through which the gainer gear 107 is driven in succession comprises the following: a pinion sun gear 271 carried on the end of a short stub shaft which may either be keyed or otherwise secured against rotation within an axial bore within the shoulder bolt 237. The pinion sun gear 271 meshes with a reducing gear 273 fixed to a smaller gear 275 the latter meshing with an idler gear 277 in turn meshing with a reducing gear 279 having a smaller gear 281 fixed thereto. The smaller gear 281 meshes with an idler 283 in turn meshing with gear 285 carried by a shaft 287 bearing in the body portion 231 of the gear 105. The left-hand end of the shaft 287 as viewed in Figure 7 carries a pinion 289 which meshes with the internal teeth of the gainer gear 107. It will thus be seen that as gear 105 is rotated the planetary gear cluster will be rotated about the sun gear 271 which is held stationary. The planetary gear cluster, however, being in mesh with the stationary pinion 271 will have imparted thereto rotation about their own axes. The ratio of the planetary gearing is so selected that, for each revolution of the gear 105 the gainer gear 107 will be advanced by a slight increment. This increment of advance of gear 107 in respect to the gear 105 is transmitted through the cam gear 223 which drives the barrel cam 217. The amount of advance desired will of course depend upon the size of the cord being wound on the package and the closeness of the wind desired. It will also be apparent that the degree of advance desired can be obtained by a selection of the proper gear ratio of the planetary gear cluster used to advance the gainer gear 107. By making appropriate gear changes the relative rotation of the gainer gear 107 with respect to the gear 105 may be obtained. In practise I have found it desirable to make changes in the gear ratio by using gears 271 of different diameter. To avoid making changes in the diameter of the mating reducing gear 273 the latter gear together with gears 275 and 277 are mounted on a swing arm 291 disposed for swinging movement about a stub shaft 293 on which gears 279 and 281 are journaled. A stub shaft 295 mounted on the swing arm 291 carries the gears 273 and 275 whereas a stub shaft 297 secured to the swing arm carries a single gear 277. It will thus be seen that all gears in the planetary train will remain in mesh as the swing arm 291 is moved to different positions except gears 271 and 273. Consequently swing arm 291 may be swung either toward or away from the pinion gear 271 so as to accommodate the meshing engagement of the reducing gear 273 with sun gear pinion 271 of different size. Once the swing arm 291 is adjusted for a given size sun pinion gear 271 it may be held in clamped

position against the body 231 of gear 105 by a clamping plate 299 releasably secured to the body portion 231 by screws 301.

In order to wind a precision package within the limited space of the balloon generatrix, special consideration must be given to the cord guide means leading from the capstan drive members 125 and 135 to the traverse mechanism 189. Movement of the traverse arm 191 back and forth lengthwise of the package constitutes a movement that is alternately toward and away from the capstan drive members which advance the cord at a constant speed. Consequently the tension in the cord will oppose traverse movement away from the capstan drive members and aid such movement in a direction toward the capstan drive members. In fact the length of the cord path would be changed at each traverse movement. These difficulties are overcome through the use of a compensator actuated in accordance with traverse movement of the arm 191 to provide a cord path of constant length between the capstan drive and the traverse arm for all positions of the traverse arm.

With reference to Figures 2 and 3 the cord leaving the capstan member 135 is directed about a captive guide pulley 303 supported from end plate 55 and directed inwardly about a movable guide pulley 305 carried by a movable rider member 307 slidably disposed about a guide bar 309 the ends of which are respectively attached to end plates 55 and 57. A coil spring encircling the guide bar 309 and having its left-hand end attached to the end plate 57 and its right-hand end secured to the rider 307 urges the rider to the left as viewed in Figure 2. From the movable guide pulley 305 the cord is advanced about a fixed guide pulley 313 adjacent the pulley 303 also mounted on end plate 55. From the pulley 313 the cord is successively directed over fixed pulleys 315 and 317 to the linear path compensator as defined by traveling pulleys 319, 321 and 323 about which the cord passes in succession in its path to the guide eye 193 of the traverse arm 191.

The structural arrangement of the compensator is best illustrated in Figures 4, 9 and 10. As shown in Figure 9 two parallel rack bars 325 are mounted in spaced relation with respect to each other on the inner wall of the hollow casing member 63 facing the flat side of the movable elongated angular bracket 199. The flat side of the bracket 199 is provided with a rack 327 of substantially the same length of the rack bars 325. The pinion 329 is disposed between the rack bars 325 and the rack 327 and is of such diameter so that it operatively engages the teeth of the rack bars and the rack simultaneously on opposite sides thereof. A bolt 331 axially disposed through the pinion 329 provides a suitable mount for a retaining washer 333, a spacing sleeve 335 and the movable pulley 319. An elongated slot-like opening 337 in the casing 63 of width sufficient to accommodate the sleeve 335 permits traversing movement of the pinion 329 in a path parallel to the rack bars 325 and the rack 327. The retaining washer 333 is preferably of diameter large enough to overhang the teeth of the upper rack bars 325 and the rack 327 so as to prevent downward movement of the pinion out of alignment with the rack bars 325 and the rack 327. The transverse portion of casing member 63 in which the slot 337 is located serves as a bearing surface for the retaining washer 333 to prevent upward displacement of the pinion 329 during its operation. A strip 338 of felt or other porous material suitable for retaining a lubricant is disposed between the rack bars 325 and which may be saturated from time to time with a lubricant to lubricate the coacting rack and pinion teeth.

Movable guide pulley 321 is carried by the movable bracket 199 which supports the traverse arm 191 and is disposed adjacent the right-hand end of the bracket as viewed in Figure 2. Consequently the pulley 321 is always positioned a fixed distance from the guide pulley

323 carried by the flared portion 195 of the arm 191. During the traversing movement of the arm 191 pulley 321 will move alternately toward and away from the fixed guide pulley 317. The cord being advanced to the traverse arm 191, however, is directed from pulley 317 to pulley 319 and then back to pulley 321. Since pulley 319 is carried by the pinion 329 operatively engaged by the rack bars 325 and the rack 327, a movement of the rack 327 and pulley 321 a given distance to the right for example, will cause the pinion 329 and the pulley 319 carried thereby to be moved to the right only half the given distance. Also a similar proportionate movement of the pulley 319 in respect to movement of the pulley 321 in the opposite direction is obtained. Consequently the linear distance from the fixed guide pulley 317 to the movable pulley 321 by way of movable pulley 319 is of constant length throughout the traverse of the arm 191.

With particular reference to Figure 9 it will also be noted that the arrangement and proportioning of the several elements of the linear path compensator such as the rack and pinion structure coacting with the movable bracket 199 and the sleeve 201 are such as to maintain the sleeve 201 properly oriented angularly about the guide shaft 205 to provide an operative driving relation between the barrel cam 217 and follower 213.

The movable guide pulley 305 biased to the left as viewed in Figure 2 by the spring 311 provides the required resilience in the linear path of the cord to the traverse arm and particularly compensates for any irregularities in the wind on the package occurring at the end of the traverse stroke.

The operation of my combined ply twisting and winding machine in connection with the production of a precision wound cord package will now be described.

In threading the machine preparatory to plying, twisting and winding a plurality of yarn ends Y as more particularly shown in Figure 1 are withdrawn from individual cones or packages 339 carried by a pin type magazine creel generally indicated at 341. The separate yarn ends Y are drawn through respective eyelet guides 343 that are mounted on an upright support 345. As the individual yarn ends Y emerge from the eyelet guides 343, each end is directed through a suitable tension device 347 which may be of the twin disk type and carried by the upright support 345. From the individual tension devices 347 the yarn ends Y are directed through a roller guide 349 from which they proceed in generally parallel relation to a multiple eyelet separator guide 351 mounted on a bracket 353 carried by the platform 17 of the twister frame. From the eyelet guide 351 the yarn ends are directed to a compression trumpet 355 which also for convenience may be mounted on the bracket 353.

Entry of the combined yarn ends Y into the spindle is best visualized with reference to Figure 2. With the aid of a flexible wire threading tool the combined yarn ends Y are drawn through the bore 37 in the tubular spindle shaft 21 and brought out through the orifice 39 onto yarn storage device surface 33. A sufficient length of the yarn is then drawn forward to pass the group of ends about the flyer 29 and forward to the yarn apex guide 139. At this point a threading tool is again used to bring the combined ends through the trumpet guide 139, through the bore in support tube 141 and about the guide pulley 145 carried at the end of the support tube 142. From the pulley 145 the combined yarn ends are drawn forward, through a pigtail guide 165 (see Figure 3) carried by the outside end plate 55 and are wound about the capstan members 125 and 135. I prefer to make five or six passes about the double capstan members to insure a positive drive of the cord through the machine. From the capstan members 125 and 135 the group of yarn ends are successively threaded about the guide pulleys 303, 305, 313, 315, 317, 319, 321 and 323. From the pulley 323 the yarn ends are threaded through the eyelet guide 193 in the traverse arm 191 and then onto a winding core 79

which has been placed on the sleeve 73. The combined yarn ends are then secured about the winding core 79 in such manner that when the sleeve and core are driven, normal cord pickup will occur.

The electric driving motor 23 for the spindle may be placed into operation and controlled by means of either of two start-stop switches 357 and 359 located at different positions on the frame as more clearly shown in Figure 1. By actuating either switch 357 or 359 the spindle is placed into operation and continues to operate at a constant speed causing the combined yarn ends Y to balloon about the floatingly supported frame 51 and be drawn through the machine by the capstan members 125 and 135. In order to protect the operator from the rotating cord in the balloon a cage-like guard 361 supported by brackets 363 and 365 mounted on the frame 15 are provided as more clearly shown in Figure 1.

As the individual plies or strands of yarn Y are advanced beyond the compression trumpet 355 through the axial bore 37 and the orifice 39 of the spindle shaft 21, it is both plied and twisted to form cords by the rotative action of a flyer 29 throwing the cord in balloon-like manner about the floating frame 51. It is to be understood that yarn ends Y route through the spindle and apex guide in such manner as to produce two complete twists in said yarns for each revolution of the spindle and flyer. The rate of advance of the cord into the machine is maintained substantially constant being controlled by the capstan drive members 125 and 135 which are positively driven from the spindle shaft 21. Tension within the balloon is automatically controlled to maintain the desired balloon dimensions by action of the cord on the wrap-around storage device surfaces 33 and 35 in a manner more fully disclosed in my copending application Serial Number 417,619, filed March 22, 1954.

The uniformly twisted and plied cord is evenly and precisely laid on the winding core as a result of the precision drive for the traverse arm 191 incorporating the herein described gainer gear mechanism. Good package shape without end bulges or surface distortion is achieved by winding the initial turns on the winding core under high tension and gradually decreasing the winding tension as the package increases. The gradual decrease in winding tension is achieved by the use of a constant torque slip clutch coupling between the gears 101 and 105 of a construction above described. The combined twisting and winding operation is started with an initial clutch tension setting to cause the gear 101 to transmit to the mandrel driving gear 105 sufficient torque to obtain the requisite high tension on the cord forming the initial layers of the package. As additional layers are added and the package grows in diameter the tension on the cord automatically decreases since the driving torque to the winding core mandrel remains constant. This gradual decrease in winding tension creates a firmer package than would be possible if a constant tension had been employed throughout.

Package shape may also be improved by the use of a pair of presser rolls 367 and 369 arranged as more clearly shown in Figures 2 and 3 to apply pressure adjacent the outer marginal edges of the package. The rolls 367 and 369 preferably have a surface layer of rubber or other suitable resilient material and are journaled in spaced axial relationship on a shaft 371. The shaft 371 is preferably attached intermediate the rolls 367 and 369 to the free end of a swing arm 373 the other end of which is attached to a pivotally mounted shaft 375 having its ends respectively journaled in support plates 55 and 57. Helical springs 377 disposed about the shaft 373 and having their respective ends anchored to the support plates 55 and 57 and to the shaft 375 in conventional manner bias the presser rolls into contact with the package. It will thus be seen that as the package is formed presser rolls 367 and 369 follow the contour of the package and assist in the formation of a more uniform surface.

In Figures 11 and 12 an alternate cam arrangement is shown for actuating the winding traverse arm 191 and the linear cord path compensator leading to the traverse arm. In the following description like parts previously described in connection with Figures 1 to 10 will be given the same numerals of reference. With reference to Figure 11 the traverse arm 191 is hinged mounted at its lower end to a traverse block 379 slidably mounted for traversing movement on a rectangular guide rod 381 supported at its ends from support plates 55 and 57 respectively. The traverse block 379 has a downwardly projecting boss 383 to which is journaled a cam follower roller 385. The cam roller 385 is adapted to coact with the right-hand edge of an annular spiral cam 387 surrounding the floating support frame 51. The spiral cam 387 is secured to the inner surface of a hollow cup-shaped support 389 having a hub portion 391 journaled by means of a low friction bearing 393 for rotation about the sleeve 41.

A second traverse block 395 is slidably mounted upon guide rod 381 and has a depending boss 397 to which is rotatably mounted a cam follower roller 399. This latter roller is adapted to coact with the left-hand cam surface of the spiral cam 387. The left-hand cam surface has substantially one-half the pitch of the right-hand cam surface coacting with the cam roller 385. The traverse blocks 379 and 395 are provided respectively with cord guide pulleys 401 and 403 and in conjunction with fixed pulley 317 define the cord approach path to the traverse arm 191. From the stationary pulley 317 the cord being processed by the machine is directed about pulley 403 and then about pulley 401 from whence it is threaded through the guide eye 193 of the traverse arm 191 and then directed to the winding core 79. It will be noted that the normal tension on the cord will urge the cam follower rollers 385 and 399 toward their respective coacting cam surfaces. In order to provide more uniform contact of these cam rollers with the cam surfaces, I provide a spring means 405 one end of which is attached to the plate 55 and the other end to one end of a chain 407 operative about a sprocket 409 carried by the traverse block 395 and having its other end anchored to a projection 411 extending from the traverse block 379.

From the foregoing description it will be apparent that as the cam 387 is rotated cam rollers 385 and 399 will follow their respective cam surfaces causing traverse blocks 379 and 395 to be actuated back and forth along guide rod 381. Traverse arm 191 and its associated guide pulley 401 will make a full traverse defining the desired package length whereas traverse block 395 carrying the guide pulley 403 will traverse but half the distance of the pulley 401. In consequence the linear cord path from the stationary guide pulley 317 about pulley 403 to pulley 401 always remains constant in length during traverse of the arm 191.

The cam 387 is adapted to be driven in timed relation with respect to the mandrel 73 from the gainer gear assembly previously described to insure the production of a precision wound package of high density. A spur gear 413 in mesh with the gainer gear 107 is carried by a shaft 415 journaled in a bearing housing 417 piercing the support plate 49 and having a support flange 419 thereon which may be bolted to the support plate 49 as shown. The shaft 415 also carries a second gear 421 which makes driving engagement with a gear 423 disposed about the bearing 393 and fixed to the hub 391 of the rotatable support 389.

The performance of the traverse mechanism and linear cord path compensator as shown in Figures 11 and 12 is comparable to that obtained with the arrangement described in connection with Figures 1 to 10. It is to be noted however that in some respect the alternate form as illustrated in Figures 11 and 12 is simpler in design and less costly to construct.

While I have shown and described a specific machine for plying, twisting and winding a plurality of strands of flexible fibrous materials, it is to be understood that the same is for the purpose of illustration and not by way of limitation and that numerous changes and modifications can be made by those skilled in the art without departing from the spirit and scopes of the appended claims.

I claim:

1. In a machine for twisting and winding a flexible strand, a rotatable spindle, a rotatable winding core support carried by said spindle, a winding core slidably mounted on said support, means for advancing a strand into the machine at a rate in direct proportion to the rate of rotation of said spindle, means for directing said strand to said winding core including a traverse mechanism driven in timed relation with respect to said rotatable support, and means driving said rotatable support from said spindle through an adjustable constant torque coupling for controlling the tension on the strand being wound on said core, said adjustable constant torque coupling comprising a slip clutch having driving and driven members urged into coactive engagement with each other through resilient means, and means for adjusting the operative tension of the resilient means for controlling the torque transmitted by the driving and driven members.

2. In a machine for twisting and winding a flexible strand, a rotatable spindle, a rotatable winding core support carried by said spindle, a winding core slidably mounted on said support, means for advancing a strand into the machine at a rate in direct proportion to the rate of rotation of said spindle, means for directing said strand to said winding core including a traverse mechanism driven in timed relation with respect to said rotatable support, and a gear train interposed between the spindle and the rotatable support for driving said rotatable support from said spindle, said gear train including an adjustable slip coupling adapted to deliver a constant torque whereby the strand is wound on said core with decreasing tension as the strand package increases in diameter.

3. In the machine of claim 2 wherein the slip coupling is disposed between two coaxially positioned gears of the gear train, the face of one of said gears constituting a driving surface, a coating driven surface carried by the adjacent face of the other of said gears, and spring means for urging said coating surfaces into engagement with each other.

4. In the machine of claim 3 wherein one of said two gears is rotatably mounted on a shaft and held against axial displacement along the shaft and the other of said two gears is rotatably mounted on the same shaft and adapted for axial displacement between predetermined limits along the shaft, and spring means disposed in surrounding relation to the shaft and having one of its ends fixed with respect thereto and the other end engaging said axially displaceable gear to urge said two gears toward each other and bring the driving and driven surfaces thereof into driving engagement.

5. In the machine of claim 4 wherein engagement of one end of said spring means with said axially displaceable gear is made through an annular member having peripheral threadwise engagement with the threaded cylindrical surface of a coaxial cylindrical cavity in the web of said gear, and means for rotating said annular member to cause it to be displaced axially within said cavity and thereby adjust the position of engagement of the said spring means with respect to said gear and the operative tension of said spring means.

6. In the machine of claim 5 wherein the peripheral threaded portions of the annular member have gear teeth thereon and a pinion rotatably mounted in the web portion of the axially displaceable gear makes meshing engagement with said gear teeth, and means for rotating said pinion to adjust the position of said annular member.

7. In the machine of claim 2 wherein the strand traverse mechanism comprises a movable strand guiding member for directing said strand onto the winding core in helical fashion, and means for driving said movable strand guiding member through one of the gears of the gear train driving said rotatable support.

8. In the machine of claim 7 wherein the gear drive for said traverse mechanism includes means for continuously advancing by a predetermined increment the relative position of the strand guiding member circumferentially with respect to the winding core during each excursion of the strand to provide a strand lay on the core of predetermined pattern.

9. In the machine of claim 8 wherein the means for continuously advancing the strand guiding member includes a differential gainer gear assembly whereby the speed ratio between the rotatable winding core support and the strand guiding member is increased by an increment sufficient to provide accurate processional displacement of adjacent turns.

10. In the machine of claim 3 wherein the traverse mechanism is driven by a third gear coaxially positioned with respect to said two coaxially positioned gears of the gear train, and means coupling the second and said third gear in driving relation.

11. In the machine of claim 10 wherein the coupling between said second and third coaxially positioned gears comprises a speed reducing planetary gear cluster.

12. In the machine of claim 11 wherein the planetary gear cluster is carried by the web portion of the second coaxially positioned gear.

13. In a machine for twisting and winding a flexible strand, a rotatable spindle, a rotatable winding core support carried by said spindle, a winding core slidably mounted on said support, means for advancing a strand into the machine at a rate in direct proportion to the rate of rotation of said spindle, means for directing said strand to said winding core including a traverse mechanism driven in timed relation with respect to said rotatable support, and means driving said rotatable support from said spindle through an adjustable constant torque coupling for controlling the tension on the strand being wound on said core, said means for advancing the strand being driven from the spindle through a gear train to provide a twist of predetermined amount in a given direction in the strand, and at least two gears of a three-gear succession in the gear train being adjustably disposed with respect to each other and the third gear to enable the order of the succession to be changed thereby permitting the spindle to be driven in the opposite direction to impart a twist in the strand in the opposite direction without change in direction of the strand advancing means.

14. In the machine of claim 13 wherein one of the said two gears in the three-gear succession is interchangeable with gears of different size to change the gear ratio of the gear train to accordingly obtain different amounts of twist in the strand.

15. In a machine for twisting and winding a flexible strand, a rotatable spindle, a rotatable winding core support carried by said spindle, a winding core slidably mounted on said support, means for advancing a strand into the machine at a rate in direct proportion to the rate of rotation of said spindle, means driving said rotatable support from said spindle through an adjustable constant torque coupling for controlling the tension on the strand being wound on said core, and means for directing said strand to said winding core including a traverse mechanism having a movable strand guiding member driven in timed relation with respect to said rotatable support, rotatable cam means for reciprocating said strand guiding member to cause said strand to be laid on said winding core, a stationary strand guide member positioned adjacent one end of the path of travel of said movable strand guiding member, a second mov-

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able strand guiding member actuated by said first named strand guiding member in a direction and at a rate with respect to said first member and said stationary member to provide a strand approach path from said stationary member to said first member of constant length throughout the traverse of said first member.

16. In the machine of claim 15 wherein said movable strand guiding member carries a rack, a pinion operatively engaging said rack and carrying said second strand guiding member, and a fixed rack engaging said pinion on the side diametrically opposite to the movable rack whereby traversing movement of said first strand guiding member causes said second strand guiding member to be traversed at the same frequency as said first member but with half the amplitude.

17. In the machine of claim 16 wherein the movable strand guiding member is slidably mounted on a guide rod and said cam means comprises a barrel cam having a continuous reversing cam groove therein, said barrel cam being disposed with its rotational axis generally parallel to said guide rod, a cam follower carried by said movable strand guiding member and coacting with said cam groove whereby rotation of said cam imparts traversing movement to said strand guiding member, said movable rack being rigidly mounted with respect to said movable strand guiding member, and said fixed rack being disposed with its longitudinal axis generally parallel to said guide rod whereby the movable rack and said movable strand guiding member are maintained in orientation about said guide rod and with respect to said barrel cam.

18. In a machine for twisting and winding a flexible strand, a rotatable spindle, a rotatable winding core support carried by said spindle, a winding core slidably mounted on said support, means for advancing a strand into the machine at a rate in direct proportion to the rate of rotation of said spindle, means driving said rotatable support from said spindle through an adjustable constant torque coupling for controlling the tension on the strand being wound on said core, and means for directing said strand to said winding core including a traverse mechanism having a movable strand guiding member driven in timed relation with respect to said rotatable support, an annular spiral cam supported from the spindle and disposed for rotation about said winding core support and said movable strand guiding member, and a cam follower associated with said strand guiding member and coacting with said annular cam to traverse said strand guiding member.

19. In the machine of claim 18 wherein the annular spiral cam has two spiral cam faces, one of said cam faces having a substantially greater pitch than the other and said cam follower associated with said strand guiding member coacting with the cam face having the greater pitch.

20. In the machine of claim 19 wherein the traverse mechanism includes a stationary strand guiding member positioned adjacent one end of the path of travel of said first named movable strand guiding member, a second movable strand guiding member having a cam follower associated therewith and adapted to coact with the spiral cam face having the least pitch for actuating the second movable strand guiding member in a direction and at a rate with respect to the first strand guiding member and the stationary guide member to provide a strand approach path from the stationary strand guide member to the first strand guiding member of constant length for all positions within the traverse of said first strand guiding member.

21. A winding machine comprising a rotatable driving member, a rotatable winding core support carried by said member, a winding core slidably mounted on said support, means for advancing a flexible strand into the machine at a constant rate, means for directing said strand to said winding core including a traverse mech-

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anism driven in timed relation with respect to said rotatable support, and means driving said rotatable support from said driving member through an adjustable constant torque coupling for controlling the tension on the strand being wound on said core, said adjustable constant torque coupling comprising a slip clutch having driving and driven members urged into coactive engagement with each other through resilient means, and means for adjusting the operative tension of the resilient means for controlling the torque transmitted by the driving and driven members.

22. A winding machine comprising a rotatable driving member, a rotatable winding core support carried by said member, a winding core slidably mounted on said support, means for advancing a flexible strand into the machine at a constant rate, means for directing said strand to said winding core including a traverse mechanism driven in timed relation with respect to said rotatable support, and a gear train interposed between the rotatable driving member and the rotatable support for driving said rotatable support from said rotatable driving member, said gear train including an adjustable slip coupling adapted to deliver a constant torque whereby the strand is wound on said core with decreasing tension as the strand package increases in diameter.

23. The machine according to claim 22 wherein the gear train includes two coaxially positioned gears, a slip coupling disposed between said gears, the face of one of said gears constituting a driving surface for said coupling, a coacting driven surface carried by the adjacent face of the other of said gears, and spring means for urging said coacting surfaces into engagement with each other.

24. The machine according to claim 23 wherein one of said two gears is rotatably mounted on a shaft and held against axial displacement along the shaft and the other of said two gears is rotatably mounted on the same shaft and adapted for axial displacement between predetermined limits along the shaft and spring means disposed in surrounding relation to the shaft and having one of its ends fixed with respect thereto and the other end engaging said axially displaceable gear to urge said two gears toward each other and bring the driving and driven surfaces thereof into driving engagement.

25. The machine according to claim 24 wherein engagement of one end of said spring means with said axially displaceable gear is made through an annular member having peripheral thread-wise engagement with the threaded cylindrical surface of a coaxial cylindrical cavity in the web of said gear, and means for rotating said annular member to cause it to be displaced axially within said cavity and thereby adjust the position of engagement of the said spring means with respect to said gear and the operative tension of said spring means.

26. The machine according to claim 25 wherein the peripheral threaded portions of the annular member have gear teeth thereon and a pinion rotatably mounted in the web portion of the axially displaceable gear makes meshing engagement with said gear teeth, and means for rotating said pinion to adjust the position of said annular member.

27. A winding machine comprising a rotatable driving member, a rotatable winding core support carried by said member, a winding core slidably mounted on said support, means for advancing a flexible strand into the machine at a constant rate, means for driving said rotatable support from said driving member through an adjustable constant torque coupling for controlling the tension on the strand being wound on said core, and means for directing said strand to said winding core including a traverse mechanism having a movable strand guiding member driven in timed relation with respect to said rotatable support, rotatable cam means for reciprocating said strand guiding member to cause said strand to be laid on said winding core, a stationary strand

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guiding member positioned adjacent one end of the path of travel of said movable strand guiding member, a second movable strand guiding member actuated by said first named strand guiding member in a direction and at a rate with respect to said first member and said stationary member to provide a strand approach path from said stationary member to said first member of constant length throughout the traverse of said first member.

28. The machine according to claim 27 wherein said movable strand guiding member carries a rack, a pinion operatively engaging said rack and carrying said second strand guiding member, and a fixed rack engaging said pinion on the side diametrically opposite to the movable rack whereby traversing movement of said first strand guiding member causes said second strand guiding member to be traversed at the same frequency as said first member but with half the amplitude.

29. The machine according to claim 28 wherein the movable strand guiding member is slidably mounted on a guide rod and said cam means comprises a barrel cam having a continuous reversing cam groove therein, said barrel cam being disposed with its rotational axis generally parallel to said guide rod, a cam follower carried by said movable strand guiding member and co-acting with said cam groove whereby rotation of said

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cam imparts traversing movement to said strand guiding member, said movable rack being rigidly mounted with respect to said movable strand guiding member, and said fixed rack being disposed with its longitudinal axis generally parallel to said guide rod whereby the movable rack and said movable strand guiding member are maintained in orientation about said guide rod and with respect to said barrel cam.

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