July 3, 1962 C. J. HAUG ET AL 3,042,330

BALLING HEAD

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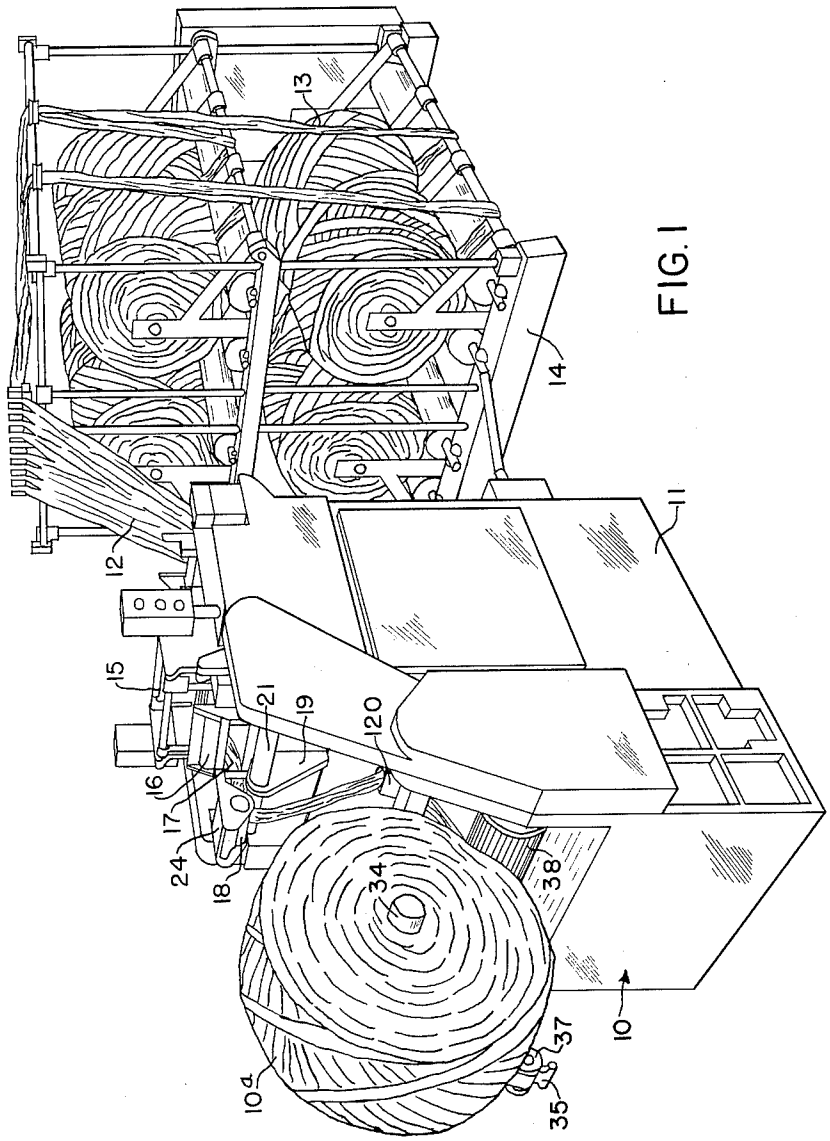


FIG. 1

INVENTORS
CHESTER J. HAUG
GEORGE S. VAN DEUSEN
BY JACK H. SELBY
Hudson, Coughton, Williams, David & Hoffman
ATTORNEYS

July 3, 1962 C. J. HAUG ET AL 3,042,330

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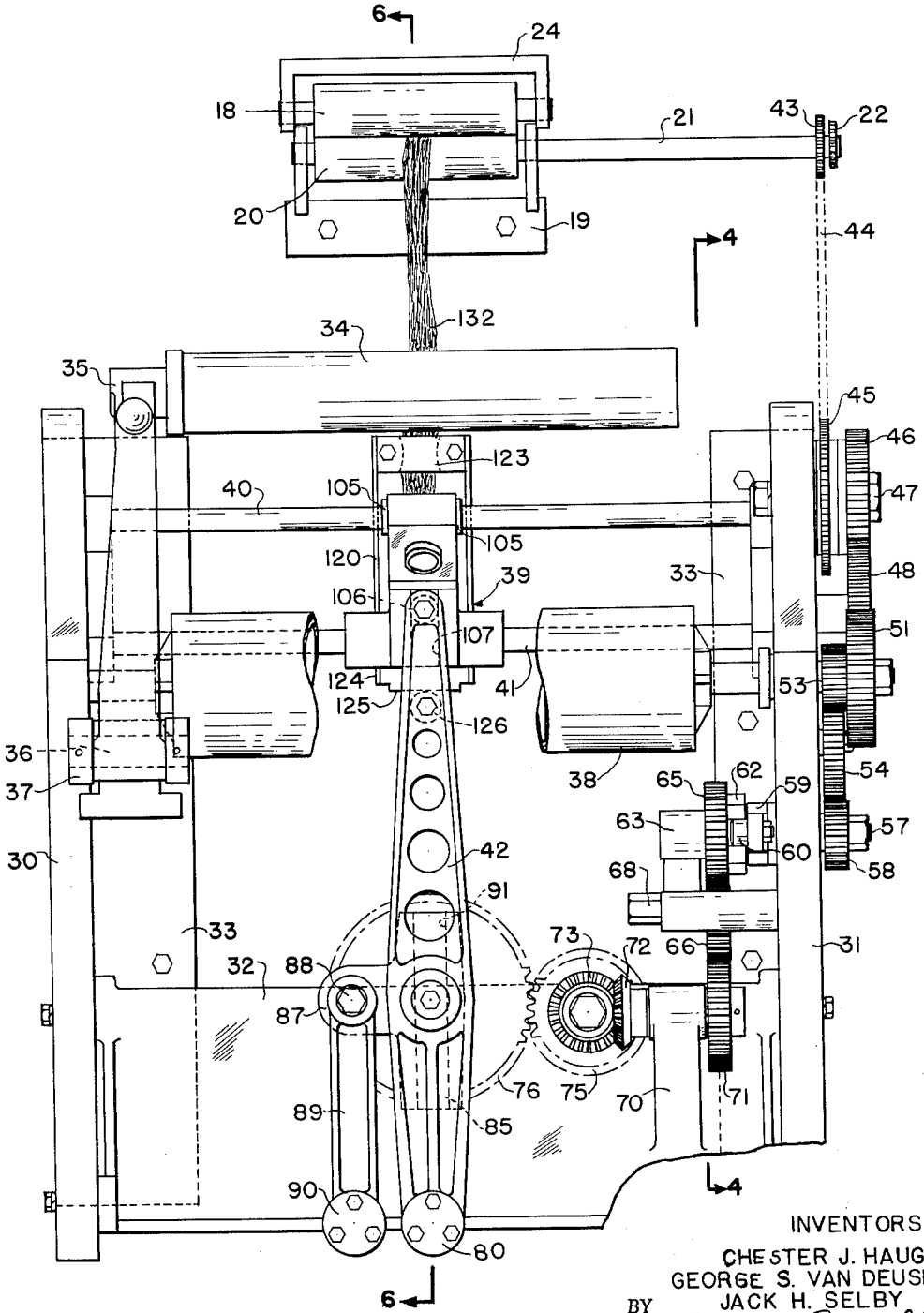


FIG. 2

INVENTORS
CHESTER J. HAUG
GEORGE S. VAN DEUSEN
JACK H. SELBY
BY Hudson, Boughton, Williams, David & Hoffmann
ATTORNEYS

July 3, 1962 C. J. HAUG ET AL 3,042,330

BALLING HEAD

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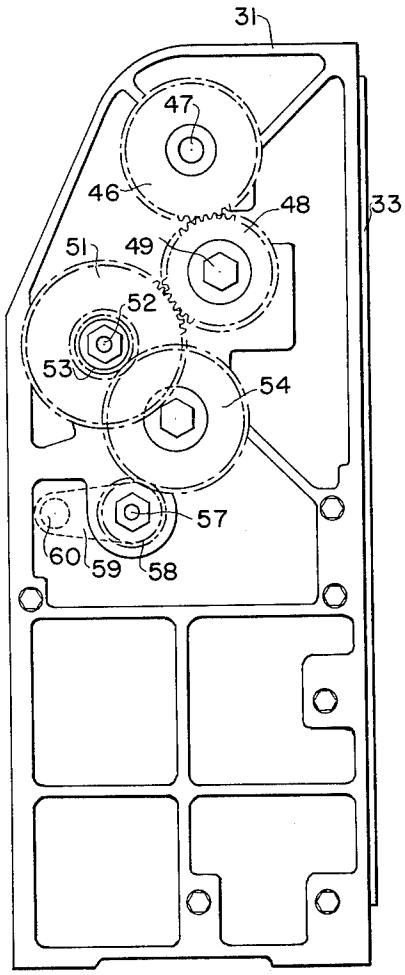


FIG. 3

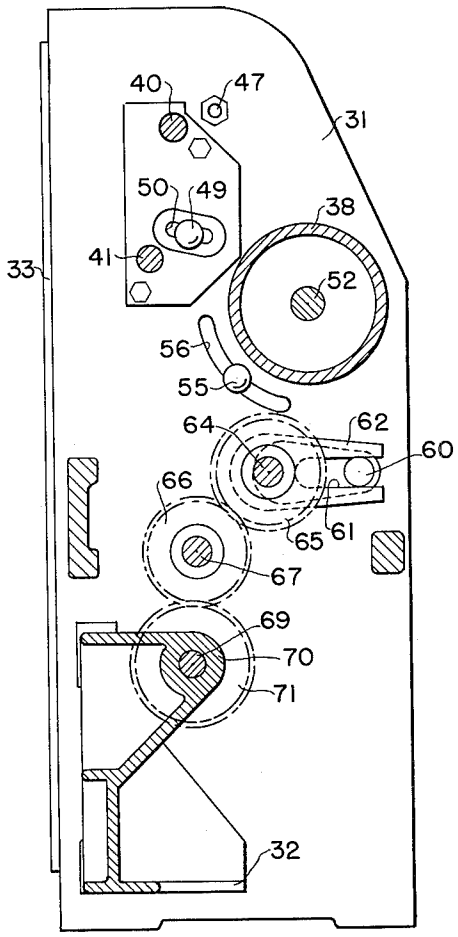


FIG. 4

INVENTORS
CHESTER J. HAUG
GEORGE S. VAN DEUSEN
BY JACK H. SELBY
Hudson, Boughton, Williams, David & Hoffmann
ATTORNEYS

July 3, 1962 C. J. HAUG ET AL 3,042,330

BALLING HEAD

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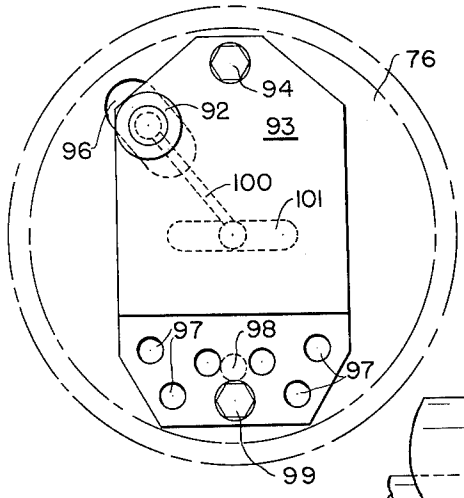


FIG. 8

FIG. 5

INVENTORS
CHESTER J. HAUG
GEORGE S. VAN DEUSEN
JACK H. SELBY
BY Hudson, Boughton, Williams, David & Hoffmann
ATTORNEYS

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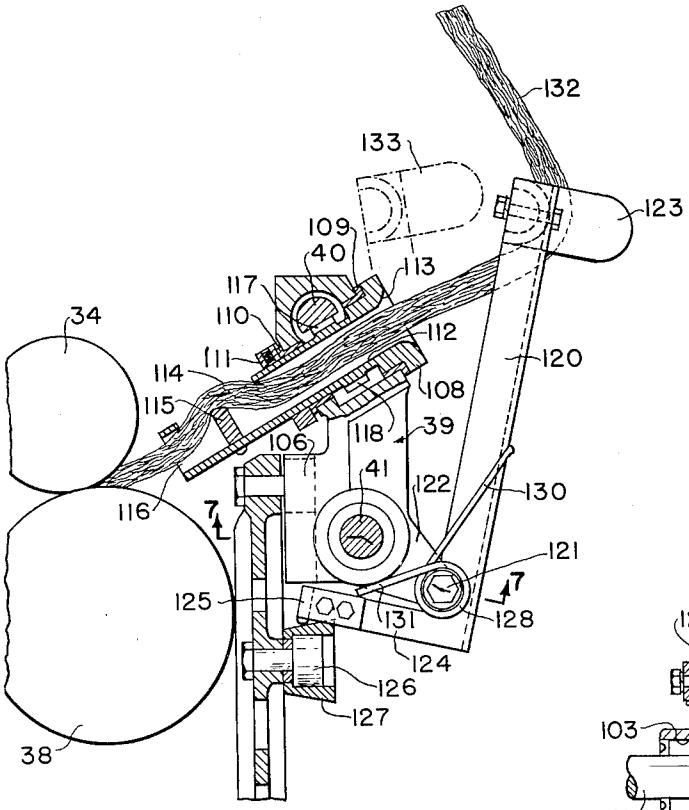


FIG. 6

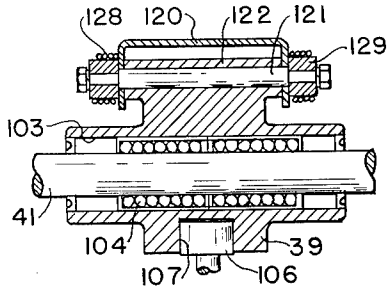
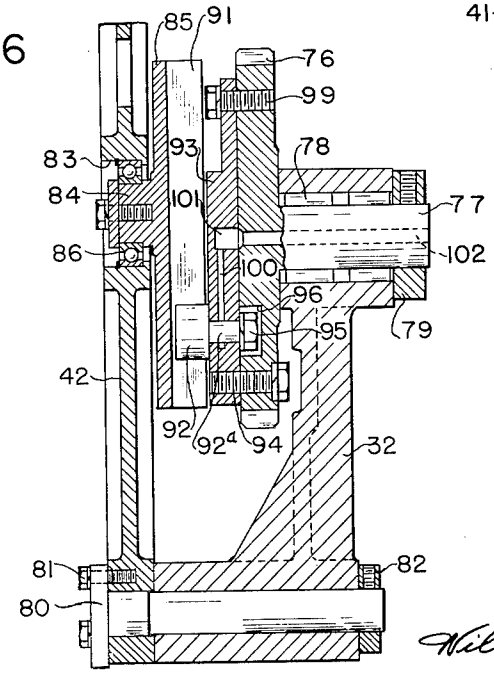


FIG. 7



INVENTORS
CHESTER J. HAUG
GEORGE S. VAN DEUSEN
BY JACK H. SELBY
Hudson, Boughton, Williams, David & Hoffmann
ATTORNEYS

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3,042,330

BALLING HEAD

Chester J. Haug, Cleveland, George S. Van Deusen, Lyndhurst, and Jack H. Selby, Chagrin Falls, Ohio, assignors to The Warner & Swasey Company, Cleveland, Ohio, a corporation of Ohio

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3 Claims. (Cl. 242—54.4)

This invention relates to a balling head mechanism such as is used with carding machines, gill boxes and other similar machines to wind sliver of textile fibers into ball-like packages.

When sliver is being wound onto a ball, the mechanism which handles the sliver may exert an uneven tension at certain times during operation by virtue of a change in the path length of the sliver. Since sliver composed of the usual textile fibers is relatively fragile, any tension beyond a certain minimum tends to draft the fibers and lessen the weight of the sliver at that point. The resulting unevenness is very undesirable for its presence during the early stages of fiber preparation may mean that additional gilling and doubling operations will be required. The occurrence of such unevenness in the final gilling operations usually means that an inferior yarn will be produced.

The use of ball type sliver packages has many advantages for purposes of storage, dyeing, and handling but balls have found little use beyond the initial operations because of the problems of forming the balls at the high speeds of modern gilling, carding and combing machines without introducing objectional unevenness in the sliver.

Therefore, an object of this invention is to provide a novel balling head mechanism which maintains a relatively constant sliver path length at all times during balling without any drafting effect on the sliver.

Another object of the invention is to provide a balling head mechanism having a compensating device for maintaining constant sliver path length between the calender rolls and twister trumpet regardless of the traverse position of the crosshead and which is easily adjustable as required for various fibers and sliver weights.

Another object of the invention is to provide in a balling head mechanism a compensating device for maintaining constant sliver path length, to enable constant tension to be maintained, which is self-contained with the crosshead and traversing mechanism.

Another object of the invention is to provide in a balling head mechanism a traverse drive which produces a compact, symmetrical ball having a uniform diameter and square ends.

A further object of the invention is to provide a balling head mechanism having a minimum of reciprocating mass to allow operation at high speeds with a minimum of power and wear and which is readily adjustable as to ball length, traverse rate, and ball density.

Still another object of this invention is to provide a novel balling head mechanism capable of producing a large ball yet requiring a minimum of floor space and having a relatively simple construction.

Further and additional objects and advantages of the invention will become apparent from the following detailed description and accompanying drawings which show one embodiment of the invention as used in combination with a gill box, although it is understood that it may also be used in combination with other types of machines which produce sliver or a sliver-like material as a product.

In the drawings:

FIG. 1 is a perspective view of a gill box and creel provided with a balling head embodying the present invention;

FIG. 2 is a front elevational view of the balling head shown in FIG. 1, but with certain parts broken away to better show the traverse drive;

FIG. 3 is a side elevational view with covers removed showing the drive mechanism to the drive roll and traverse mechanism.

FIG. 4 is a sectional view taken on line 4—4 of FIG. 2 looking in the direction of the arrows;

FIG. 5 is a developed sectional view of the drive mechanism taken along a broken plane from center to center of the various shafts but with parts rotated from the position of FIG. 4 to facilitate the showing of the parts;

FIG. 6 is a sectional view taken on line 6—6 of FIG. 2 and showing the traverse arm, crosshead and compensating arm;

FIG. 7 is a sectional view taken along line 7—7 of FIG. 6 and showing the mounting of the compensating arm; and

FIG. 8 is a schematic view showing the method of adjusting the stroke of the traverse arm.

Referring now to the figures in detail, FIG. 1 shows the balling head, indicated generally at **10**, attached to the front of a gill box **11**. The gill box may be constructed in the usual manner and is supplied with sliver ends **12** taken from supply balls **13** mounted in creel **14**. The sliver ends **12** are delivered to the faller section **15** of the gill box where they are combined and drafted in the manner well understood in the art, the finished sliver **17** emerging from the nip rolls **16**. From the nip rolls the sliver passes between a pair of upper and lower calender rolls **18**, **20**, and then to the balling head **10** where it is formed into a bill **10a**.

The calender rolls **18**, **20** are mounted on a bracket **19** attached to the front of the gill box **11**, and the lower calender roll **20** is journaled in the bracket **19** and has an extension shaft portion **21** (see also FIG. 2) on the end of which is secured a sprocket **22** which is driven from the nip rolls **16**. In practice, the lower calender roll **20** is driven at the same angular velocity as the nip rolls but is slightly larger in diameter to have a slightly greater surface speed so that no slack will develop in the sliver between the nip rolls and the calender rolls. To keep the sliver from slipping on the lower roll, the upper calender roll **18** is journaled in a movable bracket **24** which is resiliently mounted to hold the upper roll **18** in contact with the sliver passing over lower roll **20**. Further description of the gill box and calender rolls is not deemed necessary as they form no part of the present invention and are shown and described only to show equipment with which the balling mechanism may be used.

The balling head mechanism, shown in greater detail in FIG. 2, has a pair of upright frame members **30** and **31** which are spaced apart at the bottom by a cross beam **32**. When used with a gill box, it may be attached thereto by means of mounting plates **33** which are fastened both to the backs of the upright frame members **30** and **31** and to the frame of the gill box.

From the calender rolls **18**, **20**, the ball of sliver is wound on an arbor **34** which is freely rotatably journaled at one end on one end of an arbor arm **35** and extends substantially perpendicularly to the arm. The other end of arbor arm **35** is journaled on a shaft **36** which is pinned to a forked bracket **37** mounted on the front of upright frame member **30**. The axes of the arbor **34** and shaft **36** are parallel to each other and to a drive roll **38** which is journaled at each end in the frame members **30** and **31** and is disposed below the

arbor 34. The sliver from the calender roll forms a ball on the arbor 34 and the ball is rotated by contact with the rotating drive roll 38. When there is no ball on the arbor 34, it rests on top of the drive roll 38 and as the ball builds up thereon, the arm 35 swings about the shaft 36 to raise the arbor 34 and accommodate the ball.

To insure that the ball will be built up evenly, the sliver must be traversed lengthwise of the arbor to move from side to side across the face of the ball during winding. The traversing of the sliver is accomplished by a crosshead 39 through which the sliver passes and which is slidably mounted on rails 40 and 41 for movement parallel to the axes of the drive roll 38 and arbor 34, the rails being supported between the uprights 30, 31. The crosshead 39 is reciprocated along the rails to move the crosshead and sliver back and forth across the ball by an oscillating traverse arm 42 as explained more fully hereinafter.

The drive for the balling head mechanism which rotates the drive roll 38 and oscillates the traverse arm 42 is shown in detail in FIGS. 2 through 5. As stated above, the lower calender roll 20 is driven from the gill box through a chain drive to sprocket 22 on extension shaft 21. The shaft 21 also carries another sprocket 43 which drives a chain indicated at 44 which, in turn, drives a sprocket 45. The sprocket 45 is secured on a gear 46 journaled on stub shaft 47 secured in frame member 31. An idler gear 48 mates with the gear 46 and a driven gear 51 is mounted on a stub shaft 49 which is adjustably secured in an arcuate slot 50 in frame member 31. This adjustment does not change the engagement between gear 46 and idler gear 48, but allows the use of different sized gears for driving the drive roll gear 51 which is keyed to a shaft portion 52 of the drive roll 38 to effect rotation of the drive roll. By varying the number of teeth on drive roll gear 51, the surface speed of the drive roll 38 will be varied with respect to that of the lower calender roll 20. The drive roll is driven with a surface speed slightly greater than that of the lower calender roll 20, the exact amount depending upon the nature of the sliver, to give the ball the necessary compactness.

Also keyed on shaft portion 52, between driven gear 51 and frame member 31, is a drive gear 53 from which the drive to the traverse arm is taken. An idler gear 54 is driven by drive gear 53 and is mounted on a stub shaft 55 which is adjustable in a second arcuate slot 56 in frame member 31 in a manner similar to the mounting of idler gear 48. The idler gear 54 meshes with a driven gear 58 removably fixed to a shaft 57 journaled in frame member 31. By substituting gears on the shaft 57 with different numbers of teeth, the rate of traverse of the traverse arm 42 and crosshead 39 can be varied relative to the speed of the drive roll 38.

A bracket 63 is fastened to the inside of the frame member 31 adjacent the shaft 67 and is spaced away therefrom to support part of a double crank or fast motion mechanism. This mechanism is incorporated into the drive from the shaft 57 to the traverse arm to reduce the amount of dwell at each end of the traverse, as explained in greater detail hereinafter. Shaft 57 passes through frame member 31 into the space adjacent to the bracket 63 and on its inner end carries a crank arm 59. A roller 60 is secured on the projecting end of crank arm 59 and is received in a slot 61 of a crank yoke 62, journaled on a stub shaft 64 secured in bracket 63 and having an axis parallel to the shaft 57.

The axis of stub shaft 64 is offset from the axis of shaft 57 toward the front of the machine and therefore the yoke 62 and crank 59 rotate about offset axes. The amount of the offset is a small portion of the spacing of the axis of the roller 60 on the end of crank 62 from the axis of shaft 57. The drive train, as described hereinabove, rotates shaft 57 and crank arm 59 at a uniform velocity, and since the roller 60 is fixed on crank arm 59, the roller will move in a circular path at a constant velocity. However, since the axis of crank yoke 62 is displaced from the center of the circle described by roller 60, and since the point of contact where roller 60 rotates the crank yoke 62 must lie on the aforesaid circle, which is eccentric with respect to the axis of stub shaft 64, the crank yoke 62 is rotated by a lever arm of varying effective length. When the effective lever arm of crank yoke 62 is the shortest, as is shown in FIG. 5, it is rotated at an angular velocity greater than that of the crank arm 59, and conversely, when the effective lever arm is the longest, as is shown in FIG. 4, the crank yoke 62 has an angular velocity less than that of crank arm 59. In one revolution, the angular velocity of the crank yoke 62 varies from a maximum to a minimum and back to a maximum during one revolution, but the number of revolutions of both the crank yoke and the crank arm remain constant. Increasing the offset of the stub shaft 64 or shortening the lever arm of roller 60 will produce a greater variation between the maximum and minimum angular velocities, and vice versa.

The drive from this double crank mechanism to the traverse arm is taken from a gear 65 which is keyed on the shank portion of the crank yoke 62. An idler gear 66 meshes with the gear 65 and is journaled on a stub shaft 67 secured in the wall of bracket 63 by a nut 68. A jackshaft 69 is journaled in a projecting support 70 on cross beam 32 and has a gear 71 keyed to its one end and in mesh with the idler gear 66. The other end of the jackshaft 69 has a bevel gear 72 keyed thereto which meshes with a mating bevel gear 73 journaled on a stub shaft 74 secured in the wall of cross beam 32. A drive gear 75 is keyed on the shank of the bevel gear 73 and drives a traverse gear 76. The traverse gear 76 has an integral shaft portion 77 which is journaled in the cross beam 32 by a roller bearing 78 and secured against axial movement by a locking collar 79.

The mounting of the traverse arm 42 and its operative connection with traverse gear 76 is best shown in FIGS. 2 and 6. The traverse arm 42 is secured at its lower end to a rocker shaft 80 by screws 81, the rocker shaft 80 being journaled in cross beam 32 and secured against axial movement by a lock collar 82. The traverse arm 42 has spaced from the lower end thereof a bore 83 wherein a shaft portion 84 of a drive yoke 85 is journaled by a ball bearing 86. The drive yoke 85 has fixed to it a lug portion 87 which projects from it horizontally and transversely to portion 84 and has at its outer end a pivotal connection 88 with an idler link 89 whose lower end is fastened to a second rocker shaft 90 journaled in cross beam 32 adjacent to but spaced from rocker shaft 80 and parallel thereto. From this it will be seen that a parallelogram linkage is formed by the traverse arm 42, idler link 89, lug portion 87 of the drive yoke 85 and the cross beam 32, which serves to maintain the drive yoke 85 in a fixed, non-rotating orientation during oscillation of traverse arm 42 about the axis of the rocker shaft 80. The connection of the lug portion 87 to idler link 89 prevents free rotation of the yoke 85 on the axis of shaft portion 84 and relative to the traverse arm 42.

The drive yoke 85 has a vertical groove or slot 91 in the face adjacent the traverse gear 76. This slot receives a roller 92 which is secured on a carrier plate 93 which in turn is fastened on the traverse gear 76. Roller 92 is positioned a suitable radial space outwardly from the axis of rotation of the traverse gear and is free to move vertically within slot 91. As the traverse gear rotates, roller 92 thus oscillates the drive yoke 85 in response to the horizontal displacement of the roller to reciprocate the traverse arm 42. Since the traverse arm 42 is in the vertical position whenever the roller 92 is in the same vertical plane as the axis of the traverse gear 76, it will be seen that the resulting oscillation of the traverse arm will be symmetrical with respect to this vertical position, not only in position, but also in velocity. It should be noted

that in the disclosed drive, the traverse arm 42 is reciprocated from its mid or vertical position to a limit position in one direction and back to vertical position by 180° rotation of the roller 92 with the traverse gear 76 while a second 180° rotation in the same direction moves the traverse arm 42 from its mid or vertical position to a limit position and back to vertical position. In the case of a crank and connecting rod type of drive, the connecting rod must be of infinite length to approach this result.

If traverse gear 76 were rotated at a constant angular velocity, the horizontal displacement and velocity of roller 92 would be sinusoidal in form. In such case, the inherent characteristics of the sinusoidal motion would result in a period of relatively slow movement effectively amounting to a dwell at each reversal of direction. Since the sliver is being wound on the ball at a constant rate, this slow up at each reversal would result in a build-up of sliver to a larger diameter at the ends of the ball. This objectionable build-up can be eliminated by reciprocating the crosshead 39 at a more nearly uniform velocity.

As stated hereinabove, the double crank mechanism has an output characteristic wherein the output velocity varies from a maximum to a minimum and back during one complete revolution. The above-described drive by means of gears 65, 66 and 71, bevel gears 72 and 73 and gear 75 from the double crank mechanism to the traverse gear 76 results in a two to one reduction in speed and the position of roller 92 with respect to crank yoke 62 is timed so that the point of maximum angular velocity occurs when the traverse arm is reversing its direction of travel at each end of the stroke. The point of minimum angular velocity of the traverse gear 76 then occurs at the center of the stroke where a pure sinusoidal motion would produce the maximum traverse velocity. Thus, the effect of the double crank mechanism is to vary the rotational speed of the traverse gear to produce a more nearly uniform movement of the crosshead while not affecting the symmetrical nature of the motion.

The length of stroke of the crosshead 39 is adjusted by varying the stroke of the traverse arm 42, which, in turn, is done by changing the effective lever arm of the roller 92. To this end, the carrier plate 93 (see FIG. 8) is secured at one end on traverse gear 76 by a pivot bolt 94. Roller 92 has a threaded shank 92*a*, which passes through the carrier plate and is secured by a nut 95 which fits within a clearance recess 96 in the traverse gear. At the other end of the carrier plate 93, there are a plurality of holes 97, which may be selectively brought into alignment with one of several tapped holes 98 by rotating the carrier plate about the pivot bolt 94. The carrier plate 93 can then be locked in the selected position by a bolt 99. It will be noted that the roller 92 is located on the carrier plate 93 so that as it is shifted to and from the center of the traverse gear 76, by rotation of the carrier plate about the pivot bolt 94, the roller retains very closely the same angular position on the traverse gear. Thus, for all lengths of the traverse stroke, the timing of the double crank mechanism and the traverse arm position remains effectively constant. To allow lubrication of the roller 92, a passage 100 connects the roller shank 92*a* with an elongated recess 101 in the carrier plate 93, which recess, in turn, connects with a passage 102 in the shaft portion 77 of the traverse gear. Because some portion of the recess 101 will always register with the passage 102 regardless of the position of the carrier plate 93, the roller 92 can be lubricated in a suitable manner through passage 102 regardless of the stroke setting of the carrier plate.

As stated hereinabove, the crosshead 39 is slidably mounted on cross rails 40 and 41 which are secured at their ends to the frame members 30 and 31. The lower rail 41 passes through a bore 103 in the lower part of the crosshead and ball bushings 104 (see FIG. 7) are disposed between the rail and crosshead to facilitate its sliding movement, while a pair of plain bushings 105 (see FIG. 2) are disposed between the upper rail 40 and the crosshead. The upper end of traverse arm 42 carries a roller 106 which is received in a vertical slot 107 in the crosshead 39 to provide the driving connection between the traverse arm and the crosshead so that the oscillation of the arm 42 reciprocates the crosshead along the rails 40, 41.

A twister 108 is journaled in the upper part of the crosshead just below upper rail 40 by a pair of bushings 109 and 110 and is secured therein against axial movement by a lock collar 111. The twister has a lower end adjacent the arbor 34 or the ball thereon and an axial bore 112 which opens into a trumpet 113 at the upper end of the twister. The wall of the twister adjacent its lower end has a cut out portion 114 and in the center of this cut out 114 a transverse bridge 115 is secured to fit across the bore 112. As is well understood in the art, the sliver enters bore 112 through trumpet 113, passes over the bridge 115 and through the cut out portion 114 and leaves bore 112 at its lower end 116 adjacent the drive roll 38 to be wound on the arbor 34. The purpose of the transverse bridge 115 is to supply a certain amount of gripping force to the sliver so that rotation of the twister can supply a false twist to the sliver. To rotate the twister 108, the lower surface of upper rail 40 is cut to form a rack 117 which engages a pinion gear 118 secured on twister 108 between bushings 109 and 110. As the crosshead reciprocates, rack 117 will rotate pinion gear 118 and twister 108 first in one direction and then in the other, but since the amount of these rotations are equal, only a false twist is imparted to the sliver, as is well understood in the art.

It will be seen that when the crosshead 39 is in the center position, trumpet 113 of twister 108 is directly beneath and at its closest distance to the calender rolls 18, 20 and as the trumpet is moved toward either side, this distance necessarily increases because of the resulting angularity. Since the sliver comes out of the calender rolls at a constant rate, and since the sliver is drawn through the twister 108 and wound on the ball at a constant rate, as this distance changes, the sliver length between the calender rolls and the trumpet must necessarily change. If the sliver is placed under the proper tension at the shortest distance, then as the distance increases, the sliver tension wil increase and usually result in a slight but definite and very undesirable drafting of the sliver. Conversely, if the sliver is properly adjusted at the traverse ends, at the center there will be a slack in the sliver which will interfere with proper forming of the ball. This problem is accentuated by the increased angularity produced by a longer traverse width or by bringing the calender rolls and trumpet closer together as must be done to produce a smaller and more compact machine.

To eliminate this problem, the present balling head mechanism maintains a constant path length for the sliver regardless of the position of the crosshead. To this end, a compensating arm 120 is rotatably mounted at its lower end on a pivot shaft 121 which is secured in a projecting lug 122 of the crosshead 39. The compensating arm 120 extends upwards above trumpet 113 between the trumpet and the calender rolls 18, 20 and is provided with a compensating sliver guide 123 around which the sliver passes. At the lower end of the compensating arm 120 and disposed on the side of shaft 121 remote from the upper end of the compensating arm are a pair of projecting arms 124 between whose outer ends is attached a cam follower bar or element 125 forming part of an inverse cam mechanism. (See FIG. 2.) An element in the form of a driver roller 126 is secured on the traverse arm 42 below roller 106 and includes a conical roller sleeve 127 which is contacted by the lower edge of cam follower bar or element 125. A double torsion or rat trap spring 128 has its coiled portion mounted on bushings 129 on pivot shaft 121 and has a looped arm 130 which passes around compensating arm 120

while the other two arms **131** rest against the underside of crosshead **39** so that the spring rotates the compensating arm **120** to maintain cam follower bar or element **125** in contact with roller sleeve **127** of the cam element on the traverse arm **42**.

When the crosshead is in the center of the traverse with the traverse arm in the vertical position as shown in FIG. 6, the compensating arm is tipped away from the crosshead and the sliver guide means carried thereby. This is due to the roller **126** and sleeve **127** being at the highest point of their arcuate movement with the traverse arm **42**. To pass around the compensating sliver guide **123**, the sliver **132** must bend outward around the guide and form effectively a bow or loop. As the traverse arm **42** oscillates to either side of vertical position it moves the crosshead toward either side from central position, the resultant angularity of the traverse arm lowers roller **126** and sleeve **127** forming the cam element of the traverse arm away from the crosshead. The spring **128** rotates the compensating arm toward the trumpet or sliver guide means on the cross head to maintain contact between cam follower bar or element **125** and roller sleeve **127** of the traverse arm cam element so that at the end of the traverse the compensating sliver guide is in the position shown in dashed lines **133**. By shortening the loop in the sliver produced by compensating sliver guide **123**, the angularity produced by the traverse of the crosshead is compensated for and the effected sliver path remains constant at all times.

Because the loop formed at the center position must not only be of a predetermined length, but also must be taken up at a prescribed rate during traverse to maintain constant sliver tension, the compensating arm must follow a definite path, which needs to be adjustable for different slivers. It can be seen that the angularity of the traverse arm which lowers the driver roller **126** also causes the sleeve **127** to roll lengthwise along the lower edge of cam follower bar **125**. By making this edge curved, the path of the sliver guide can be adjusted to the desired shape. The same effect can also be produced by varying the conical slope and diameter of sleeve **127**, since rotation of the compensating arm necessarily causes a relative movement between the follower bar and sleeve along the axis of the sleeve.

Although only the preferred embodiment of the invention has been shown and described, it will be understood that various changes and modifications to meet specific requirements can be made without departing from the scope of the invention as defined in the following claims.

Having thus described our invention, what we claim is:

1. In combination with a continuous textile sliver delivering mechanism having sliver exit means, a balling mechanism operatively associated with said delivering mechanism and comprising an arbor upon which the sliver is to be wound, means for rotating said arbor in timed relationship to the operation of said delivering mechanism, a crosshead supported for reciprocation in opposite directions from a central position and longitudinally of said arbor and provided with sliver guide means, the distance between said sliver guide means and said sliver exit means being shortest when the crosshead is in central position and variably greater when said crosshead reciprocates therefrom, an oscillating traverse arm operatively connected to said crosshead, means to oscillate said arm in timed relationship to said delivering mechanism to reciprocate said crosshead, and a sliver compensating mechanism for maintaining constant sliver length between said exit means and said guide means and comprising a lever carried by said crosshead on the side thereof that is toward said delivering mechanism and rockable transversely of said crosshead on an axis parallel to said arbor, said lever on one side of its pivotal connection being provided with compensating sliver guide means located intermediate said exit means and said crosshead sliver guide means, said lever on the opposite side of said pivotal connection having an actuating portion including a cam element, said traverse arm at a point spaced from said crosshead mounting a cam element cooperating with said first cam element, and spring means carried by said crosshead and acting on said lever to maintain both of said cam elements in contact with each other whereby when said crosshead is in central position said compensating sliver guide means is located a maximum distance from the crosshead sliver guide means to provide a substantial bow in said sliver but when said crosshead reciprocates from central position the arcuate movement of the traverse arm cam element lowers said element and said spring means acts on said lever to rock the latter to move the compensating sliver guide means transversely of and toward said crosshead to positions closer to the crosshead sliver guide means to reduce the magnitude of said sliver bow in proportion to the increase in the distance between said exit means and said crosshead sliver guide means.

2. The combination defined in claim 1 and wherein said traverse arm is vertically disposed and is pivotally mounted at its lower end while its upper end is operatively connected to said crosshead, and said cam element that is carried by said traverse arm is spaced downwardly below the upper end of the traverse arm and said crosshead.

3. The combination defined in claim 1 wherein said lever of said sliver compensating mechanism is an L-shaped lever and its pivotal connection to said crosshead is at the junction of the L-shaped portions of the lever while the compensating sliver guide means carried by the lever is at the upper end of the longer portion of the lever and the cam element which cooperates with the cam element of the traverse arm is at the free end of the actuating or shorter portion of the lever and said spring means is a rat trap spring surrounding the pivotal axis of said lever and having its opposite ends respectively engaging the longer and shorter portions of said lever.

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