part
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63

THE

PRINCIPLES AND PROCESSES

OF

COTTON YARN MANUFACTURE

BY

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CHAPTER VI

COMBING

As has been previously mentioned, when yarn of a very fine quality is desired, it is customary to use the process of combing. All very fine yarn as well as coarse yarn which is to be used for thread and the better class of hosiery is subjected to combing. The position of combing in the sequence of processes is between carding and drawing. Before the combing machine proper there come two preparatory machines, whose duty it is to get the cotton in proper condition for the comber. Their names are the "sliver lap machine" and the "ribbon lap machine."

SLIVER LAP MACHINE

A view in perspective of the sliver lapper made by the Whitin Machine Works is shown in Fig. 73, and a diagram of gearing and plan in Fig. 74. To this machine the slivers from the card are fed, from 12 to 16 at a time, usually 16. They enter the machine in much the same manner as on a drawing frame. After passing over spoon levers, whose duty is to stop the machine should a sliver break, the combined strand comes within the action of three sets of drawing rollers at the right in Fig. 74. These rollers are just like those on a drawing frame, the top ones being leather covered, while the bottom ones are fluted steel rolls. These draw the strand very slightly. It then passes between two sets of large heavy calender rolls. These rolls press the slivers into a sheet. The sheet of cotton next passes over the two large lap rolls seen at the front of the machine, and is rolled up into a lap in a manner similar to that in which a lap is formed on a lapper. A stop motion is provided to stop the machine when the lap has reached the desired size. The lap when finished is about 9 inches in diameter and about 7½ or 8¾ inches in width. If the lap is to be fed directly to the comber, as is sometimes the case, it has the greater width. If, however, the more usual practice of using both the sliver lapper and the ribbon lapper is in vogue, laps are made narrower

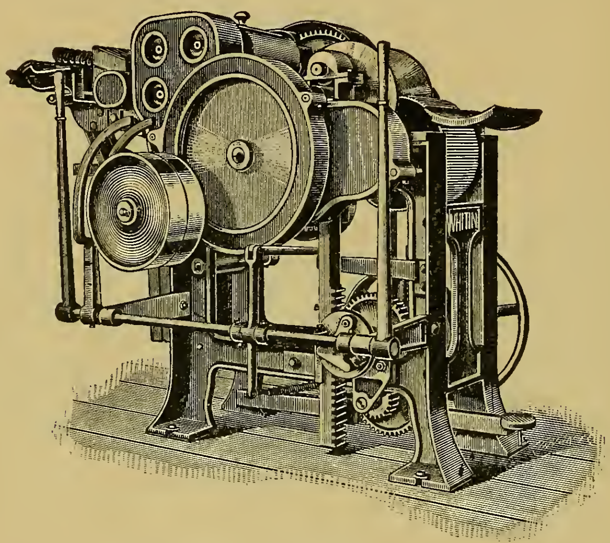

FIG. 73.

in order to allow for the spreading in the ribbon lapper. The usual weight per yard of the lap is between 200 and 300 grains. One machine may produce between 300 and 1,000 pounds per day of ten hours.

CALCULATIONS

The necessary calculations are exactly the same as those on a card. They are performed in the same manner except that there is no such allowance for waste. The sliver lapper does no cleaning, and consequently there is practically no waste while the cotton is passing through the machine. Of course, there is the small amount of ever-present fly seen on all machines in a cotton mill, but this is so small that no allowance need be made for it in the draft calculations.

There is on the machine one change gear known as the draft gear, or the front-roll change gear. It is on the end of what is called the front roll. The word "front," of course, refers to its position relatively to the two other sets of small drawing rolls. The calender rolls and the lap rolls are really in front of it. The change gears that may be used on the machine shown in Fig. 74 vary in size from 20 teeth to 43 teeth. The

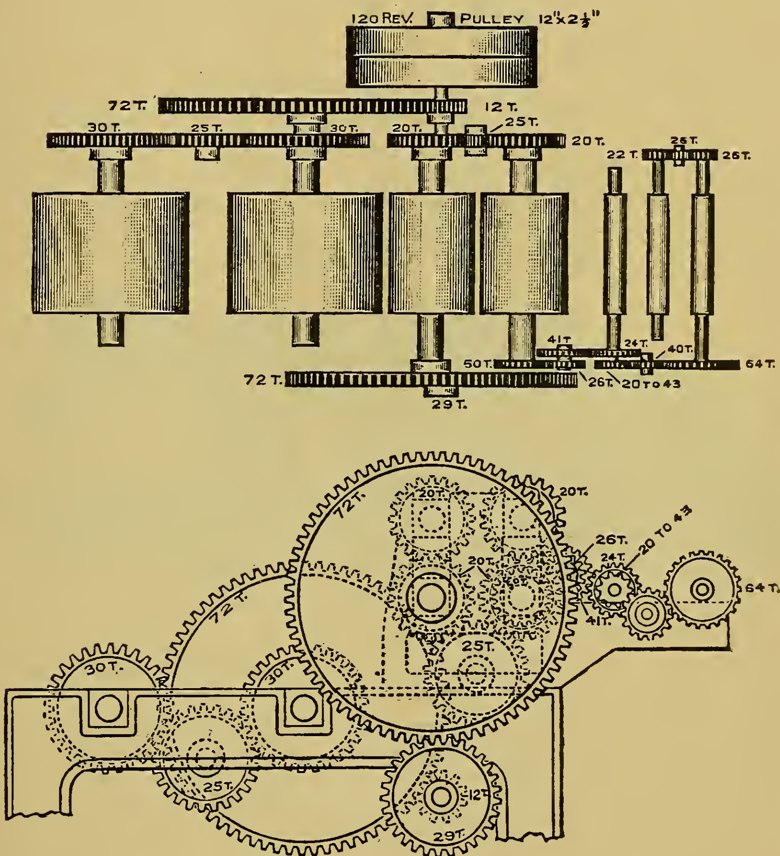

FIG. 74.

following examples are practically the only ones needed on a sliver lapper. The gears used are those in Fig. 74. The large lap rolls are 9 inches in diameter, and the back drawing roll is 1 inch in diameter.

Draft.

The total draft is figured from the first small drawing roll to the lap rolls, inclusive.

Examples:—

1. $$\text{Draft} = \frac{64 \times 24 \times 26 \times 20 \times 72 \times 12 \times 9}{\underset{\text{draft gear}}{20} \times 41 \times 50 \times 20 \times 29 \times 72 \times 1} = 3.22$$

2. $$\text{Draft constant} = \frac{64 \times 24 \times 26 \times 20 \times 72 \times 12 \times 9}{41 \times 50 \times 20 \times 29 \times 72 \times 1} = 64.40$$

3. Production equals surface speed of lap rolls — 10 per cent., allowance for stopping, oiling, cleaning, etc.

Theoretical production in pounds for ten hours of a lap weighing 200 grains per yard, when the calender rolls' speed is 100 revolutions =

$$\frac{100 \times 5 \times 3.1416 \times 60 \times 10 \times 200}{36 \times 7000} = 748 \text{ pounds}$$

Actual production = 748 pounds — 10 per cent. = 748 — 74.8 = 673.2.

RIBBON LAP MACHINE

The ribbon lap machine is not always used, but its use certainly does insure the production of a comber lap of much more uniformity, and with the fibers in a more nearly parallel condition than is possible if only the sliver lapper be used. It certainly is desirable to have the fibers in a comber lap as nearly parallel as possible. The more nearly parallel the fibers are, the less likelihood is there that the needles of the comber will injure the fibers. Consequently the unnecessary waste on a comber may be decreased. The functions of the ribbon lapper are therefore two; namely, to combine six or eight laps from the sliver lapper into one of greater uniformity, and in the production of this new lap to draw the fibers into a parallel condition in much the same manner in which a drawing frame does.

Fig. 75 shows the front of a ribbon lapper, and Fig. 76 a sectional view. It will be seen in Fig. 75 that a machine consists usually of six deliveries. It receives six narrow laps from the sliver lapper, and treats each one separately. They are then all combined at the front of the machine. Each lap rests upon two corrugated wooden lap rolls *l* in Fig. 76. By their revolution it is unrolled. The sheet of cotton then passes along an inclined plate beneath a small tension roll *s*; then over a stop-motion lever, to be referred to later, and then further along the plate to some drawing rolls. Of these there are four sets just as on a drawing frame. The bottom rolls are of steel, fluted longitudinally on their bosses, while the top rolls are usually leather covered. Solid, shell and metallic rolls have been used with considerable success. Metallic rolls are not, however, very common. There is a draft of about six introduced by the rolls, so that when the cotton emerges from the front set it is in the form of a thin ribbon, hence the name of the machine. Each ribbon then

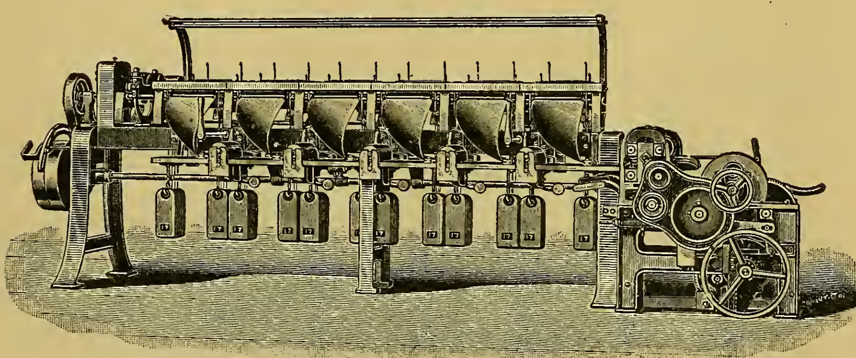

FIG. 75.

passes around a highly-polished curved plate, seen in Fig. 75. By these plates the ribbons are directed into a course at right angles to the one which they took in passing through the rolls. The ribbons then pass along a so-called sliver plate to the lap head of the machine. During their passage they are compressed into one sheet by five smooth self-weighted calender rolls, seen in Fig. 75. The sheet is drawn on by two sets of heavy 5-inch calender rolls, seen in Fig. 78. Their duty is, of course, to compress the sheet very tightly, after which it is rolled up into a compact lap, just as on a sliver lapper. A stop motion is actuated when the lap has reached the required size in a manner similar to that on pickers. There is also a stop motion to ensure the stopping of the machine if a lap breaks at the back or runs out. This stop motion as used on a Hetherington lapper is shown in detail in Figs. 76 and 77. The sheet of cotton, while being pulled on by the back rolls, has sufficient tension to hold the lever *p* in the position shown in the drawing. The creation of this tension is aided by the small roll *s* under which the sheet passes. *p* is balanced

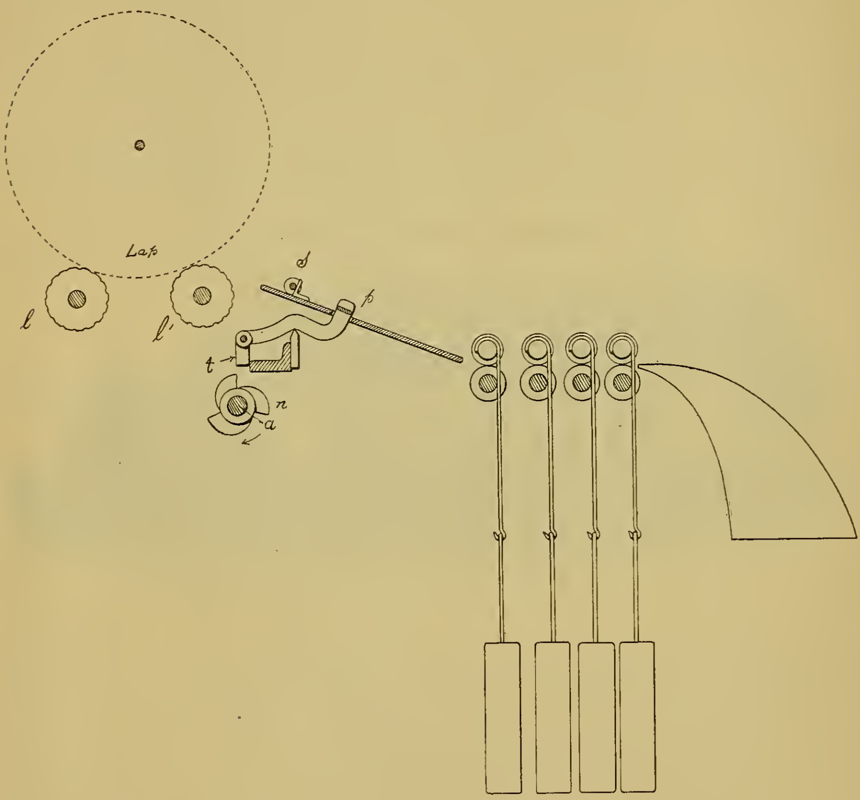

Fig. 76.

on a knife edge, and has a small weighted tail piece. Directly beneath the tail piece *t* is a continually revolving three-winged notched wheel *n*. *n* is fast on the shaft *a*, and is driven by gearing in the direction of the arrow. At the other end of the shaft *a* (Fig. 77) is a toothed clutch *c c'*. The half *c'* has formed upon it a gear and may run loosely around the shaft *a*. *g* is driven from the end of the machine, and when the clutch *c'* engages with *c*, it imparts motion to the shaft *a*. The weighted lever *q* fulcrumed at *o* keeps *c'* in engagement with *c* by pressing against the collar *h*. The working of the mechanism is as follows: When the sheet at the back breaks, the pressure on *p* is removed. *t* then falls into the path of *n*. *t* engages with one of the wings of *n* and stops the shaft *a*. The gear *g*, of course, continues to revolve. But since *a* is firmly held, the notches in *c'* must mount upon the teeth in *c* so that *g* and *h* are

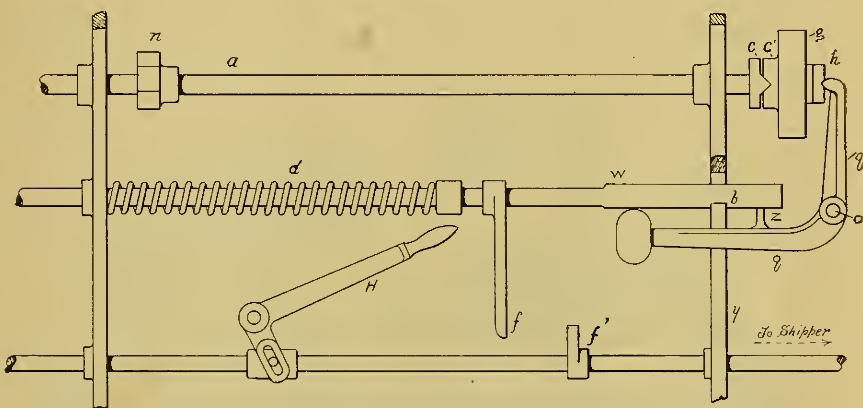

FIG. 77.

pushed to the right. *q* is then rocked around *o*. The finger *z* on *q* is thereby raised. As *z* rises it raises the rod *w* so that the notch *b* is disengaged from the casting *y*. The spring *d* then moves *w* to the right. A finger *f* attached to *w* pushes against *f'* which is attached to the belt shipper rod. By turning the handle H the belt may be later transferred to the fast pulley. At that time *f* would be pushed backward, *b* would re-engage with *y* and the other parts occupy their normal positions. The belt may also be moved to or from the fast pulley without affecting the stop motion.

CALCULATIONS

The calculations needed on the ribbon lapper are just like those on a drawing frame, and are performed in exactly the same manner. In the few examples below the gears used are those shown in Fig. 78.

1. Draft.

For convenience the draft gear in use may be assumed to have 50 teeth.

Example:—

$$\text{Draft} = \frac{56 \times 70 \times 100 \times 68 \times 19 \times 14 \times 21 \times 12 \times 4}{30 \times 50 \times 25 \times 72 \times 40 \times 20 \times 50 \times 11} = 6.02$$

2. Draft Constant.

Example:—

$$\text{Draft constant} = \frac{56 \times 70 \times 100 \times 68 \times 19 \times 14 \times 21 \times 12 \times 4}{30 \times 25 \times 72 \times 40 \times 20 \times 50 \times 11} = 300.81$$

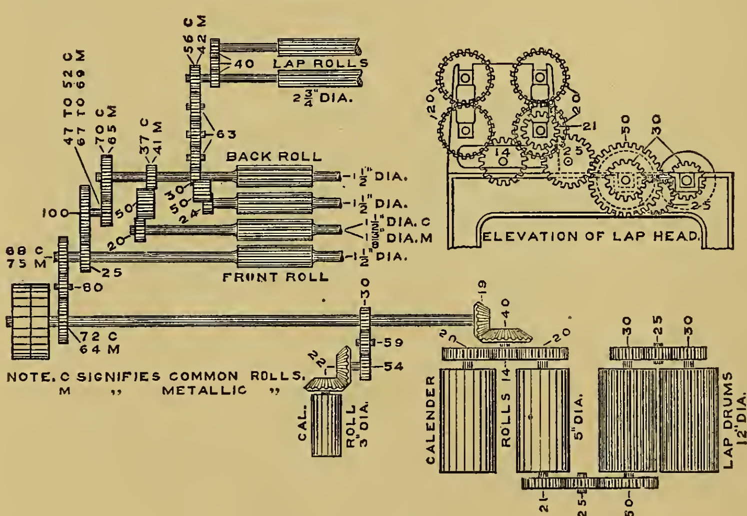

Fig. 78.

3. Draft Gear equals draft constant divided by draft.

Example:—

$$\text{Gear} = \frac{300.81}{6.02} = 50 \text{ teeth}$$

4. Production.

The speed of the 5-inch calender roll from which the production is usually computed is within the limits of 85 and 105 revolutions per minute. In practice the speed may, of course, be ascertained by calculation or more easily with help of a speed indicator. In the examples below the speed is assumed to be 100, the time ten hours, and the weight per yard of the lap delivered 200 grains.

A. Theoretical Production.

Example:—

$$\text{Pounds in 10 hours} = \frac{100 \times 5 \times 3.1416 \times 60 \times 10 \times 200}{36 \times 7000} = 748$$

B. Actual Production.

A deduction of 10 per cent. from the theoretical production for claning, oiling and other stoppages gives good practical results.

Example:—

Pounds in 10 hours = 90 per cent. of 748 = 673.2

COMBER

The combing machine most commonly in use is known as the "Heilman Comber." It is manufactured by several firms, both here and abroad. Front and back views are shown at Figs. 79 and 80, respectively. Fig.

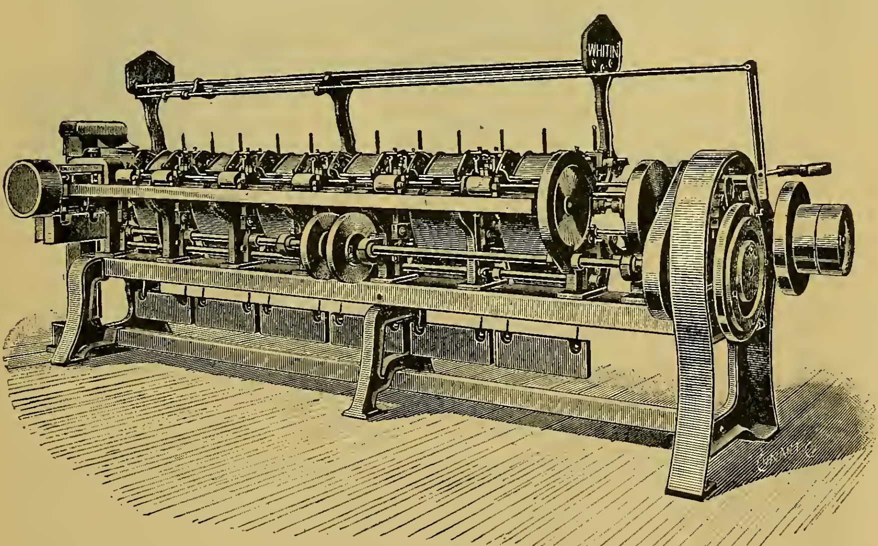

FIG. 79.

81 shows a section. The laps produced on the ribbon lapper are placed at the back of the comber. The machine is usually made so as to receive six

laps. These are put side by side and each receives separate treatment, so that a comber really consists of six separate heads coupled together. They are each driven from the same source, but each in its action on the cotton is complete in itself. A description will therefore be given of one head.

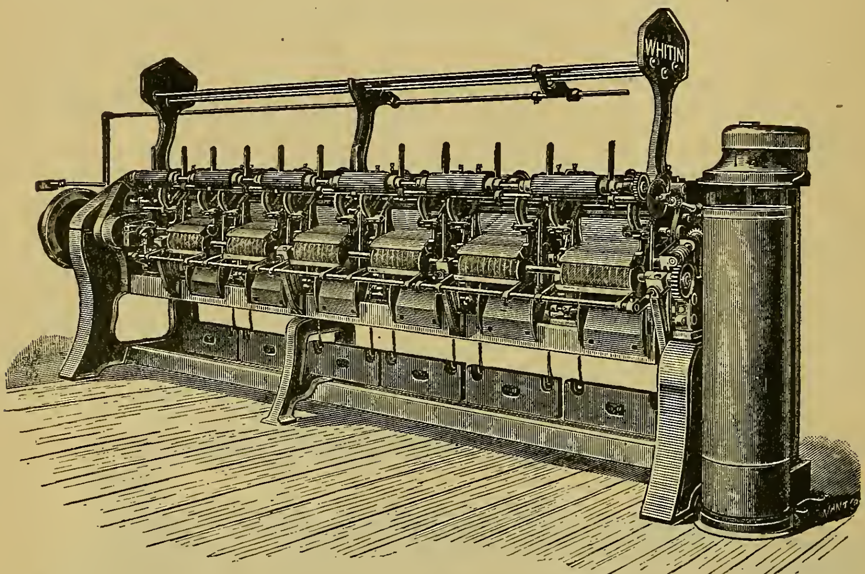

FIG. 80.

The lap (Fig. 81) rests on corrugated wooden lap rollers *l*, is unrolled by their revolution, and then passes down a smooth metal trough to two feed rollers L, K. These rollers have an intermittent motion, *i. e.,* they revolve for a certain time and then remain stationary for a certain other time. The manner in which they receive their power will be shown later. By their revolution the end of the lap is pushed forward between two nippers T, R; to speak more correctly, the upper one T is called the nipper and the lower one R the cushion plate. After the feed rollers have fed a short length of lap into the machine they cease to move, and the nipper closes down upon the cotton, holding it tightly between itself and the cushion plate. The styles of the nipper and cushion plate are quite clearly shown in the figure. The cushion plate is so called because its edge is covered with leather, which does form a sort of cushion. Such an arrangement prevents injury to the cotton when it is being firmly held by the nipper. When the rollers are not moving and the nippers are closed, a small part of the lap protrudes from the nippers, and is in a position to be acted upon by the main working organ of the machine. This main working organ is the cylinder C. It has at A′ a segment in which are set seventeen rows of fine needles. The pitch, or distance apart, of the needles becomes less from left to right. The length of the cylinder is equal to the width of one head of the machine; the rows of needles, of course,

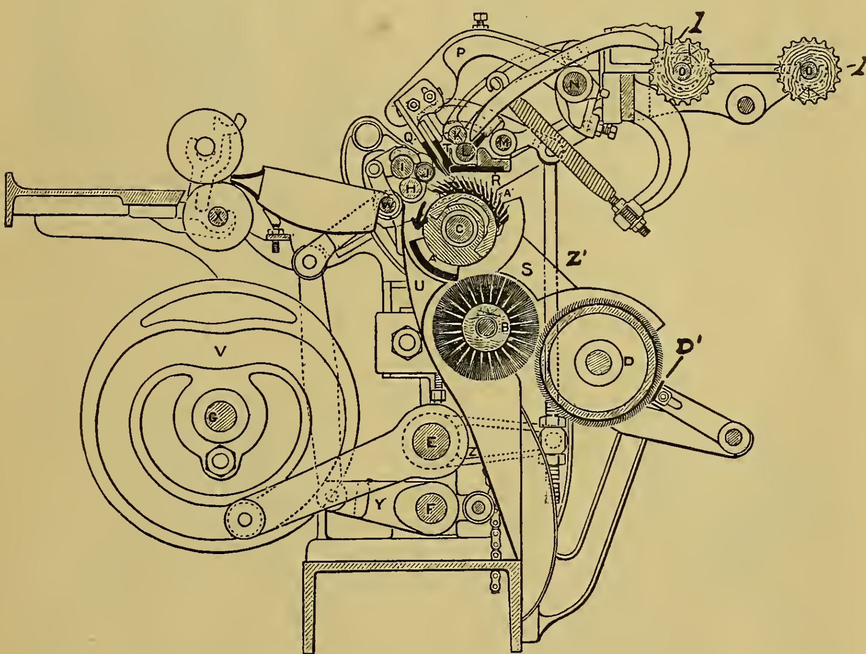

FIG. 81.

extend the length of the cylinder. Opposite the needles is another segment A, which is fluted lengthwise, and is technically called the "fluted segment." The outline of the cylinder between the needles and the fluted segment is shown in the figure. It is perfectly smooth. The cylinder revolves in the direction shown by the arrow. Hence the needles pass through the protruding end of the lap. The length of the lap that is allowed to protrude from the nippers is so regulated in a manner hereinafter described that the needles as they pass through the end will comb out all fibres below a given length. All these short fibres, together with neps and whatever other waste matter the needles may remove are carried around and removed from the needles by the bristle brush B. The brush lays them on a doffer D, similar to the doffer on a card, except in size. The doffer is in turn stripped by a comb D′. The waste then falls into a can provided for it.

Reference to Figs. 82, 83, 84 and 85 may now be made to follow the course taken by the combed end of the lap. In these figures the direction of motion of the cotton is opposite to that shown in Fig. 81. In Fig. 82 the first needles are about to penetrate the cotton. The nippers are closed and the feed rollers are stationary. In Fig. 83 the last row of needles has passed through the lap. The nippers are still closed and the rollers are still not moving. The end of the cotton then falls into the space between the needles and the fluted segment. At the same time the nipper begins to rise. The cushion plate also rises with it for a very short distance, when the nipper leaves it entirely, thereby releasing the cotton. At the same time a comb Q (Figs. 81, 82, 83, 84 and 85) having a double row of needles all along its lower edge begins to descend, and by the time that the fluted segment reaches the cotton, the top comb, as Q is called, penetrates the cotton as shown in Fig. 84. In all of these figures can be seen three rollers H, J and I, known respectively as the steel detaching roller, the leather detaching roller and the piecing roller. H is a fluted steel roller about an inch in diameter. J is a leather-covered roller about ¾-inch in diameter, similar in construction to the leather top roller on a drawing frame. It is held down in place by weights. I is a solid fluted brass roller 1 inch in diameter and of considerable weight. While the fluted segment is approaching these rollers the detaching roller H makes about one-third of a revolution backwards, bringing the cotton which had previously passed over it into the position shown in Fig. 84. As H makes its backward movement the leather roller rolls backward around it until it comes in contact with the fluted segment. Meanwhile the fluted segment will have pressed the cotton upward into the position shown in the figure. Therefore, as J rolls backward it really comes in contact with the cotton. J is of course suitably supported at the ends, and the weight upon it establishes a firm grip upon the cotton between it and the fluted segment. It will be noticed that while the parts are in the positions described, the cot-

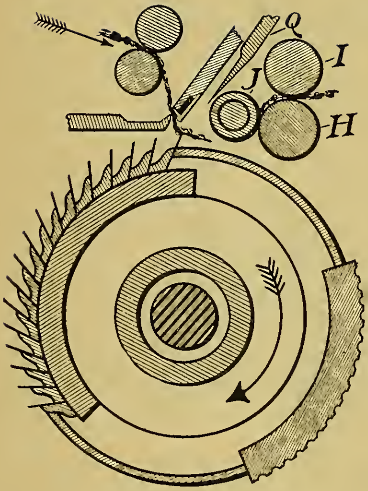

Fig. 82.

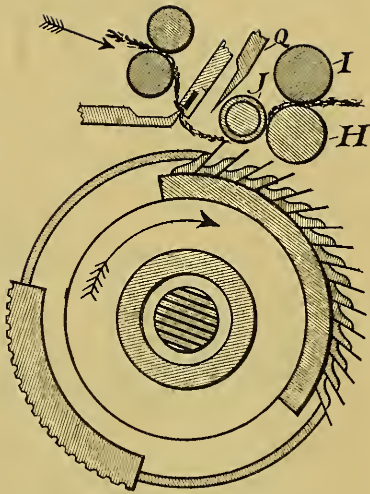

Fig. 83.

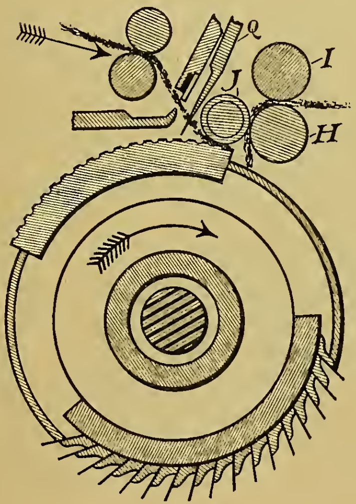

Fig. 84.

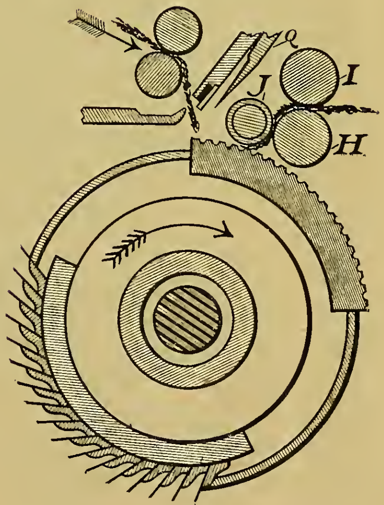

Fig. 85.

ton which has previously been combed hangs down in the space in front of the fluted segment. As the cotton between the segment and the roller is brought forward it overlaps that which is hanging back from the roller H. H then begins to move forward, continuing for about two-thirds of a revolution. The pressure of J upon H makes the new cotton cling to the old. The piecing is further continued as the cotton passes between H and I. When it is ready to leave H and I, it is a continuous evenly-pieced strand. It then passes through a trumpet hole forward between two calender rolls to a smooth trough at right angles to its previous direction of motion. Along this trough the slivers from the six heads of the machine pass, are combined into one at the end, drawn a little, and coiled into a can exactly as on a card. At the end of the sliver plate the drawing is performed by three sets of drawing rolls just like those on a drawing frame. The top ones are leather covered, while the bottom ones are fluted-steel rolls.

While the fluted segment carries the combed cotton forward, the feed rolls move slowly forward delivering new cotton through the nippers. The cotton held between the segment and the leather roller is pulled away from the uncombed lap. Since the top comb Q is at this time penetrating the cotton which has been partially combed by the circular comb, the top comb combs the ends which were in the grip of the nippers. The top comb remains down until the top nipper descends the next time. When it rises it leaves some short fibres, etc., in the protruding end of the lap. These are removed by the circular comb during the next cycle of movements. When the segment has passed the parts are as shown in Fig. 85. The rollers H, I, J stop, the feed rolls stop, too, and the nippers close upon the new cotton just fed by the feed rolls. The circular comb approaches the cotton, and all the previously described operations are repeated.

It will be seen that the motions of most of the parts are intermittent. All the operations are, however, performed so quickly that the only apparent effect is a jerky movement of the sliver as it comes from the piecing roller. The calender rolls and coiler move continuously, so that the delivery is continuous. No breakage occurs, because the length delivered by the detaching rollers at each forward movement is as much as the calender rolls can take up during that same length of time and the subsequent period of arrestation.

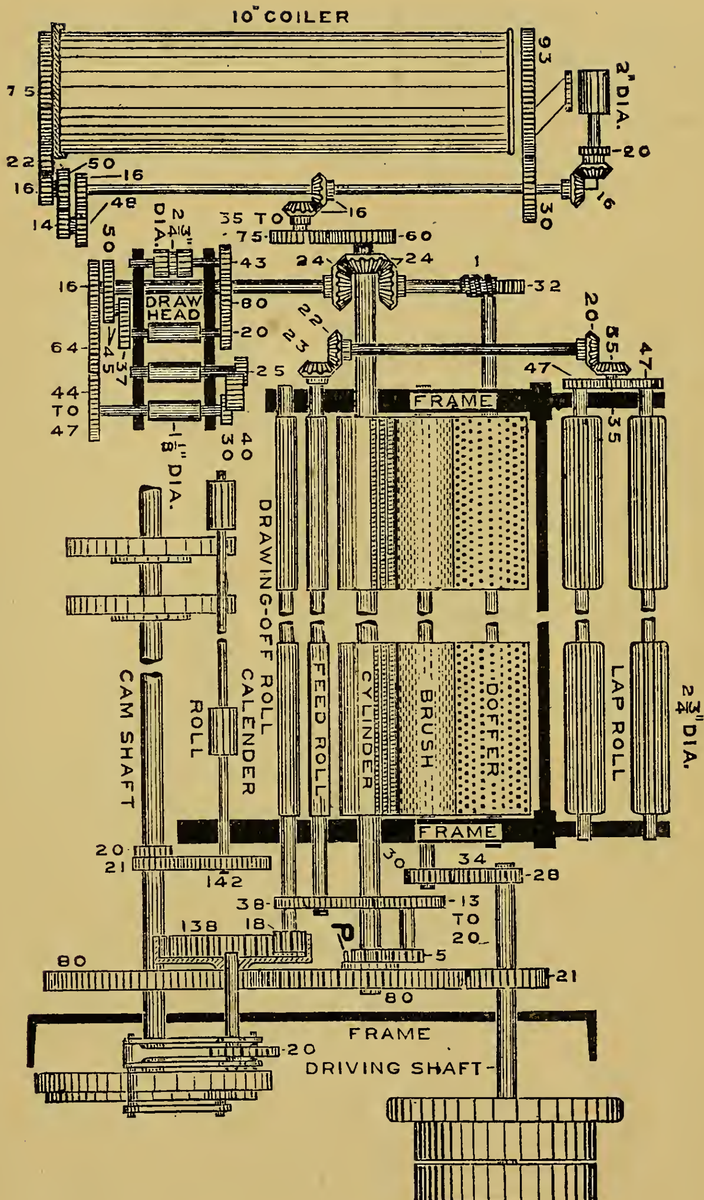
10" COILER
2" DIA.
2¾" DIA.
DRAW HEAD
1⅛" DIA.
FRAME
DRAWING-OFF ROLL
CALENDER ROLL
CAM SHAFT
FEED ROLL
CYLINDER
BRUSH
DOFFER
LAP ROLL
2¾" DIA.
FRAME
FRAME
DRIVING SHAFT

FIG. 86.

A diagram of all the gearing on the machine is shown in Fig. 86. It must be noted, however, that this diagram shows the gears stretched out without enabling one to see clearly whether one gear is above or below another. The connection of one with another is readily seen. The same driving mechanism drives all six heads in a machine, although only two heads are shown in the figure.

On the main shaft of the machine a pinion with 21 teeth meshes with a large 80-toothed gear on the shaft carrying the cylinder. On the boss of the cylinder shaft is a pin or tooth P, whose duty is to engage with the notches in a star wheel marked 5 in the figure. A clearer view of this star wheel will be referred to presently. Suffice it here to say that whenever the star wheel is made to move the change gear on the same stud with it, having between 20 and 13 teeth, is also made to revolve. The change gear meshes with a 38-toothed gear on the end of the lower feed roll. From the other end of the feed roll the lap rolls receive their power by the bevel gears and side shaft seen in the figure. It must be noted that the feed rolls are actually above the cylinder, while they appear here to be beside it. The parts already mentioned may be seen in their proper relations in Fig. 87. C is the cylinder shaft. On the boss of a gear on this shaft is the pin P. It, of course, revolves with the shaft. At each revolution it engages with a notch in the star wheel D. It therefore advances the star wheel one tooth at each revolution of the cylinder. The star wheel is prevented from moving except when actuated by P by the gapped disc F, which is firmly attached to the cylinder shaft. In dotted lines are shown the change pinion E on the stud with D and the gear G on the lower feed roll. The wooden lap rolls are driven from the opposite side of the machine in the manner easily understood from the drawing. It will be readily understood that at every complete revolution of the cylinder shaft the feed rolls move a fixed distance. It will also be seen that the movement of all these parts is not continuous. They move only for the length of time occupied by the pin P in moving the star wheel through the distance between two notches. The desired length of movement of the feed rolls depends upon the cotton being combed. At times it will be desirable to have them feed into the machine more cotton than at other times. The length of the fibres which the circular comb removes depends directly upon the length of cotton protruding from the nippers when the needles pass through. All fibres below any given length can be combed out by making the feed rolls feed sufficient cotton each time that they move. Regulation of this feeding can be had by changing the change gear already mentioned. The longer the fibres which one wishes to remove the larger must be the change gear. It will be understood that the

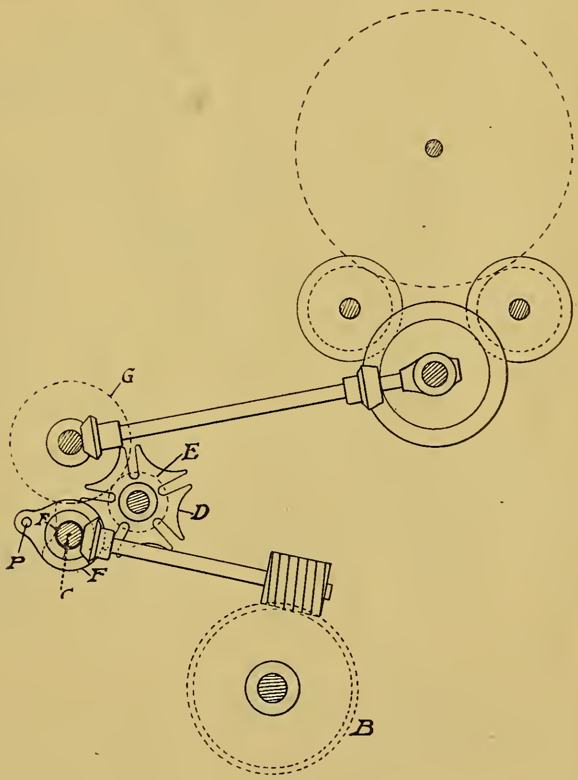

FIG. 87.

top leather feed roll is driven from the bottom by frictional contact, being held in contact with it by springs. B in the figure is a worm wheel on the end of the doffer, which is driven from the cylinder shaft as shown.

Nipper.

We shall next describe the driving of the nipper. On the shaft marked "cam shaft" may be seen two cams at about the middle of the machine (see also Fig. 79). One of them controls the movement of the nipper. In Fig. 88 the shape of this cam and its connection with the nipper may be traced.

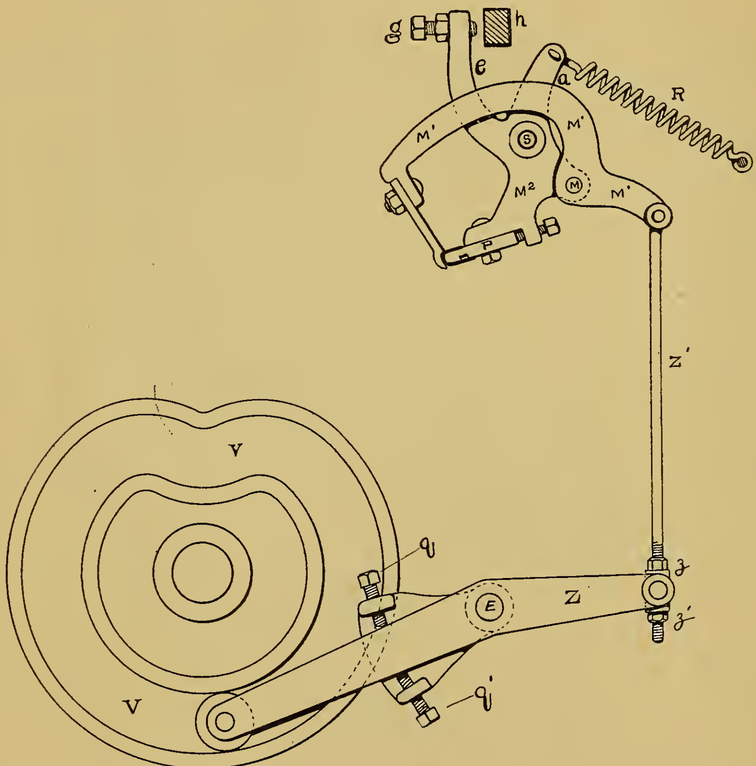

Fig. 88.

per may be traced. A bowl at the left end of a lever fulcrumed at E fits into the cam course V. The lever Z is attached by the nuts z and z' shown to the upright rod Z′. Z′ is attached to the nipper, which is fulcrumed at M. One part of the cam course V is circular. While the bowl on the lever is in this portion of the course, as it is shown to be in Fig. 88, the nipper is pressing tightly on the cushion plate and is being held firmly in that position. When with the revolution of the cam the bowl enters the flattened part of the course, the bowl is raised, Z′ is lowered, the casting carrying the nipper is oscillated around M, and the nipper is raised away from the cushion plate. The casting M′ which carries the nipper is fulcrumed at M to a second casting M^2, not to the frame work of the machine. M^2 is however fulcrumed to the frame work at *s*. Two ears, *e* and *a*, are formed in the top of M^2. The spring R having its other end attached to the frame work is connected with one. Through the ear *e* is the screw *g*, which rests against a shoulder on the frame work. R is continually tending to revolve M^2 around *s*. It can do it only when the screw *g* is not pressing against the shoulder. When the nipper presses down upon the cushion plate it goes so far down that it pushes the cushion plate with it, rocking it slightly around *s*. The spring R is thereby stretched and the screw pulled away from the shoulder *h*. When the pressure of the nipper on the cushion plate is released the spring pulls the cushion plate back into its normal position. It will be readily seen that by having the cushion plate movable for a short distance a firmer grip is established on the cotton than would be the case if P were rigid. The cushion plate is adjustable backward and forward in the casting M^2. Adjustment of the throw of the nipper can be had by the screws *q*, *q′*, *z* and *z′*.

Reference to Fig. 86 will show that on account of the size of the gears for every revolution of the cylinder shaft the cam shaft revolves once. Therefore for one revolution of the circular comb the nippers open and close once. They, of course, have to be timed so as to open and close at the proper time.

Top Comb.

A detached view of the top comb may be seen in Fig. 89. P, holding the top comb Q, is fulcrumed at M and may move loosely around M. Attached to the shaft M is a downward extending arm L carrying a bowl B, which rests on a cam W on the cylinder shaft. In the position shown in the drawing the bowl is resting on the low part of the cam. In that case the weight of the combs is sufficient to make them penetrate the cotton. The shaft M extends the whole length of the machine, and to it the six levers P, carrying the several combs, are fulcrumed. When the high part of the cam raises the bowl B the end of the screw S presses against the lower end of the lever P and raises the comb. By the screw S the length of time that the comb may remain in the cotton can be regulated, and S′ the distance which it will sink into the cotton. P is set

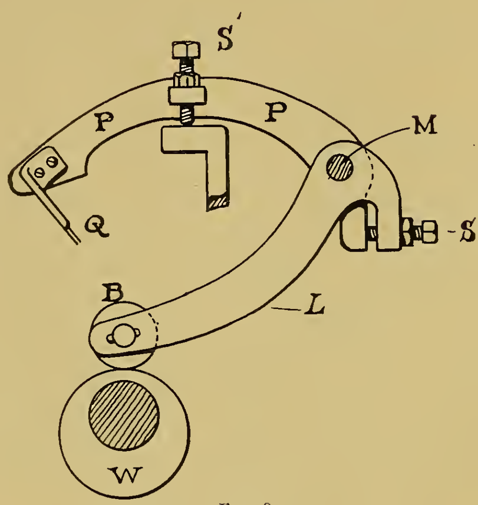

Fig. 89.

loosely upon M so that any comb may be raised, when one wishes to get at the inner part of the machine.

Rocking Motion of Top Detaching Rolls.

It has been previously stated that the steel detaching roll makes a backward movement through about one-third of a revolution just as the newly-combed piece of cotton is brought forward by the fluted segment. The leather detaching roll revolves backward, too, since it is held firmly in contact with the steel roll by weights. The leather roll also is rocked backward as the segment approaches in order to establish a firm grip upon the cotton between itself and the fluted segment. The rocking motion of the leather roll will first be described. The movement is controlled by the second cam on the cam shaft. The connection between the cam and the roll is shown in Fig. 90. In the cam course C fits a bowl attached to the bell-crank lever Y, which is fulcrumed at F. Y is connected by the connecting rod *r* to the second bell-crank lever Y′. Y′ is fulcrumed at W.

The ends of the leather roll J are supported by shoulders on the lever Y′. The weight applied to the roll keeps it in contact with the shoulder as well as with the steel roll. The shape of the cam is clearly shown. For a little more than half a revolution the bowl is in a part of the course that is concentric with the cam shaft. Therefore, for that length of time the levers and connections do not move. When the bowl enters the other part of the course, Y begins to rock around F, and Y′ around W. J is thus brought down upon the fluted segment. It remains down until the seg-

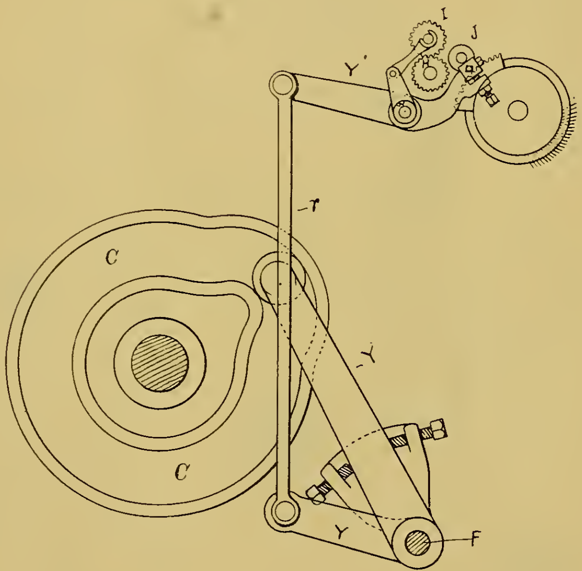

Fig. 90.

ment passes, and then rises again, by which time the bowl has re-entered the concentric part of the cam course. The brass piecing roll is also supported in slots in the lever Y′. Consequently it is rocked backward with the leather roll, always remaining in contact with the cotton on the steel detaching roll.

In Fig. 90 the cylinder needles are only roughly indicated, and the exact positions of the rolls in relation to each other are not so accurately shown as in Figs. 82, 83, 84 and 85. The object of the Fig. 90 is to show mainly the shape of the cam and its connection with the rolls.

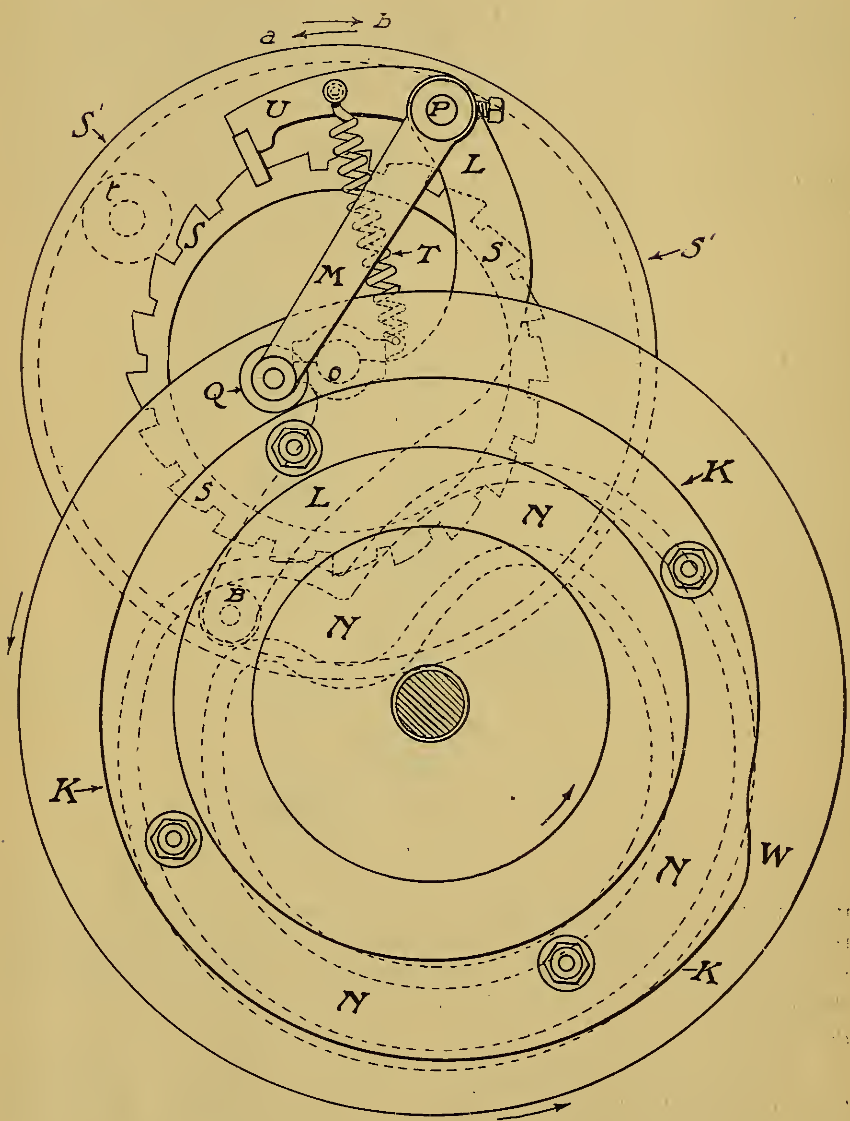

FIG. 91.

Rotary Movement of the Steel and Other Detaching Rolls.

The steel detaching roll has a rotary movement only. It does, however revolve both backward and forward. Since the other detaching rolls are both driven by friction from the steel roll, their rotary motions correspond to its. These movements are controlled by cams on the cam shaft at the head of the machine close to the driving pulleys. They may be seen in Fig. 79, and more clearly in their correct relation to each other in Fig. 91. The connection between the cams and the detaching roll is as follows: The notched wheel S in Fig. 91 receives a backward and forward movement in a manner to be described. On the shaft with this notched wheel is a large gear S′ having teeth on its inner instead of its outer edge. In Fig. 86 this gear is marked 138. This internal gear meshes with a small pinion *t* on the end of the steel detaching roll. The levers L and M connect the cams with the notched wheel. One lever L is fulcrumed at O, and has a bowl B at its end, which runs in the cam course N, shown in dotted lines. The second lever M is attached to the first at P, but is free to move around P. The lever M has integral with it a catch U, the end of which is so shaped as to fit the notches in the notched wheel. A spring T is attached to the catch U and to the frame work of the machine. Its duty is to pull the catch down towards the notched wheel. On the other end of the lever M is a bowl Q, which rests against the outside edge of the cam K. Since the catch U and the lever M are practically one piece, the spring will keep the bowl Q always in contact with the cam K. It is evident that the lever M will be made to rock on the centre P, while at the same time the lever L will rock on the centre O. Thoroughly to understand the working of this mechanism the shape of the cams must be carefully observed, and the position of the bowls at different times noted. Fig. 91 shows the relation of the parts as the notched wheel, and consequently the detaching roll, is making its backward movement. The cams revolve continually in the direction indicated by the accompanying arrows. It will be observed that the bowl B is entering the part of the cam course nearest the axis of the cam. While the bowl B continues to approach the axis of the cam, the lever L will be rocked around O, the top of L will move towards the left, the catch U will move to the left and the notched wheel will be rotated in the direction indicated by the arrow *a*. As the cam N continues to revolve, the bowl B moves away from the axis of revolution towards the higher part of the cam course. The notched wheel is thereby rotated in the direction shown by the arrow *b*. During all this time the bowl Q has been resting on the concentric part of the cam K. The catch has therefore been engaged with the notched wheel, as was of course necessary in order that S′ might revolve. When the bowl Q reaches the projection W on the cam K it is raised. U is thereby removed from the notched wheel. Directly after this the lever L changes its direction of motion, and the top of it begins to go to the left. The catch U is held sufficiently high by the cam K to clear the top of the next tooth to the left. By the time that the catch has reached he next notch in its backward movement, Q will have reached the concentric part of the cam K and will allow the catch again to engage with the notched wheel. S′ is then rotated backwards and the cycle of operations is repeated. If the outline of the cams and their effect upon the time of action of the motions be carefully observed it will be seen that the forward movement of the detaching roll is about twice the backward. This is due mechanically to the fact that for about half of the backward movement of the catch it, the catch, is out of engagement with the notched wheel.

Other Driving Mechanism.

The driving of the coiler and draw box will be readily understood from Fig. 86. Both of these pieces of mechanism are driven from the cylinder shaft on the side of the machine opposite to the driving pulleys. It must here be noted that there are on a comber four change gears, situated as follows: The first, which has already been mentioned, is on the stud with the star wheel in the train of gears driving the feed rolls. The range of sizes used on the machine shown in Fig. 86 is indicated. The larger the gear the greater will be the amount fed in by the feed rollers at each revolution of the cylinder. The second change gear is on the end of the back roll in the draw head. One of the proper size must be used to make the back roll of the draw head take up as much cotton as is delivered by the calender rolls. By changing this gear the draft in the draw head is of course incidentally changed. The third change gear is the one on the end of the front roll in the draw head, and is marked 37 in the drawing. It is changed whenever it is desired to change the draft in the draw head. The larger the gear the smaller the draft in the draw head. Whenever this gear is changed it is necessary to change the fourth change gear, which is marked 35 to 75 in the drawing, and is in the train between the cylinder shaft and the coiler. It must be of the proper size to make the coiler calender rolls take up all that is delivered by the draw head. The brush and doffer driving mechanism is readily seen in the drawing.

Timing and Setting.

The most important consideration in the running of a comber is the timing of the various parts. The machines are of course accurately put together and adjusted by the machine builders when they are set up in the mill. Yet even after the proper settings have been made there are parts which need readjustment on account of changes in the cotton used or in the quality of the product desired. It is with those settings and adjustments that we shall here have to do. The actual timing of the parts needs very little change. It is the distance of the various organs from each other that is more often varied. The changes in timing and the changes in distance apart of the organs depend upon two factors; first, the length of the staple, and second, the amount of waste desired. The following parts require setting:

Cushion Plate to Detaching Roll.

The distance depends upon the length of the staple. The distance is measured from the forward edge of the cushion plate to the flutes of the detaching roller. Good average settings are $1\frac{3}{16}''$ for Egyptian cotton, $1\frac{7}{16}''$ for Sea Island, and $1\frac{1}{16}''$ for Peeler.

Feed Rolls to Detaching Rolls.

This distance depends upon the length of the staple. Good average distances between flutes of feed rolls and flutes of steel detaching roll are $1\frac{11}{16}''$ for Peeler, $1\frac{13}{16}''$ for Egyptian, and $2\frac{1}{16}''$ for Sea Island.

Nipper to Cylinder Needles.

The distance depends upon both the previously-mentioned factors. Different wire gauges are usually provided to facilitate the setting. For Sea Island cotton a so-called 21's gauge is commonly used. For Egyptian cotton a 19's gauge, and for Peeler a 17 or 18. The gauge is inserted between the edge of the nipper and the cylinder needles. By making closer settings the amount of waste may be increased.

Top Comb to Fluted Segment.

The same gauges are used to set the top comb to the fluted segment as are used for setting the nipper to the cylinder needles. The comb is set at an angle of about 28 degrees. An angle gauge is provided for the purpose. The greater the angle at which the comb is set the greater the amount of waste.

Changes in Timing.

By moving the position of cams on the cam shaft the time at which the nippers close may be delayed. The feeding may also be delayed. In either case there would be more waste produced.

All the remarks relating to the care of the rollers on a drawing frame apply with equal force to the rolls on a comber.

Speed and Waste.

The speed of a comber is stated in terms of the number of "nips" per minute. The number of nips made varies between 75 and 100 per minute. The amount of waste depends upon the quality of product desired. It is between 15 and 20 per cent.

Duplex Machine.

There is manufactured a so-called "Duplex Machine," of which Fig. 92 shows a section. It has two sets of cylinder needles, two fluted segments and all the operations of the single-nip machine occur twice during each revolution of the cylinder. It is possible for it to make about 120 nips per minute, so that the production thereon is considerably increased. It has not been thought to produce quite such a good quality of work, and is not nearly so extensively used as the other type.

CALCULATIONS

The calculations on a comber are not numerous. The essential ones are here given.

I. Feed.

A. Length delivered by feed rolls at each nip.

It is here merely necessary to calculate the surface speed of the 1-inch feed rolls for one revolution of the cylinder. The sizes of the gears are shown in Fig. 86. Let it be assumed that a 20-toothed change gear is being used.

$$\text{Length} = \frac{1 \times 20 \times 1 \times 3.1416}{5 \times 38} = .33 \text{ inches}$$

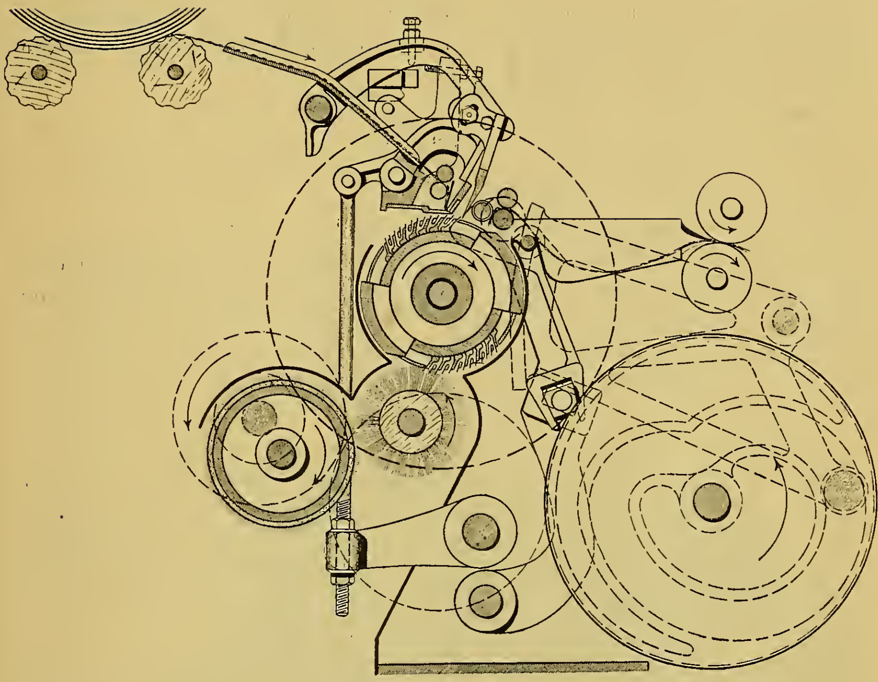

FIG. 92.

B. Gear required to make feed rolls deliver a certain length.

First carry out the above calculation, omitting the change gear. Second, divide the length which must be delivered by this result.

Example:—Find the gear needed to make the feed rolls deliver .215 inches per nip.

$$\frac{1 \times 3.1416}{5 \times 38} = .01653$$

$$\frac{.215}{.01653} = 13.007$$

Answer, 13 teeth.

II. TOTAL DRAFT.

The total draft is calculated from the lap rolls to the calender rolls in the coiler in the usual manner.

Example:—Find total draft, using the gearing shown in Fig. 86, assuming that the feed-change gear has 20 teeth and the change gear on coiler connecting shaft 75 teeth.

Total draft =

$$\frac{47 \times 55 \times 22 \times 38 \times 5 \times 60 \times 16 \times 16 \times 2 \times 4}{35 \times 20 \times 23 \times 20 \times 1 \times 75 \times 16 \times 16 \times 11} = 19.52$$

III. Draft in Draw Head.

Example:—Assume that front-roll change gear has 37 teeth, and that back-roll change gear has 44 teeth, the others being the same as in the drawing.

$$\text{Draft} = \frac{44 \times 50 \times 45 \times 20 \times 11 \times 8}{16 \times 45 \times 37 \times 43 \times 4 \times 9} = 4.24$$

IV. Weight of Sliver Delivered.

All the calculations relating to the weight per yard of the sliver and lap, and to the draft may be made on a comber exactly as on a drawing frame. It is, however, a good plan to arrange the sizes of the change gears according to the principles previously stated, and from the total draft thereby secured to calculate the weight per yard of the sliver delivered. Consequently one calculation only is given here.

Example:—Find the weight per yard of the sliver delivered, if the weight per yard of each of the laps fed is 280 grains, the draft is 30.32, and the waste 20 per cent.

$$\text{Weight per yard of sliver} = \frac{280 \times 6 \times .80}{30.32} = 44.33 \text{ grains.}$$

V. Production.

The production is calculated from the surface speed of the coiler calender rolls. An allowance of about 5 per cent. should be made for stoppages.

Example:—Find the production of a comber for 58 hours if the sliver being produced weighs 40 grains per yard, the speed of the machine is 90 nips, and the change gear on the coiler-connecting shaft is the same as before, namely, 75.

Production in pounds =

$$\frac{90 \times 60 \times 2 \times 3.1416 \times 60 \times 58 \times 40 \times .95}{75 \times 36 \times 7000} = 237.38$$

CHAPTER VII

FLY FRAMES

The machine used directly after the last drawing frame is the first of a series of machines classified as "fly frames." In the language of the mill they are also called "speeders" and "roving frames." The number of

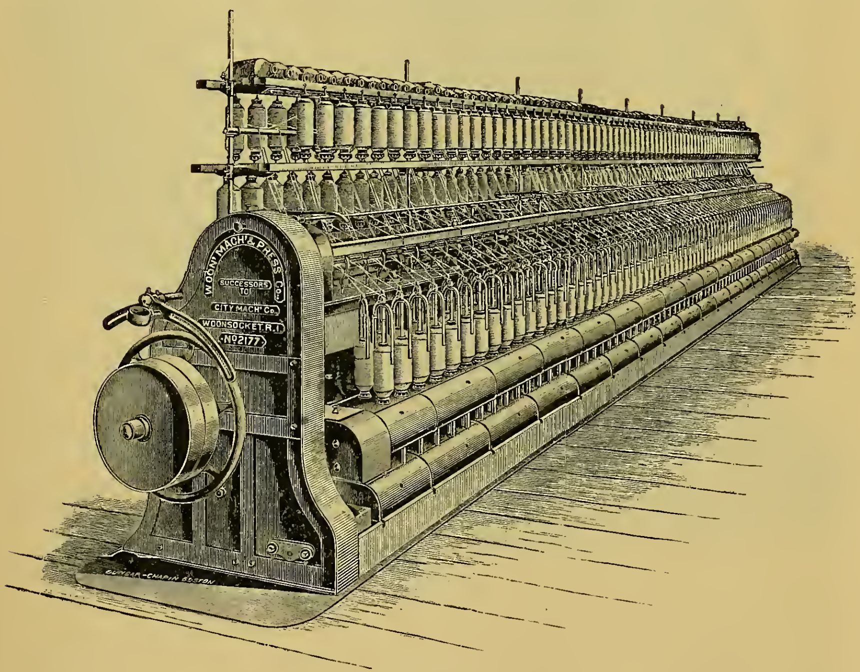

FIG. 93*a*.

this sort of frames used varies from two to five according to the fineness of the yarn to be spun. They are all very nearly alike in construction, differing from one another mainly in size. The first is always called a "slubbing frame" or a "slubber." If two frames are used the second is merely called the roving frame. If three or more are used, the second is called an "intermediate," the third the "roving frame," or "second intermediate," and the fourth a "jack frame." A slubber receives the cotton in the form of a sliver from the drawing-frame cans, and delivers it in a

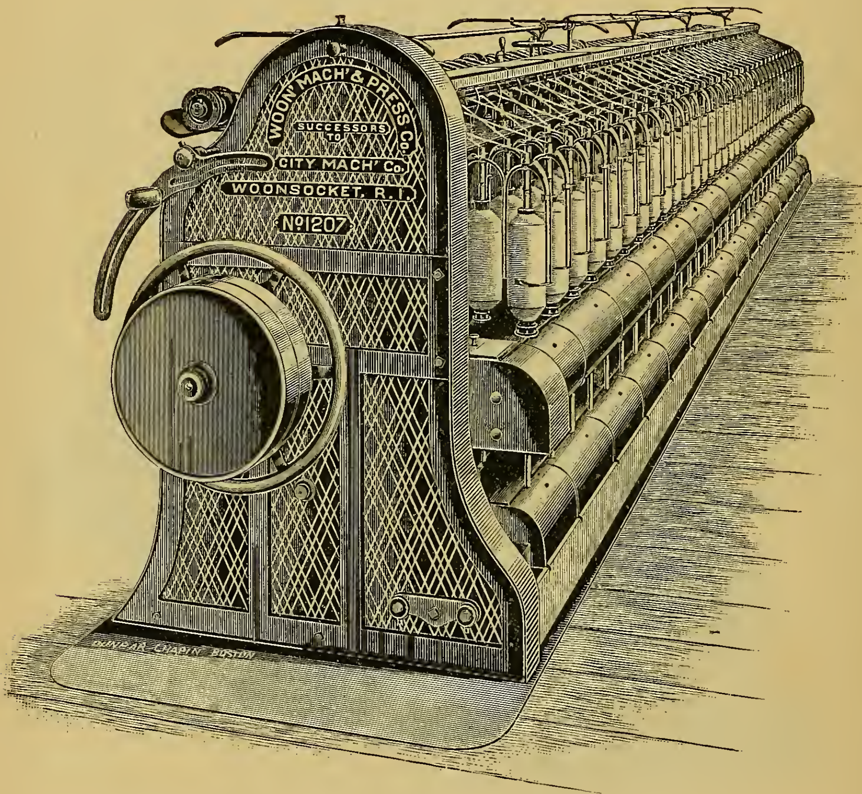

FIG. 93.

thinner strand coiled upon a cylindrical wooden barrel called a "bobbin." There must therefore be a slight difference between the feeding arrangement of a slubber and that of the fly frames which follow. So far as the working of the various fly frames is concerned they are all alike; they have exactly the same functions to perform. These duties are three, drawing out of the strand, the introduction of twist and winding upon a bobbin. The preceding machines have had the work of preparing the cotton for the process of spinning. They clean the cotton, attempt to make the sliver uniform in weight and thickness, and lay the individual fibres in an approximately parallel position. By the fly frames the size of the strand is gradually reduced. By the time it passes through the slubber it is so fine that the cohesion of the fibres is not sufficient to prevent the strand from breaking, therefore it has to be twisted slightly. Convenience in handling is more satisfactorily obtained by winding the strand on a bobbin than in any other way. The actual process is one of drawing and twisting, gradually to decrease the diameter of the strand of cotton, and at the same time to preserve sufficient strength to facilitate feeding to the machines without rupture. After emerging from the slubber the strand of cotton is called "slubbing," and that produced by the subsequent fly frames has the name of "roving."

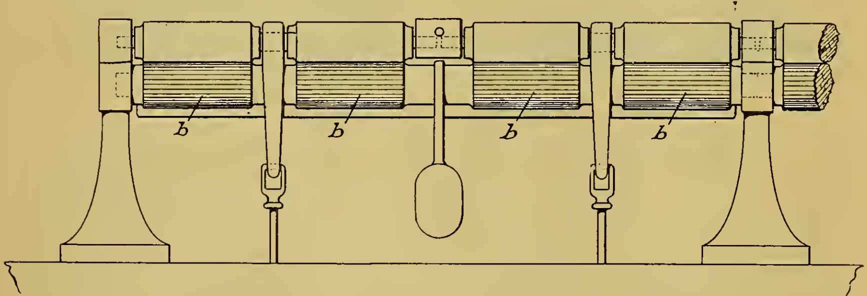

Fig. 94.

DRAWING

Fig. 93 is a front view of a slubber, and Fig. 93a is a front view of a roving frame. The cans of sliver are placed behind the frame, and each sliver is fed separately into the machine; there is no doubling. The slivers pass over a tin or wooden carrying roll shown, thence through guide holes in a strip of wood extending the whole length of the frame, and then they come within the action of three sets of drawing rollers. All the draft is introduced by these rollers on exactly the same principle as previously described. The rollers resemble closely drawing-frame rollers. The bottom ones are of steel made in short lengths and coupled together in the manner previously described. Their bosses are fluted longitudinally, but the pitch of the flutes is less and the bosses shorter than on a drawing frame. A front view of a portion of the drawing rolls of a slubber is shown in Fig. 94. These bosses can be seen at *b;* there are usually four between roller stands on a slubber. The roller stands which support the bottom rollers can well be understood from Fig. 95. The necks of the

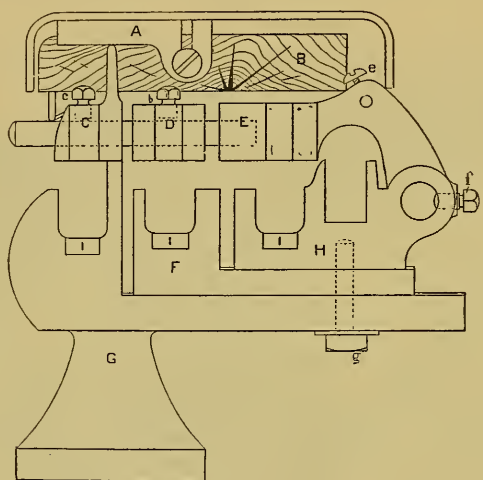

Fig. 95.

rolls rest in the steps I. The stand G is bolted to the frame of the machine and has attached to it the movable stands F and H, which support the middle and back rolls, respectively. By loosening the nut *g* the stands F and H can be moved forward or backward, thus regulating the distance between the different sets of rolls. B is a wooden clearer, to the bottom

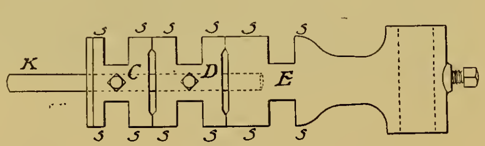

Fig. 96.

of which a strip of flannel is attached to catch what little fly may escape from the cotton. A is a metal clearer cover to which B is fastened.

The top rolls resemble very closely those of a drawing frame. They are invariably leather rolls, and each has two bosses, with the weight suspended from an arbor in the middle. The top rolls rest on the bottom rolls and are held in position by "cap bars." Side, front and top views of one cap bar are shown in Figs. 95, 94 and 96, respectively. The "nebs" C and D are movable upon the finger K, being held in position by the set screws shown. By reference to the top view it will be seen that the nebs are so shaped as to furnish shoulders S, against which the ends of the top rolls may rest. By adjusting the nebs the distances apart of the top rolls may be made to conform with those of the bottom rolls. The principles governing the setting of the rolls are practically the same as those on a drawing frame. The rolls need about the same attention that do drawing-frame rolls. They are, however, never varnished.

In addition to the types already described there is sometimes used on roving frames a roller with a ball-bearing shell. This has not, however, come into general use. A view of one is shown in Fig. 97.

Fig. 97.

In order to prevent wear of the top rolls in one place the guide rail through which the cotton passes before entering the rolls is given a traverse motion.

TWIST

On the frame under discussion we meet for the first time the principle of twist. The size of the strand by the time that it has passed through the slubber is so small and its strength so little that some twist must be introduced in order that the strand will not be broken. To twist any strand it is always necessary that one end or one point be held firmly, while the other end, or some other point, makes a turn out of the axial lines and is made to revolve. In this manner the strand is turned around on its own axis or twisted. Various mechanical means are employed, of which that in use of fly frames is probably the oldest. The strand of cotton, after leaving the front roller, first passes through a hole in the top of what is called a "flyer," then emerges from a hole in the side, is given a half turn around the top of the flyer, passes down through its hollow arm, is coiled three times around a presser finger and then is wound on a wooden bobbin. The flyer, presently to be described in detail, is made to revolve very rapidly. It therefore carries one end of the strand around while the other is held in the nip of the front roll. The result is the introduction of twist.

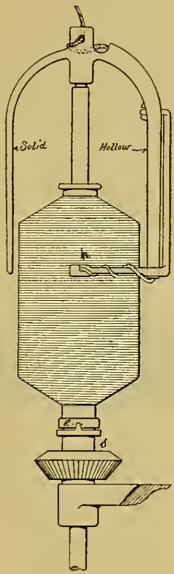

FIG. 98.

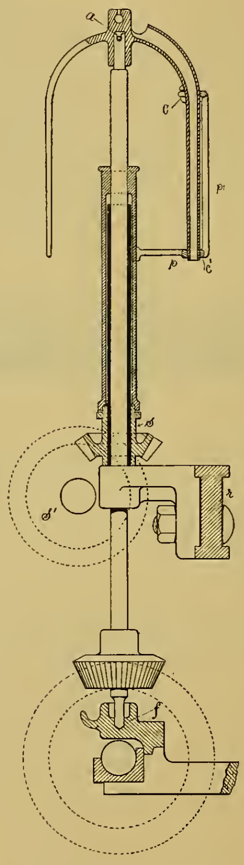

FIG. 99.

Full and sectional views of the flyer and its contiguous parts are shown in Figs. 98 and 99. The flyer is itself a device having two arms, one of them solid and the other hollow. The cylindrical part between the two arms is bored to fit the top of a long upright spindle. In Fig. 99 may be seen a pin *a*, which fits into a slot on the top of the spindle, thereby holding the flyer firmly in place. It therefore receives every movement that the spindle receives, being practically integral with it. Attached to the hollow arm of the flyer at the points C and C′ is an unevenly-balanced right-angular lever *p*, of which the vertical leg is the heavier. Around the horizontal part the cotton is wound before passing to the bobbin. The device is called a presser, and its function is to assist in making a tightly-wound bobbin in a manner to be hereafter described. Motion is of course given to the flyer from the spindle, so a description of the method of driving the spindle may well be given here.

Reference to Figs. 99, 100 and 101 will assist in understanding the

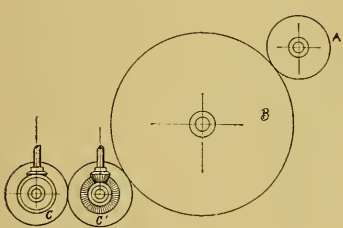

FIG. 100.

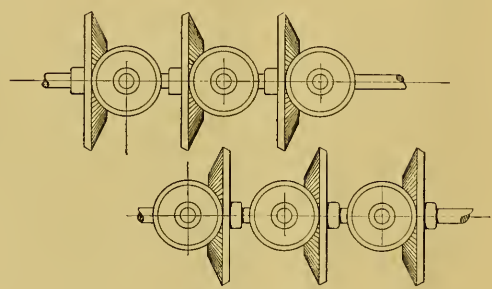

FIG. 101.

arrangement of spindle and its surrounding parts. Its lower end is pointed and rests upon a footstep *f*. At a point further up a hollow cylindrical casing, shown in heavy black, called a "bolster," surrounds and supports it. A bevel gear at the bottom receives power from another bevel on a long shaft running the whole length of the machine, known as a "spindle shaft." Since there are two lines of spindles there are two spindle shafts, parallel to each other, and driven one from the other. If one is driven directly from the other they must revolve in opposite directions. But since the spindles all must go in the same direction, the arrangement shown in Fig. 101 is adopted. One spindle shaft has its bevel gears on one side of the spindles, while the other has them on the opposite side. The spindle shafts themselves are driven from the main shaft of the machine by a short train of gearing in the head of the machine close to the driving pulley. The gearing is shown in Fig. 100, where A is a gear on the main shaft, B is a large carrier, and C and C′ are spur gears on the spindle shafts.

WINDING

In Fig. 99, surrounding the lower part of the bolster, may be seen a short sleeve S. On the bottom of this sleeve is formed a bevel gear, above which is a flange used to support the wooden bobbin barrel. A small lug *l* on this flange engages with a slot in the bottom of the bobbin as shown in Fig. 98. Since the sleeve may be made to revolve loosely around the bolster, on a portion of which it rests, it will be readily seen that the revolution of the bobbin barrel is entirely independent of any movement of the spindle. It is important to understand this point at the very beginning. Upon it depends the whole principle of winding the bobbin. It must be carefully noted, too, that not only is the bobbin driven independently of the spindle, but it is also driven positively, *i. e.,* not by the pull of the strand of cotton. The source from which the bobbin receives its power need not be considered until the principle on which the winding depends is set forth. The problem is an interesting one, and one the careful consideration of which is well worth the attention of all students.

The shape of a bobbin filled with cotton can be seen in Fig. 98. By forming a taper at each end it has been found that the various layers have no tendency to slip off, but on the contrary the bobbin can be handled pretty roughly without injury to the cotton. In order to form a bobbin of the shape shown it is necessary (1) to cause the bobbin to revolve faster than the flyer so as to pull the strand away from the flyer; (2) to give the bobbin an upward and downward movement for a distance which gradually decreases at both ends as the bobbin increases in size. Such in general are the two necessary operations, their mechanical performance and the theory underlying the former need considerable discussion in detail. Operation 1, which may be classed as the speed of the bobbin, will be discussed first.

The strand of roving passes directly from the hole in the flyer eye to the bobbin itself. The revolution of the flyer serves only to twist the roving without pulling it at all away from the front roller. The front roll is delivering all the time, hence some provision has to be made for disposing of this roving. Evidently if the bobbin revolves at the same speed as the flyer, although the roving is attached to the bobbin, it cannot take up any of the roving delivered by the front roll. The bobbin must revolve either faster or slower than the flyer in order that any winding whatever may take place. Again it is not only necessary that there be a difference between the speed of the bobbin and that of the flyer, but also that this difference be a certain amount, no more and no less. Consider, for example, that 800 inches of roving must be wound upon a cylindrical surface 5 inches in circumference. The number of coils that there would be upon the cylinder is 800 divided by 5, or 160. In other words, the cylinder must make 160 revolutions in order to take up 800 inches of roving. The problem on a roving frame is similar to this, yet at the same time somewhat different. The front roll may deliver 800 inches of roving and the bobbin may be 5 inches in circumference, but at the same time the flyer is carrying the roving around about 800 times. It is not sufficient, therefore, that the bobbin make, as in the previously supposed case, 160 revolutions. It must go either as fast as the flyer and 160 revolutions faster, or it must go 160 revolutions slower than the flyer. In the former case the bobbin would pull the roving away from the flyer, and in the latter case the flyer would be allowed to wrap the roving around the bobbin. The important thing to be borne in mind is that if the bobbin and flyer go at the same speed there can be no winding. Since it is possible to perform winding either by having the bobbin go a certain amount faster or slower than the flyer, machines have been constructed on both principles. The two types of frames have been distinguished from each other by the names "bobbin lead" and "flyer lead." At present, however, all roving frames are constructed on the bobbin-lead principle. Practical considerations are the cause. The excess speed must always be equal to the surface speed of the front roll, divided by the circumference of the bobbin. As the bobbin increases in size its circumference of course increases. If this size increases from 5 inches to 8 inches the excess speed will evidently decrease from 160 revolutions to 100 revolutions. The problem of driving the bobbin may be sub-divided into two: (1) that of making the bobbin revolve faster than the flyer, and (2) that of decreasing the excess speed after the winding of each layer. Many simple methods might accomplish the first; the mechanical solution of the second is far more difficult. The manner in which both are actually performed on a roving frame will now be described.

The loose sleeve *s* in Fig. 99 is supported by a flange on the casting, and receives its motion from the gear *s'* on a long shaft called a bobbin shaft. These shafts, of which there are two, resemble the spindle shafts in that they are parallel to each other, and extend the whole length of the front of the machine. In Fig. 99 the bolster may be seen to be bolted to a rail *r*. This rail is called a "bolster rail." To it the bearings for the bobbin shafts are also fastened. The exact method is not shown in the drawing. It will be seen, however, that the bobbin shaft and the sleeve carrying the bevel gear *s* are not in the same plane. In such a case it is necessary to use gears as shown in Fig. 102. Such gears are called skew

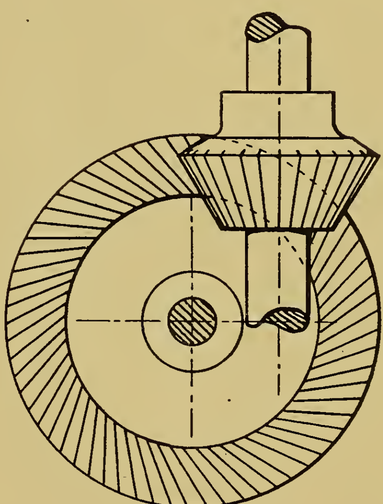

Fig. 102.

bevels. The bolster rails with their connected parts form what is called a carriage. This carriage is given an upward and downward motion, and to the bobbin shafts power of course has to be transmitted from the fixed frame work of the machine. Several methods of doing this are in use, of which the following, shown in Fig. 103, is new and very good. On the end of one of the bobbin shafts is a bevel gear that meshes with one on the upright shaft. The gear G is not fast upon the upright shaft, but is held in the carriage, and of course moves up and down with it. It slides up and down the upright shaft, being made to revolve with it by a key, which fits in the key slot *k*. On the top of the upright shaft is a bevel meshing with another on an oblique shaft. Another bevel on the other end of this shaft meshes with a larger bevel A, known as the last gear of the differential motion. A revolves loosely around the main shaft of the machine. The gear A gets its motion partly from the main shaft through gearing to be described hereafter, and partly through the gear marked 30,

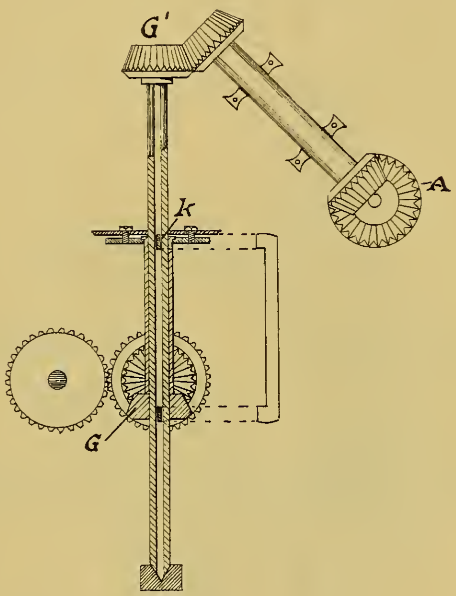

FIG. 103.

seen in Fig. 104. Tracing the train of gearing back through the pair of belt-connected cone drums we finally reach a gear on the end of the main shaft.

Differential Motion.

It will be easily seen that gearing can be conveniently arranged so that the bobbin may have at the beginning of its winding any desired speed. This entire speed of the bobbin should, however, always be thought of as being made up of two parts, as has been previously stated. The first is equal to speed of the flyer, while the second is the excess speed, and varies as the bobbin increases in size. It is of course desirable

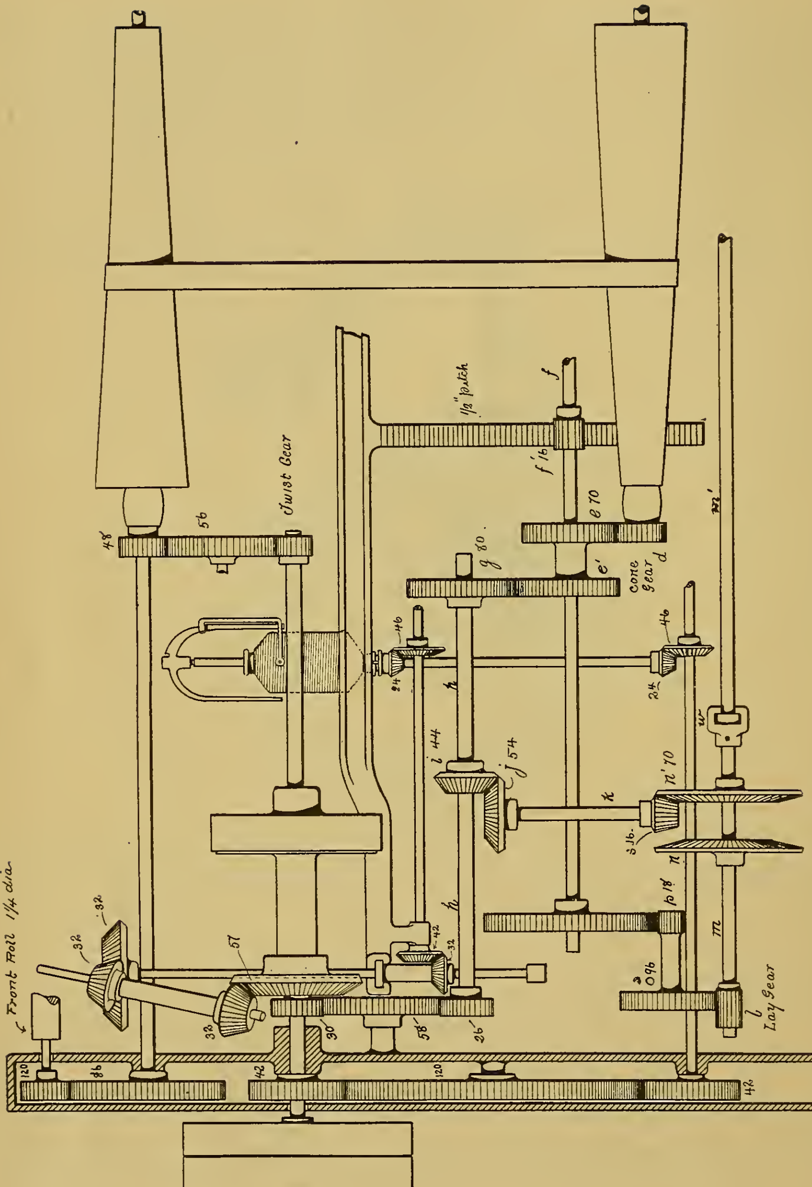

FIG. 104.

that these two parts of the bobbin's speed come from different sources. In this manner they can more easily be controlled and regulated. Before reaching the bobbin, however, the power from the two separate sources may well be combined. Such conditions actually exist on all fly frames. The mechanism for combining the power from the two sources is called the "differential motion." Besides the combination of the two powers there must be a chance for one of them to be varied, while the other

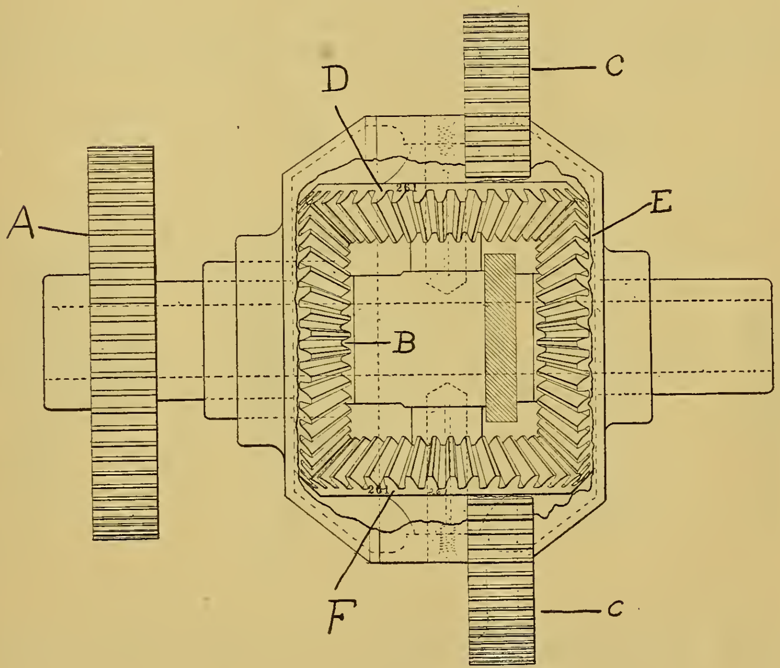

FIG. 105.

remains constant. There are several types of differential motions, all of which conform to these conditions. The oldest and best known type is shown in Fig. 105. It is called Holdsworth's motion. It consists essentially of four bevel gears of equal size and an encircling spur gear. The main shaft of the machine is shown by the two parallel horizontal dotted lines. To the jack shaft the gear E is made fast. The gears D and F, which mesh with E, are carried on studs held in castings which form a part of the large spur gear C. C is loose upon the jack shaft. The bevel gear B is also loose upon the jack shaft, but entirely independent of C. A is on the same sleeve with B, and by it power is transmitted to the bobbins. It may now be seen that B, and consequently A, receives its power through the spur gear C, and from the jack shaft through the medium of E, F and D. C receives its power from the cone drums previously mentioned in a manner to be described hereafter. The power received by B through E, F and D must remain constant, since E is fast upon the main shaft of the machine. That coming through C is variable. If C were held stationary, while all the other gears, including those which connect A with the bobbins, moved at their normal speed, the bobbins would go

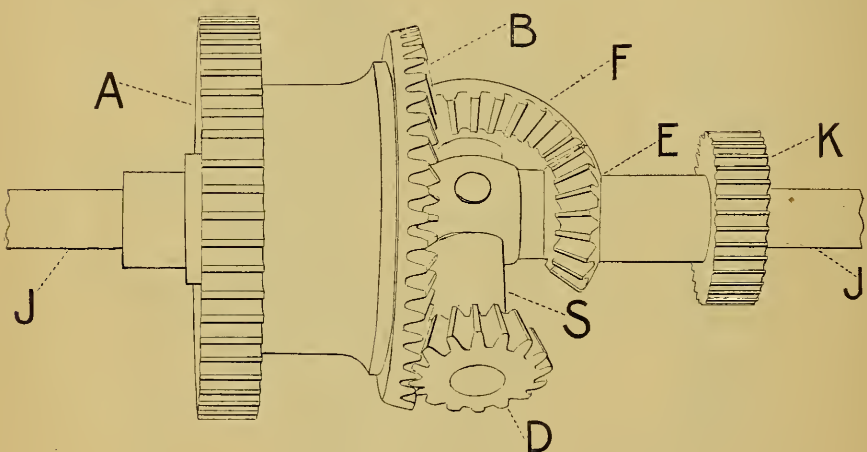

Fig. 106.

just as fast as the flyers. On the other hand, if E were stationary and C revolved at its normal speed, the speed of the bobbins would be equal to the surface speed of the front roll divided by the circumference of the bobbin. In actual practice C and E both revolve. The combined power that is then transmitted to the bobbins is just sufficient to make them take up all the roving that is delivered by the front roll. When the machine is running, the gear C and the gears A and B revolve in the opposite direction from the jack shaft.

A second style of differential called Tweedale's is shown in Fig. 106. The jack shaft is there represented by J. A link is formed in the jack shaft for the reception of the short shaft S. S passes through the jack shaft, being suitably held in place. It may revolve on its own axis, while at the same time it will always be carried around bodily by the jack shaft. Fastened to the shaft S are the two bevel gears F and D. D meshes with the large bevel B, while F meshes with E only. B is sufficiently large to allow F to revolve without coming in contact with it. The spur gear A is cast with B, and the two combined are called a bell wheel. A transmits power to the bobbins. A and B revolve loosely around the shaft J. E also is loose upon J and is compounded with the spur pinion K. K receives its power from the cone drums. In this style of motion the jack shaft and the gears which are loose upon it all revolve in the same direction.

In Fig. 107 is shown Daly's differential motion. In this motion all

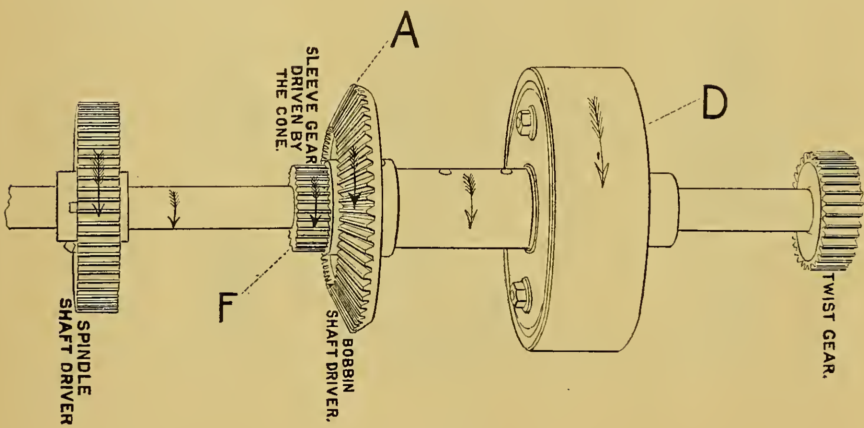

FIG. 107.

the gears are spur gears. Fig. 108 shows the motion separated so that all the gears may be seen. Made fast to the jack shaft is a large internal gear D. Running loosely on the shaft is a sleeve carrying the gears E and F. Surrounding this sleeve is another, which has the large bevel gear A formed on one end of it, while on the other it is formed into a disc. To the disc are bolted two studs which serve as axes for the two gears C and C′. C and C′ mesh with E and also with the internal gear D. F receives its motion from a train of gearing connected with the cone drums. The bevel A transmits power to the bobbins. In this motion all parts revolve in the same direction.

Several other styles are in use, but the three above mentioned are those most commonly seen in this country.

All these so-called motions are constructed on the principle of what is called epicyclic trains of gears. An epicyclic train is one in which one of the gears is carried bodily around the circumference of another. In Holdsworth's the gears D and F are carried around E by the revolution of C. In Tweedale's the gears F and D are carried around E, since their shaft is carried by the jack shaft. In Daly's the internal gear D meshes with C and C′ and carries them around E. For convenience in explanation, the part of the mechanism which carries one gear around the

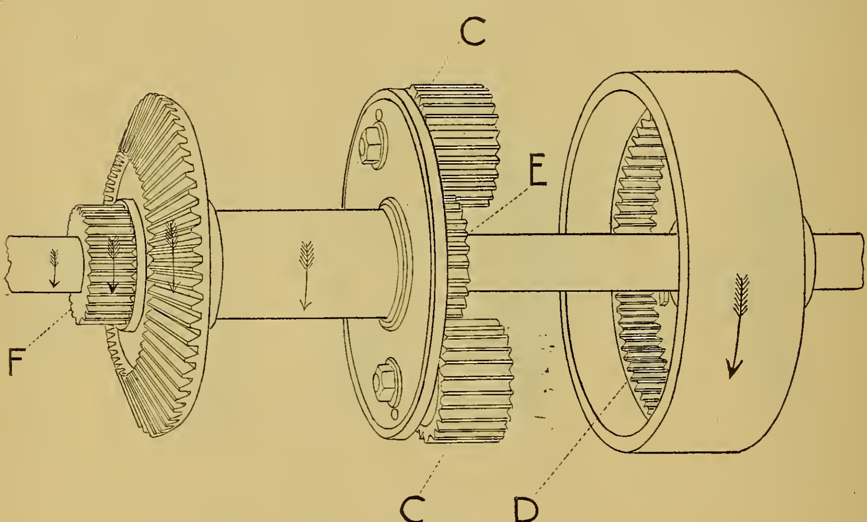

FIG. 108.

periphery of another may be called the "arm." In any of the motions one could readily determine the speed of the arm by the usual methods for finding the speed of any gear. Moreover, it is possible to find the speed of one other gear in the motion. In all three types shown this second gear of which it is possible to find the speed is the one marked E. In Fig. 105 E has the same speed as the jack shaft; in Figs. 106 and 107 its speed may be figured from some gear or shaft whose speed may be known. In each the gear E may be called the "first gear" in the motion. The bobbin driving gear A is in Figs. 105 and 106 compounded with another gear B, and of course has the same speed as B. B may be called the "last gear" in the motion.

In Fig. 107 the gear A is compounded with the disc carrying C and C′. This disc, therefore, corresponds to the last gear in the motion. To find the speed of this last gear we cannot proceed in exactly the same way as in the case of a straight train of gears. A general formula may be derived which will apply to all differential motions constructed on the epicylic principle. It is only necessary to know the speed of the arm and the speed of the first gear. The action of the arm always has some effect upon the speed of the last gear in the motion. If we subtract the speed of the arm from the speed of the last gear or from the speed of the first or any other gear in the motion, we have the speed of the respective gear uninfluenced by the arm; *i. e.*, as it would be if the train of gearing were of the ordinary type. In the ordinary type of gearing where gears mesh one with another without the intervention of a carrying arm, the value of the train is defined as the ratio between the speed of the last gear and the speed of the first gear, *e. g.*, if the first gear of a train makes one revolution and the last seven, the value of the train is $7 \div 1$ or 7. Therefore,

let $t =$ value of the train in any differential motion.
$l =$ speed of last gear.
$f =$ speed of first gear.
$a =$ speed of arm.

Then speed of last gear if influence of arm be removed $= l—a$ and speed of the first gear under the same condition $= f—a$.

$$\text{Consequently } t = \frac{l—a}{f—a}$$

$$\text{or } ft—at = l—a$$

$$\text{whence } l = a—t\ (a—f).$$

If this formula is too abstruse as written it may be expressed in words as follows: The speed of the last gear in a differential motion equals the speed of the arm, minus the product obtained by multiplying the ordinary value of the train by the difference between the speed of the arm and the speed of the first gear.

We can in the case of any differential motion obtain values for all the letters in the above formula except l, and can thereupon solve for l. It is necessary in solving to note carefully whether the speeds are positive or negative, *i. e.,* whether the gears whose speeds the letters represent revolve in the same direction as, or in the opposite direction to the first gear, whose speed we may for convenience consider to be positive.

We shall now apply the formula to Holdsworth's motion. In this case B is the last gear, E is the first gear and C is the arm. The value of the train is found by considering that the gearing is of the ordinary sort, and then by finding the speed of the last gear while the first makes one revolution. B, D, E and F are all of the same size; hence for one revolution of E, B would make one revolution, but in the opposite direction. The value of the train t is therefore -1. We may then modify the general formula for this particular motion by substituting (-1) in the place of t, at the same time considering whether the first and last gears and arm go in the same direction. As a matter of fact, the arm C revolves in the opposite direction to E, as does B, consequently a is negative.

Substituting in (1),

$$
\begin{aligned}
l &= -a - (-1)(-a-f) \\
&= -a + (-a-f) \\
&= -2a-f \\
\text{or } l &= -(2a+f).
\end{aligned}
$$

Therefore, the speed of A equals twice the speed of C, plus the speed of E. The minus sign simply indicates that the direction of revolution of the last gear is opposite to that of the first.

Suppose, for example, that C has a speed of 9, that the speed of the jack shaft is 300. Then the speed of the last gear B equals 300 plus two times 9, which equals 318. Then knowing that 318 is the speed of A we can easily calculate the speed of the bobbins themselves.

Formula (1) applies with equal force to Tweedale's motion. The jack shaft itself is the arm, E is the first gear, B is the last gear, and the value of the train is $\frac{E \times D}{F \times B}$, the letters of course indicating in this case the number of teeth in the gears. t is positive. All values for the letters in formula (1) are positive; consequently the substitution may be made very easily. E has 18 teeth, F has 30, D has 16, and B has 48. Assume the speed of the jack shaft to be 310, and the speed of E to be 262. Then the speed of B equals $310 - \frac{18 \times 16}{30 \times 48}(310 - 262) = 300.4$.

In applying formula (1) to Daly's motion the only point requiring discussion is the value of the train. The value of the train t is the speed of the sleeve carrying A while the first gear E makes one revolution. The case is slightly different from a straight train of gears, since even when the arm is stationary two influences are brought to bear upon the carriers C and C′, one by E and the other by the internal gear D. The proportionate amount of the whole power that is exerted by E may be found by dividing the speed which it would give to the carriers if the gearing were of the ordinary type by the sum of this speed and the speed which the internal gear would give to the carriers under the same condition. That is, the value of the train

$$t = \frac{\frac{E}{C}}{\frac{E}{C} + \frac{D}{C}} = \frac{E}{E + D}$$

E and D representing the number of teeth in the gears. Knowing the values for a, the speed of the jack shaft; f the speed of E; and t the value just derived, substitution in the formula

$$l = a - t\,(a - f)$$

is not difficult.

In actual practice the gear E has 24 teeth, C has 25, and D 80. Applying formula $t = 28 \div (28 + 80) = \frac{28}{108} = \frac{7}{27}$. Therefore, the value of the train is $\frac{7}{27}$. Then by substitution in formula (1) l equals a minus $\frac{7}{27}$ of (a minus f). If we suppose that the speed of the first gear is 100 and the speed of the internal gear D is 300, then since D corresponds to the arm, and E to the first gear, the speed of the last gear or the sleeve carrying the carriers and the bobbin wheel will be $300 - \frac{7}{27}$ of $(300 - 100)$, which equals 248.15.

From the bobbin wheel of any differential motion it is easy to find the speed of the bobbins in the usual manner.

Decrease in Bobbin Speed.

We have next to consider the manner in which the speed of the bobbins is decreased at the end of the winding of each layer. It has been mentioned that one part of every differential motion receives its power through the medium of a pair of long cone drums. The upper one or driving cone is connected by a train of three gears with the jack shaft of the frame, and the two cones are connected by a belt. The exact arrangement may be seen in Fig. 104. This belt may be moved from one end of the cones to the other. The large diameter of one cone is directly over the small diameter of the other, so that by shifting the belt the speed of the driven cone may be varied. The belt is moved along automatically after each layer of roving has been wound. Since the driven cone controls the bobbin's excess speed, its speed must decrease in proportion to the increase in the diameter of the bobbin, *i. e.*, must vary inversely as the diameter of the bobbin. To make this possible merely by shifting a belt a fixed distance at the end of each layer, the cones have to have very carefully designed outlines. It is necessary to assume diameters for the full and empty bobbin and the fixed sum of the corresponding diameters of the cones. From these data and by the assumption of diameters of the bobbin at intermediate points, the necessary diameters of the cones can be calculated. In the algebraical proof which follows it has been aimed to use as simple mathematics as possible. No sufficiently general proof can be given by the use of arithmetic merely. The proof follows:

Let E = diameter of empty bobbin.
" F = " " full bobbin.
" L = " " large end of driving cone.
" S = " " small end of driving cone.

Since large end of driving cone is to be same size as large end of driven cone:

L = diameter large end driven cone.
S = diameter small end driven cone.

Let D = sum of the diameters of the two cones.

This sum must of course always be constant, since the cones are to be connected by a belt of a certain length.

Then $\frac{L}{S}$ = speed of bottom cone when belt is on one end.

And $\frac{S}{L}$ = speed of bottom cone when belt is at other end.

It has been previously stated that the speed of the bottom cone varies inversely as the diameter of the bobbin.

$$\text{Hence } \frac{L}{S} : \frac{S}{L} :: F : E$$

$$\text{Or } \frac{LE}{S} = \frac{SF}{L}$$

$$L^2E = S^2F$$

$$\text{Whence } L = S\sqrt{\frac{F}{E}} \qquad (1)$$

$$\text{And } S = L\sqrt{\frac{E}{F}} \qquad (2)$$

$$\text{Also } L + S = D \text{ and } S = D - L$$

The above mathematical proof is necessary in order to arrive at the correct sizes for the diameters of the cones at the ends. The first proportion depends upon the fact that the speed of the bottom cone is always in inverse proportion to the diameter of the bobbin. The diameters of the cones at intermediate points may be determined by proportion by a similar course of reasoning. In addition to the above letters

Let I = diameter of top cone at any intermediate point.

Let I' = diameter of bottom cone at corresponding point.

At which time

Let b = diameter of bobbin.

Then a proportion depending upon the same principle previously stated may be formed as follows:

$$\frac{L}{S} : \frac{I}{I'} :: b : E$$

$$\frac{LE}{S} = \frac{Ib}{I'}$$

$$I = \frac{I'LE}{bS}$$

$$I + I' = D \text{ and } I' = D - I$$

Let us now take a concrete example to illustrate the result of the above somewhat abstract proof.

Suppose that $E = 1\frac{3}{4}$ inches

$F = 6$ inches

$D = 10$ inches

$$\text{Then from (1) } L = S\sqrt{\frac{\frac{24}{4}}{\frac{7}{4}}}$$

$$\text{Or } L = (10 - L)\sqrt{\frac{24}{7}}$$

$$\text{Or } L = 18.51 - 1.851\,L$$

$$\text{Whence } L = 6.5 \text{ inches}$$
$$\text{And } S = 3.5 \text{ inches}$$

These would then be correct diameters for the ends of cones. If D, E and F had different values, the sizes of the ends would be different. To exemplify the formulas for the diameters at intermediate points it is

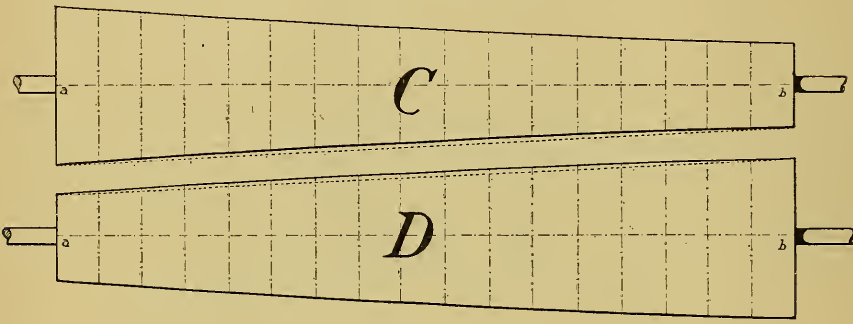

FIG. 104a.

merely necessary to assume several different values for b to represent different diameters of the bobbin. We shall first assume that the diameter of the bobbin has been increased merely by ¼ inch.

$$\text{So let } b = 2 \text{ inches}$$

$$\text{Then } I = \frac{I' \times 6.49 \times 1.75}{2 \times 3.5}$$

$$\text{Or} \quad I = (10 - I)\ 1.622$$
$$I = 16.22 - 1.622\ I$$
$$\text{Whence } 2.622\ I = 16.22$$
$$\text{And} \quad I = 6.18 \text{ inches}$$
$$\text{And} \quad I' = 3.82 \text{ inches}$$

If this same operation be carried out with a different value for b between 1¾ and 6 inches, as many diameters of the top and bottom cones may be obtained as is desired. The larger the number, the more accurate will be the outline of the cones to be made.

In the following table diameters have been worked out corresponding to 18 different diameters of the bobbin. The bobbin diameters differ from one another by ¼ inch:

Diameter of Driving Cone, Inches.	Diameter of Driven Cone, Inches.	Diameter of Bobbin Cone, Inches.
6.50	3.50	1.75
6.18	3.82	2.00
5.91	4.09	2.25
5.65	4.35	2.50
5.41	4.59	2.75
5.20	4.80	3.00
5.00	5.00	3.25
4.81	5.19	3.50
4.64	5.36	3.75
4.48	5.52	4.00
4.33	5.67	4.25
4.19	5.81	4.50
4.06	5.94	4.75
3.94	6.06	5.00
3.82	6.18	5.25
3.71	6.29	5.50
3.60	6.40	5.75
3.50	6.50	6.00

These above diameters are the ones which are needed at different stages in the building of the bobbin. It is now not hard to see what the shape of the cones must be. To make this clear, we shall arrange these various diameters side by side at a fixed distance apart, and have them bisected by the lines a — b in Fig. 104a.

Now, if the ends of these diameters be joined the result is the outlines shown at C and D. C, then, represents the shape of the top or driving cone, and D that of the bottom or driven cone. The driving cone will be seen to be concave, while the driven is convex. In the small drawing shown the variation of the outline from a straight line is not very pronounced. Dotted lines show straight lines, and with them comparisons may be made.

Traverse of the Carriage.

The moving of the cone belt through the fixed distance after each layer has been wound is done by the so-called "builder motion." The working of the builder motion is, however, so intimately connected with the performance of the second operation in winding, namely, the upward and downward traverse of the bobbin for the required distance that a description of the manner of giving the bobbin its traverse may well be described first. Reference to the view of the entire gearing of a slubber in Fig. 104 will clarify the description. There the connection between the main shaft and the bottom cone may be easily seen. On the shaft with the bottom cone is a pinion *d*, which meshes with the compound carrier *e e'*, which revolves loosely around the long "lifter shaft" *f*. *e'* meshes with the large gear *g* on the end of a short shaft *h* called the "spider shaft." A bevel gear *i* on the spider shaft meshes with the bevel *j* on the upright shaft *k*. On the lower end of *k* is the pinion *s*, called the "strike pinion," which meshes with either one or the other of the two large bevel gears *n* and *n'*. *n* and *n'* are called "strike bevels," and are fast upon the short shaft *m*. On the end of *m* is the pinion *l* called the "lay gear." *l* meshes with the gear *o* on the stud with which is the pinion *p*, which meshes with a large gear *q* on the previously-mentioned lifter shift *f*. *f* extends the whole length of the machine, and has, at a certain distance apart, several pinions which mesh with racks attached to the carriage. By the movement of this train of gearing the carriage gets its motion. At the end of the winding of each layer the strike pinion is thrown out of gear with *n* and into gear with *n'*, or vice versa, in a manner to be described. The direction of motion of the carriage will easily be understood to depend upon which bevel *s* happens to be meshing with. The weight of the carriage is counterbalanced by several heavy weights attached to the carriage by chains.

The Builder Motion.

The function of the motion is to make all the changes that occur after a layer of roving has been wound. These changes are three in number: (1) the changing of the direction of motion of the carriage; (2) the moving of the cone belt a certain distance; (3) the shortening of the traverse of the carriage. These operations will be taken up in their order. Fig. 109 shows the American style of builder motion and its relation to the rest of the machine. Two rectangular plates P and P' are threaded upon a screw S. The inclination of the thread on one-half of the screw is opposite to that on the other. Consequently when the screw is revolved the plates P and P' are moved either towards or away from each other, depending upon the direction of revolution of the screw. The screw is supported by a casting made fast to the carriage. The screw and the plates, therefore, move up and down with the carriage. The two plates occupy the position shown in the figure, and form a bearing surface for the so-called "tumbling dog" D. D is keyed to the upright shaft E. Fastened to E is the horizontal disc F, downward from which extend two pins shown at *f* and *f'*. A finger H is held in contact with *f* or *f'* by

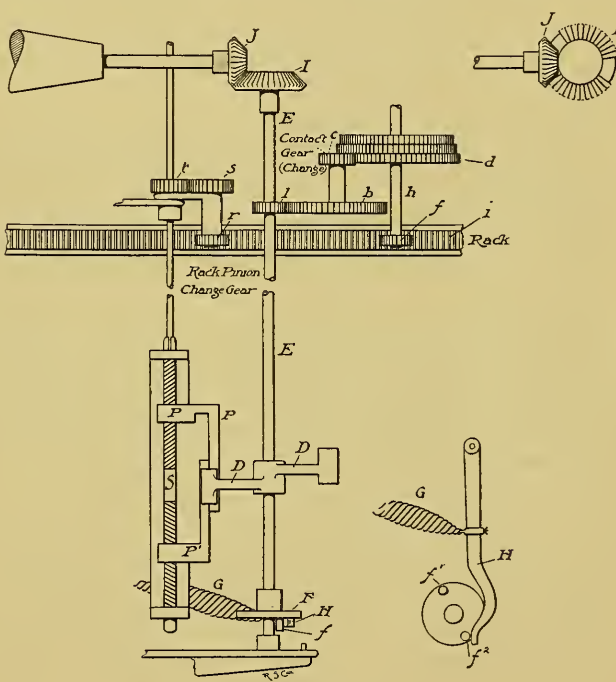

FIG. 109.

the spring G. In this manner one part of the tumbling dog is held against the plane surface formed by the plates P and P'. Since the plates move up and down with the carriage the surface against which one arm of D rests will at one time be removed entirely from D. D will then slide off the bottom or the top, as the case may be. As D will then have nothing against which to rest, the pressure of the spring G will tend to revolve the upright shaft E. The pressure of G is not strong enough, however, to turn E through a half revolution. Therefore there is on the top of E a gapped bevel gear I, capable of meshing with the gear J on the end of the top cone. When the builder motion is not working, or better when the upright shaft E is not revolving, the gear J fits into the gap in the gear I without touching the teeth in I. When, however, the shaft E is partially revolved by the pressure of the spring G the teeth in I are brought into contact with those in J, and E is thereby turned quickly through half a revolution. It is of course prevented from moving farther by the pressure of the other part of the tumbling dog against the plates P and P′. The finger and spring act also as a sort of brake, preventing the tumbling dog from striking the plates too forcibly.

It has now been shown that the tumbling dog makes a half revolution at the end of the winding of each layer. The effect of this movement is as follows: On the bottom of the shaft E is a cam which fits into an elliptical slot in a casting fastened to the long rod *m′* seen in Fig. 104. In the end of *m′* is formed a swivel joint W to connect the rod *m′* with the moving shaft *m* (Fig. 104). When E makes its half revolution the cam presses against one side of the elliptical slot in the end of *m′*, thereby moving *m′* either to the right or to the left, and with it the shaft *m* and the gears thereon. One of the strike bevels is in this way thrown out of contact and the other into contact with the strike gear *s*. The lay gear *l* is made so wide that it is never out of gear with the gear *o*. So much for the reversal of the direction of motion of the carriage.

In Fig. 109 near the top of the shaft E is a pinion *c* which meshes with a gear *b*, which is carried in a movable arm. On the stud with *b* is a pinion *c* meshing with a gear *d* on a second upright shaft *h*. A small pinion on this shaft engages with a long rack *i*. Attached to the rack *i* is a fork for moving the cone belt. *i* is called the cone rack. It will be readily understood that every partial revolution of the shaft E will set in motion the just-mentioned train of gears, thereby moving the cone rack a certain distance. In this manner the cone belt is moved along the cones at the end of the winding of each layer.

The third change, namely, the shortening of the traverse of the carriage, is accomplished as follows: Fixed to the frame work of the machine is a casting carrying the two small gears *r* and *s* seen in Fig. 109. *r* meshes with the cone rack. *s* is on the stud with *r*. *s* meshes with *t*. The screw S has an upward extending shaft with a square cross section.

This shaft protrudes through a square hole in the gear *t*. It is free to move up and down through this gear when the screw is carried up and down by the motion of the carriage. When the cone rack is given its lateral motion in the manner previously described the gears *r*, *s* and *t* are set in motion. The screw is thereby partially revolved on its own axis. The partial revolution of S moves the plates P and P′ a short distance towards each other. Therefore at the next traverse of the carriage, upward or downward as the case may be, the tumbling dog will not have such a long surface upon which to bear. It will make its half revolution a little sooner. Since both plates P and P′ are moved towards each other the same distance, the effect upon the winding of the bobbin will be that each layer will be shorter than the preceding one, and that the length of the layer will be shortened both at the top and at the bottom.

It will be observed that all the changes now described occur at the same time, *i. e.*, when the carriage reaches the highest and lowest points of its traverse.

STOP MOTIONS

All fly frames are fitted with stop motions, of which those shown in Figs. 110, 111 and 112 are good samples. One device is used to stop the machine when the bobbins have reached a certain size. It is possible to make adjustments so that the full bobbins may be of any desired size not greater than the space between the two arms of the flyer. It is, however, customary to make a bobbin as large as the flyer will allow, so that there is scarcely ever need of adjustment. The other stop motion is to stop the machine if by some chance the carriage should move too far up or too far down. Their working is as follows: In Figs. 110 and 111 may be seen a weighted lever N fulcrumed at *n* to the frame work of the machine. One arm of the lever extends upward, and by means of a ring formed in the end engages with a knob *b* on the rod S attached to the belt shipper. Pivoted to the lever N is a flat rod, near the end of which is formed a notch *c*, clearly seen in Fig. 111. When the parts are in their normal position while the machine is running the notch engages with the casting *q* shown in cross section, thereby holding the weighted lever N in approximately the position shown in Fig. 110. The top of N is prevented from pushing against the belt-shipper rod S. Attached to the cone rack, only a part of which is shown in Figs. 110 and 111, is an adjustable casting *c*′.

It may be noted in passing that the cone rack is some distance in front of the weighted lever, although they appear to be nearly connected in the drawing. During the building of a bobbin the cone rack moves in the direction shown by the arrow. The projection seen on the casting c' gradually approaches the curved part of the rod x. It finally raises the rod, disengaging the notch from the casting q. The weight W then swings the lever N around the fulcrum n. The belt shipper is thereby moved to the left and the belt transferred from the fast to the loose pulley. The parts then occupy the position shown in Fig. 111.

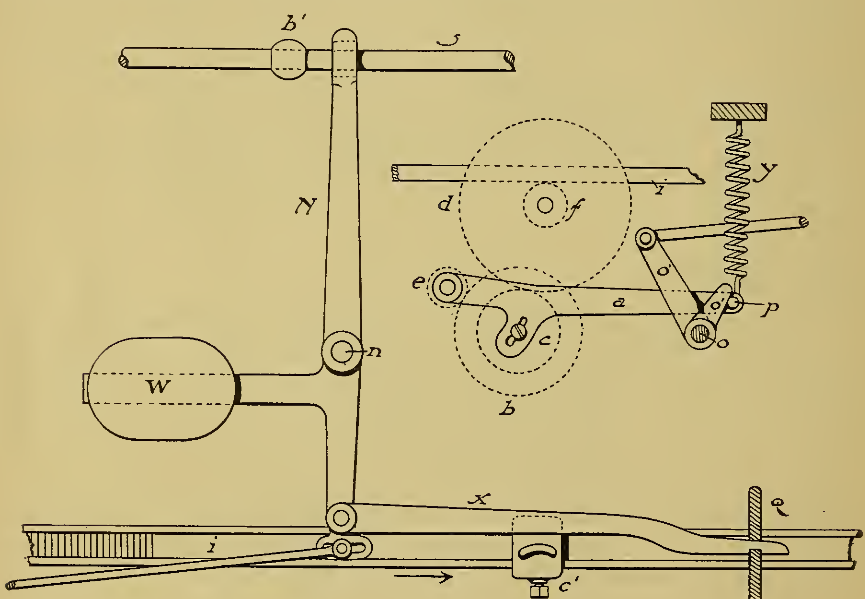

FIG. 110.

By the stop motion is actuated other mechanism, namely, that seen in small drawing at the right in Figs. 110 and 111. These detailed drawings represent top views of the gearing which connects the tumbling-dog shaft with the cone rack. The function of the gearing has been described, and the same letters are here used as far as possible. e is the pinion on the tumbling-dog shaft. b and c are on the same stud and are carried on the arm a, which may swing around the tumbling-dog shaft. c meshes with d, which, together with f, is on an upright shaft, as previously described. *f* meshes with the teeth in the cone rack. A spring *y* attached to the frame work of the machine and to the arm *a* holds *c* in gear with *d*. Pivoted on the stud *o*, which is attached to the frame work of the machine, is a lever *o′*. *o′* is connected to the weighted lever N by the connecting rod, a part of which is shown in each drawing. One arm of the short lever rests against a pin *p* on the arm *a*. Now, when the stop motion is actuated in a manner previously described, the connecting rod is moved to the right. The short lever *a* is turned about the tumbling-dog shaft.

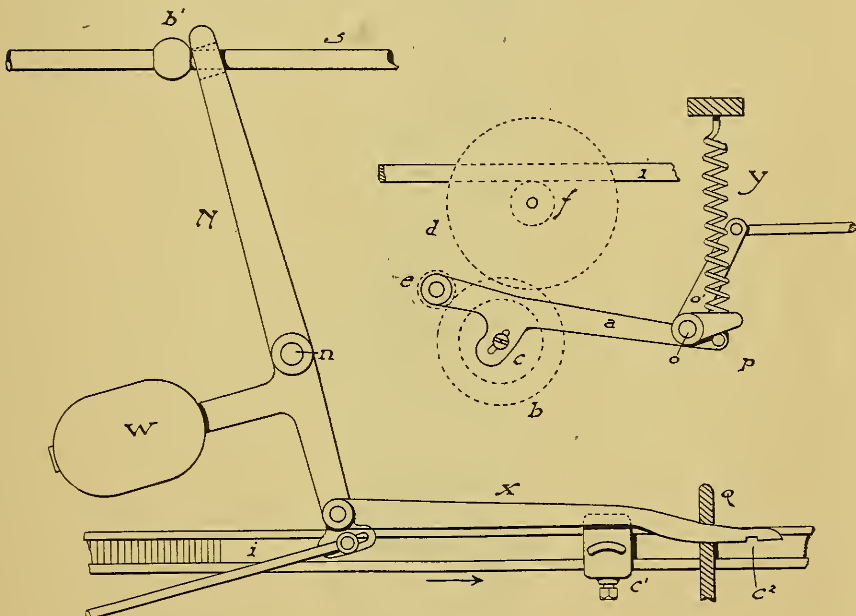

Fig. 111.

The pressure of *o′* upon the pin *p* moves the gear *c* out of mesh with *d* and stretches the spring *y*. The position of these parts is clearly shown in Fig. 111 at the right. The object of all this action is to free the upright shaft on which are the gears *d* and *f*. By means of a hand wheel on the top of said shaft the cone rack may be moved back to its first position, to which operation reference will hereafter be made. If *c* should remain in gear with *f* it would be impossible to turn the shaft, for in order to turn it one would have to set in motion the train of gearing between it and the tumbling dog shaft. But since at all times one part of the tumbling dog is pressing against the previously-mentioned plates P and P′, it is impossible to move the train of gearing without breakage. By pulling one of the gears out of mesh the desired result is easily secured.

For a description of the second stop motion reference may be made to Fig. 112. Here is shown a side view of the motion and its actuating

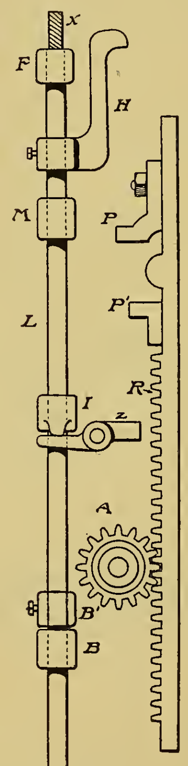

Fig. 112.

parts. A is one of the pinions on the lifter shaft. R is the rack attached to the carriage. At x in cross section is shown the end of the rod x of the stop motion already described. The upright rod L passes loosely through two castings M and B attached to the frame work of the machine. It is prevented from falling to the floor by the collar B′. The two peculiarlyshaped castings H and I are screwed to the rod L, while Z is a short, slightly-weighted lever pivoted to the frame work of the machine. If by any chance, due to a derangement of the parts or some other cause, the carriage should rise too high the projection P on the rack would raise H and consequently the rod L. x would then be raised and the stop motion actuated. In this case a different sort of device is employed for raising the notched lever, the effect is of course the same. If the carriage should move too far down P′ would push one end of Z down. The other end would then push up against I and raise the rod L with the same result as before.

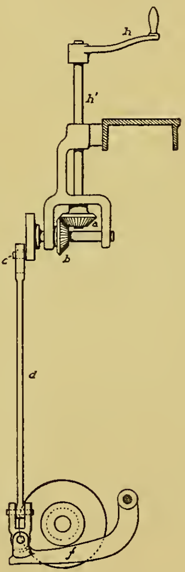

FIG. 113.

DOFFING

Doffing, or the process of removing the full bobbins and replacing them with empty ones, is not difficult to perform, but it necessitates the careful observance of a few simple rules. In the first place a handle *h*, seen in Fig. 113, is turned. This handle is easily accessible from the front of the machine. The handle is at the top of an upright shaft, on the bottom of which is a bevel gear *a* meshing with another bevel *b*. *b* has formed upon it a disc carrying a pin *c*. Around the pin *c* is a slot in the connecting rod *d*. The connecting rod is fastened at its lower end to a casting loosely hung upon the lifter shaft beneath an extended part of the bottom cone shaft. Any revolution of the shaft *h'* tends to raise the connecting rod, and consequently the casting *f*. *f* pushes up under the bottom-cone shaft and raises it. The cone belt is thereby made slack. If the machine is now started, the bottom cone and all the parts driven therefrom will not move. It will be recalled that the power which gives the bobbins their excess speed over the flyers comes to the differential motion from the bottom cone. If the bottom cone fails to move the bobbins will go no faster than the flyer. Consequently no roving will be wound on the bobbin. The second operation in the process of doffing is then to start the machine in order to allow more roving to be delivered by the front roll than is taken up by the bobbin. This slack, as it is called, is used later to start the empty bobbins.

By means of the hand wheel referred to on page 161 the cone rack is next turned back to its original position with the belt at the end of the cone. The handle *h* in Fig. 113 is then turned, allowing the bottom cone to drop to its normal position. The lever N in Fig. 111 is then pushed back to allow the catch *c'* in the rod *x* to engage with the casting. The machine will then be ready to restart after the bobbins have been removed and empty ones put on in their places. The removing of the bobbins and the piecing up of the ends is done as follows:

The machine is turned by hand until the flyers are all parallel with the front roll. The front flyers are then removed from the spindles and placed upon the top of the clearer covers. There is just enough room to hold all the front flyers on the machine. In this operation the roving is of course broken at the point where it leaves the flyer eye. Enough empty bobbins are then provided for the whole machine. These may be placed on the carriage between the spindles, two in each space. Or one may be placed in each space and one on the top of each of the full bobbins on the front. Next the attendant begins at one end of the machine, takes off a full bobbin from the front row, and drops an empty one in its place. At the same time the full bobbin is thrown into a large box provided for it. The corresponding back flyer is then raised, the full bobbin removed and replaced by an empty one. After which the back flyer is put back into place. This method of procedure is carried out until all the bobbins have been doffed. The back ends are then pieced up. The end of the roving that protrudes from the flyer eye is trained around the bobbin, the bobbin is raised a little to allow the roving to overlap, and the winding is continued until all the slack is taken up. After all the back ends have been pieced up the front flyers are put on and the ends pieced. The machine is then ready to start.

When the machine is stopped by the stop motion the carriage is either at the top or at the bottom of the traverse. When it is down, or, in other words, when the roving is being guided to the top of the bobbin, it is advisable to run the machine, holding the handle on until the carriage has traversed about a quarter or a third of its entire distance. This is done to insure that the roving will be broken when the flyers are subsequently removed.

CALCULATIONS

There are more calculations to be made on fly frames than on any of the other machines in a mill. Most of them are somewhat more difficult as well. An attempt will be made here to give as simple rules as possible for all the important calculations, and to illustrate each with an example. The sizes of the gears used are those in Figs. 104 and 105.

I. DRAFT.

The arrangement of the draft gearing is seen in Fig. 114. It will be observed that it resembles the arrangement on a drawing frame. The two back sets of rolls are driven from the front roll. A front-roll pinion A meshes with the crown gear *c*. The draft pinion is on the stud with *c* and meshes with the back-roll gear *b*. The middle roll is driven from the back roll as shown.

1. The rules for draft and draft constant are of course the same as previously given. Assume that the draft gear has 30 teeth.

Example:—

$$\text{Draft} = \frac{56 \times 100 \times 5}{35 \times 40 \times 1 \times 4} = 5$$

$$\text{Draft Constant} = \frac{56 \times 100 \times 5}{40 \times 1 \times 4} = 175$$

2. The rules relating to the size of the strand at the back and front of the machine, and the draft needed to produce strands of certain sizes, need some little modification in their application to all the machines which follow. This is due to the fact that the customary method of speaking of the size of roving, slubbing and yarn is not the same as in the case of sliver and laps.

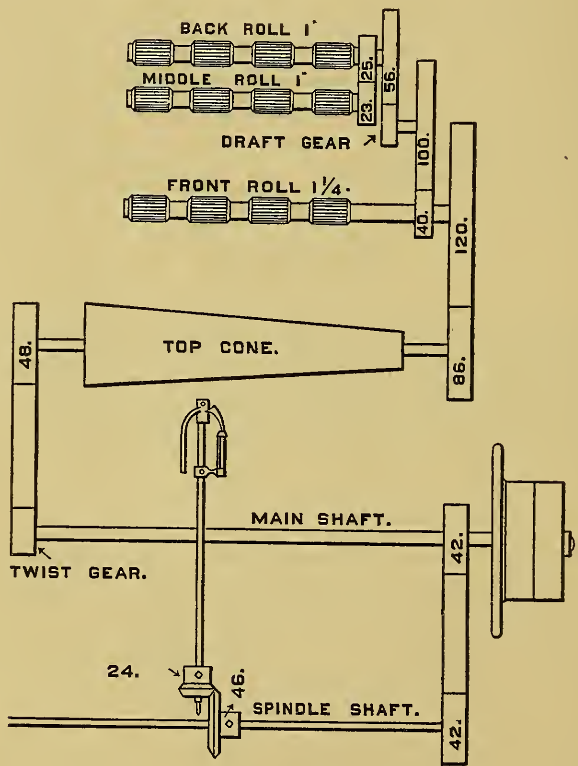

FIG. 114.

On all the preceding machines we have spoken of the strand of cotton as weighing a certain number of ounces or grains per yard. On the following machines we shall refer to the size of the strand by numbers. The method of numbering cotton roving and yarn is as follows: Instead of basing the size on a yard a higher denomination called a "hank" is used. A hank equals 840 yards. The "size" or "count" of a given piece of roving or yarn is the number of hanks that would be required to make one pound in weight. If two hanks, *i. e.,* 2 x 840 yards, of a strand of roving weigh just one pound the roving is called No. 2, or No. 2-hank. If ½ of a hank weighs just one pound the roving is No. ½ or ½-hank. It will be observed that as the roving increases in fineness more hanks will be needed to make a pound weight; consequently the finer the roving the greater is the number. It is customary to refer to roving as No. 5-hank, etc., while in the case of yarn merely the number is used, as No. 40 yarn.

The size of a given strand is found in practice by weighing a certain number of yards of it, and then by making a simple calculation. Twelve, 24, 60 or 120 yards are variously taken, depending upon the fineness of

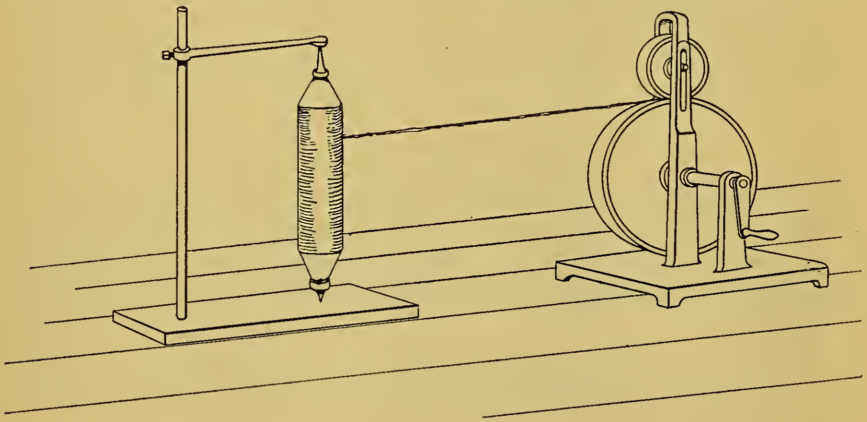

FIG. 115.

the strand being tested. Let it be assumed that we wish to find the size of some roving, which has just been made. It would in this case be sufficient to take 12 yards. A skewer would be placed through the bobbin and supported in a device like the one in Fig. 115. By means of a reel a yard in circumference the desired length would be measured. On a small pair of scales the weight of the 12 yards would be found. The weight is most accurately found in grains, and the scales are arranged to weigh as fine as tenths of grains. Assume that the 12 yards weigh 30 grains. In order to find the size we must find how much one hank of the roving in question would weigh. If 12 yards weigh 30 grains, 1 yard would weigh $\frac{1}{12}$ of 30 grains. Eight hundred and forty yards, or one hank, would weigh 840 x $\frac{1}{12}$ of 30 grains. Now, in one pound there are

7,000 grains. If one hank of the roving in question weighs 840 x $\frac{1}{12}$ x 30 grains, the number of hanks in a pound would equal 7000 ÷ (840 x $\frac{1}{12}$ x 30)

$$= \frac{7000 \times 12}{840 \times 30} = \frac{8\frac{1}{3} \times 12}{30} = 3.33$$

If the last fraction in the above example be observed it will be seen that the numerator consists of the number of yards weighed times 8⅓. The denominator is the weight in grains. Therefore, we may derive the following rule:

Size or count of roving or yarn equals number of yards weighed times 8⅓, divided by the weight in grains.

If only one yard be used

$$\text{Size} = \frac{8\frac{1}{3}}{\text{weight in grains}} \qquad (1)$$

$$\text{Weight in grains} = \frac{8\frac{1}{3}}{\text{size}} \qquad (2)$$

So on account of this change in nomenclature the rules for the size of the strand in their relation to draft must be changed to the following:

(*a*) Size of roving at back of machine equals size produced at front divided by draft, multiplied by the number of strands fed in together to the machine.

Example (1):—Find the size of the roving at the back of a roving frame if the roving produced at the front is No. 5.7, the draft 5.5 and the number of ends fed to the machine 2.

$$\text{Size at back} = \frac{5.7 \times 2}{5.5} = 2.07$$

Example (2):—Find the weight per yard of the slivers fed to the slubber, if the size of the slubbing produced is .64-hank, the draft 4.5 and the number of ends fed to the machine 1.

NOTE:—

On a slubber each strand fed to the machine is operated upon separately, while on the other roving frames it is customary to feed two strands together. The number of strands fed in together is commonly, although erroneously, called the number of "doublings."

$$\text{Size at back} = \frac{.64}{4.5} = .142 \text{ hank.}$$

The example called for a result in grains per yard, since it is usual to refer to all slivers in those terms. By applying rule (2) above,

$$\text{Grains per yard} = \frac{8\frac{1}{3}}{.142} = 58.68 \text{ grains.}$$

(*b*) Size of roving at the front of a machine equals size fed divided by number fed, times the draft.

Example:—Find the size delivered by a roving frame if that fed is 2.14-hank, the number of ends fed 2 and the draft 5.6.

$$\text{Size} = \frac{2.14}{2} \times 5.6 = 5.99 \text{ hank.}$$

(*c*) Draft required to produce roving of a certain size equals the size desired, divided by the size fed divided by the number fed together.

Example (1):—Find the draft required to produce No. 9-hank roving from two strands of No. 3-hank.

$$\text{Draft} = \frac{9}{\frac{3}{2}} = \frac{9 \times 2}{3} = 6$$

Example (2):—Find the draft required to produce .52-hank slubbing from a 65-grain sliver.

Here the grains must be reduced to a number by an application of formula (1).

$$65 \text{ grains} = \frac{8\frac{1}{3}}{65} = .128\text{-hank.}$$

$$\text{Draft} = \frac{.52}{.128} = 4.06$$

On account of the twist that is introduced into roving the strand contracts in length. As a result the roving when it is wound on the bobbin is coarser or, in other words, of a smaller number than it was when delivered by the front roll. In all the last three rules the expression "size of roving produced" really referred to the size of the roving delivered by the front roll. The contraction varies according to the amount of twist between 1 and 4 per cent. Two and one-half is nearly the average, and will be used in the following calculations. If "size of roving produced" in the foregoing rules is made to mean the actual size of the roving after it is wound on the bobbin, allowance must be made for the contraction as follows:

(*d*) 1. Deduct the percentage of contraction from the result of (*a*) if the answer is to be in grains per yard.

2. Divide the result of (*a*) by 100 per cent. minus the percentage of contraction if the answer is to be in size.

(*e*) Deduct the percentage of contraction from the result of (*b*).

(*f*) Divide the result of (*c*) by 100 per cent. minus the percentage of contraction.

The following examples show the corrected results of the foregoing examples if the contraction is 2½ per cent.

Example (2) under (a):—

$$\text{Size on bobbin} = \frac{2.07}{.975} = 2.12$$

Example (2) under (a):—

Actual weight per yard of sliver = .975 x 58.68 = 57.31

Example under (b):—

Size on bobbin = .975 of 5.99 = 5.84-hank.

Example (1) under (c):—

$$\text{Actual draft} = \frac{6}{.975} = 6.15$$

Example (2) under (c):—

$$\text{Actual draft} = \frac{4.06}{.975} = 4.16$$

3. Draft gear needed in any case equals the draft constant divided by the actual draft required.

Example:—Find the draft gear needed to make 1.6-hank roving if the size of one of the two strands fed is .6, the draft constant is 175 and the contraction by twist 2.4 per cent.

$$\text{Actual draft required} = \frac{\dfrac{1.6}{.6} \cdot 2}{.976} = 5.46$$

$$\text{Draft gear} = \frac{175}{5.46} = 30.05 \text{ or } 30 \text{ teeth.}$$

4. When the draft gear that is producing a certain draft is known it is possible to find the gear needed to produce any other draft by means

of the same proportion that was used on the card for the same purpose. The draft gear varies inversely as the draft.

Example:—Find the gear needed to produce a draft of 5.24 if a gear with 40 teeth gives a draft of 5.8.

$$5.8 : 5.24 :: x : 40$$
$$5.24\ x\ 232$$
$$x\ 44.3$$

Answer, 44 teeth.

II. TWIST.

By the word "twist" is meant technically the actual number of turns or twists that are introduced into each inch of roving delivered by a front roll. It is possible on all fly frames to regulate this very accurately. The actual amount of twist desired will of course vary with the fineness of the roving and the length of the staple of the cotton used. This will be readily understood when the object of introducing any twist whatever is considered. Twist is introduced for the purpose of preventing the roving from breaking when it is being fed to the next machine, and also to aid the next machine in keeping the fibres parallel while draft is put in. Long staple cottons require less twist than short staples, and fine roving must have more than coarse roving. It has been found that a certain standard amount of twist gives very good results on average American cotton. A relation has been established between the size of the roving and the turns needed per inch. Some use the following amounts: On slubbers twist equals square root of the size; on intermediates, 1.1 times the square root of the size, and on roving frames of smaller gauge, 1.2 times the square root of the size. Some invariably use 1.2 times the square root of the size. Of course the amount should be adapted to the conditions existing at the time.

Calculations may be made to determine the actual twist being introduced at any given time, as well as the size of the change gear needed to produce any desired amount of twist. A calculation will first be made to find the actual twist being put into some slubbing with a certain arrangement of gearing. Fig. 114 shows a detached diagram of the gearing having to do with twist. It will be observed that the change gear for the regulation of twist is on the end of the jack shaft. Through it power is transmitted to the top cone. On the end of the top-cone shaft in the head of the machine is a large gear meshing with another on the end of the front roll. The gearing driving the spindle is also shown. Attention is called to the fact that there is no change gear between the main shaft of the machine and the spindle. Therefore all regulation of the twist per inch must be made by changing the speed of the front roll. The theory underlying the determination of the twist per inch is as follows: Suppose that 100 inches of roving is delivered in one minute by the front roll. Suppose also that within the same length of time the spindle makes 500 revolutions. Now, since for every revolution of the spindle one twist is introduced into the roving, the 500 turns must be put into the 100 inches delivered. The number of turns in each inch must then be 500 divided by 100 or 5 turns.

1. Twist Per Inch.

The twist per inch may therefore be defined as the speed of the spindle divided by the surface speed of the front roll. Since the twist is only a relation between these two terms it matters not for how long a time we compute the twist. We may take any unit whatever. So in this case we may adopt a method similar to the one used in the finding of draft, and take for the unit of time the time required by the front roll to make one revolution. Then by considering that the front roll drives the rest of the machine we can find the speed of the spindle. This speed divided by the circumference of the front roll gives the twist per inch.

Example:—Find the twist per inch with the gearing shown in Fig. 114 if the twist gear being used has 26 teeth. Considering that the front roll makes one revolution and drives the rest of the machine.

$$\text{Speed of spindle} = \frac{120 \times 48 \times 42 \times 46}{86 \times 26 \times 42 \times 24} \qquad (1)$$

$$\text{Surface speed of front roll} = 1 \times \frac{5}{4} \times 3.1416 \text{ inches.} \qquad (2)$$

Twist per inch = (1) divided by (2).

$$\frac{120 \times 48 \times 42 \times 46 \times 4}{86 \times 26 \times 42 \times 24 \times 5 \times 3.1416} = 1.26 \qquad (3)$$

Note:—There is actually some contraction while twisting goes on, as has previously been stated. The result is that there is really more twist per inch than the above calculation shows. Since it is the amount delivered that contracts, the truer divisor or denominator in (3) should be a certain per cent. less than the one used. Consequently the actual twist per inch = result of (3) divided by 100 per cent. minus the per cent. of contraction, *e. g.* contraction 2 per cent., actual twist = 1.26 divided by .98 = 1.28. In practice it is unnecessary to consider the contraction in figuring the twist on fly frames, although it must be regarded in the calculation for draft.

2. Twist Constant.

By omitting the twist gear in the preceding calculation a constant number may be obtained, from which the size of the twist gear needed to produce any desired amount of twist may be easily found. From above example

$$\text{Twist constant} = \frac{120 \times 48 \times 42 \times 4}{86 \times 42 \times 24 \times 5 \times 3.1416} = 32.689$$

$$\text{Twist gear} = \frac{\text{twist constant}}{\text{twist per inch desired}}$$

$$\text{Twist per inch at any time} = \frac{\text{twist constant}}{\text{twist gear in use}}$$

The following examples are practical and are likely to occur often:

Examples:—

(*a*) Find the twist gear needed to produce No. 6-hank roving if the twist constant is 71.26.

As previously stated the twist per inch wanted may be considered to be 1.2 times the square root of the size of the roving to be made.

$$\text{Then twist per inch} = 1.2 \times 6 = 2.94.$$

$$\text{Twist gear} = \frac{71.26}{2.94} = 24.2. \qquad \text{Answer, 24 teeth.}$$

(*b*) Find the twist gear needed to produce .72-hank slubbing, if in making .5-hank slubbing a gear with 30 teeth has been used.

This example may be done by proportion. It is possible to find the gear needed without knowing either the turns per inch or the constant. It is important to know, however, what sort of a proportion to use. A larger twist gear would increase the amount delivered by the front roll in a given length of time, and would consequently decrease the number of turns per inch. Mathematically considered a larger twist gear would increase the size of the denominator in the calculation for twist, consequently decreasing the twist. Therefore the twist gear varies inversely as the twist per inch. Now the twist per inch is always equal to a constant, usually 1.2 times the square root of the size of the roving. So in other words the twist per inch varies directly as the square root of the size of the roving being made. Hence the twist gear varies inversely as the square root of the size of the roving being made. In the present example the proportion is as follows:

$$\sqrt{.5} : \sqrt{.72} :: x : 30$$

The above proportion may be with mathematical accuracy expressed as follows:

$$.5 : .72 :: x^2 : 30^2$$

$$\text{Whence } x^2 = 625$$

$$x = 25 \qquad \text{Answer, 25 teeth.}$$

III. CONE GEAR.

The cone gear is the change pinion on the end of the bottom cone and may be seen in Fig. 104. It will be recalled that through the cones and their connections all the power which makes the bobbins go faster than the flyers is transmitted. This power is compounded with the other at the differential motion. The cone gear is changeable in order to offer a chance for regulating the excess speed of the bobbins. To be sure it is seldom necessary to change it after the right one has once been secured. It is our purpose here to show how the first one may be found. In this case we shall assume a certain speed actually found by a speed counter for the main shaft and for the bottom cone. It is safest to find by actual count the speed of the bottom cone since it is difficult to find the diameters of the cones at the exact points at which the belt effectively drives, and since there is some little slip of the belt.

The gearing used is that in Fig. 104. Additional data will be given as required.

The calculation for the cone gear may be divided into several operations.

1. Find the necessary excess speed of the bobbin. This necessary excess speed depends upon the diameter of the bobbin. In the calculation we always take the diameter of the empty bobbin barrel. In the present case the diameter is 1¾ inches. Now the excess speed of the bobbin

equals the surface speed of the front roll divided by the circumference of the bobbin.

So the necessary excess speed of the bobbin, if speed of main shaft is 264,

$$= \frac{264 \times 34 \times 86 \times 5 \times 4}{48 \times 120 \times 4 \times 7} = 95.73$$

2. Find the speed of the spindle from the speed of the main shaft and the gearing in the usual manner.

$$\text{Speed of spindle} = \frac{264 \times 42 \times 46}{42 \times 24} = 506$$

3. Add excess speed of bobbin to speed of spindle to find the necessary actual speed of the bobbin.

Necessary actual speed of bobbin = 601.73.

4. Now let it be considered that the bobbin drives all the rest of the machine. By the usual method next find the speed of the last or "bobbin wheel" in the differential motion. The operation is as follows:

Speed of last gear in differential motion =

$$\frac{601.73 \times 24 \times 42}{46 \times 57} = 231.33$$

5. Next find the speed of the first gear in the differential motion. This must be done by an application of the formula for the particular motion that is on the machine in use. We shall assume for the present example that the differential in use is Daly's. In that case the speed of the last gear equals $\frac{20}{27}$ of the speed of the main shaft plus $\frac{7}{27}$ of the speed of the first gear in the motion. Consequently the speed of the first gear in the motion must equal $\frac{27}{7}$ of the difference between the speed of the last gear and $\frac{20}{27}$ of the speed of the main shaft.

So speed of first gear = $\frac{27}{7}$ of $(231.33 - \frac{20}{27}$ of $264) = 138$.

6. Now consider that the sleeve carrying the first gear drives back to the bottom cone. Then by tracing through the train of gearing we can get an expression for the speed of the bottom cone. It will be recalled that on the sleeve with the first gear of the differential motion is the gear marked 30 in the drawing. The connection from that point is plain. So

speed of bottom cone $= \dfrac{138 \times 30 \times 80,}{26 \times C}$ where C represents the size of the cone gear needed. Now the actual speed of the bottom cone with the belt on the small end, or at the point where one wishes to have it start may be found most accurately by means of a speed counter. In that case no allowance need be made for the inevitable slipping of the cone belt, and at the same time the difficulty of finding the length of the effective working diameters of the two cones is done away with. In the present example the speed of the bottom cone has been found to be 387 with a 34-twist gear, which is the same size that was assumed in finding the surface speed of the front roll. It is evident then that whatever the size of the cone gear the speed of the bottom cone will be 387. The right cone gear must be obtained to make the parts which are actually driven from it go at their necessary speed. In order to find the gear needed we have only to substitute in the last expression 387 in the place of C. If with the right cone gear in the place of C the result of the expression would be 387, it is evident that the result of the expression divided by 387 would give the size of the gear required.

$$\text{So cone gear needed} = \frac{138 \times 30 \times 80}{26 \times 387} = 32.91$$

Answer, 33 teeth.

The answer was seen to be nearly 33. To make the speed exactly right it might be necessary to move the cone belt a trifle towards the large end of the bottom cone. When the result is so near as in this case, however, no change is really necessary. If now the speed of the bobbins be calculated with a 33-cone gear in use it will be found to be approximately that found to be necessary in the first part of the calculation.

Speed of first gear in differential motion =

$$\frac{387 \times 33 \times 26}{80 \times 30} = 138.35$$

Speed of last gear of differential =

$$\tfrac{7}{27} \text{ of } 138.35 \text{ plus } \tfrac{20}{27} \text{ of } 264 = 231.42$$

$$\text{Speed of bobbin} = \frac{231.42 \times 57 \times 46}{42 \times 24} = 601.96$$

The result with a 33-gear is only .23 of a revolution a minute too much, none too great to prevent good work. The cone gear need only be changed when it is desirable to change the size of the bobbin barrel or when the belt is to be started at a point further along the cones. Both conditions rarely occur.

IV. THE LAY GEAR.

Reference to Fig. 104 will show the position of the lay gear. It is made changeable in order that the speed of the carriage may at different times be increased or decreased. Such a change in the speed of the carriage is necessary whenever the size of the roving to be made on the machine is changed. It will be readily understood that if roving of a smaller diameter is wound on a bobbin, it will be necessary to make the carriage move up and down more slowly to wind the layers just as near together as when the roving was coarser.

The calculation for the lay gear consists in finding the size of the gear needed to give the carriage such a speed that a certain number of coils will be wound in each inch. The coils that may be made to lie side by side in an inch vary with the size of the roving and with the twist. For the same number of roving it is possible to have the number of coils per inch different, if the twist per inch is different. Good results may be gotten if the number of coils that will lie side by side in an inch be taken to be 10 times the square root of the size of roving being made for intermediates, 9 times the size for slubbers and 11 times the $\sqrt{\text{size}}$ for finer frames. Some find it practicable to use 9 times the $\sqrt{\text{size}}$ throughout. In the present example on a slubber the number taken is 9 times the $\sqrt{\text{size}}$. The operation is as follows:

First find the surface speed of the front roll for the time taken by the carriage to move one inch. This may be done by considering the number of teeth in one inch of the lifter rack as the first driver. From that the train of gearing may easily be traced to the front roll. In going through the cones it is possible either to use the effective diameters if they be known, or to use in place of the diameters the speeds which will already have been found in the calculations for the cone gear. At least the speed of the bottom cone will have been found and the speed of the main shaft. The speed of the top cone may be found through the gears. In this example the speed of the bottom cone is taken as 387, and the speed of the top cone has been found by calculation to be 187. In going from the lifter rack to the front roll the bottom cone is of course a driver, whereas during the running of the machine it is driven. Therefore, if the speeds be used, the speed of the top cone must be used in place of the diameter of the bottom cone, and vice versa. The number of teeth in one inch of the rack on the slubber used for the calculation is two. The surface speed of the front roll for one inch traverse of the rack =

$$\frac{2 \times 86 \times 96 \times 70 \times 54 \times 80 \times 187 \times 86 \times 5 \times 22}{16 \times 18 \times L \times 16 \times 44 \times 33 \times 387 \times 120 \times 4 \times 7} \qquad (1)$$

The number of coils that would be coiled in one inch on the bobbin equals the length delivered by the front roll divided by the circumference of the bobbin. The circumference of the bobbin in the present case equals 1¾ times 22/7. It was said that the number of coils desired was 9 times the $\sqrt{}$ size. We shall assume that the size to be made is .64-hank. The square root of .64 is .8. .8 x 9 = 7.2. Substituting 7.2 in the place of L in (1) and dividing by the circumference of the bobbin as given above we have lay gear =

$$\frac{2 \times 86 \times 96 \times 70 \times 54 \times 80 \times 187 \times 86 \times 5 \times 22 \times 4 \times 7}{16 \times 18 \times 7.2 \times 16 \times 44 \times 33 \times 387 \times 120 \times 4 \times 7 \times 7 \times 22} = 25.64$$

Answer, 26 teeth. (2)

The following will show why the number of coils was substituted in the place of the lay gear. If the calculation had been carried out, using the actual size of the lay gear, the answer would have been the number of coils per inch. Now, if something or other divided by the lay gear equals the coils per inch, then the size of the lay gear must equal that something divided by the coils per inch. It is merely a case of having given the dividend and quotient to find the divisor. If expression (2) be examined it will be found that nothing there is variable except the square root of the size of the roving, which went to make up the 7.2. All the other numbers remain the same no matter what size of roving or what size lay gear may be used. Therefore, if we carry out the calculation, omitting everything that is variable, we shall obtain a constant which will help us in finding the lay gear desired at any subsequent time. If, therefore, instead of using 7.2 we use merely 9 we shall have a constant, which, when divided by the square root of the roving to be made at any time, will give the size of the gear needed. Hence the lay constant =

$$\frac{2 \times 86 \times 96 \times 70 \times 54 \times 80 \times 187 \times 86 \times 5 \times 22 \times 4 \times 7}{16 \times 18 \times 9 \times 16 \times 44 \times 33 \times 387 \times 120 \times 4 \times 7 \times 7 \times 22} = 20.67$$

$$\text{Lay gear} = \frac{\text{lay constant}}{\sqrt{\text{size of the roving}}}$$

The speed at which the carriage goes up and down depends upon the diameter of the roving being made. If one should take the trouble to work out the diameter of roving one would find that the diameter could be expressed in terms of the square root of the size. This fact explains why the number of coils that will lie side by side in an inch is expressed in terms of the square root of the size. It is for the same reason, by the way, that the twist per inch is expressed in terms of the square root of the roving. The diameter then varies inversely as the size. The coils per inch vary inversely as the diameter, and consequently directly as the square root of the size. Again, since the lay gear is a driving gear when the machine is running, the speed of the carriage varies directly as the size of the gear. Now, the faster the carriage goes the fewer will be the coils per inch. So we may say that the coils per inch vary inversely as the size of the lay gear. Hence the size of the lay gear varies inversely as the square root of the size of the roving. We may find the lay gear then in the same manner as the twist gear, provided we already have a gear that is giving good results on a certain size of roving.

$$\sqrt{\text{Size of roving being made}} : \sqrt{\text{size wanted}} :: \text{gear wanted} : \text{gear in use.}$$

V. TENSION GEAR.

The tension gear lends itself less readily to mathematical calculation than any other gear on the machine. Temperature, humidity, twist and the number of coils around the presser finger make the problem complicated. If, however, all the conditions are known, it is possible to find with tolerable ease the size of the tension gear needed for any size of roving. Fig. 109 shows the position of the tension gear on the machine. It has already been incidentally referred to. Its function is to regulate the distance traversed by the cone belt at the end of the winding of each layer. It, therefore, controls the amount by which the excess speed of the bobbin is gradually decreased. It, of course, has no work until the first layer has been wound upon the bobbin. It is necessary to change the tension gear every time the size of the roving is changed. If roving of a smaller diameter is wound on a bobbin the increase in size of the bobbin will be less than when the roving was coarser. Hence, the excess speed must be decreased at the end of each layer by a smaller amount. The decrease is controlled by the movement of the cone belt. The so-called tension gear directly controls the distance moved.

It is the custom with machine makers to try to make as solid bobbins as possible, that is, to get as much cotton on a bobbin as is consistent with good work. In order to attain this result dependence should be put upon the action of the presser rather than upon the tension on the roving. As the roving is pulled away from the front roll its tension on the presser tends to hold the presser close to the bobbin. At the same time the rapid revolution of the flyer tends to throw the weighted part of the presser p in Fig. 99 away from the centre of revolution. p is then turned around c and c' so that the end h presses tightly against the bobbin. The more times the roving is wrapped around the presser the tighter will be the bobbin. Usually three turns give the best results. It is, under this condition, possible to wind about as many layers in an inch as could be laid side by side in an inch, if the roving were a perfectly cylindrical strand of solid cotton. It is convenient and safe in practice to work on this basis. And this leads us to the interesting, though not difficult, problem of determining the diameter of a cotton thread. This may be done as follows:

Assume that the thread is a perfectly cylindrical mass of cotton of a certain length. Take as that length the length that would weigh just one pound. Then determine the number of cubic inches in that mass, assuming that its diameter be d, and its size or number or count n.

In other words, let $d =$ diameter.

$n =$ size.

Then $n \times 840 =$ number of yards in the length that would weigh a pound.

$n \times 840 \times 36 =$ number of inches in the length that would weigh a pound.

$\frac{n \times 840 \times 36 \times d^2 \times 3.1416}{4} =$ cubic inches in the length that would weigh a pound.

If now the weight of the above number of inches of cotton be compared with the weight of the same number of cubic inches of water, an equation may be formed from which the diameter may be expressed in terms of the size.

1 cubic inch of water weighs .03608 pounds.

Cotton weighs 1.5 times as much as water.

Hence the above number of cubic inches of cotton would weigh

$$\frac{n \times 840 \times 36 \times 3.1416 \times d^2 \times .03608 \times 1.5}{4} \text{ pounds.}$$

But by our hypothesis the weight of the number of cubic inches in question is just one pound.

$$\text{Hence } \frac{n \times 840 \times 36 \times d^2 \times 3.1416 \times .03608 \times 1.5}{4} = 1.$$

$$\text{Whence } d^2 = \frac{1}{1285.368 \times n}$$

$$\text{And } d = \sqrt{\frac{1}{1285.368 \times n}}$$

$$\text{Or } d = \frac{1}{35.85 \times \sqrt{n}}$$

Therefore, the diameter equals 1 divided by 35.85 times the square root of the size. So the number of strands that would lie side by side in an inch is 35.85 times $\sqrt{\text{size}}$. In practice we may then take this as the number of layers possible per inch.

1. It is now necessary to determine the number of layers that would practically be wound on a bobbin of a given diameter.

For example, let diameter of full bobbin = 4.85 inches.

Let diameter of empty bobbin barrel = 1.75 inches.

The actual thickness of the cotton equals, of course, one-half of the difference between these two diameters.

$$\text{Then thickness of cotton} = \frac{3.10 \text{ inches}}{2} = 1.55 \text{ inches.}$$

In that number of inches there could be wound $1.55 \times 35.85 \times \sqrt{n}$ layers where n equals the size of the roving being made. We have then the number of layers wound on the full bobbin in terms of the size of the roving.

2. We shall next find by means of the gears the number of layers that would be wound on the same bobbin. In this case we must first determine the distance that the cone belt moves during the winding of the entire bobbin. This may be done in two ways:

(*a*) If the machine has been running and has been making bobbins of a certain diameter, the actual distance traversed by the cone belt may be measured along the cone rack. This distance should be measured from the starting point of the belt to the position of it at the time that the stop motion is actuated. Care should be taken to get this distance accurate.

(*b*) If it is not possible to use the above method the distance may be found as follows:

Proceed in much the same manner as when finding the cone gear. First, determine the necessary excess speed of the bobbin for the diameter in question. Second, add that to the speed of the spindle to get the actual speed of the bobbin. Third, find the necessary speed of the last gear of the differential motion by considering that the bobbin drives the rest of the machine. Fourth, find the speed of the first gear of the differential motion from the formula for the motion in use. Fifth, find the necessary speed of the bottom cone, using as the cone gear the one already found to be correct. Sixth, find the effective diameter of the bottom cone at which the belt must be in order to give the cone the correct speed. The last operation may be done in the following manner: Suppose that by going through the five steps just mentioned the necessary speed of the bottom cone, when the diameter of the bobbin is 4.85 inches, has been found to be 132.09.

It has been previously found that the speed of the top cone on the slubber in question is 187. Let it be assumed that the sum of the diameters of the cones, which must always be constant, is 9.875 inches.

Then let x = diameter of top cone at desired point.

" y = diameter of bottom cone at desired point.

Then $x + y = 9.875$. (1)

$$\text{And } \frac{187\,x}{y} = 132.09 \qquad (2)$$

From (1) $x = 9.875 - y$

Substituting in (2)

$$y = 5.82 \text{ inches.}$$

During the winding of the last layer the belt must be at a point where the diameter of the bottom cone is 5.82 inches. That is to say, the middle of the belt should be at this point. It is then easy to measure the distance from the position of the belt at the beginning of the winding to this last position.

In the present example we shall assume that either one of the foregoing methods has shown that the belt traverses 18.375 inches.

3. We next suppose that the cone rack drives the gearing between itself and the builder motion. In one inch of the rack there are, by measurement, $\frac{32}{13}$ teeth. Reference to Fig. 109 will now show that the number of revolutions made by the tumbling-dog shaft, while the cone belt makes its full traverse, is as follows:

$$\frac{18.375 \times 32 \times 80 \times 68}{16 \times 13 \times T \times 16}$$

where T represents the size of the tension gear.

Now the tumbling-dog shaft makes a half revolution for every layer wound on the bobbin. The number of layers wound, if the size of the roving is .64, would also be equal to

$$\sqrt{.64} \times 35.85 \times 1.55 = 44.46 \text{ layers.} \qquad (3)$$

We then have two expressions for the total number of layers on the bobbin. These may be equated as follows:

$$\frac{18.375 \times 32 \times 80 \times 68 \times 2}{16 \times 13 \times T \times 16} = 44.46 \qquad (4)$$

By substituting 44.46 in the place of T and carrying out the operation we have T = 43.2.

Answer, 43 teeth.

It will be observed that in the last two expressions there were two variable quantities, one the tension gear itself, and the other the size of the roving. Everything else remains constant for every gear and every size of roving that may be used. Instead of using .64 we shall now use the letter n, and shall therefore have to retain the letter T to represent the size of the tension gear.

Expression No. 4 then becomes the following:

$$\frac{18.375 \times 32 \times 80 \times 68 \times 2}{16 \times 13 \times T \times 16} = 1.55 \times 35.85 \times \sqrt{n}$$

Substituting 1.55 x 35.85 in the place of T, we have

$$\frac{18.375 \times 32 \times 80 \times 68 \times 2}{16 \times 13 \times 55.57 \times 16} = T \times \sqrt{n}$$

By working out the first half of the equation we have what is called the tension constant. This in the present case equals 34.59.

So $T \times \sqrt{n} = 34.59$

Whence $T = \dfrac{34.59}{\sqrt{n}}$

Therefore, the tension gear equals the tension constant divided by the square root of the size of the roving to be made.

If with a certain size roving a gear of a certain size is giving good results the gear needed for any other size may be found by the use of a proportion similar to the one used to find the twist gear or the lay gear, as follows:

$\sqrt{\text{size of roving being made}} : \sqrt{\text{size wanted}}$: gear wanted : gear in use.

VI. PRODUCTION.

The production of fly frames is expressed as hanks or as pounds per spindle in a certain length of time, usually a week of sixty or fifty-eight hours.

$$\text{Production per spindle in hanks} = \frac{\text{speed of front roll in yards.}}{840}$$

$$\text{Production in pounds} = \frac{\text{above result}}{\text{size}}$$

An allowance for doffing and other stops should be made from the theoretical production. The per cent. of allowance is usually between 7 and 10, it depending quite largely on the efficiency of the help.

Applied to the front rolls of fly frames are clock dials which register the production in actual hanks per spindle. The pay of the help depends upon the amount produced. The actual production of fly frames for sixty hours will vary from 350 pounds per spindle of No. 2-hank slubbing to .9 pounds per spindle of No. 32-hank fine roving.

GENERAL REMARKS

The preceding examples and descriptions have related mainly to one type of machine. But it is thought that thereby a sufficiently clear understanding of the principles upon which all roving frames are constructed could be obtained. The American style of builder motion was chosen. The English type may be found clearly explained in any English book on the subject. It has been stated that there is some difference between the various roving frames. On all frames except the slubber the cotton is fed from bobbins held in a "creel" by wooden sticks called "skewers," which rest in glass steps, and act as the axis of revolution for the bobbins. The machines decrease in size also. On slubbers the number of spindles ranges from 24 to 96, depending upon the size of the bobbin to be made. Intermediates are made with anywhere from 48 to 168 spindles. On fine roving and jack frames the number will run as high as 200. The size of a machine is usually designated by the size of the full bobbin that it is intended to make. If the full bobbin produced on a frame is 6 inches in diameter and the length of the traverse of the carriage while the first layer is being wound is 12 inches, the frame is called a 12 by 6. This is the largest size slubber that is made. The range is from 12 by 6 to fine jack frames, as small as 4½ by 2¼.

The number of spindles on a slubber must be a multiple of 4, for it is customary to make the number of spindles for one section of roll 4. On intermediates the number is usually 6, and on finer frames 8.