

[54] BAG-MAKING MACHINE

3,334,550 8/1967 Craig 93/8

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[22] Filed: **Oct. 4, 1973**

[21] Appl. No.: 403,484

[30] Foreign Application Priority Data

Oct. 4, 1972	Germany.....	2248613
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[52] U.S. Cl. 93/8 WA, 93/35 H, 156/519,
156/552

[51] Int. Cl. B31b 1/86

[58] **Field of Search**..... 93/8 R, 8 WA, 35 H, 61 A;
53/128, 134; 156/361, 362, 519, 520, 552

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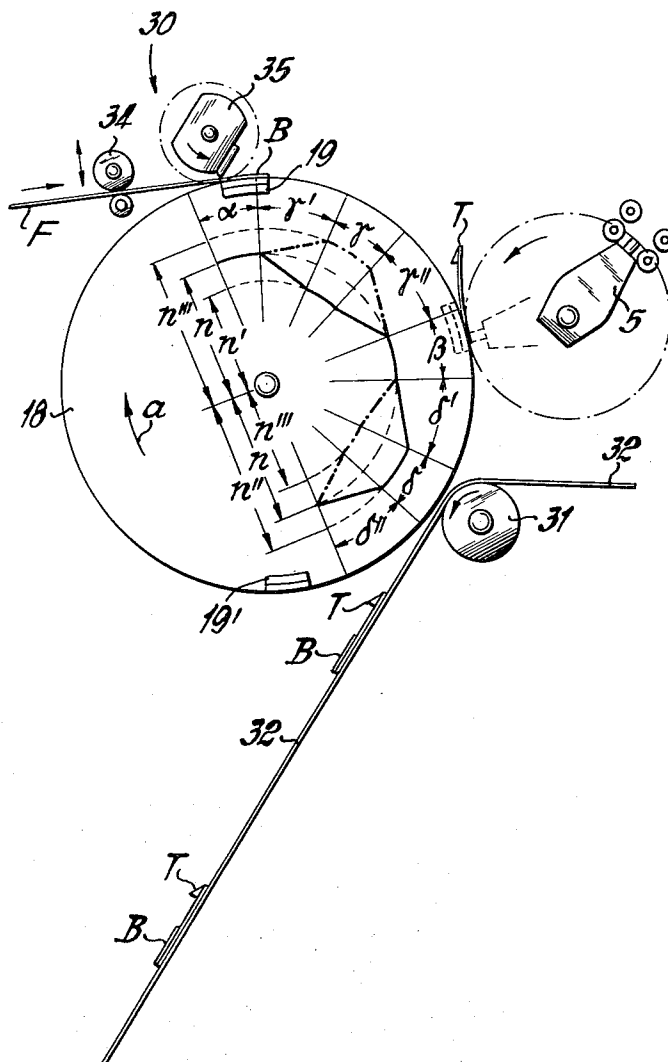
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[57] **ABSTRACT**

In the manufacture of carrier bags, a continuously travelling web of bag-making material is provided at intervals with carrier handles which are stuck thereto by means of reinforcing sheets. The reinforcing sheets are successively fed to a rotary transfer drum with which the web of bag-making material is in contact. The reinforcing sheets are coated with adhesive on one side. During rotation of the transfer drum, and before each reinforcing sheet makes contact with the web of bag-making material, a carrying handle is fed to the transfer drum at the location where a reinforcing sheet is being held on it. To enable the pitch of the carrying handles on the web to be varied without changing the transfer drum, the latter is turned by variable gearing.

8 Claims, 4 Drawing Figures



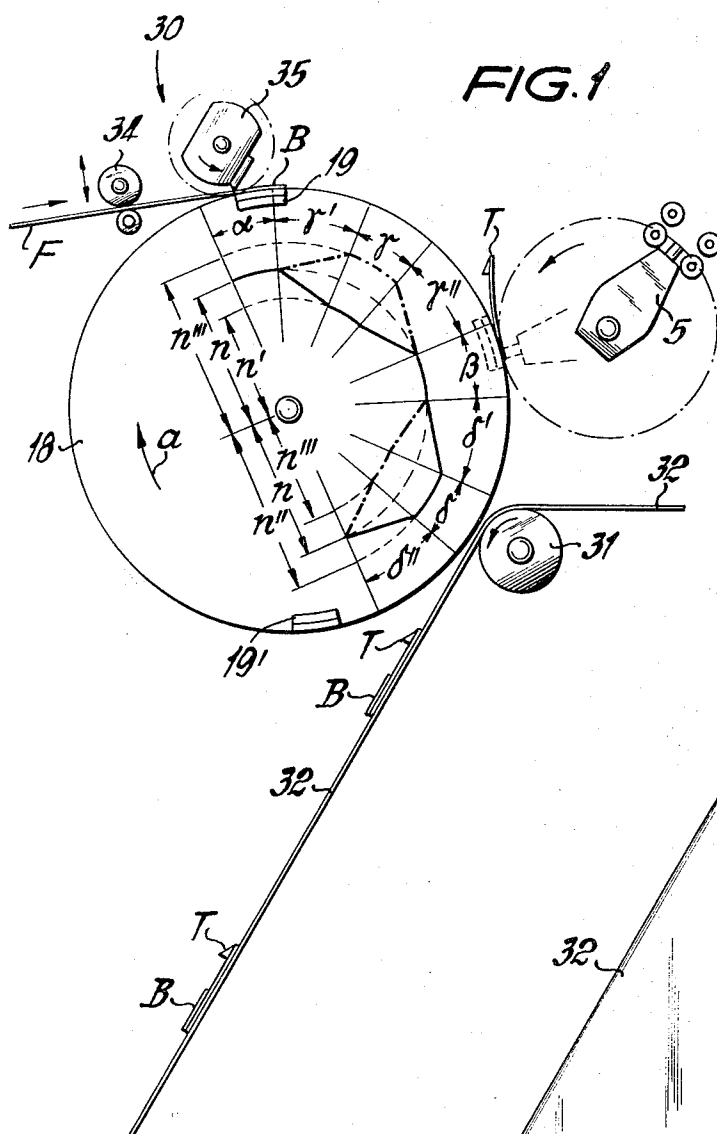
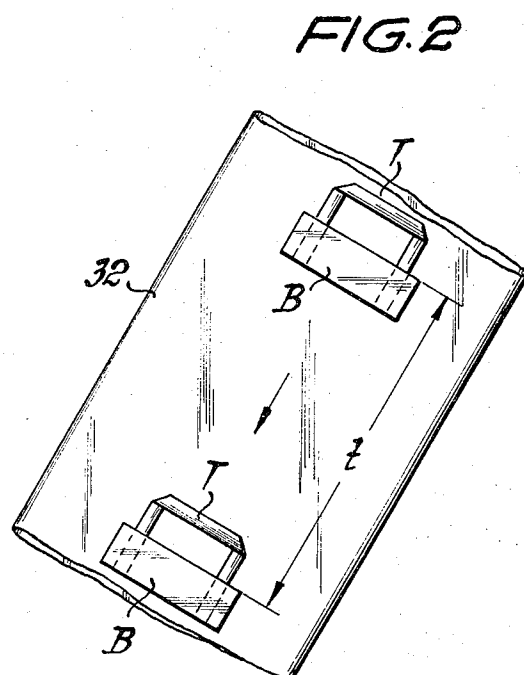


FIG. 3

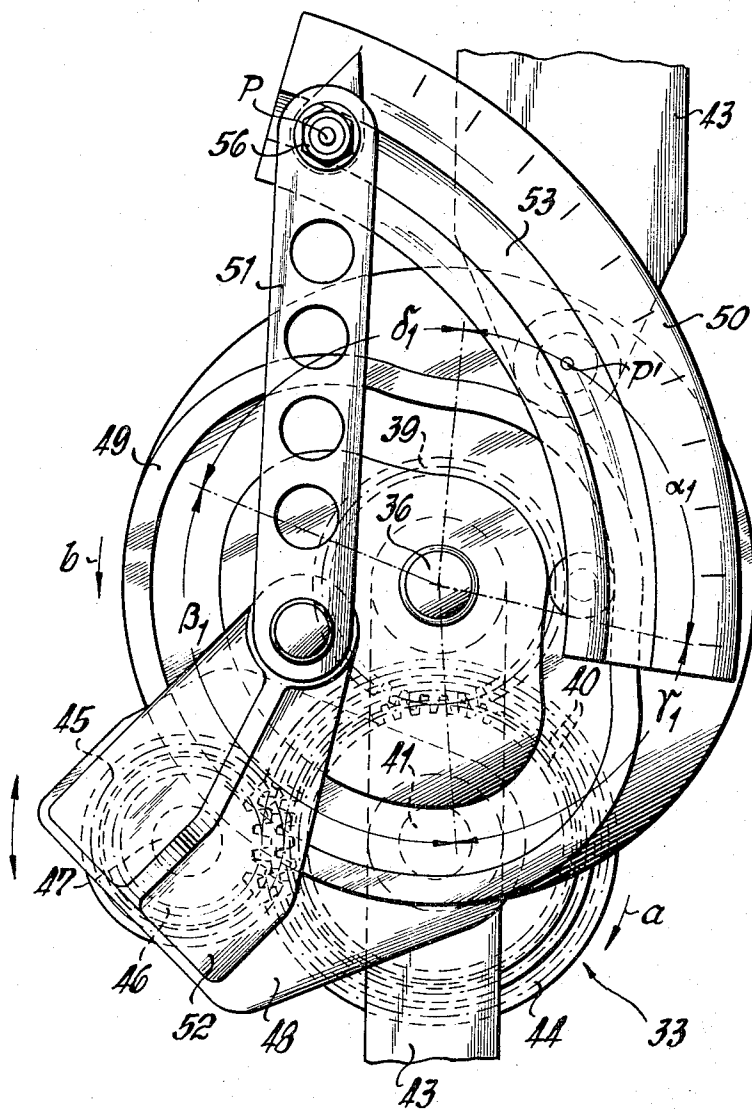
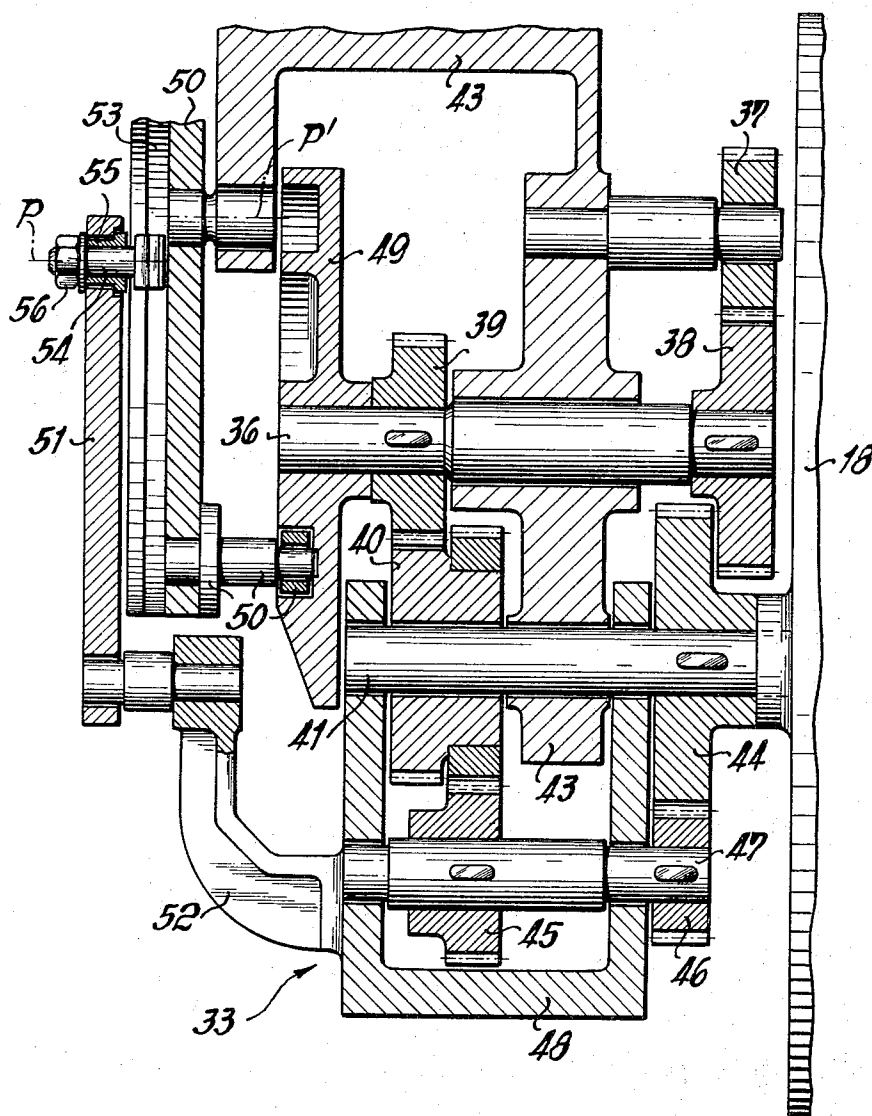


FIG. 4



BAG-MAKING MACHINE

The invention relates to a machine used in the manufacture of carrier bags and operative in cycles for successively adhering handles at intervals along a travelling web of bag-making material with the aid of individual reinforcing sheets which are each provided with adhesive on one side.

In such machines, provision is made for first means effective to sever and successively feed the reinforcing sheets to a holder carried by a rotary transfer drum in contact with the web, and separate means for severing and forming the handles and successively feeding same to a holder provided on a different transfer drum which is also in contact with the web. The use of two transfer drums makes the machine expensive and heavy. This complicated construction used to be chosen because the speed at which each reinforcing sheet and each handle is fed to the associated transfer drum should depend only on the cycle rate of the machine, not on the dimensions of the bags to be made. On the other hand, the peripheral speed of the respective drum at the moment of transfer should be equal to the speed of web travel (which is dependent on the size of bag being made). Having regard to the uniform and constant rotation of the transfer drums, it is therefore necessary to make their periphery equal to the pitch at which the handles are to be stuck to the web or a whole number multiple of such pitch. This interdependence of the periphery or diameter of the transfer drums and the pitch of the carrier handles calls for a different drum diameter whenever the pitch of the handles is to be changed. Apart from the fact that different drums must therefore be provided whenever a different bag size is being made, the machine is of costly construction because the means for feeding the reinforcing sheets and the means for feeding the carrier handles have to be adjustable in position and adjustably driven so as to conform to the different transfer drum diameters and the different peripheral speeds. Converting the machine to different carrier bag sizes is also time consuming. Further, the separate application of the reinforcing sheets and handles with the aid of separate transfer drums leaves room for errors because each reinforcing sheet and handle may not arrive on the web at the desired location relatively to one another.

The invention aims to avoid this source of error as far as possible and also to provide a smaller, lightweight and cheaper machine which can be more easily converted whenever production is changed to a different bag size.

According to the invention, in the manufacture of carrier bags, a machine operative in cycles for successively adhering handles at intervals along a travelling web of bag-making material with the aid of individual reinforcing sheets which are each provided with adhesive on one side comprises first means for successively feeding said reinforcing sheets at a first supply station substantially tangentially to a rotary transfer drum contacting said web at a transfer station, a holder mounted on said drum and adapted to hold one of the reinforcing sheets on the drum during rotation thereof, further means for successively feeding said handles at a second supply station substantially tangentially to the same said transfer drum at a location where a reinforcing sheet is being held, whereby the handle becomes stuck to the reinforcing sheet held by the holder, and variable

gearing for rotating said transfer drum such that, when the said holder thereon is passing through the first and second supply stations, the drum is turned at a mean peripheral speed which is proportional to the machine cycle and conforms to the tangential feeding speeds of the reinforcing sheets and handles at the supply stations and, when the said holder is passing through the transfer station for transfer of the reinforcing sheet and handle to the web, the drum is turned at a peripheral speed which is equal to the speed of web travel and proportional to said intervals and, when the said holder is passing through an intermediate arcuate path portion between said first and second stations, the drum is turned at a compensating peripheral speed which is different from the mean speed and is calculated so that the product of the difference between the web travel and mean peripheral speeds and the arc transcribed by the holder while the drum speed differs from the mean speed in favour of the web travel speed is equal to the negative product of the difference between the compensating and means speeds and the arc transcribed by the holder while the drum speed differs from the mean speed in favour of the compensating speed.

According to the invention, therefore, there is a single transfer drum and the variable gearing drives the drum so that some of the time it turns at a constant mean angular speed which is only proportional to the cycle rate of the machine, some of the time it turns at an angular speed which is only proportional to the speed of web travel (which varies depending on the bag size being made), and some of the time it turns at a compensating speed. The angular ranges at which the drum turns at the speed of web travel and the angular ranges at which the drum turns at the compensating speed, as well as the compensating speed itself, are selected so that the time integral of the speed differential between the actual speed and the mean speed is equal to zero during one complete machine cycle. In other words, the compensating speed and the associated angular region are always selected so that the time gain or time loss relatively to the travel at the mean speed is just equal to the time lack or time advance within the period of one cycle as occasioned by the angular speed corresponding to the speed of web travel.

In order that the supply stations can be suitably accommodated in the machine from a construction point of view and will not be too close to one another so that their accessibility will not be impaired, two said holders are preferably mounted on the transfer drum at diametrically opposite positions, the first and second supply stations and the transfer station being disposed about a peripheral portion of the drum subtending no more than 180°.

The variable gearing may comprise planet gearing comprising an input member driven at a constant angular speed proportional to the cycle rate of the machine, an output member positively connected to the transfer drum, and a superposed member driven by a drive capable of executing an operating cycle whenever the or each holder on the transfer drum passes a fixed point, the operating cycle imparted to the superposed member being composed of the phases: standstill, acceleration, constant angular speed, deceleration in one direction of rotation, standstill, acceleration, constant angular speed, deceleration in the opposite direction of rotation. To achieve the drive with only mechanical means which can be handled by unskilled labour, the

drive desirably comprises a four-part linkage having two rocker arms pivotable about respective fixed points and a coupling member interconnecting the two rocker arms, the said drive being operated by a cam connected to an input gear which turns at a rate proportional to the cycle rate of the machine, the cam being operatively coupled to one of the rocker arms, the input gear being connected to the input member of the planet gearing, and the other rocker arm being positively connected to the superposed member, it being also preferred that the supply stations be disposed adjacent peripheral positions of the transfer drum at which the superposed member is at a standstill and the transfer station be disposed adjacent a peripheral position of the transfer drum at which a constant angular speed is imparted to the superposed member. The input member may comprise a set of central gears and the output member a further set of central gears, the superposed member being constituted by the planet carrier of the planet gearing.

At a given cycle rate for the machine, the web speed is dependent on that dimension of the bags being made that extends lengthwise of the web. In order to permit the angular speed of the transfer drum in the angular range at which transfer takes place to be adapted to the speed of web travel corresponding to the size of bag being made, the coupling ratio between the two rocker arms of the four-part linkage is desirably variable, preferably from positive to negative values. In one form of the invention, the one rocker arm that is coupled to the cam comprises a groove extending through the fixed pivot point of the said one rocker arm, the groove containing a slide block which can be clamped therein and which is provided with a pivot for the coupling member. If the groove is arcuate and has a radius of curvature defined by the length of the coupling member, then variations of the coupling ratio from a central position of the four-part linkage will avoid a change in the angular position of the rocker arms.

Depending on whether the handle spacing on the web is larger or smaller than the spacing of the holders on the transfer drum, the speed of web travel will be higher or lower than the mean peripheral speed of the transfer drum and the peripheral speed of the transfer drum during transfer of the handle to the web must be set to be equal to the speed of web travel.

Thus, for higher web speeds corresponding to larger spacings of the handles on the web, after a handle has been supplied to the transfer drum the latter must be accelerated to the higher speed of web travel and this higher peripheral speed must be maintained during an angle of rotation of the drum that will suffice to transfer the handle to the web and stick it thereto by means of the reinforcing sheet. This, nowever, causes the transfer drum to advance relatively to the machine cycle and such advance can be balanced out only by a temporary reduction in the angular speed of the drum to below its mean angular speed. The negative acceleration (deceleration) imparted to the transfer drum by means of the invention so that the drum turns at an angular speed below its mean angular speed, all this taking place between the first and second supply stations, gives the drum a time lag relatively to the machine cycle and this time lag can be set to be equal to the advance that occurs later when the drum is accelerated to the higher speed of the web and then decelerated back to its mean angular speed. Accordingly, subsequent to transfer of

the reinforcing sheet and handle to the web, only one deceleration of the transfer drum to its mean angular speed is necessary in order to revert to the machine cycle.

5 Analogously, when setting to lower web speeds the drum is positively accelerated instead of decelerated between the two supply stations and, after the drum has received the handles, it is subjected to a negative acceleration (deceleration) to the lower web speed, such acceleration having a value so that the advance of the drum brought about by the positive acceleration is equal to the subsequent time lag brought about by the negative acceleration.

The means for non-uniformly driving the transfer drum in the manner described are preferably planet gearing in which the planet carrier is oscillated by means of a cam. The cam permits the output shaft of the planet gearing to be set into any desired non-uniform motion consisting of one acceleration and one deceleration during each full turn, the angles during which the acceleration and deceleration take place being selected by appropriately shaping the cam. In particular, by appropriate shaping of the cam, the latter can during its forward stroke bring the output to a speed above or below the angular speed of the input shaft, maintain this higher or lower speed at a constant value over a certain angle, and then return it to its original angular speed by appropriate deceleration or acceleration and, during its return stroke, it can decelerate or accelerate the angular speed of the output shaft to below or above that of the input shaft and then return it to its original mean angular speed; i.e., it can superimpose an accelerating and decelerating cycle onto each turn of the output shaft and these cycles balance each other out.

An example of the invention will now be described with reference to the accompanying diagrammatic drawings, wherein:

FIG. 1 is a side elevation of a machine for adhering carrier handles at equal intervals on a travelling web with the aid of reinforcing sheets;

FIG. 2 is a fragmentary plan view of the web after the handles have been applied thereto;

FIG. 3 is an axial side view of variable gearing for driving the transfer drum of the FIG. 1 machine, and

FIG. 4 is a radial side section of the FIG. 2 gearing but showing the components in a different position for the purpose of clarity.

An important component of the machine according to the invention is a transfer drum and this is shown at 18 in the example of FIG. 1. Mounted at the periphery of the drum there are two diametrically opposed holders comprising suction segments 19, 19'. The drum is rotated in the direction of the arrow *a* at a mean speed of *n* revolutions per unit time and this rotary speed is exactly half the machine cycle. Distributed along no more than half the periphery of the drum there is a conventional device 30 for supplying reinforcing sheets B which are provided with adhesive on one side, a continuously rotating carrier 5 for holding carrier handles T and transferring them to the reinforcing sheets B, and a guide roll 31 for a web 32 onto which the reinforcing sheets with handles are to be stuck at equal intervals *t*, the guide roll being effective to lead the web tangentially past the transfer drum.

The drum 18 is driven by planet gearing 33 shown in FIGS. 3 and 4 at a variable angular speed represented

by a polar coordinate diagram which has been included in the FIG. 1 illustration.

The device 30 comprises a pair of feed rollers 34 which are operated in sequence with the operating cycle of the machine to feed a web F, which is coated with adhesive on the top, to a rotary transverse severing device 35 which is effective to cut the reinforcing sheets B from the web F. The feeding speed of the web F is set so that it is advanced by the length of one reinforcing sheet during each machine cycle and fed by that amount to the severing device of which the tangential speed is substantially equal to the mean angular speed of the transfer drum 18 so that transfer of the reinforcing sheets B to the suction segments 19, 19' takes place substantially in synchronism with the transfer drum.

The carrier 5 which forms the carrying handles T and sticks them to the reinforcing sheets B arriving on the transfer drum 18 is turned at an angular speed corresponding to the cycle rate of the machine so that adhesion of the handles likewise takes place in synchronism with the transfer drum. Adhesion of the handles T and reinforcing sheets B to the web 32 of bag-making material by means of the guide roll 31 should likewise take place in synchronism with the transfer drum 18. The speed of travel of the web 32 (which at this stage is in the form of a flattened tube) should therefore be the same as the peripheral speed of the drum 18. With constant rotation of the illustrated transfer drum 18 at a rotary speed corresponding to the cycle rate of the machine, the spacing t of the carrying handles along the web 32 would therefore be equal to exactly one half of the drum periphery. Unless special provisions are made, the apparatus would thus be able to produce only those carrier bags having a length or width equal to half the drum periphery. Different pitches t corresponding to different bag sizes would necessitate different diameters for the transfer drum 18 and all the equipment associated with the drum would have to be correspondingly adjusted and repositioned. To avoid this, the drum is non-uniformly rotated by means of the planet gearing 33 shown in FIGS. 3 and 4 so that, as shown from the polar coordinate diagram included in FIG. 1, the drum is turned at its mean angular speed n only in the angular ranges α and β , at which a reinforcing sheet B is transferred to one of the suction segments and a handle T is transferred to the reinforcing sheet, respectively. At other times during the machine cycle, when each segment 19 or 19' is not passing through the angular ranges α or β , the drum is turned at a speed differing from the mean angular speed n . Thus, in the angular range γ disposed between the ranges α and β , the drum is turned at an angular speed n' . In the angular range δ , at which each reinforcing sheet and handle are together applied to the web 32 in synchronism therewith, the drum is turned at an angular speed n'' . In order that the rotary cycle of the transfer drum 18 can be accurately maintained at a constant value despite the aforementioned speed differences, the ranges γ and δ and their associated acceleration or deceleration ranges γ' , γ'' and δ' , δ'' , respectively, are so adapted to one another that the advance in the cycle rate of the drum 18 brought about in one of the angular ranges is exactly equal to the retardation occurring in the other range.

If, for example, the pitch t of the handles on the web 32 is to be increased relatively to that obtained by

means of the mean angular speed n of the transfer drum, then, as shown in full lines in FIG. 1, the angular speed of the drum must be increased to n'' in the angular range δ and the web 32 must be fed at a speed corresponding to the higher peripheral drum speed as occasioned by the higher rotary speed n'' . This temporary increase in speed in the angular range δ and the associated ranges δ' and δ'' gives rise to an advance by which the drum 18 departs from its normal operating cycle. To avoid this, a time lag exactly equal to the advance caused in the angular range δ' - δ - δ'' is brought about in the angular range γ' - γ - γ'' by temporarily reducing the angular speed of the drum to n' .

A reduction in the pitch t of the handles on the web 32 calls for the mean angular speed n of the drum to be reduced to n''' in the angular range γ (see the chain-dotted lines of the co-ordinate diagram) and to be increased to n'''' in the angular range γ' . Thus, again, the time lag in the drum cycle brought about in the angular range δ' - δ - δ'' is exactly equal to the advance brought about in the range γ' - γ - γ'' .

The speed differences from the mean angular speed n of the drum as hereinbefore described for the angular ranges δ' - δ - δ'' and γ' - γ - γ'' are imparted to the transfer drum by the planet gearing 33 of FIGS. 3 and 4.

Referring to FIGS. 3 and 4, the frame of the machine is indicated at 43. The planet gearing comprises an input shaft 36 driven at the rate of the single machine cycle by means of pinions 37, 38. By means of a gear 39, the input shaft drives a central set of input gears 40 that is loosely mounted on the output shaft 41 of the planet gearing. The shaft 41 is also mounted in the frame 43 and it has fixed to it a central output gear 44 and the transfer drum 18. The planet gears 45, 46 are fixed to a common shaft 47 and engage in the two central gears 40, 44. This shaft 47 is mounted in a rocker arm 48 which constitutes the planet carrier of the gearing 33 and which is pivotable about the output shaft 41. The total transmission ratio between the gears 39 to 44 is in the present example equal to 2:1 so that the drum 18 executes half a turn during each machine cycle and thereby assumes its mean angular speed n .

In order to impart to the drum the previously described speed differences n' , n'' , n''' , n'''' in the angular ranges γ' - γ - γ'' and δ' - δ - δ'' , there is fixed to the input shaft a cam plate 49 which oscillates the rocker arm 48 by means of a roller lever 50 mounted in the machine frame 43 and constituting the second rocker arm of a four-part linkage, a coupling member 51 and an arm 52 which is connected to the rocker arm 48. This oscillating movement imparted to the rocker arm or planet carrier 48 causes the output shaft 41 to be advanced or retarded (depending on the direction of oscillation), the angular position, size and speed of the advance or retardation being determined by the shape of the cam plate 49. Thus, in the present example, the cam portion $\alpha 1$ of the cam plate 49 (see FIG. 3) corresponds to the angle α of the drum in FIG. 1 and the cam portion $\beta 1$ corresponds to the angle β . Since the rocker arm or planet carrier 48 is stationary whilst the cam portions $\alpha 1$ and $\beta 1$ are operative, the transfer drum 18 will be turning at its mean angular speed n in the angular ranges α and β . The cam portions $\gamma 1$ and $\delta 1$ disposed between the portions $\alpha 1$ and $\beta 1$ correspond to the drum angles γ' - γ - γ'' and δ' - δ - δ'' , respectively. They are so designed that the drum will execute the speed variations n' and n'' or n''' and n'''' . The actual

values of these speed differences can be changed by exchanging the cam plate 49 and this would result in a change in the pitch t of the handles on the web 32.

To avoid replacement of the cam plate 49, the links 50, 51 interposed between the cam plate and the rocker arm 48 are constructed to make it possible to vary their transmission ratio (coupling ratio). The roller lever 50 contains a longitudinal groove 53 in which the pivot P of the coupling member 51 is adjustable. The cross-section of the groove 53 is for this purpose of swallow-tail or T-shape and the pivot pin 54 is of corresponding shape so that it is displaceable along the groove but secured against rotation. A clamping bush 55 and a nut 56 enable the pivot pin 54 to be fixed in any desired position along the groove 53, thereby determining the radial spacing of the pivot pin 54 from the fixed pivotal axis P' of the roller lever 50. As shown particularly clearly in FIG. 3, the roller lever 50 is a double-armed lever, i.e., its fixed pivotal axis P' is disposed intermediate its length. The groove 53 passes through the axis P'.

As indicated by the arrows a and b in FIG. 3, the input shaft 36 and thus the cam plate 49 are turned anti-clockwise whilst the output shaft 41 and thus the transfer drum 18 are turned clockwise by means of the gears 39 to 41.

If the pivot P of the coupling member 51 is disposed on the pivotal axis P' of the roller lever 50, the rocker arm or planet carrier 48 will not oscillate. The transfer drum 18 will then turn at its mean angular speed n which, as already described, corresponds to two machine cycles in the present example. The web 32, which in this case will be moved past the transfer drum in synchronism therewith at a speed corresponding to the cycle rate of the machine, will then be furnished with carrying handles at intervals t corresponding to half the periphery of the transfer drum.

By pivoting the coupling member 51 to the roller lever 50 on the lefthand side of the pivotal axis P' of the lever 50 as shown in FIG. 3, the drum 18 will undergo the speed changes n' and n'' as shown in full lines in the polar coordinate diagram included in FIG. 1. Pivoting the coupling member 51 to the lever 50 at the other side of its pivotal axis P' results in a reversal of the oscillating direction of the rocker arm 43 and consequent speed changes n''' and n'''' of the transfer drum. The larger the spacing between the pivotal axes P and P', the larger will be the differences of the speeds n' , n'' , n''' , n'''' from the mean angular speed n . Construction of the roller lever 50 as a double-armed lever with the groove passing through its pivotal axis therefore permits the angular speed of the transfer drum to be smoothly changed in the angular ranges γ and δ from a value below the mean angular speed n up to one exceeding the mean angular speed. Accordingly, within the limits determined by the lengths of the two lever arms any desired pitch t can be obtained for the carrying handles on the web 32. Since each handle spacing and thus each bag size corresponds to a certain spacing between the pivotal axis P' of the roller lever 50 and the pivot point P of the coupling member 51, the roller lever may, as shown in FIG. 3, be provided with a scale graduated in bag sizes. The scale permits the pivot point P of the coupling member 51 to be rapidly and accurately set.

I claim:

1. In the manufacture of carrier bags, a machine operative in cycles for successively adhering handles at intervals along a travelling web of bag-making material with the aid of individual reinforcing sheets which are each provided with adhesive on one side, said machine comprising first means for successively feeding said reinforcing sheets at a first supply station substantially tangentially to a rotary transfer drum which is contacting said web at a transfer station, a holder mounted on said drum and adapted to hold one of the reinforcing sheets on the drum during rotation thereof, further means for successively feeding said handles at a second supply station substantially tangentially to the same said transfer drum at a location where a reinforcing sheet is being held, whereby the handle becomes stuck to the reinforcing sheet held by the holder, and variable gearing for rotating said transfer drum such that, when the said holder thereon is passing through the first and second supply stations, the drum is turned at a mean peripheral speed which is proportional to the machine cycle and conforms to the tangential feeding speeds of the reinforcing sheets and handles at the supply stations and, when said holder is passing through the transfer station for transfer of the reinforcing sheet and handle to the web, the drum is turned at a peripheral speed which is equal to the speed of web travel and proportional to said intervals and, when the said holder is passing through an intermediate arcuate path portion between the said first and second stations, the drum is turned at a compensating peripheral speed which is different from the mean speed and is calculated so that the product of the difference between the web travel and mean peripheral speeds and the arc transcribed by the holder while the drum speed differs from the mean speed in favour of the web travel speed is equal to the negative product of the difference between the compensating and mean speeds and the arc transcribed by the holder while the drum speed differs from the mean speed in favour of the compensating speed.

2. A machine according to claim 1, wherein two said holders are mounted on the transfer drum at diametrically opposite positions, the said first and second supply stations and transfer station being disposed about a peripheral portion of the drum subtending no more than 180° .

3. A machine according to claim 1 or claim 2, wherein the variable gearing comprises planet gearing comprising an input member driven at a constant angular speed proportional to the cycle rate of the machine, an output member positively connected to the transfer drum, and a superposed member driven by a drive capable of executing an operating cycle whenever the or each holder on the transfer drum passes a fixed point, the operating cycle imparted to the superposed member being composed of the phases: standstill, acceleration, constant angular speed, deceleration in one direction of rotation, standstill, acceleration, constant angular speed, deceleration in the opposite direction of rotation.

4. A machine according to claim 3, wherein the drive comprises a four-part linkage having two rocker arms pivotable about respective fixed points and a coupling member interconnecting the two rocker arms, the said drive being operated by a cam connected to an input gear which turns at a rate proportional to the cycle rate of the machine, the cam being operatively coupled to one of the rocker arms, the input gear being connected

to the input member of the planet gearing, and the other rocker arm being positively connected to the superposed member, and wherein the supply stations are disposed adjacent peripheral positions of the transfer drum at which the superposed member is at a standstill and the transfer station is disposed adjacent a peripheral position of the transfer drum at which a constant angular speed is imparted to the superposed member.

5. A machine according to claim 4, wherein the input member comprises a set of central gears, the output member comprises a central gear, and the superposed member is the planet carrier.

6. A machine according to claim 4 or claim 5, wherein the coupling ratio between the two rocker arms is variable.

7. A machine according to claim 6, wherein the one rocker arm that is coupled to the cam comprises a groove extending through the fixed pivot point of the said one rocker arm, the groove containing a slide block which can be clamped therein and which is provided with a pivot for the coupling member.

8. A machine according to claim 7, wherein the groove is arcuate and has a radius of curvature defined by the length of the coupling member.

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