Principles of Weaving

R. Marks, F.T.I.
Principal Lecturer, Bolton Institute of Technology

and

A. T. C. Robinson, M.Sc.Tech., F.T.I.
*Former Head of the Department of Textiles,
Bolton Institute of Technology*

The Textile Institute
Manchester
1976

THE TEXTILE INSTITUTE
10 Blackfriars Street, Manchester M3 5DR

First published 1976
ISBN 0 900739 25 8

Text set in 10 pt. Photon Times, printed by photolithography, and bound in Great Britain at The Pitman Press, Bath

Introduction

This book attempts to explain the principles involved in the various mechanisms found on a loom. It is intended to provide a background for students pursuing Groups 1 and 2 of the A.T.I. examination and also for many areas of study in the TEC schemes.

In some instances, a mechanism is peculiar to one loom maker, but in most cases the principles are common to many manufacturers. In this latter respect, certain manufacturers are instanced in an attempt to guide the reader to a particular example of the mechanism or principle described. No disrespect is intended to the loom makers who use certain principles that are not mentioned. An attempt has been made to cover a wide cross-section of machinery makers. The popularity of specific looms and the authors' particular experience are obviously, however, an influencing factor in the examples chosen.

One of the major difficulties in referring to specific mechanisms is to ensure being up to date. This is always a problem in machine detail, and it is the reason why this book is basically aimed at principles. A detailed study of any mechanism will require the reader to develop the lines of thinking established in the following text and then proceed with a more detailed analysis of the manufacturer's literature.

CW 487

Contents

		PAGE
Introduction		
Chapter 1: An Outline of the Weaving Process		
1.1	Basic Mechanisms	1
1.2	Shedding	3
1.3	Picking	5
1.4	Beating-up	7
1.5	Other Loom Mechanisms	7
1.6	Classification of Weaving Machinery	8
Chapter 2: The Motion of the Healds, Sley, and Shuttle		
2.1	A Method of Indicating Loom-timing	11
2.2	The Motion of the Heald Shafts	12
2.3	The Motion of the Sley	13
2.4	Acceleration of the Shuttle: Picking	21
2.5	Retardation of the Shuttle: Checking	28
2.6	Shuttle Flight and its Timing	33
2.7	The Timing of the Primary Motions	37
2.8	The Geometry of the Warp Shed	38
Appendix: The Picking Mechanism as a Simple Elastic System		43
Chapter 3: Shedding Mechanisms		
3.1	The Scope of Tappet, Dobby, and Jacquard Shedding	47
3.2	Tappet Shedding	49
3.3	Dobby Shedding	59
3.4	Types of Shed	72
3.5	Jacquard Shedding	74
Chapter 4: Picking and Checking Mechanisms		
4.1	Types of Picking Mechanism	88
4.2	Conventional Picking Mechanisms	97
4.3	Power Required for Picking	103
4.4	Shuttle-checking Devices	105

PAGE

Chapter 5: Rapier, Jet, and Continuous Methods of Weft Insertion

5.1 Comparative Factors 109
5.2 Selvedges 116
5.3 Rapier-weaving 123
5.4 Jet-weaving 130
5.5 Continuous Weft Insertion 134

Chapter 6: Warp and Cloth Control

6.1 Cloth Formation 140
6.2 The Take-up Motion 151
6.3 The Let-off Motion 162
6.4 Temples 174

Appendix: Forces Acting at a Floating Back Roller 178

Chapter 7: Stop-motions

7.1 Warp Protection 182
7.2 Warp Stop-motions 186
7.3 Weft Stop-motions 190

Chapter 8: Weft-patterning

8.1 Comparison of Systems Available 195
8.2 Box Motions 197
8.3 Weft-patterning on Unconventional Looms 204

Chapter 9: Weft Replenishment

9.1 The Work of the Weaver 211
9.2 Feelers 215
9.3 Feeler Position 216
9.4 Types of Feeler 216
9.5 Single-shuttle Automatic Bobbin Change 220
9.6 Bobbin Loaders 225
9.7 Loom-winders 227
9.8 Multishuttle Weft Replenishment 230

Chapter 10: Loom Drives

10.1 Methods of Driving the Loom 234
10.2 Clutch Systems 236
10.3 Loom Control 239

References 244

Index 245

CHAPTER 1

An Outline of the Weaving Process

1.1 Basic Mechanisms

In order to interlace warp and weft threads to produce fabric on any type of weaving machine, three operations are necessary:

(*a*) *shedding*: separating the warp threads, which run down the fabric, into two layers to form a tunnel known as the *shed*;
(*b*) *picking*: passing the weft thread, which traverses across the fabric, through the shed; and
(*c*) *beating-up*: pushing the newly inserted length of weft, known as the *pick*, into the already woven fabric at a point known as the *fell*.

These three operations are often called the primary motions of weaving and must occur in a given sequence, but their precise timing in relation to one another is also of extreme importance and will be considered in detail later.

Two additional operations are essential if weaving is to be continuous:

(*d*) *warp control (or let-off)*: this motion delivers warp to the weaving area at the required rate and at a suitable constant tension by unwinding it from a flanged tube known as the *weaver's beam*; and
(*e*) *cloth control (or take-up)*: this motion withdraws fabric from the weaving area at the constant rate that will give the required pick-spacing and then winds it onto a roller.

In order to give some reality to these generalities, the diagram in Fig. 1.1 shows the passage of the warp through a loom.

The yarn from the warp beam passes round the back rest and comes forward through the drop wires of the warp stop-motion to the healds, which are responsible for separating the warp sheet for the purpose of shed formation. It then passes through the *reed*, which holds the threads at uniform spacing and is also responsible for beating-up the weft thread that has been left in the triangular warp shed formed by the two warp sheets and the reed. *Temples* hold the cloth firm at the fell to assist in the formation of a uniform fabric, which then passes over the front rest, round the take-up roller, and onto the cloth roller.

The mechanisms of a power-driven loom receive their motion from shafts that traverse from side to side in the loom and are driven from a motor. Their relative speeds are of importance since they govern the mechanisms that they drive.

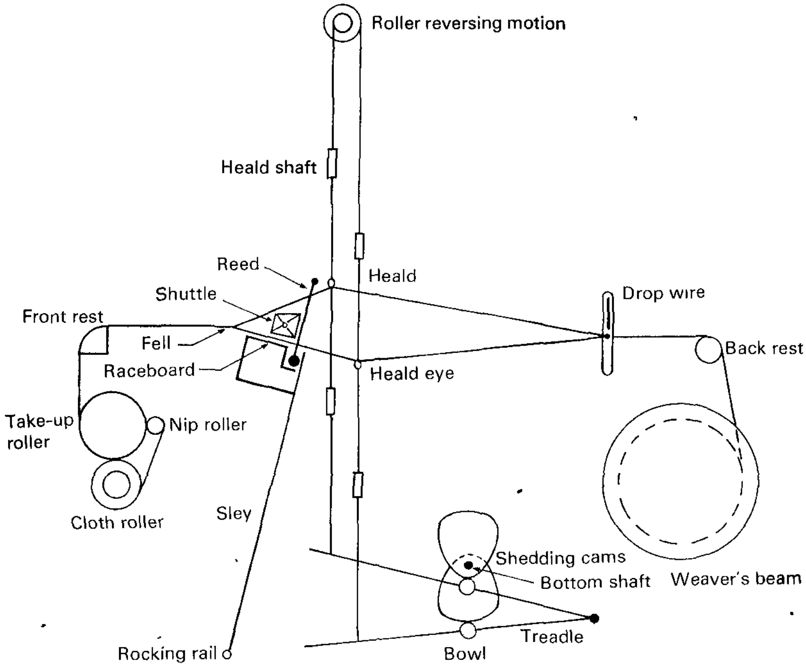


Fig. 1.1 A cross-section through the loom

Fig. 1.2 illustrates the three main shafts, the crankshaft being driven from the motor and making one revolution per pick. The ratio of the teeth of the gear wheels connecting this shaft to the bottom shaft is always 2:1 (40:20, 48:24, and 60:30 are typical examples), so that the bottom shaft will make one revolution every two picks.

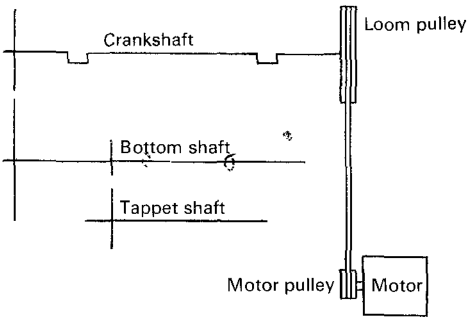


Fig. 1.2 The arrangement of loom shafts

Now consider the frequency with which each of the primary motions completes its action. The heald shafts (when producing plain weave) lift on alternate picks and thus complete their cycle once every two picks. There are two picking mechanisms on a loom, one at each side, and they operate alternately and so complete their cycle once every two picks. The *sley*, however, reciprocates once for every pick cycle to perform the beat-up. If the frequency of action of each mechanism is now related to the speed at which the loom shafts rotate, then the most appropriate driving source can be determined (Table 1.1).

Table 1.1

Mechanism	Frequency of Action	Loom Shaft
Shedding	Once every two picks	Bottom
Picking	Once every two picks	Bottom
Beating-up	Once every pick	Crank

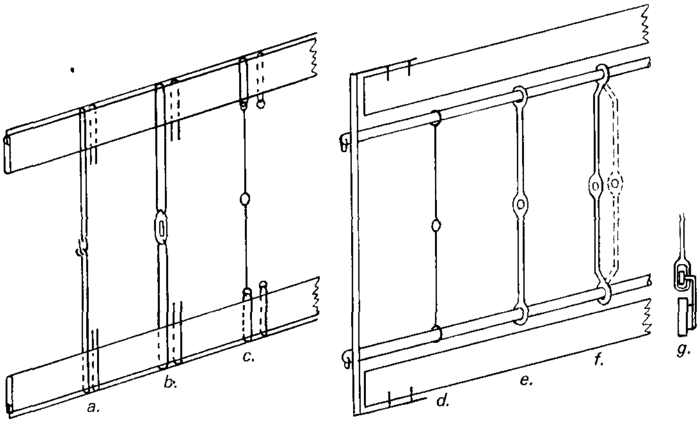
However, not all fabrics are in plain weave, and only plain-weave and weft-rib fabrics repeat their weave cycle on two picks. If the number of picks per repeat is increased from two, than it is no longer possible to control the shedding tappets from the bottom shaft, and an auxiliary shaft becomes necessary. It is also necessary to be able to change the speed at which this shaft can rotate so that fabrics repeating on different numbers of picks can be woven. A series of gear wheels is usually mounted on the auxiliary shaft for this purpose, but the size of the gear wheel on the bottom shaft should not be too small. Its diameter should be large enough to ensure that, when it is geared with the smallest wheel on the auxiliary shaft, the largest wheel on this latter shaft can rotate without touching the bottom shaft. A suitable arrangement of wheels is indicated in Table 1.2.

Table 1.2

Number of Picks per Repeat	Fraction of Revolution of Auxiliary Shaft per Pick	Ratio of Auxiliary to Crank Shaft	Ratio of Auxiliary to Bottom Shaft	Wheel on Bottom Shaft	Wheel on Auxiliary Shaft
2	1/2	1:2	1:1	36	36
3	1/3	1:3	2:3	36	54
4	1/4	1:4	1:2	36	72
5	1/5	1:5	2:5	36	90

1.2 Shedding

A simple cam-shedding motion is illustrated in Fig. 1.1. One of the two cams mounted on the bottom shaft depresses a bowl and treadle. The treadle is fulcrummed towards the back of the loom so that its front end will move down and pull its corresponding heald shaft down because they are joined by a series



Knitted Healds

a. Cord
b. Mail-eye
c. Wire

Metal Healds

d. Twisted-wire
e. Flat-steel (Simplex)
f Flat-steel (Duplex)
g. Riderless

Fig. 1.3 Types of heald

of connexions. Further connexions above the heald shafts cause the roller motion to rotate partially so that the other heald shaft and its treadle will be raised. As the cam unit continues to rotate, the second cam will cause the whole motion to be reversed.

The heald shaft is a wooden or metal frame, the width of which is slightly greater than that of the warp, and it is usually between 36 and 48 cm deep. *Healds* (Fig. 1.3) were originally made of twisted cord, but, although these were cheap, they had a relatively short life and could not be used, dismantled, and reassembled for use in the production of another quality of fabric. Twisted-wire healds attached to twisted cord around the top and bottom bars or staves of the heald frame lengthened the lifetime of the healds, but twisted-wire or flat-steel healds, which are free to move sideways on bars mounted just inside the framework of the heald frame, are now much more popular. Although they are more expensive, they have a much longer life and can be reassembled on the heald frames to suit any weaving requirements.

In weaving plain-weave fabrics, only two heald shafts are theoretically needed, the odd ends being drawn through the eyes of the healds on one heald shaft and the even ends being drawn through the eyes of the other shaft. If there are too many healds per unit length on one heald frame, however, there may be insufficient space to allow the ends controlled by the other heald shaft to move freely up and down as the two heald shafts change position. This may result in incorrect thread-interlacing (which is generally known as *stitching*) and will certainly increase the friction between the healds and warp threads, the result being an increase in the end-breakage rate and a reduction in the life of the healds. Furthermore, when one heald shaft controls too many ends, the forces required to drive the heald shaft from one position to the other become undesirably high. A general guide would suggest a maximum of 12 ends/cm per heald shaft, with 10 an even more realistic figure, although higher values could be more realistic in the weaving of very fine continuous-filament yarns, and lower figures are more common in the weaving of fabrics from coarse spun yarns. By using the quoted figures as a guideline, it will be realized that, in weaving fabrics having more than 20 but fewer than 40 ends/cm, it is more usual to use four heald shafts, while six heald shafts will be necessary in weaving fabrics having between 40 and 60 ends/cm.

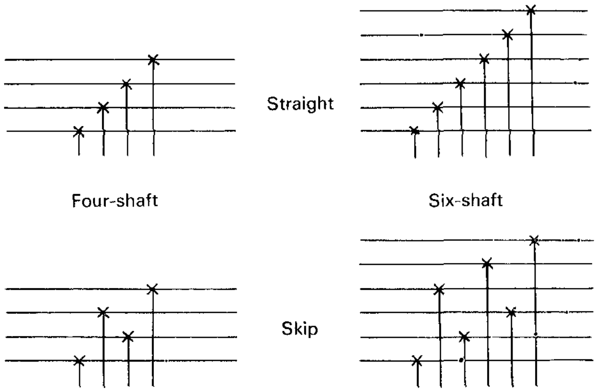


Fig. 1.4 Straight and skip drafts

Under these circumstances, it is often preferable to use skip rather than straight draft (Fig. 1.4), so that the odd ends are controlled by the heald shafts nearest to the front of the loom and the even ends will be controlled by the back heald shafts. The front shafts can then have common connexions to the treadle controlled by one of the cams, and the back shafts will be controlled by the other cam. Each group of heald shafts also have common connexions to the rollers of the reversing motion.

1.3 Picking

It has already been mentioned that the two picking mechanisms operate on alternate picks. The cams that activate them are mounted on the bottom shaft of the loom and are set at 180° to one another.

As the cam rotates, it will eventually push a bowl attached to the back end of the picking shaft in an upward direction (see Fig. 2.6). The partial rotation that this movement gives to the picking shaft will create a sharp inward movement of the lug straps, which are wrapped round the picking stick at their outer end. Since the picking stick is fulcrummed at its lower end, then the upper end will move quickly inwards, and the picker mounted near to the top of the stick will project the shuttle across the loom.

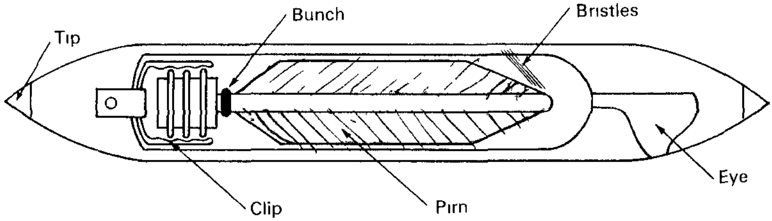


Fig. 1.5 The shuttle

The *shuttle* (Fig. 1.5) is a rectangular piece of wood, tapered at each end to a point so that entry into a partly opened shed is easier and more accurate. The main body of the shuttle is hollowed out to accommodate the package known as the *pirn*, which contains the weft yarn. The insides of the walls of the shuttle are lined with fur, bristles, or loops to control the yarn as it unwinds from the pirn. There is a clamp arrangement at one end of the shuttle to hold the pirn steady during weaving (or alternatively the pirn is mounted on a spindle), and at the other end of the shuttle there is a unit known as the *shuttle eye*, in which there is an arrangement to control the weaving tension of the weft thread as it is delivered from the shuttle. A groove along the front wall of the shuttle prevents

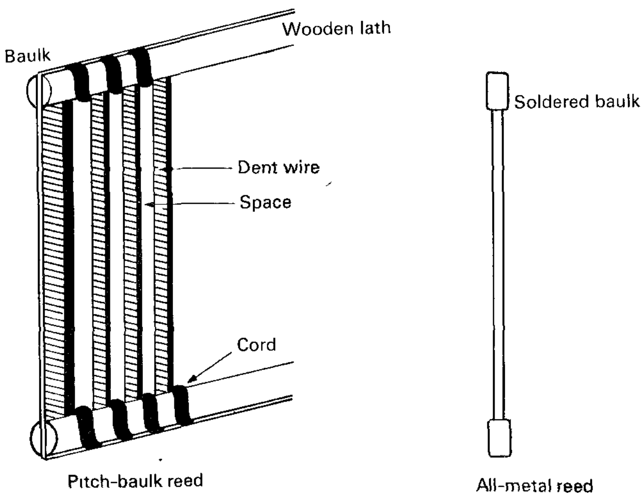


Fig. 1.6 Reeds

the weft from being trapped between the shuttle and the shuttle-box front, and a second groove along the base reduces abrasion by the shuttle on the bottom warp sheet as it traverses the loom from one box to the other.

Correct shuttle flight is essential if the shuttle is to arrive successfully in the opposite box, and to a large extent the reed and raceboard, which are integral parts of the sley, are responsible for ensuring satisfactory flight. At each end of the raceboard, there is a shuttle-box, and correct flight is achieved initially by ensuring that the back and base of the shuttle-boxes are perfectly aligned with the reed and raceboard, respectively.

The reed (Fig. 1.6) is a closed comb of flat metal strips, which are uniformly spaced at intervals corresponding to the required spacing of the warp ends. The top and bottom baulks of the reed, which close the comb, can be made of wood wrapped with string and set in pitch (*pitch-baulk reed*) to produce a cheaper product, or alternatively the wires may be soldered in position on a metal bar (*all-metal reed*) to produce a unit of much greater accuracy. The spaces between the metal strips through which the ends pass are known as *dents*. The main functions of the reed are to hold the warp threads at uniform spacing and to beat-up the newly inserted picks of weft in addition to supporting the shuttle durings its traverse of the loom.

1.4 Beating-up

The sley must reciprocate for the reed to push the weft into the fell of the cloth, and the two sley swords therefore extend down from the raceboard to a fulcrum point known as the *rocking shaft* (see Figure 2.1). A connecting rod (*crankarm*) is connected to the back of each of the two sley swords by a pin (*swordpin*) just below the level of the raceboard, and its other end fastens round the bend in the crankshaft, which is known as the *crank*. As the crankshaft rotates, the crankarm and thus the top end of the sley are made to reciprocate with a movement that approximates to simple harmonic motion.

1.5 Other Loom Mechanisms

Uniform tension in the warp sheet is essential during weaving, and this is achieved by controlling the rate at which the warp beam is allowed to rotate by the let-off motion. Similarly, strict control of the rate of cloth withdrawal from the fell is essential if uniform pick-spacing and thus regular cloth appearance are to be achieved, and for this purpose a take-up motion is incorporated.

A series of other mechanisms is used in the interests of productivity and quality. The warp-protector motion will stop the loom to prevent excessive damage to the warp threads, cloth, and reed if a shuttle becomes trapped between the top and bottom shed lines and the reed by failing to complete its traverse. Warp and weft stop-motions will stop the loom almost immediately a warp end or a weft thread breaks. Automatic replacement of the weft package in the shuttle, when almost all the yarn on the pirn has been used up, is achieved with the aid of a detecting-feeler motion and a bobbin- or shuttle-transfer mechanism. Furthermore, it is possible to vary the weft being inserted by having more than one shuttle-box in a unit at the end of the sley. The shuttle in each shuttle-box may contain a weft of different colour or character, and the appropriate shuttle-box is positioned opposite the raceboard just before insertion of the pick.

1.6 Classification of Weaving Machinery

So far, only looms using a weft-carrying shuttle have been considered. These looms have a wide degree of versatility and fall into four main classes:

(*a*) *hand looms*: still used in relatively large quantities for the production of all types of fabrics in the less-developed countries, but also used in the United Kingdom for the production of certain classic brocades, tapestries, and tweeds;

(*b*) *non-automatic power looms*: these machines are being used in ever-decreasing numbers, especially in the developed countries, but they seem likely to retain a certain usefulness in the production of specialist fabrics, such as industrial fabrics woven from heavy coarse wefts on wide looms;

(*c*) *conventional automatic looms*: machines that have gained world-wide popularity because of their advantages of versatility and relative cheapness; and

(*d*) *circular looms*: strictly limited in their applications (i.e., tubular fabrics for hose-pipes and sacks), but they do achieve the ideal of high weft-insertion rates from relatively low shuttle speeds because insertion of the weft is continuous.

The inherent problems of pirn-winding for shuttle looms and the dynamic problems created by the picking and checking mechanisms in these looms have encouraged loom makers to investigate and develop various alternative means of weft insertion:

(*e*) *single or multiple grippers or projectiles;*

(*f*) *gripper heads mounted on rapiers*, which may be rigid or flexible;

(*g*) *needles*;

(*h*) *fluid jets of air or water*; and

(*i*) *various other methods.*

Of the many various types of gripper-shuttle loom developed, the multiple-gripper Sulzer weaving machine was the first, and to date it is the only loom of its type to become widely established. Initially, it was suitable for only a limited range of relatively plain fabrics, produced from spun yarns. Over the years, it has been developed to weave dobby, jacquard, and terry fabrics, from continuous-filament as well as spun yarns, and with up to six different colours of weft. It is probably the most versatile of the unconventional looms and has a combination of width and high speed that gives it very high productivity attainments.[1]

Although many loom makers have in the past attempted to develop single-gripper looms, their efforts now seem to have been discontinued, probably because their only advantages appear to be low price and ease of conversion from a conventional shuttle loom. They still require picking and checking mechanisms and only share the advantage of random weft selection from stationary supply packages at the side of the loom with multiple-gripper and rapier looms. The one area in which they may be used to advantage is the weaving of very coarse soft-spun wefts, for which maximum weft velocities may cause weft breaks in rapier looms or the drag on the carrier may create problems in projecting the grippers at the required velocity in a multiple-gripper loom. Their apparent trend towards obsolescence precludes further mention in this book.

Rapier looms are made in a variety of types. The rapiers may be required to extend across the full width of the warp, in which case they will be of a rigid construction, or alternatively two rapiers may enter the shed from opposite sides of the loom and transfer the weft from one rapier head to the other at a point near the centre of the loom. In the latter case, the rapiers may be either rigid or flexible. The future of this method of weft insertion would appear to lie mainly in the field of multi-colour work, since their rates of weft insertion are generally only comparable with those of conventional automatic looms. It should also be remembered that only 50% of the rapier movement is utilized in weft insertion, and, for single-rapier looms, this wasted movement is also a time loss.

Needle insertion is used primarily in the weaving of Axminster carpets and for the production of certain narrow fabrics, such as tapes, but, since the weft is threaded through the eye of the needle, each pick is a double one, and this severely restricts the potential of this method in other applications.

The technique of inserting weft in a fluid jet achieves weft-insertion rates comparable with those achieved by the Sulzer weaving machine, but there are width limitations. Water-jet looms have been shown to be most economical for producing certain types of plain continuous-filament fabric, especially when they are produced from hydrophobic synthetic-fibre yarns. Air-jet looms are more popular for the weaving of plain cotton-type fabrics, although they are not entirely unsuitable for continuous-filament yarns.

Various other methods of weft insertion have been proposed from time to time. One or two are currently being developed, and of these probably the most advanced is the Rüti TWR ripple-shedding loom, which achieves high weft-insertion rates from low projectile speeds. The system is similar in principle to that used in circular looms, but the loom is flat, and each of several weft carriers, which are traversing the loom successively at the same time, is supplied with one pre-measured pick length for each traverse of the loom.

Since none of the shuttleless looms is as versatile as a conventional automatic loom, it is evident that their success must rest on other criteria. They tend to have two main advantages when compared with the conventional shuttle loom in that they will have a higher production rate and also that a given production rate will require less labour, although the advantage of a free choice of weft selection in certain types of loom is not insignificant.

The productivity of a loom is most conveniently expressed as its rate of weft insertion (i.e., the number of metres of weft inserted per minute when the loom runs without stopping). A high rate of weft insertion can be achieved either by using very wide looms or by a high rate of pick insertion, or by a combination of both. Sulzer weaving machines, for example, are wide, and, for cotton-type fabrics, the normal width of machine allows the production of fabric up to 330 cm wide (machines of 389 and 541 cm are also available but have been introduced mainly with the tufted-carpet-backing trade in mind). The machine runs at approximately twice the speed of a conventional loom of the same width, which explains its high rate of weft insertion. The width of air- and water-jet looms is strictly limited, and their high productivity results from the high speeds of which they are capable. All three types of loom show a saving in space as compared with a conventional loom of the same width. There is an even greater saving in space per unit output. The saving in labour already referred to results partly from the direct supply of weft from a large stationary package and partly from a lower yarn-breakage rate associated with the method of weft insertion.

Overcoming the technical problems essential to the success of a new method of weft insertion is not enough. The economics of production, which include initial cost, maintenance, depreciation, and labour costs, must be competitive. It is usually found that a new method of weft insertion is most competitive over a limited range of fabric types and constructions. This can only be determined after large-scale production trials, and the process of establishing new methods is therefore slow and potentially very costly.

CHAPTER 2

The Motion of the Healds, Sley, and Shuttle

2.1 A Method of Indicating Loom-timing

The numerous motions and mechanisms of an automatic loom must be set in the correct timing relationship to each other. We therefore need a simple and unambiguous method of identifying and stating these timings. The timings of most of the events in the loom cycle are governed by the positions of the reed and the sley. For example, the shed must be open and the reed well on its way towards the back of the loom before the shed is large enough to admit the shuttle. This determines the timing of the picking mechanism. Again, automatic replenishment of the weft bobbin can occur only when the sley is momentarily at rest in its most forward position. The warp-protector motion must operate when the sley is about halfway through its forward motion. These timings are directly related to the position of the reed and sley. Others are related to it indirectly. For example, the timing of the weft-break stop-motion is related to the flight of the shuttle, which is governed by the position of the reed.

The loom overlooker often adjusts and times loom mechanisms with the reed or sley at a particular distance—measured by a steel rule or a gauge—from a fixed mark on the loom frame. This is convenient for practical purposes but not for studying the principles of weaving, for which it is better to state timings in relation to the angular position of the crankshaft from which the reed and sley derive their motion. In Fig. 2.1, the circle represents the path traced out by the axis of the crankpin. This path is called the *crank circle*. The arrow on the crank circle in the main diagram shows the usual direction of rotation of the crankshaft. In this diagram, the crank and crankarm are in line, and the sley is in its most forward position. The crank circle is graduated in degrees from this point in the direction of rotation of the crankshaft. Any timing can be stated in degrees, as, for example, 'healds level at 300°.

In certain makes of loom (e.g., Draper and Crompton & Knowles), the crankshaft turns in the opposite direction, as indicated by the inset diagram in Fig. 2.1. Since the crank circle is graduated in the direction of rotation, the timing referred to above would be stated as 'healds level at 300°' as before.

With reference to the main diagram in Fig. 2.1, the terms *front*, *bottom*, *back*, and *top centre* are sometimes used to correspond to 0, 90, 180, and 270° on the crank circle. The timing quoted above as 'healds level at 300°' would then become 'healds level at 30° after top centre'. If, however, the crankshaft rotates in the opposite direction, as in the inset in Fig. 2.1, the same timing becomes 'healds level at 30° after bottom centre'. For a timing stated in this

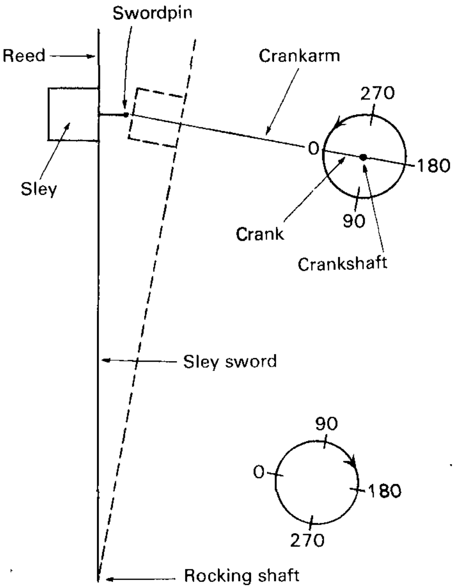


Fig. 2.1 Loom-timing

way to be meaningful, the direction of rotation of the crankshaft must therefore be stated. Furthermore, the inclination of the line through the axes of the swordpin and crankshaft causes the timing circle to be tilted, so that 270° in the main diagram is not at the top of the crank circle. Both these ambiguities are avoided by stating timings in degrees measured in the direction of rotation from the position of the axis of the crankpin when the reed is in its most forward position. This is the method we shall use.

Some modern looms are provided with a graduated disc or wheel on the crankshaft and a fixed pointer to enable settings to be made in relation to the angular position of the crankshaft, and no doubt this tendency will grow. In the absence of such provision, one may use a timing gauge consisting of a disc graduated in degrees and having a weighted pointer mounted freely and coaxially with the disc. The rear of the disc carries a permanent magnet, which enables it to be attached to the end of the crankshaft. With the reed in its most forward position, the disc is adjusted so that the pointer is opposite 0° on the graduated scale. The loom may then be turned to any desired position, the disc turning with it and the pointer remaining vertical and indicating the angular position of the crankshaft. Such a timing device is essential for teaching the principles of weaving.

2.2 The Motion of the Heald Shafts

It will be shown in the next section that the passage of the weft through the shed usually occupies rather more than one-third of the loom cycle (i.e., rather more than 120° of crankshaft rotation). Provided that there are no other overriding conditions, it is sensible to arrange for the shed to remain fully open during this period. This is easy to arrange if the heald shafts are operated by cams, as they are when tappet-shedding or cam-operated dobbies are used, by designing the cams to give a period of dwell of about 120° of crankshaft rotation. During this period, the shed will remain fully open, and, if the cams are to give simple harmonic motion to the heald shafts between the periods of dwell, the design of their profile involves only a simple geometrical construction, which can be found, for example, in Hanton's 'Mechanics for Textile Students'[1].

In weaving plain cloth, the upper and lower warp sheets that form the shed interchange position every pick. The ends pass each other when the heald shafts are level and when their relative velocity is a maximum. The rubbing action between ends in the same dent of the reed as they pass each other tends to cause end-breakages, especially when the warp is fine and closely set. There is some evidence that the frequency of end-breakages due to this cause tends to increase as the relative velocity of the healds increases. For a given loom speed, the relative velocity of the healds will increase if the period of dwell is increased, which thus reduces the time available for the movement of the healds. On the other hand, a longer period of dwell may be desirable in order to facilitate the passage of the weft, especially if other considerations require the shed timing to be such that the passage of the weft does not coincide exactly with the period of dwell (see Fig. 2.11*B*). For most types of conventional loom, a dwell of about 120° offers a suitable compromise, but this may be increased to 180° in wide looms (e.g., sheeting looms) in order to avoid excessively high shuttle speeds.

2.3 The Motion of the Sley

2.3.1 Factors Affecting the Motion

In most looms, the sley is operated by cranks and crankarms, and its motion approximates to simple harmonic. The extent to which it deviates from simple harmonic motion has practical significance and is governed by the following factors:

(*a*) the radius of the arc along which the axis of the swordpin reciprocates,
(*b*) the relative heights of the swordpin and crankshaft, and
(*c*) the length of the crank in relation to that of the crankarm.

(*a*) The swordpin (see Fig. 2.1) travels along an arc of a circle centred upon the rocking shaft. This modifies the movement of the swordpin and hence of the reed, but, since the radius of the arc is large—usually about 0·75 m—the effect is small enough to be neglected here.

(*b*) In Fig. 2.1, the axis of the crankshaft is on a line passing through the extreme positions of the axis of the swordpin, and the reed is vertical at beat-up.

This may be regarded as the normal arrangement. Raising or lowering the crankshaft from its normal position affects both the extent and the character of the motion of the swordpin. This has been discussed by Hanton[2] on p. 136 of his book 'The Mechanics of Textile Machinery'. Hanton showed that moving the crankshaft 10 cm up or down from its normal position increases the distance moved by the swordpin by about 8%. This has no practical significance, since the increase could easily be obtained, if required, by slightly lengthening the cranks.

The other effect of raising the crankshaft is to increase the swordpin's velocity as it approaches its most forward position and to decrease it as it approaches its most backward position. This tends to have the double effect of increasing the effectiveness of beat-up and of allowing more time for the passage of the shuttle. If, however, in Hanton's example, we restore the displacement of the swordpin to its original value by shortening the cranks, the difference between the 'normal' and '10-cm-above-normal' displacement curves largely disappears. Since a 10-cm movement is extreme, we may reasonably conclude

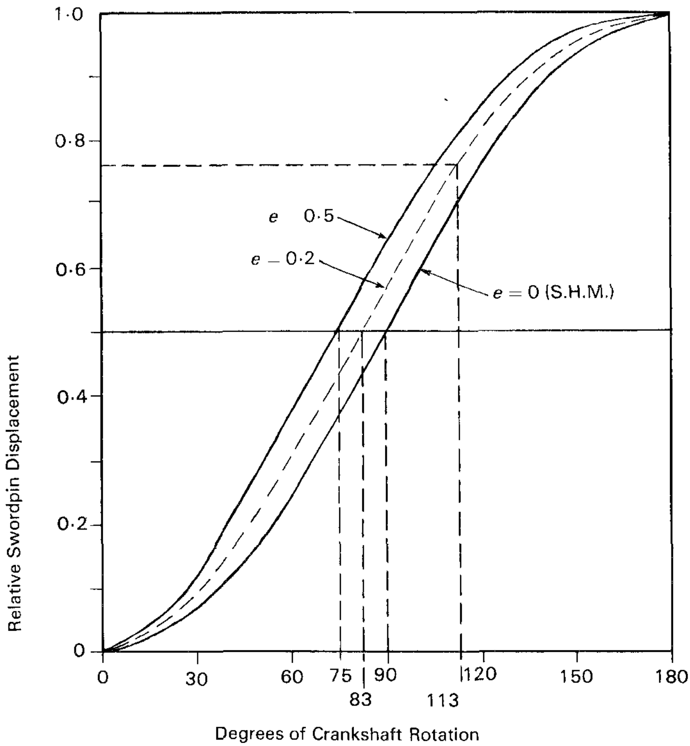


Fig. 2.2 The motion of the sley

that the effect of raising the crankshaft above its normal position is not large. Such effect as it has is advantageous, so the crankshaft tends to be positioned either in the normal position or somewhat above it.

(*c*) The ratio r/l, where r is the radius of the crank circle and l is the length of the crankarm, is called the *sley-eccentricity ratio, e.* The larger it is, the greater is the deviation from simple harmonic motion. Sley eccentricity has effects that need to be considered in some detail.

2.3.2 Sley Eccentricity and its Effects

The curves in Fig. 2.2 show the displacement of the swordpin, expressed as a fraction of its total displacement, for half a revolution of the crankshaft with the crankshaft in the normal position. With this normal position, the curves for the second half of the crankshaft's rotation would be mirror images of those for the first half.

With simple harmonic motion (corresponding to $e = 0$ and indefinitely long crankarms) and a normal position of the crankshaft, the swordpin attains its maximum velocity and exactly half its maximum displacement at 90° and again at 270°. With a finite value of e, that is to say, with any arrangement possible in practice if the sley is crank-driven, the swordpin attains its maximum velocity and half its maximum displacement earlier on its backward movement and later on its forward movement. This is illustrated in Table 2.1.

Table 2.1

Eccentricity Ratio (e)	Positions of Crankshaft at Half Maximum Displacement	Period during which Displacement is at Least Half Maximum
0·0	90° and 270°	180°
0·2	83° and 277°	194°
0·5	75° and 285°	210°

Thus, as the sley-eccentricity ratio increases, the sley remains longer nearer its most backward position, and more time is available for the passage of the shuttle. Furthermore, since the velocity of the swordpin is proportional to the slope of the curves in Fig. 2.2, it is clear that increasing the sley-eccentricity ratio increases the velocity of the sley around beat-up. We therefore have the same two effects as were noted in the preceding section in connexion with the height of the crankshaft. The effects of altering the sley-eccentricity ratio within the practicable limits are, however, greater than those obtained by altering the height of the crankshaft. We may summarize the advantages of a high sley-eccentricity ratio as follows:

(*a*) it facilitates the passage of the shuttle; and
(*b*) it tends to increase the effectiveness of beat-up.

The disadvantage associated with a high sley-eccentricity ratio are of a mechanical kind. A high value implies rapid acceleration and deceleration of

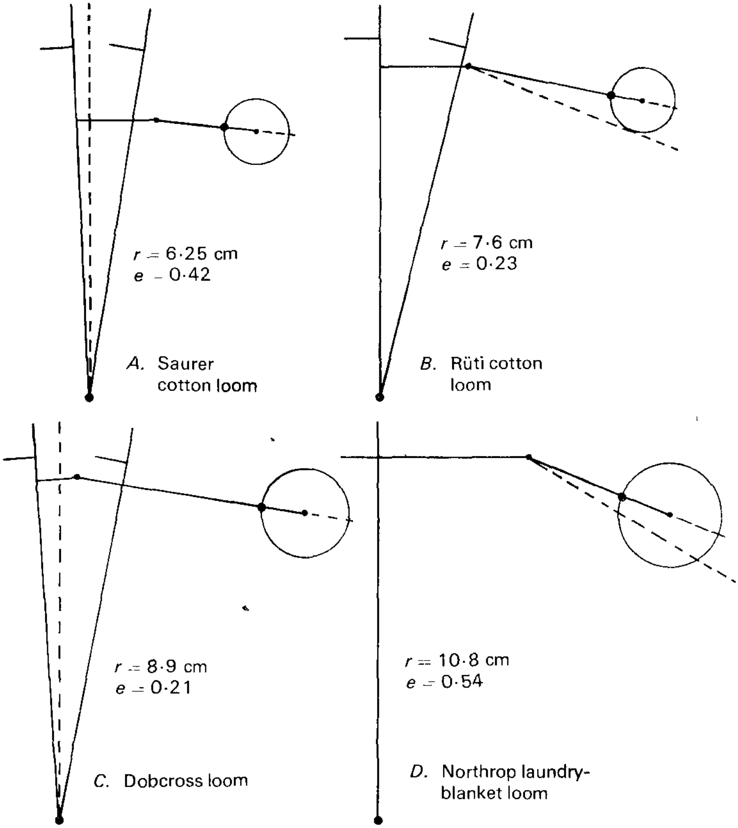


Fig. 2.3 The geometry of the sley

apparently very little room for the heald shafts, but this is not so because the crankshaft does not extend across the loom. The crankshaft position is normal, and the reed has a slight forward inclination at beat-up, as is indicated by the vertical dotted line.

Table 2.3 Some Typical Loom Dimensions

Loom Maker	Type	R (cm)	L (cm)	$e = R/L$
(A) Saurer	Cotton, tappet	6·25	15·0	0·42
(B) Rüti	Cotton, dobby	7·6	33·5	0·23
(C) Picanol	Cotton, tappet	7·2	32·4	0·225
(D) Prince (water-jet)	Rayon, tappet	3·33	22·9	0·145
(E) Dobcross	Worsted, dobby	8·9	43·2	0·21
(F) Northrop	Industrial-blanket, tappet	10·8	20·3	0·54

2.3.3.3 Rüti Cotton Loom (B)

Room is required to accommodate twenty heald shafts, and this is obtained partly by fairly long extensions behind the sley swords and partly by relatively long crankarms, which give a rather low eccentricity ratio. The crankshaft is above its normal position, which would lie on the dotted line. The reed is vertical at beat-up.

2.3.3.4 Dobcross Woollen and Worsted Loom (E)

This loom requires more heald-shaft space than a cotton dobby loom for two reasons. The wider, heavier fabrics require more rigid, and hence thicker, heald shafts, which occupy more space, and in addition the dobby is designed to control up to 28 heald shafts. The cranks are longer than in the previous examples in order to accommodate larger shuttles. The reed has a forward inclination at beat-up, and the crankshaft position is normal.

2.3.3.5 Northrop Industrial-blanket Loom (F)

This is a heavy, wide loom with a reedspace of 5·33 m. Its very high eccentricity ratio (0·53) is practicable because the loom speed is only 65 picks/min. As we have seen, a high eccentricity ratio gives more time for the passage of the shuttle, which has a long way to travel in this loom. The extensions behind the sley swords are exceptionally long (35 cm) so as to leave room for the heald shafts in spite of the very short cranks (20·3 cm). The reed is vertical at beat-up.

In three of the looms whose particulars are tabulated, the reed has a slight forward inclination at beat-up. One school of thought holds this to be beneficial, although the evidence for this opinion is not altogether conclusive. In a conventional loom, the fell of the cloth makes contact with the reed about one-third of the distance upwards from the bottom of the reed wires. The forward inclination of the reed would tend to prevent the cloth from rising on impact with the reed. If this happened, the fell would strike the reed nearer the middle of the reed wires, where they would be less rigid, and hence beat-up might be less effective. It seems more likely, however, that the reason for using a forward inclination of the reed is connected with the formation of the cloth at the fell. This is discussed in another connexion in Section 2.8.2 of this chapter and illustrated in Fig. 2.16*A*. With the configuration shown in this diagram, it seems likely that a forward inclination of the reed at beat-up would be effective in securing a closer packing of the picks if this were required.

2.3.4 Sleys that Dwell

The high rate of weft insertion that is one of the main objectives in the design of shuttleless looms may be obtained in several ways, including the use of:

(*a*) a very wide loom running comparatively slowly (e.g., a wide rapier loom);

(*b*) a relatively narrow loom running at high speed (e.g., an air- or water-jet loom);
(*c*) a wide loom that runs faster than a shuttle loom of similar width (e.g., the Sulzer loom);
(*d*) a low-speed loom in which several picks are inserted during each loom cycle (e.g., multiphase flat or circular looms).

In (*b*) and (*c*) above, which cover a large number of shuttleless looms, the designer is seeking to achieve high loom speeds. To be successful, he must keep the mass of the sley and the distance through which it reciprocates as low as possible. Since the device used to carry the weft through the shed will have a smaller cross-section than a shuttle, he can use a smaller shed and hence a smaller sweep of the sley.

In order to minimize the weight of the sley, the designer will avoid mounting on it any heavy parts associated with weft insertion, and he will mount them on the loom frame instead. He may, indeed, be forced to do this because they would not perform satisfactorily if subjected to the reciprocating motion of the sley. We may therefore expect the sley in most shuttleless looms to be free of parts associated with weft insertion, except perhaps for some means of guiding the weft through the shed.

If the sley provides a means of guiding the weft-insertion device through the shed, as in the Sulzer gripper loom and some flexible-rapier looms, then the sley must dwell in its most backward position during the whole of the time occupied by weft insertion. If there is no means of controlling the weft insertion device within the warp shed, as in rigid-rapier looms, the Draper DSL, and other

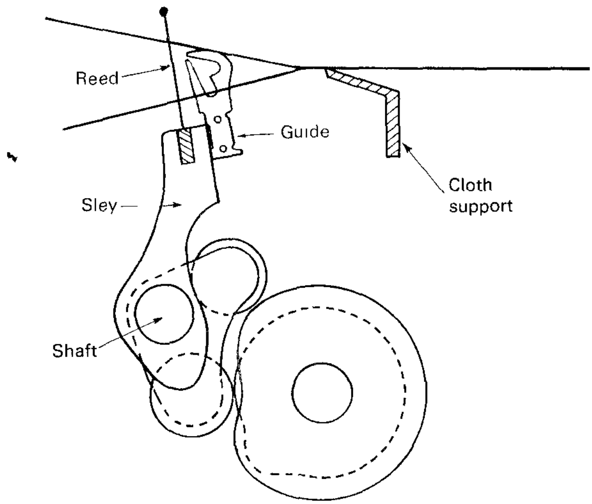


Fig. 2.4 A cam-driven sley

flexible-rapier looms, then it is possible to reduce the sley dwell period, but only by a small amount.

The introduction of a period of dwell in the motion of the sley precludes driving it by cranks and crankarms and requires the use of positively acting cams.

The Sulzer sley drive uses several pairs of matched cams, spaced at intervals across the width of the sley. Each unit (Fig. 2.4) has a pair of cams, with an anti-friction roller for each cam. The rollers are carried by an L-shaped lever, fixed to a horizontal shaft, which also carries short arms to which the reed and gripper guides are fixed. The mass of the reciprocating parts is small compared with that of the sley in a shuttle loom. The motion of the sley, including its period of dwell, is positively controlled. The diagram shows the parts in the position they occupy during the period of dwell when the gripper is passing through the shed. As the reed moves forward to beat-up the pick of weft, the motion is such that the gripper guides retract from the shed and lie below the cloth near the fell during beat-up.

In most models of the Sulzer weaving machine, the whole of the sley movement is completed in 105° of the pick cycle, which thus allows the sley to dwell in its back position for 255°. In the narrow, single-colour machines, which run at the highest speed, the sley movement is spread over 140° to prevent excessive vibration, which leaves a dwell period of 220°. This reduction in the number of degrees of the pick cycle available for weft insertion is acceptable because the weft carrier does not have as far to travel as in the wider machines.

The advantages of a period of sley dwell, in order to increase the time available for weft insertion and thus allow a higher loom speed to be achieved, are so obvious that the possibility of including a similar feature on a shuttle loom should be considered. The main difficulty is that the amplitude of the sley's movement necessary to allow a large enough shed would produce very large forces of acceleration and retardation if its movement were confined to a relatively small part of the pick cycle. A secondary consideration is that the movement of the sley, especially during the latter part of the shuttle's transit, helps to control the flight of the shuttle. During the latter part of its flight, the sley is moving forward and rising, and this helps to maintain contact between the shuttle and the reed and raceboard and thus ensure that the shuttle is boxed correctly. This effect would be absent if the sley dwelt during the period of shuttle flight.

2.4 Acceleration of the Shuttle: Picking

2.4.1 An Experimental Study

The acceleration of the shuttle during picking and its retardation during checking were studied experimentally by Thomas and Vincent[3–5] at the Shirley Institute, which was then the British Cotton Industry Research Association. This was a pioneering piece of work because it was the first time on record that an effective study of the running loom had been successfully undertaken. The method used was an optical one, and, although the science of electronics nowadays affords more elegant and convenient methods, Thomas and Vincent were able to plot with some accuracy the displacement of the shuttle during picking and checking against the angular position of the crankshaft and also against a time scale, which enabled them to take account of variations in loom speed during the loom cycle.

The displacement curve for the shuttle obtained in this way, with the loom running at full speed, is quite different from the curve plotted from observations made while the loom is turned over slowly by hand. The nature of this

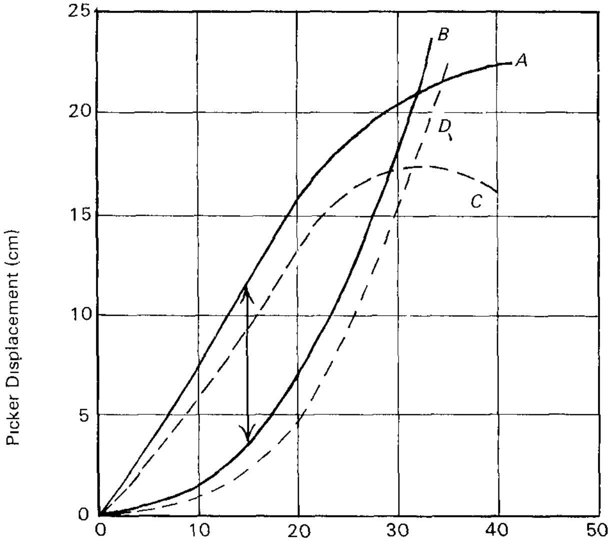


Fig. 2.5 Nominal and actual picker displacement

difference is shown in Fig. 2.5. Curves A and B relate to a non-automatic cone-overpick loom—a type common in Lancashire and elsewhere when the results of the investigation were published in 1949. Curve A, obtained from measurements made when the loom was turned over by hand, is called the 'nominal' displacement curve. Curve B is the 'actual' displacement curve obtained from observations of the running loom. On this graph, 0° on the abscissa corresponds to the position of the crankshaft when the picker and the shuttle are just about to begin to move. This usually occurs at about 75° on the normal timing circle as defined in Section 2.1.

Comparing the two curves, we note that the nominal and actual displacements of the picker and shuttle are the same at the beginning (before they have begun to move) and again after about 30° of crankshaft rotation (when the shuttle leaves the picker). Between 0 and 30° on the graph (about 75 and 105° on the normal timing circle), the actual position of the picker and shuttle lags behind their nominal position. The lag increases from 0 to about 15°, where it reaches a maximum value as indicated by the double-headed arrow. Thereafter, the lag decreases as the actual displacement begins to catch up with the nominal displacement and finally overtakes it at about 30°, where the curves cross.

When the loom is turned over slowly by hand, the force exerted on the shuttle by the picker is sufficient only to overcome frictional resistance, which is due mainly to pressure exerted by the swell on the shuttle. If, for example, the swell exerts a force of 65 N on the back wall of the shuttle, there will also be a force of 65 N between the front wall of the shuttle and the box front. If the coefficients of friction between the swell and the shuttle and between the shuttle and the box front are both 0·25, the force exerted by the picker on the shuttle in overcoming this fraction will be: 65(0·25 + 0·25) N = 32·5 N.

A similar force will be required to overcome friction when the loom is running at speed, but, in addition, a much larger force will be required to accelerate the shuttle against the resistance due to its inertia. Suppose, for example, that the mass of the shuttle is 0·5 kg and it is uniformly accelerated from rest to a speed of 12 5 m/s over a distance of 0 2 m. Then, since $v^2 = 2as$ and $f = ma$, we have:

$$f = \frac{mv^2}{2s},$$

where v is the final velocity in m/s,
a is the uniform accleration in m/s^2,
f is the force in N,
m is the mass of the shuttle in kg, and
s is the distance over which acceleration occurs in m.

In the above example, therefore:

$$f = \frac{0\ 5 \times 12{\cdot}5^2}{2 \times 0{\cdot}2} = 195 \text{ N}.$$

On the basis of the above calculations, the force required to overcome inertia is six times that required to overcome friction. As we shall see later in this section, the acceleration is never uniform, and the peak force generated in overcoming picking may be as much as twice what it would be if the acceleration were uniform.

In the discussion that follows, the force required to overcome friction is neglected. This is justified for our present purpose because it is substantially the same whether the loom is turned over by hand or is running at speed and because it is relatively small compared with the force required to overcome inertia.

Reverting to curves A and B in Fig. 2.5, we can now appreciate that the lag between the nominal and actual displacements is a result of the force required to overcome the inertia of the shuttle. The picking mechanism is not rigid, and the stresses set up in the mechanism by the shuttle's resistance to acceleration result in strains in the picking mechanism: the picking band or lug strap stretches, the picking stick bends, and the picking shaft twists. In the early part of the shuttle's acceleration (0–15° in Fig. 2.5), these stresses and strains are building up because the picker is trying to move more quickly than the shuttle. In the latter part (15–30° in Fig. 2.5), the stresses and strains diminish and eventually disappear when the shuttle leaves the picker where the curves across at about 30°.

Thomas[6] suggested that 'a close analogy exists between the action during the second stage of picking (from 15 to 30°) and the action of a catapult. The missile represents the shuttle, the leather part of the catapult represents the picker, and the elastic band the picking band and picking stick. When the elastic is stretched and then released, the force due to the stretch causes the leather missile to gain speed until the elastic becomes slack and the missile is projected. Because of this analogy, the shuttle may be said to be catapulted by the picking mechanism, and the effect is known as the "catapult effect".'

In this analogy, the fully stretched elastic corresponds to the position of maximum lag at about 15° in Fig. 2.5. The force exerted by the elastic is greatest when it is released and falls to zero when it is slack. At this instant, the missile is projected and loses contact with the leather. This corresponds to the position in Fig. 2.5, where the two curves cross at about 30°. At this point, the shuttle leaves the picker.

If there is a linear relation between stress and strain during picking, the lag at any instant will be proportional to the force exerted on the shuttle by the picker at that instant. This is only approximately true because there are complicating factors, which will be mentioned later. If we accept it as a rough approximation, it is clear that the accelerating force increases from zero at the start of the pick to a maximum about halfway through the pick and then decreases to zero when the shuttle leaves the picker. Its peak value must be much larger than it need be if it were uniform. The prospect of reducing the peak value of the accelerating force by a closer approach to uniform acceleration is attractive. Vincent and Catlow[7] investigated this problem, which is discussed more fully in the next two sections.

Since Thomas and Vincent[3] published their first papers in 1949, interest has shifted to automatic looms, which use cone-underpick mechanisms (see Fig. 2.6). These are more rigid, and acceleration is complete over a shorter distance but over about the same angular movement of the crankshaft. This is illustrated by curves C and D in Fig. 2.5, which relate to a 1·12-m-reedspace automatic loom running at 216 picks/min. In this case, the effective length of the stroke is about 16·5 cm compared with about 21·5 cm for the non-automatic cone-overpick loom.

A study of the nominal and actual displacement curves gives a better understanding of the behaviour of the picking mechanism. It is important to the loom designer because it affects the design of the picking cams, but it does not appear to have any immediate practical application in the weaving room. Thomas and Vincent, however, were able to show experimentally that the speed at which the shuttle leaves the picker is approximately proportional to the distance moved by the picker during the first 30° of crankshaft rotation after the picker begins to move when the loom is running. This information is of no direct use because it is impracticable to measure the actual displacement of

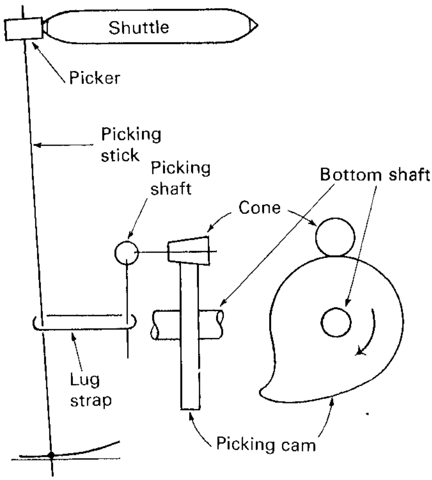


Fig. 2.6 Cone-underpick

the picker in the weaving room. From Fig. 2.5, however, we know that the nominal and actual displacements are the same after 30°. It follows that the shuttle speed will be approximately proportional to the nominal displacement of the picker after 30°. This nominal displacement is easy to measure by using only a timing disc and a steel ruler. If we have several identical looms weaving similar cloths, we can determine the nominal movement that gives the most suitable shuttle speed and set the looms to give this nominal movement. All the looms should then have the same shuttle speed. This led to one of the earliest attempts to systematize loom-settings. The idea was sound, but it was not successful at the time because it was found to be impossible to standardize one loom-setting in isolation from all the rest. Nevertheless, it prepared the way for the standardization of settings and the practice of setting to gauges, which is the basis of modern loom-overlooking.

2.4.2 The Elastic Properties of the Picking Mechanism

When the stiffness of a picking mechanism is known, and it is easy to measure it under static conditions, it can be used in conjunction with the nominal and actual displacement curves to estimate the forces exerted by the picker on the shuttle during picking and to compare these with the constant force that would produce the same final shuttle speed. The following calculation is based on data quoted by Thomas and Vincent[3] in their Fig. 5 and presented now in Table 2.4 and relates to a loose-reed cone-overpick loom.

Since a force of 3·65 kN acting on the picker parallel to the axis of the shuttle produces a deflexion of 1 m, the force required to produce a deflexion of 0·075 m

Table 2.4

Mass of shuttle	0·32 kg
Stiffness of mechanism	3·65 kN/m
Shuttle speed	12·2 m/s
Stroke of picker	0·20 m
Maximum lag	0·075 m

is $0{\cdot}075 \times 3650$ N = 274 N. This is the peak force exerted by the picker on the shuttle during picking. Now, the uniform force that would produce the same final speed over the same distance is given, as before, by:

$$f = \frac{0{\cdot}32 \times 12{\cdot}2^2}{2 \times 0{\cdot}20} = 119 \text{ N}.$$

The actual peak force (274 N) includes the force required to overcome friction. An example in the preceding section gave a typical value of 32·5 N for this, so we may reasonably assume that, of the actual peak force of 274 N, a force of about 240 N was used to overcome inertia. This is almost exactly twice the uniform force (119 N) calculated above.

The forces encountered in modern automatic looms are substantially greater than those calculated above because the shuttle tends to be heavier, the effective stroke shorter, and the shuttle speed higher. In the loom to which curves C and D in Fig. 2.5 relate, the shuttle speed was 13·3 m/s, the mass of the shuttle including a full pirn was 0·51 kg, and the effective length of the stroke was 0·165 m. The required uniform accelerating force would therefore be:

$$f = \frac{0{\cdot}51 \times 13{\cdot}3^2}{2 \times 0{\cdot}165} = 273{\cdot}5 \text{ N}.$$

This compares with 119 N in the preceding example, in which the actual peak force was twice as great. The actual peak force in the last example might exceed 500 N.

In the above calculations, we have been treating the picking mechanism as a simple elastic system that obeys Hooke's Law. We have also assumed that the modulus or stiffness of the system, calculated according to Hooke's Law, is constant during the period of picking and that the product of the stiffness and the lag gives the force acting on the shuttle at any instant during picking. These assumptions are not completely true.

The stiffness of the mechanism at opposite sides of the loom is likely to be different. At the driving side of the loom, the picking cam is close to the driving wheel on the bottom shaft, and twisting of the bottom shaft will contribute little to the elasticity of the mechanism. On the other side of the loom, the picking cam is remote from the driving pinion on the bottom shaft, and twisting of this length of shaft must reduce the stiffness of the mechanism, especially in wide looms.

The stiffness on the side away from the drive will also vary during the insertion of a pick because the leverage of the system alters as a consequence of the changing radius of the picking cam and the changing disposition of other parts of the mechanism. Lord[8] has shown that the stiffness may increase substantially from the beginning to the end of the insertion of a pick.

Finally, there is evidence that the nominal curve obtained by turning the loom over slowly by hand does not correspond to the curve that would be obtained if the mechanism were perfectly stiff.

For all these reasons, it is evident that the analogy of the picking mechanism to a simple elastic system is a very approximate one, which should not be pushed too far. A mathematical treatment of the picking mechanism as a simple elastic system is given in the Appendix to this chapter.

2.4.3 Factors Tending towards Uniform Acceleration

For mechanical reasons (e.g., noise, wear, and vibration), it is desirable to reduce as much as possible the forces acting during picking. As loom speeds increase, so also do shuttle speeds, and the force required to accelerate the shuttle over a given distance is proportional to the square of its final speed. The speed of single-shuttle automatic looms of about 120-cm reedspace has tended to increase over the last ten or fifteen years from about 180 to as much as 240 picks/min. For the same weight of shuttle, this would involve an increase of over 75% in the force generated in picking. In fact, higher speeds demand more robust and therefore heavier shuttles, so that the actual increase in picking force would be greater than 75%. The need to reduce the peak forces generated in picking therefore becomes more urgent as loom speeds increase. For a given shuttle speed and weight of shuttle, there are two possibilities of reducing these peak forces. These are either to increase the distance over which the shuttle is accelerated or to obtain a closer approach to uniform acceleration.

The first possibility, that of increasing the distance over which the shuttle is accelerated, is not very fruitful. One reason for this is that the picker in modern automatic looms is fixed to the picking stick and constrained to move in a straight, horizontal path by causing the fulcrum at the bottom of the picking stick to rise and fall in a controlled manner (see Chapter 4, Figures 4.8 and 4.9). It is impracticable to obtain straight-line motion of the picker over an extended stroke, and, as already mentioned in Section 2.4.1, the stroke tends to be shorter rather than longer on modern automatic looms.

The second possibility—more uniform acceleration—was thoroughly investigated theoretically by Vincent and Catlow[7], and Catlow[9] subsequently carried out experimental work to discover to what extent the theories fit the facts. In the past, picking cams have commonly been designed to give a straight line when the nominal picker displacement is plotted against the first 30° of crankshaft rotation from the commencement of picking. This is illustrated by curves A and C in Fig. 2.5. Owing to the catapult effect, a linear nominal movement gives a markedly non-uniform acceleration. Vincent and Catlow[7] investigated several alternative forms of nominal movement, and Catlow[9] selected the most promising one for further investigation.

By working along these lines, it was found possible to design picking cams that would give a much closer approach to uniform acceleration. Cams designed in this way were made available for fitting to existing cone-underpick looms and were later incorporated in the picking mechanism of the Shirley loom. It is beyond doubt that loom makers all over the world have benefited from the pioneering work done at the Shirley Institute.

2.5 Retardation of the Shuttle: Checking

2.5.1 General Consideration of Checking

The shuttle normally emerges from the shed and enters the shuttle-box with a velocity only slightly less than its velocity on leaving the picker. It must be brought to rest reasonably smoothly over a distance about the same as, or rather less than, that over which it was accelerated. This becomes progressively more difficult to achieve as loom speeds increase, since the energy that has to be dissipated—the kinetic energy of the shuttle—is proportional to the square of its velocity. Shuttle-checking is therefore a vital factor in the design of high-speed looms, and it is one of the main factors that limit speeds.

Shortly after its leading end enters the shuttle-box, the shuttle contacts and begins to displace the swell. The swell is usually at the back of the shuttle-box and is displaced by the back wall of the shuttle. The swell is sometimes at the front of the shuttle-box, but in either case the swell performs two functions, namely:

(*a*) it operates the fast-reed warp-protector motion, which prevents a shuttle trap if the shuttle does not arrive or is delayed; and
(*b*) it helps to reduce the velocity of the shuttle.

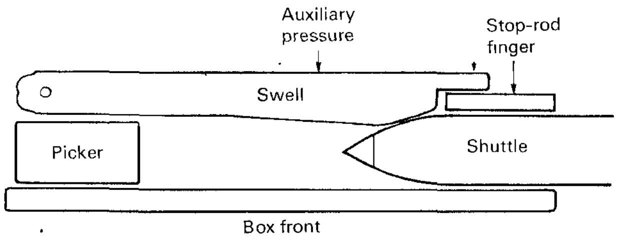


Fig. 2.7 A simple swell

A simple arrangement is illustrated in the plan view in Fig. 2.7. In this example, the swell is pivoted at its outer end. As the shuttle displaces the swell, its inner end moves a finger, which operates the warp-protector motion. The finger is usually spring-loaded, and this, together with the inertia of the moving parts, causes the swell to resist displacement. Pressure therefore develops between the shuttle and the swell and also between the shuttle and the box front (the swell being assumed to be at the back of the box). In order to improve the braking action of the swell without unnecessarily increasing the frictional forces that have to be overcome during picking, supplementary spring pressure may be applied to the swell during checking and relieved during picking. The mechanism responsible for this is called a *swell-easing motion.*

Thomas and Vincent[4] obtained retardation curves for this type of swell for both fast- and loose-reed looms. One of their curves for a fast-reed, 1·5-m-reedspace, non-automatic loom running at about 180 picks/min is reproduced in Fig. 2.8. The shuttle had a velocity of 14·3 m/s when it contacted the swell at

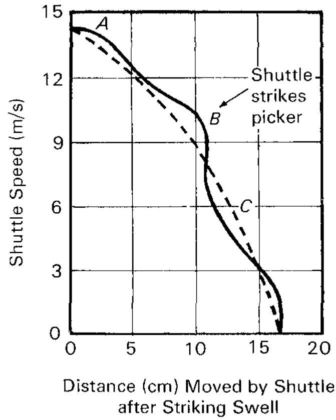


Fig. 2.8 Shuttle-retardation curve (hinged swell)

A. During the next 11·4 cm of its travel (between A and B), its velocity was reduced to 9·75 m/s by the action of the swell. This corresponds to a reduction of slightly less than one-third of its velocity. At B in Fig. 2.8, the shuttle struck the picker, and from this point onwards the combined action of the swell, picker, and check straps reduced its velocity to zero over the last 5 cm of its travel. It is clear that the retardation is by no means uniform.

The dotted line in Fig. 2.8*A* represents the form the curve would take if the retardation were uniform over the total distance of 16·5 cm. The value of this uniform retardation would be about 618 m/s². The actual retardation between A and B is equivalent to a uniform retardation of about 460 m/s². Between B (where the shuttle strikes the picker) and C (where the check strap comes into action), the actual retardation is equivalent to a uniform deceleration of nearly 2000 m/s². Since the actual deceleration is not uniform, its peak value must be greater than this. It is further clear from the shape of the retardation curve that the peak retardation occurs on impact with the picker and that the value of this must be greater than 2000 m/s². Although the shuttle and weft package in this loom had a mass of only 0·34 kg, the peak force acting on the shuttle parallel to its axis must have been well over 450 N, which is substantially greater than the probable peak force generated during picking. This reinforces the statement already made that checking is a critical factor in loom design.

More efficient checking can be achieved by separating the two functions of the swell, so there is a tendency in modern looms to use two swells or a swell in

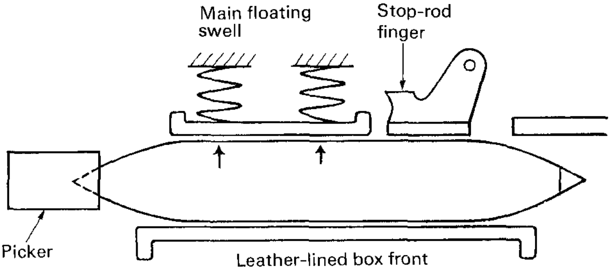


Fig. 2.9 A floating swell

two parts. One of several possible arrangements is shown in Fig. 2.9, and a shuttle-retardation curve obtained by stroboscopic observation of a loom equipped with this kind of swell is given in Fig. 2.10. In this arrangement, a small hinged swell, which is fully displaced soon after the shuttle enters the box (A in Fig. 2.10), operates the warp-protector motion. It is an advantage that this should happen as soon as possible to facilitate the stopping of the loom in the event of a shuttle trap. Note, however, that this swell has very little effect on the speed of the shuttle. The main swell is not hinged but floats against spring pressure applied at two places, which are indicated by arrows in Fig. 2.9. The shuttle begins to displace the inner end of the main swell at B in Fig. 2.10, and both ends of it are fully displaced at C. From this point onwards, the velocity of the shuttle is rapidly and fairly uniformly reduced until at D it strikes the picker, which is cushioned by a check strap.

Comparing the curves in Figures 2.8 and 2.10, we note that checking occurs over a greater distance in the latter. This has the effect of reducing the uniform deceleration that would bring the shuttle to rest over the same distance from

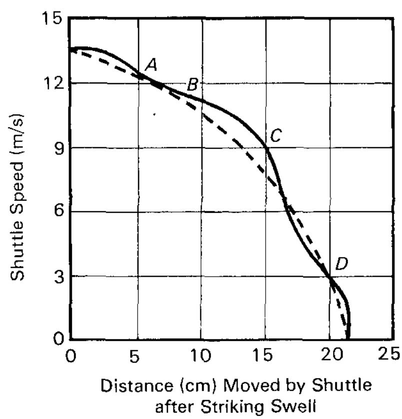


Fig. 2.10 Shuttle-retardation curve (floating swell)

about 618 m/s^2 to about 422 m/s^2. Furthermore, the retardation curve in Fig. 2.10 is nowhere very steep, and it is clear that the peak values of retardation must be much less than those in Fig. 2.8, probably of the order of 700 m/s^2. This represents a substantial improvement in the effectiveness of checking.

2.5.2 The Rest Position of the Shuttle

It is important that the shuttle be brought to rest without a violent shock at the end of its travel, since this would be likely to cause sloughing-off of the weft and rebounding of the shuttle. It is also important to bring the shuttle to rest in the same position each time, for two reasons.

(*a*) If the rest position of the shuttle varies, the effective length of the picker's stroke, and therefore the shuttle velocity on the next pick, will also vary. This may cause the loom to bang-off eventually through the operation of the warp-protector motion, since the effect of one weak pick is to promote a still weaker pick, and so on. Note that a weak pick may also follow if the shuttle rebounds.

(*b*) In an automatic loom, the empty pirn is ejected and a full one inserted in the shuttle while the loom is running at full speed. The empty pirn must be precisely underneath the full one at the instant of transfer. Faulty transfers will occur if the rest position of the shuttle varies materially.

To ensure that the rest position of the shuttle does not vary unduly, it is necessary that it should retain an appreciable fraction of its initial velocity right up to the end of its travel. In other words, there must be an impact as the shuttle forces the picker to the limit of its movement. There are several reasons for this. The shuttle's velocity will vary slightly from pick to pick and over a period of time owing to wear and tear. Different warps, warp tensions, shed-settings, and timings will cause different amounts of retardation as the shuttle passes through the shed. In order to avoid the need for having to make frequent adjustments, the final shuttle velocity must be sufficient to cover these variations.

The mass of the shuttle and its contents decreases as the pirn weaves off. The checking force, however, remains constant, and it must therefore become more effective in slowing down the shuttle as the pirn weaves off. The speed with which the shuttle strikes the picker in the receiving box will thus decrease as the pirn weaves off. The speed with which it strikes the picker when the pirn is full must be sufficient to allow for this, as well as to take care of inevitable variations in the speed of the shuttle as it enters the box. Such variations may arise at the picking side of the loom, from variations in the resistance it encounters in passing through the shed, or as a result of its arriving at the mouth of the receiving box with its axis misaligned.

Suppose, for example, that the mass of the shuttle and its contents is 0·51 kg when the pirn is full and 0·48 kg when the pirn is nearly empty. Suppose also that we require the shuttle to have an impact speed of not less than 4·5 m/s at any time while the loom is functioning correctly, that its calculated speed when it strikes the swell is 13·75 m/s, and that it is uniformly retarded over a distance of 0·20 m up to impact with the picker.

Now:

$$v^2 - u^2 = 2as,$$

where v is the initial velocity,
u is the impact velocity,
a is the uniform retardation, and
s is the distance over which retardation occurs.

The impact velocity is least when the pirn is nearly empty, i.e., when:

$$13{\cdot}75^2 - 4{\cdot}5^2 = 2a \times 0{\cdot}20,$$

from which $a = 422$ m/s^2.

Since $f = ma$, the retardation is inversely proportional to the mass being retarded, so that, when the pirn is full, we have:

$$a = 422 \times \frac{0{\cdot}48}{0{\cdot}51} = 397 \text{ m/s}^2,$$

and:

$$13{\cdot}75^2 - u^2 = 2 \times 397 \times 0{\cdot}20,$$

from which $u = 5{\cdot}5$ m/s.

In this example, the impact velocity is about 22% greater when the pirn is full than when it is nearly empty.

2.5.3 *The Inertia Effect in Checking*

If, in the above example, we assume the coefficient of friction between the swell and the shuttle and that between the shuttle and the box front both to be 0·25, we may calculate the pressure that the swell must exert on the shuttle to produce the required uniform retardation. Taking the case when the pirn is full, we have:

$$f = ma = 0{\cdot}51 \times 397 = 202{\cdot}5 \text{ N}.$$

This is the uniform force acting parallel to the axis of the shuttle that will produce the required retardation. The pressure, p, that the swell must exert on the shuttle, since friction acts on both the front and the back walls of the shuttle, must be:

$$p = \frac{202{\cdot}5}{2 \times 0{\cdot}25} = 405 \text{ N}.$$

It is easy to check the actual swell pressure with a spring balance. If this is done, it will be found to be very much less than the value calculated above. The reason for this has been examined by Morrison[10]. Without entering into the rather advanced mathematical argument, we may note that the effective swell pressure is partly due to the applied spring pressure plus any effect of gravity produced, for example, by the mass of the daggers of a fast-reed warp-protector motion. We may call this the static pressure and note that it is substantially independent of speed, that is to say, it will have the same value whether the loom is turned over by hand or is running at full speed. When the loom is running, this static pressure is augmented by the resistance due to inertia, which the swell and its associated parts (e.g., the stop-rod assembly on a fast reed loom) offer to rapid displacement.

There is a similarity between the action of the picking mechanism (discussed in Section 2.4 and the Appendix to this chapter) and that of the swell and its associated parts. In both cases, we have a simple force massively reinforced by an inertia effect that greatly modifies the behaviour of the system. It would be possible by experimental observation to obtain data from which to plot nominal and actual swell-displacement curves. If this were done, it would be found that the actual swell displacement would lag behind the nominal displacement during the first part of its movement and eventually catch up with and overshoot it. The additional pressure due to inertia would be approximately proportional to lag at any instant. It follows that the total pressure exerted by the swell during checking will increase at first and then decrease. In fact, at some stage during checking, the momentum of the swell and its associated parts is usually more than sufficient to overcome the spring pressure, and the swell loses contact with the shuttle for a time. This is one reason for using a floating rather than a hinged swell, especially if a separate braking swell is used as in Fig. 2.9. This enables the inertia effect to be reduced to a minimum, because there are no moving masses associated with the braking swell. The effective pressure is then more dependent on the applied spring pressure and therefore varies less during checking. The tendency of the swell to lose contact with the shuttle is also reduced.

The inertia effect helps to explain the characteristic shape of the first part of the retardation curves in Figures 2.8 and 2.10. In Fig. 2.8, the swell is not very effective during the first 2·5 cm of the shuttle's movement after contacting the swell. Over the next 5 cm, it becomes progressively more effective as the inertia effect builds up. Over the succeeding 2·5 cm, it becomes less effective again as the momentum of the swell begins to counteract the spring pressure. A similar effect is present, but to a less degree, in Fig. 2.10.

2.6 Shuttle Flight and its Timing

2.6.1 Loom Speed and Timing

There is a simple numerical relation between the speed at which a loom runs, the distance the shuttle travels in passing through the shed, the degrees of crankshaft rotation available for its passage, and the average velocity of the shuttle during its passage. The distance it travels is equal to the width of the warp in the reed plus the effective length of the shuttle. The latter will vary somewhat according to the type and width of the loom. Then, if:

R is the width of the warp in the reed in m,
L is the effective length of the shuttle in m,
v is the average velocity of the shuttle in m/s,
p is the loom speed in picks/min,
θ is the number of degrees of crankshaft rotation available for the passage of the shuttle, and
t is the time in seconds available for its passage,

we have:

$$t = \frac{60}{p} \times \frac{\theta}{360} = \frac{\theta}{6p}$$

and:

$$v = \frac{s}{t} = \frac{(L+R)6p}{\theta},$$

from which:

$$p = \frac{v\theta}{6(L+R)}.$$

Suppose, for example, that the maximum tolerable value for the average shuttle speed in a loom with an effective reedspace of 1·15 m is 13·75 m/s, that 135° of crankshaft rotation can be allowed for the passage of the shuttle, and that the effective length of the shuttle is 0·30 m, then the maximum permissible loom speed will be:

$$p = \frac{13{\cdot}75 \times 135}{6(1{\cdot}15 + 0{\cdot}30)} = 213 \text{ picks/min.}$$

This corresponds closely to the speed at which modern automatic looms of this width run. In the above calculation, we have assumed the effective reedspace to be fully utilized. It does not necessarily follow that the loom could run faster if the effective reedspace were not fully utilized because other considerations would then tend to impose limitations.

For a given width of loom and length of shuttle, the denominator in the equation is constant, in which case we may write $p \propto v\theta$. Thus, if we wish to increase the loom speed, we must increase the shuttle velocity or the fraction of the loom cycle available for its passage or both. It is clearly important to utilize as much as possible of the loom cycle for the passage of the shuttle, and we must therefore examine the controlling factors.

The earliest practicable time for the entry of the shuttle into the shed is determined by the position of the reed. From Fig. 2.2, we see that, for a sley-eccentricity ratio of 0·2, the reed has attained three-quarters of its total displacement from its most forward position by about 113° on the timing circle. By this time, the shed will be fully open, or nearly so (see Fig. 2.11), and large enough for the shuttle to enter without excessive displacement of the warp

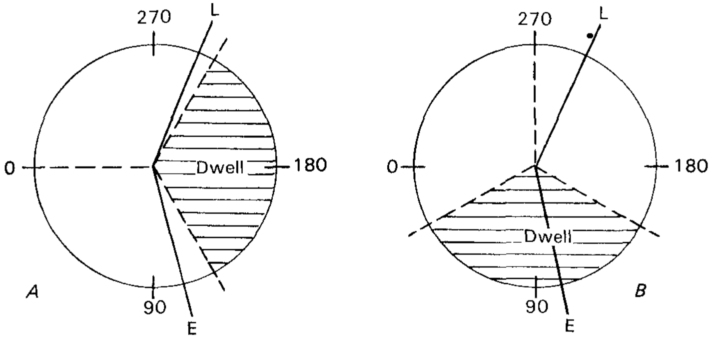


Fig. 2.11 The timing of the primary motions

threads. It is therefore the position of the reed rather than the timing of shedding that determines the earliest time for the shuttle to enter the shed. In practice, it may be permissible for it to enter a little earlier, say between 105 and 110°.

The latest time for the shuttle to come to rest in the shuttle-box on the opposite side of the loom is determined by the fast-reed warp-protector motion. When the loom is running normally, the shuttle must have fully displaced the swell in time to prevent the loom from banging-off. Now, if the shuttle does not arrive in time owing to some mishap, the daggers on the stop rod must engage the frogs on a fast-reed loom by 270° at the latest in order to prevent a shuttle trap. Hence, when the loom is running normally, the swell must have lifted the daggers clear of the frogs before 270°, say, by 260°, to allow a little latitude to cover variations in the shuttle's time of arrival. The shuttle must contact the swell somewhat earlier because the swell is not fully displaced until the shuttle has moved a few inches after contacting it. Suppose, for example, that, in a loom that is running at 216 picks/min, the shuttle moves 0·075 m between contacting the swell and displacing it fully and that its average velocity over this distance is 12·25 m/s. This will then occupy:

$$\frac{216 \times 360 \times 0{\cdot}075}{60 \times 12{\cdot}25} = 7{\cdot}9^\circ \text{ of crankshaft rotation.}$$

On this basis, the latest time for the shuttle to contact the swell will be about 250°. When the reedspace of the loom is being fully utilized, the trailing end of the shuttle leaves the shed at just about the same time as its leading end contacts the swell, so the trailing end of the shuttle must leave the shed by about 250°. The time available for the passage of the shuttle through the shed is therefore about 250 − 110 = 140°. We shall see in the next section that the shed timing shown in Fig. 2.11*B* makes it desirable for the shuttle to leave the shed a little earlier if possible. Moreover, if warp yarns that are susceptible to abrasion—such as low-twist continuous-filament yarns—are being woven, it may be desirable to delay the entry of the shuttle slightly. Again, if the sley-eccentricity ratio were higher than the value of 0·2 assumed above, it should be possible to extend the period available for the passage of the shuttle by a few degrees. The main purpose of the above argument is to show that there is a logical basis for the timing of the flight of the shuttle and to give some indication of the limits of possible variation.

2.6.2 Loom Width and Rate of Weft Insertion

The linear rate (e.g., m/min) at which weft is inserted is a useful criterion for the productivity of a loom. It follows from the equation

$$p = \frac{v\theta}{6(L + R)}$$

that, for given values of v and θ, the rate of weft insertion will increase slowly but steadily with the loom width. The figures in Table 2.5, calculated from the equation, illustrate this point. They assume, as before, that the effective length of the shuttle is 0·30 m and that its passage through the shed occupies 135° of crankshaft rotation.

Table 2.5

Effective reedspace (m)	1·0	1.5	2·0	2·5	3·0
Loom speed (picks/min)	238	172	135	110·5	93·8
Rate of weft insertion (m/min)	238	258	270	276	281

Thus, by weaving a fabric 2·0 m wide instead of 1·0 m, there is an increase of 13·4% in the rate of weft insertion and also in the area of fabric produced in a given time. The percentage increase from doubling the width falls off at greater widths. For example, doubling the width from 1·5 to 3·0 m gives an increase of 8·9%. This can be seen more clearly if the effective reedspace is plotted against the rate of weft insertion.

Wider fabric usually cuts up more economically in garment-making. During the inter-war years, it was largely pressure from the large makers-up of woven lingerie that resulted in replacing 36-in. by 54-in. fabric. About the same time, jacquard tie fabric, which was traditionally 24 in. wide, began to be woven 48 in. wide. In this case, weaving economies were the attraction. Until recently, most furnishing fabrics, especially curtain fabrics, were woven about 52 in. wide. A width and a half had often to be sewn together to make one curtain, and, to avoid this, curtain fabric began to be woven in small quantities 78 in. wide. The above examples, all from the days before metrication, illustrate some of the reasons for seeking to produce wider woven fabrics, which may also lead to economies in dyeing and finishing.

In stating above that 2-m fabric might be expected to give 13·4% more output per loom than 1-m fabric, we have tacitly assumed the same weaving efficiency for both. The wider looms would be consuming yarn more quickly, so that the frequency of warp and weft breaks might be expected to increase slightly, which would thus reduce to some extent the advantage of the wider loom.

For the production of a given area of fabric, fewer of the wide looms would be required, but the initial cost of each would be higher, and they would occupy more floor space. In order to evaluate the economic attractiveness of re-equipping with wide looms, these and many other factors would have to be investigated quantitatively. These other factors include the effect of wide looms on the productivity of weavers and ancillary labour and the probable need for re-equipment in the preparation departments and in the grey-cloth warehouse. Even if such an investigation were to yield an answer in favour of wide looms, such a project would have to be weighed against the rival claims of other plans for re-equipment, which might be found to give a better return on capital invested in re-equipment. Then again, the finisher might not be equipped to process the wider fabric, and the maker-up might have to install wider tables and modify his patterns. For all these reasons, the move towards wider fabric is likely to proceed fairly slowly.

2.7 The Timing of the Primary Motions

We have seen in Section 2.6 that the shuttle cannot normally enter the shed before about 110° and must leave it not later than about 250°. It seems sensible to make the period of heald-shaft dwell coincide as nearly as possible with the passage of the shuttle, so that the shed will be fully open while the shuttle is passing through it. With a dwell of 120° in shedding, this condition is very nearly met if we set the tappets so that the healds cross, or are level, at 0°, as shown in Fig. 2.11*A*. This timing is desirable for weaving low-twist continuous-filament yarns, which are easily damaged by abrasion such as would result if the shed were not fully open during part of the shuttle's passage. It is not always possible to use this timing for weaving continuous-filament yarns because other considerations, which will be referred to in Section 2.8.2, may intervene.

In Fig. 2.11*B*, the heald shafts are level at 270°, and the period of dwell is more than half over by the time the shuttle enters the shed. Moreover, the shed begins to close again before the shuttle is halfway across the loom. It is clear that, although the shuttle will enter the shed quite freely, there must be severe friction between the shuttle and the warp as the shed closes on it during the latter part of its passage. This timing, however, is commonly used for weaving cotton and spun-rayon fabrics. There are two reasons for this apparently illogical procedure:

(*a*) it improves the cover of the cloth, and by this is meant the apparent uniformity of spacing of the warp threads, and

(*b*) it assists in beating-up and helps to achieve the close pick-spacing required in many plain cloths.

The explanation of these statements is deferred until Section 2.8.2 because it involves consideration of the warp line.

2.8 The Geometry of the Warp Shed

2.8.1 The Size of the Shed

The width and depth of the shuttle are determined by the diameter of the weft package it is required to hold. In considering the size of shed required for a given size of shuttle, the important dimension is the depth of the shed at the front wall of the shuttle. This is C in Fig. 2.12, in which A is the width of the shuttle, B the distance from the cloth fell to the reed, and D the depth of the shed at the reed. During the passage of the shuttle, B and D both vary because

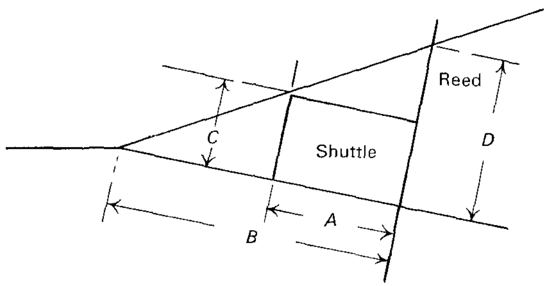


Fig. 2.12 The geometry of the shed

of the motion of the reed, and D will also vary owing to movement of the heald shafts unless the passage of the shuttle coincides with the period of dwell.

In Fig. 2.13, the depth of the shed at the front of the shuttle is plotted against the angular position of the crankshaft for a particular loom[6]. Curve B, which is symmetrical, was obtained with the heald shafts set to cross at 0°. If we assume that the shuttle enters the shed at 110° and leaves it at 240°, the depth of the shed at the front of the shuttle as it enters and leaves the shed will be as given in Table 2.6.

Table 2.6

	Depth of Shed (cm)	
	Entering	Leaving
Curve A: healds cross at 270°	2·44	0·94
Curve B: healds cross at 0°	2·36	2·54

The depth of the shed at the front of the shuttle, expressed as a fraction of the height of the front wall of the shuttle, is called the *bending factor* or *interference factor*. It indicates the extent to which the warp threads are deflected, if at all, by the shuttle. A bending factor of less than 1·0 implies deflexion. If we take the height of the front wall of the shuttle in the loom to which Fig. 2.13 relates to be 2·8 cm, the bending factors will be as given in Table 2.7.

The two extreme cases are illustrated in Fig. 2.14, in which the dotted lines represent the position the top shed would occupy if it were not deflected by the

Table 2.7

	Bending Factors	
	Entering	Leaving
Curve A: healds cross at 270°	0.87	0.34
Curve B: healds cross at 0°	0.84	0.90

shuttle. In this particular case, there would be some deflexion of the warp by the shuttle on entering and leaving with both shed-timings. The amount of bending, however, is quite small except when the shuttle is leaving with the healds set to cross at 270°. The question arises as to how much bending can be tolerated. Referring to Fig. 2.13, we see that for both curves bending occurs as the shuttle

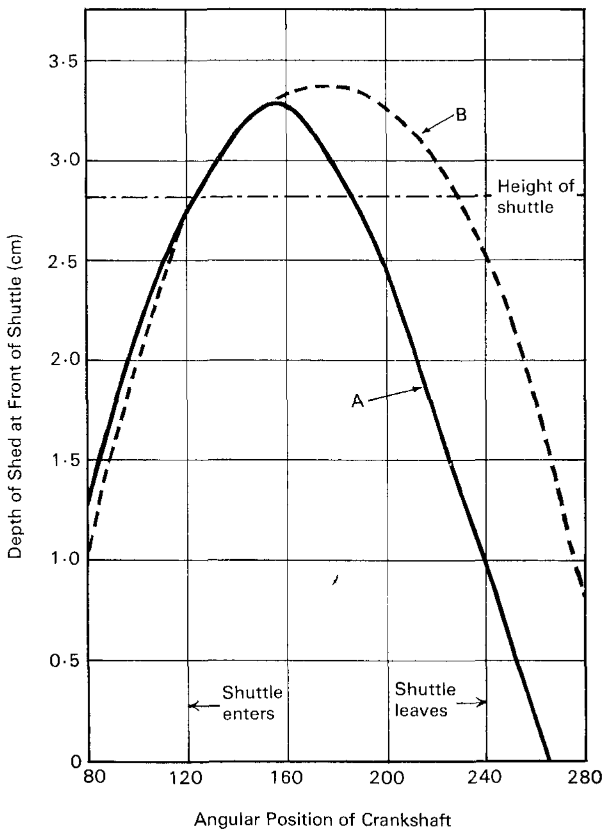


Fig. 2.13 Shed-depth curves

enters only between 110 and 120°. For curve B (healds cross at 0°), the depth of the shed is greater than the height of the shuttle between about 120 and 230°. A small amount of bending occurs between 230 and 240° as the shuttle leaves. These conditions would be acceptable even in weaving low-twist continuous-filament yarns. If need be, a slight increase in the depth of the shed would eliminate bending altogether.

Curve A represents conditions that could not be tolerated in weaving continuous-filament yarns but are quite common in weaving cotton yarns.

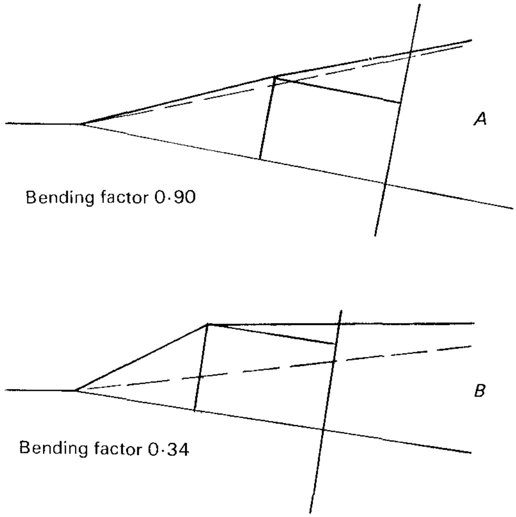


Fig. 2.14 The deflexion of the warp

There is no more bending as the shuttle enters than with curve B, but the shuttle deflects the warp from about 195° until it leaves the shed at 240°. By the time it leaves, the bending factor is 0·34 as shown in Fig. 2.14*B*, and it is clear that there must be quite severe rubbing of the warp by the shuttle during the latter part of its passage through the shed. This can usually be tolerated in weaving spun yarns in order to secure the benefits already mentioned, namely, better warp cover and more effective pick-packing.

2.8.2 The Warp Line

It is usual to set the heald shafts so that, when the sley is at 180°, the bottom warp sheet (AD in Fig. 2.15) is in contact with the raceboard, which is the name for the upper face of the sley. The lift of the heald shafts is then adjusted to give some clearance between the upper part of the shed and the top of the front wall of the shuttle. The 'normal' position of the backrest would then be with its top on the dotted line AC, which bisects angle BAD. The lengths of

Kuhele

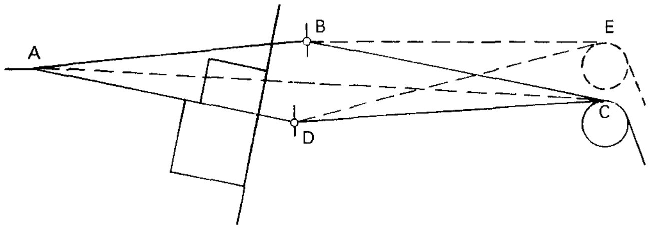


Fig. 2.15 The warp line

ABC and ADC are then equal, and the tensions in the yarns in the top and bottom sheets will also be equal. Note that the line AC is not necessarily horizontal; it usually inclines downwards towards the rear, as shown in Fig. 2.15.

It is known from experience that it is possible to obtain a closer pick-spacing (i.e., a tighter packing of the picks) if the backrest is raised above its 'normal' position as at E in Fig. 2.15, provided that the shedding is timed so that the healds cross well before beat-up. The yarns ABE in the upper shed are now shorter and therefore at less tension than yarns ADE in the lower shed. In effect, the upper shed is slack and the lower one tight. The situation at the cloth fell at the instant of beat-up is then approximately as shown in Fig. 2.16*A*. The pick just inserted (numbered 4) is forced downwards because, since we have stipulated that the heald shafts cross well before beat-up, the ends that pass over it are now in the bottom shed and are therefore tight. Conversely, the preceding pick (numbered 3) is forced upwards, but to a less extent. It may also be that picks further from the fell are affected in the same way, but to a much smaller extent. When the next pick (5), which is not shown, is inserted and the shed has changed again, pick (4) will assume the position occupied by pick (3) in Fig. 2.16*A*, and pick (3) will assume the position occupied by pick (2). Every time beat-up occurs, there is a vertical displacement of the picks near the fell of the cloth. The precise effect of this movement is not known, but it is known that

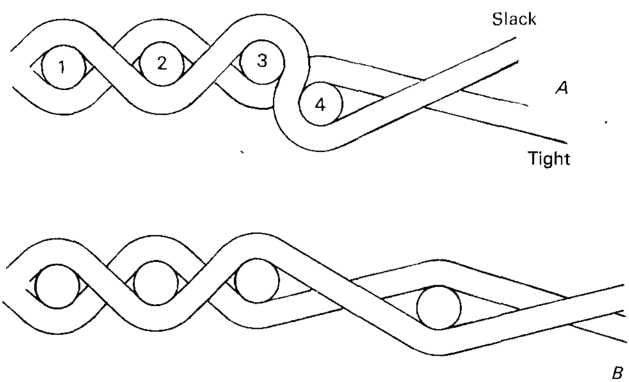


Fig. 2.16 Cloth formation

the final pick-spacing is not established until the picks have moved forward some little distance from the cloth fell. It is, however, clear from Fig. 2.16*A* that picks (3) and (4) can be placed closer together than would be possible if they were in the same horizontal plane.

An alternative to adjusting the height of the backrest is to adjust the height of the cloth fell. This is not generally practicable in shuttle looms (although facilities are often provided), since it will either raise the bottom warp sheet off the raceboard and thus interfere with the flight of the shuttle or else cause it to press heavily on the raceboard, which is detrimental to the warp.

The weft-inserting device in shuttleless looms (e.g., rapier or projectile) does not ride on the raceboard, and in these looms the height of the fell can be adjusted without affecting weft insertion. The distance from the healds to the fell is much less than the distance from the healds to the backrest, so a given movement of the fell produces the same effect as a much greater movement of the backrest. The arrangements for altering the height of the fell on the Sulzer weaving machine are shown in Fig. 2.4.

Another effect of timing shedding so that the heald shafts cross well before beat-up is illustrated in Fig. 2.16*B*, which shows a pick in the course of beating up. The heald shafts have already crossed, and this will tend to prevent the pick from springing back from the fell as the reed recedes and thus help in obtaining close pick-packing. There must be substantial friction between the warp and the pick being beaten-up in Fig. 2.16*B*, and this may help to redistribute the warp threads more evenly. Furthermore, if the backrest is above its normal position, alternate warp threads will be relatively slack and therefore freer to move laterally to achieve a more uniform spacing. It seems likely that the combination of these two effects is responsible for the improvement in warp cover that occurs when the backrest is raised and the tappets are set so that the healds cross well before beat up.

We can now appreciate why the shed-timing represented by Fig. 2.11*B*, used in conjunction with a raised backrest, is very common in cotton-weaving. It offers advantages in widely different circumstances. In weaving closely set cloths as different as typewriter ribbon, poplin, and canvas, it helps to achieve close pick-spacing, although in these cloths its effect on warp cover is unlikely to be important because the warp cover factors are high enough to ensure good cover anyway. In weaving more openly set plain cloths, pick packing is not a problem, but warp cover is, and here again the combination of early shedding and a raised backrest is likely to be beneficial.

APPENDIX

The Picking Mechanism as a Simple Elastic System

A2.1 General Analysis

If the picking mechanism behaves as a simple elastic system, it can be represented by a mass in series with a spring. In Fig. 2.17, the mass M, which rests on a smooth, horizontal surface, represents the shuttle, and the spring represents the elasticity of the picking mechanism. Fig. 2.17A represents the

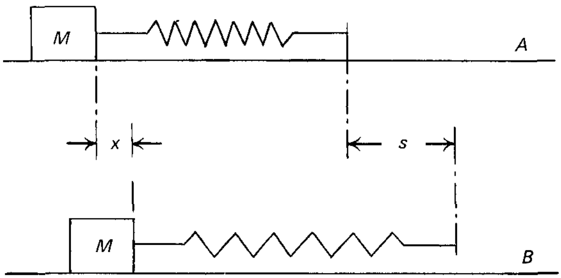


Fig. 2.17 An elastic model of the picking mechanism

situation at the start of picking. In Fig. 2.17B, the free end of the spring has been moved through a distance corresponding to the nominal movement of the picker at a given instant during picking, and the mass M has moved a distance corresponding to the actual movement of the picker at that instant.

Let:

λ = the rigidity of the mechanism (i.e., force/unit extension),
s = the nominal movement, and
x = the actual movement,

from which:

$\dot{x}$ = the actual velocity, and
$\ddot{x}$ = the actual acceleration at a given instant.

The force accelerating mass M is the tension in the spring, which is equal to the product of its rigidity and its extension, i.e.:

$$\text{force} = \lambda(s - x);$$

but:

$$\text{force} = \text{mass} \times \text{acceleration},$$

so:

$$M\ddot{x} = \lambda(s - x) \tag{A2.1}$$

or:

$$\ddot{x} = \frac{\lambda}{M}(s - x). \tag{A2.2}$$

For a given mechanism, λ and M will tend to be constant and may be regarded as constant for the purpose of this elementary treatment. For convenience in subsequent mathematical argument, Thomas and Vincent[3] write $\lambda/M = n^2$ and call n the *alacrity* of the system.

We thus have:

$$\text{alacrity} = \sqrt{\lambda/M} = n,$$

and Equation (A2.2) becomes:

$$\ddot{x} = n^2(s - x). \tag{A2.3}$$

At this stage, we may note the following points:

(*a*) no assumption has been made about the form of the nominal movement, so Equation (A2.3) should be true for any form of nominal movement, uniform or non-uniform; and

(*b*) the force accelerating the mass M is produced by stretching the spring; it follows that acceleration will be delayed and that the actual movement will lag behind the nominal movement.

The above equations show the relations between the nominal and actual movements of the picker, so they can be used to predict the actual movement that will result from a given nominal movement in a system with a known, constant alacrity.

A2.2 Straight-line Nominal Movement

The picking cams of many looms are designed to give a straight line nominal movement, the displacement of the picker being proportional to the angular movement of the crankshaft. In this case:

$$s = p\theta,$$

where p is a constant of proportionality, and
θ is the angular movement of the crankshaft.

Thus, for a constant loom speed, the nominal velocity is constant, and, if ω is the constant angular velocity of the crankshaft and θ is the angle turned after a time t, then $\theta = \omega t$, and:

$$s = p\omega t, \tag{A2.4}$$

the nominal velocity being given by:

$$\frac{\mathrm{d}s}{\mathrm{d}t} = p\omega.$$

Substituting Equation (A2.4) in (A2.3) gives:

$$\ddot{x} + n^2x = n^2p\omega t. \tag{A2.5}$$

The solution of this differential equation is:

$$x = p\omega\left(t - \frac{\sin nt}{n}\right). \tag{A2.6}$$

Differentiation of this expression gives:

$$\dot{x} = p\omega(1 - \cos nt) \tag{A2.7}$$

and

$$\ddot{x} = p\omega n \sin nt. \tag{A2.8}$$

The maximum value of $\dot{x}$ will occur when $\cos nt = -1$ and $nt = \pi$. This maximum value, which will occur after a time $t = \pi/n$, will be:

$$\dot{x}_{max} = 2p\omega. \tag{A2.9}$$

The maximum actual velocity is twice the constant nominal velocity.

The maximum value of $\ddot{x}$ will occur when $\sin nt = 1$ and $nt = \pi/2$. This maximum value, which will occur after a time $t = \pi/2n$ s, will be:

$$\ddot{x}_{max} = p\omega n. \tag{A2.10}$$

The maximum acceleration occurs at half the time required to produce the maximum velocity, that is, halfway through the effective stroke of the picker.

Note that the time to reach maximum velocity and acceleration is determined by the alacrity of the system, a more rigid system giving a shorter time. In addition, the maximum velocity is proportional to the loom speed, and the maximum acceleration is proportional to both the loom speed and the alacrity.

The distance moved when $\dot{x}$ becomes a maximum (i.e., the effective stroke of the picker) is given by substitution of $t = \pi/n$ in Equation (A2.6), from which the effective stroke L is given by:

$$L = p\omega\left(\frac{\pi}{n} - \frac{\sin \pi}{n}\right)$$

$$= \frac{p\omega\pi}{n} \text{ because } \sin\pi = 0,$$

and, because

$$p\omega = \frac{\dot{x}_{max}}{2},$$

$$L = \frac{\dot{x}_{max}\,\pi}{2n} \tag{A2.11}$$

The maximum shuttle velocity is thus proportional to the effective length of the stroke. The alacrity can be found from Equation (A2.11) if the maximum shuttle speed and effective stroke are known.

The relations between $\dot{x}$, $\ddot{x}$, and L can be derived as follows. From Equation (A2.10):

$$\ddot{x}_{max} = p\omega n,$$

and, since from Equation (A2.9):

$$p\omega = \frac{\dot{x}_{max}}{2},$$

by combining these we obtain:

$$\ddot{x}_{max} = \dot{x}_{max} \frac{n}{2}. \qquad \text{(A2.12)}$$

Substituting for n from Equation (A2.11), we have:

$$\ddot{x}_{max} = \frac{\dot{x}_{max}}{2} \times \dot{x}_{max} \frac{\pi}{2L},$$

i.e.:

$$\ddot{x}_{max} = \frac{\pi}{4L}(\dot{x}_{max})^2. \qquad \text{(A2.13)}$$

Hence, for a given maximum velocity, the maximum acceleration is inversely proportional to the effective length of the stroke.

CHAPTER 3

Shedding Mechanisms

3.1 The Scope of Tappet, Dobby, and Jacquard Shedding

The three main types of shedding mechanism are *tappet*, *dobby*, and *jacquard*. In the simplest type of tappet-shedding motion, the shedding cams or tappets are mounted on the bottom shaft, and the motion is suitable only for weaves repeating on two picks, such as plain, weft-rib, or haircord weaves. By mounting the tappets on a countershaft driven by gearing from the bottom shaft at the appropriate speed, the repeat can be extended up to eight or ten picks. Furthermore, since it is practicable to use up to eight or ten tappets, weaves with repeats up to 8 × 8 or 10 × 10 ends and picks can be woven with a straight draft. By using fancy drafts, certain weaves repeating on a much larger number of ends (e.g., pointed and herringbone twills) can be woven. A very large proportion of the total output of woven fabric can be woven on tappet looms.

Dobbies are much more versatile and usually control at least sixteen, and sometimes as many as 36, heald shafts. Since the lifting of the shafts is controlled by some form of pattern chain, there is virtually no limit to the number of picks per repeat. This, together with the use of fancy drafting, is sufficient to produce any structure, weave, or combination of weaves arranged to give stripes, checks, or designs of a geometrical character. The possibilities include handkerchiefs, tablecloths, and towels with contrasting pattern in the borders.

For designs that require the reproduction of freely drawn shapes, it is usually necessary for each end in the repeat to be separately controlled. This is provided for by a jacquard machine, so-called after its French inventor, and its associated harness. Jacquard machines are made in a wide variety of sizes to control from 100 to 2000 or more ends per repeat. A common size controls 600 ends, which, in a cloth with 30 ends/cm, gives a repeat 20 cm wide, within which the designer has complete freedom. The lifting of the ends is controlled by a chain of punched cards or by a loop of punched paper, and the length of the repeat is limited only by the cost and inconvenience of a very long pattern chain. Jacquard machines are used for a wide variety of purposes from ties to carpets.

Tappet mechanisms are simple and inexpensive with regard to both initial cost and maintenance. Being simple and robust, they are not likely to cause faults in the fabric, and they impose no limitations on the speed of the loom. Their chief disadvantage, apart from their restricted patterning possibilities, is that it is usually necessary to change the tappets, or at least to rearrange them, when the weave has to be changed. If the new weave repeats on a different number of picks, it will also be necessary to change the gearing that drives the countershaft on which the tappets are mounted. Because of the work involved in changing the weave, dobbies are sometimes used to facilitate weave changes even though the work is within the scope of tappet shedding.

Dobbies are more complicated and much more expensive initially. Maintenance costs are higher because they have many more parts, which eventually have to be replaced owing to wear. They are more liable to produce faults in the fabric than tappets, and they tend to limit the speed of the loom. This limitation is not usually a serious one, but for high-speed unconventional looms it has been necessary to develop dobbies capable of running at higher speeds. These are naturally more costly to produce. It is relatively easy to change the weave when a dobby is used. All that is needed is to change the pattern chain, which can be prepared in advance, so that less time is lost when the change is made.

Jacquard machines are normally simpler in principle than dobbies, but they contain many more moving parts. A medium-sized jacquard for controlling 600 ends in the repeat has 600 or 1200 wire hooks for raising the ends, 600 or 1200 wire needles for controlling the hooks, and as many cords as there are ends in the warp for connecting the hooks to the ends they control. This large-scale multiplicity of parts—albeit simple parts—makes the machine and its harness (i.e., the system of cords) relatively costly to install and maintain. Jacquard machines are even more liable to produce faults in the fabric than dobbies. Until recently, the jacquard machine had tended to impose limitations, sometimes quite severe, on loom speeds. For single-shuttle looms of up to about 125-cm reedspace, the jacquard was likely to be the limiting factor in fixing the loom speed. For wider fabrics and for those with more than one weft, the limiting factor might well have been the width of the loom or the multiple-shuttle-box motion. Recently, jacquard machines capable of running at speeds of up to 300 picks/min have been developed, primarily for use on unconventional looms. As one might expect, they cost more. Because of its high initial cost, its increased tendency to cause faults in the fabric as compared with tappet and dobby shedding, and its limiting effect on loom speeds, jacquard shedding is normally used only when the cloths to be woven are outside the scope of dobby shedding. Jacquards, however, are sometimes used for sampling because of their versatility, even though orders are to be executed on dobby looms.

The foregoing arguments may be summarized as follows.

Tappet Shedding
Simple, inexpensive, easy to maintain, reliable. Imposes no limitations on loom speeds. Normally limited to 8 or 10 shafts and 8 or 10 picks/repeat. Inconvenient for frequent pattern changes.

Dobby Shedding
More complicated, higher initial and maintenance costs, slightly more liable to produce fabric faults. Normally built to control 20–28 shafts. Picks per repeat virtually unlimited. Some limitation in loom speed

possible, but the most advanced types can run at speeds of up to 500 picks/min.

Jacquard Shedding

Simple in principle, but involves the use of a multiplicity of duplicate parts and is therefore costly to install and maintain. More liable than dobby shedding to produce faults. All except the most advanced types of jacquard, which can run at speeds of up to 300 picks/min, tend to limit loom speeds. Patterning possibilities virtually unlimited.

3.2 Tappet Shedding

3.2.1 Definition of Terms 'Positive' and 'Negative'

Confusion sometimes arises over the use of the terms 'positive' and 'negative' in connexion with tappet and dobby shedding. Positive shedding implies that the heald shafts are both raised and lowered by the cams or crank-and-lever system of the shedding mechanism. Shedding is taken to be negative when the heald shafts are either raised or lowered by the mechanism but are returned by the action of some external device, such as springs. The question then arises as to whether the simple tappet motion shown in Fig. 3.1 is negative or

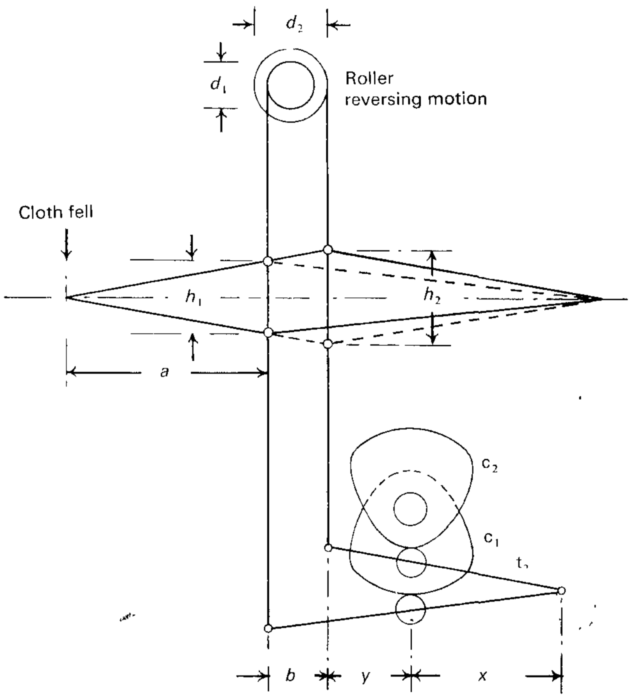


Fig. 3.1 Plain-cam shedding

positive in action. The shedding cams are negative in action because they can only push and not pull. If the mechanism as a whole is positive in action, it is made so by the action of the roller reversing motion, which in practice is very far from positive or precise in action. These ambiguities are avoided in tappet shedding if we use the terms positive and negative to refer to the action of the tappets or cams and not to the mechanism as a whole.

3.2.2 *Shedding with Negative Cams*

A pair of cams mounted on the bottom shaft is sufficient for weaving many plain cloths. A slightly simplified line diagram of an arrangement of this kind is shown in Fig. 3.1. In the lettering of this diagram, the suffices 1 and 2 refer, respectively, to the front and back heald shafts. Treadles t_1 and t_2 are fulcrummed low down near the back of the loom, and each carries an anti-friction bowl on which the corresponding cams c_1 and c_2 act. Connexions are made from the treadles to the bottom of the corresponding shaft and from the top of each shaft to the roller reversing motion. When one shaft is lowered by the action of its cam, the other shaft is raised through the roller reversing motion.

The leverage of the system is such that, if the two cams have the same throw, the front shaft will be given a greater displacement than the back shaft. This is the opposite of what is required to form identical sheds on alternate picks, and the cam c_2, which operates the back shaft, must therefore have a substantially greater throw than the cam c_1. In Fig. 3.1, if L_1 and L_2 are the throws of the two cams, then:

$$h_1 = \frac{L_1(x + y + b)}{x}$$

and

$$h_2 = \frac{L_2(x + y)}{x},$$

from which:

$$\frac{L_2}{L_1} = \left(\frac{h_2}{h_1}\right) \frac{(x + y + b)}{(x + y)}.$$

But:

$$\frac{h_2}{h_1} = \frac{(a + b)}{a},$$

so that:

$$\frac{L_2}{L_1} = \frac{(a + b)(x + y + b)}{a(x + y)}.$$

Taking a numerical example in which, for instance, $b = 3$ cm, $x = y = 22{\cdot}5$ cm, and $a = 20$ cm, we have:

$$\frac{L_2}{L_1} = \frac{(20 + 3)(45 + 3)}{20 \times 45} = 1{\cdot}23.$$

In this example, the throw of the cam operating the back shaft should be 23% greater than that of the cam operating the front shaft. The action of the mechanism will be positive provided that the connexions between the treadles and the roller reversing motion do not stretch and are taut at all times. To satisfy the latter condition, the relative diameters of the two rollers of the reversing motion must be such that, if d_1 and d_2 in Fig. 3.1 represent the diameters of the roller plus the thickness of the straps that connect them to the shafts, then:

$$\frac{d_2}{d_1} = \frac{h_2}{h_1} = \frac{(a + b)}{a}.$$

Suppose that the cams and rollers have been designed according to the principles outlined above and that the connexions are taut and inextensible and of the correct thickness; the action of the mechanism will then be precise and positive. It is necessary, however, to be able to vary the amplitude of the displacement of the heald shafts to some extent for different warps. The only way of doing this in this mechanism is to alter the point of attachment of the connexions to the treadles, but this will upset the leverage of the system. It will no longer be possible to keep the connexions taut at all times, and there will be a loss of lift when they are slack. This is a somewhat academic point, but, in combination with other factors, such as stretching of the straps and lack of rigidity in the shafts, it results in the motion's being far from precise and positive. ✓

In order to avoid overcrowding of the healds, it is usually necessary to draw the warp on four heald shafts, in which case skip-drafting will be preferred, so that shafts 1 and 2 can be connected to the first treadle and roller and shafts 3 and 4 to the second treadle and roller. With a straight draft, four shedding cams would be required.

Some warp-faced plain-weave cloths have too many ends/cm to be woven on four shafts and require six or occasionally even eight shafts. For these cloths, it is desirable also to stagger the crossing of the healds, so as to reduce abrasion between ends on the different shafts and between the ends and the heald eyes. This can be achieved by using a straight draft on six shafts and making the cams in three pairs, so designed that the three pairs of shafts that they control cross at slightly different times. This is occasionally given the misnomer 'split shedding', and it is generally more accurate to describe it as 'staggered healds'.

In general, if we wish to change the weave, we must replace one set of shedding cams with another, but it is sometimes possible, either by changing the order in which the cams are arranged on the countershaft or by altering the angular position of the cams in relation to each other, to alter the weave. More usually, however, the cams are cast in sets, and then the only way of using the same set for more than one weave is by altering the drawing-in plan. For exam-

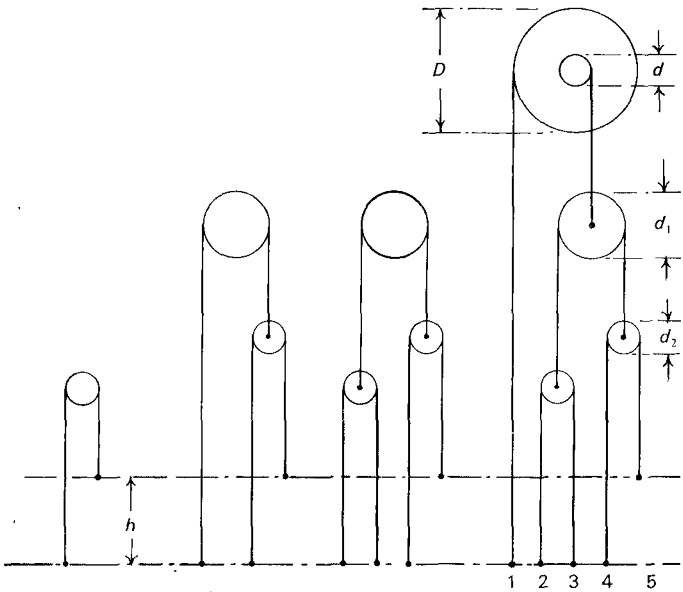


Fig. 3.2 Roller reversing motions

ple, a set of cams designed to weave 2/2 twill with a straight draft will give 2/2 broken twill with a skip draft.

3.2.3 Heald-shaft-reversing Motions

The roller reversing motion can be elaborated as indicated in Fig. 3.2 to con trol three, four, or five shafts, provided that the weave is such that the same number of shafts is lifted on each pick. This is true of twills, sateens, and some simple fancy weaves. In the mechanisms shown in all the daigrams in Fig. 3.2, the topmost roller rotates in fixed bearings, but the other rollers can rise and fall.

On considering first the arrangements for controlling three and four shafts, and neglecting the thickness of the connecting straps, it is clear that, if all the shafts are to have the same displacements, the diameter of the small rollers should be half that of the larger rollers. In the five-shaft arrangement, $D = 4d$, and $d_1 = 2d_2$. But we know that the shafts further from the cloth fell should have a greater displacement. To achieve this, each of the three floating rollers in the five-shaft arrangement, and all the rollers in the three- and four-shaft arrangements, should consist of two rollers of different diameter as in Fig. 3.1. The relation $D = 4d$ in the five-shaft arrangement would also need modifying. These reversing motions suffer from the same inherent disadvantages as the simple one for controlling two shafts, and they are seldom used on modern automatic looms.

In each of the motions represented in Fig. 3.2, one shaft is shown lifted as would happen in weaving 1/2 twill, 1/3 twill, and 1/4 twill or 5-end sateen. They could just as easily be set up to weave 2/1 twill, 2/2 or 3/1 twill, 4/1 twill, or 5-end satin. The only restriction, as already mentioned, is that the same number of shafts must be lifted on each pick. These reversing motions have to be mounted directly above the heald shafts, but there are others that use a combination of rollers and levers to attain the same objective and that can be placed above, but to one side of, the heald shafts. One of these—the Lacey top motion—can control five shafts, but, by locking one or both of the levers on which the rollers are mounted and not using all the rollers, it can be used equally well to control two, three, or four shafts.

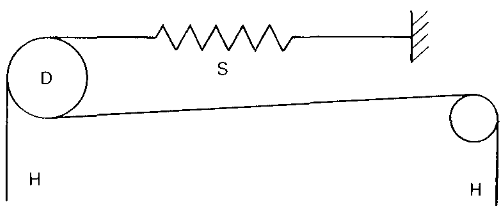


Fig. 3.3 A hypothetical reversing motion

When negative-shedding cams are used on modern automatic looms, the tendency is to use some form of spring reversing motion, with separate springs for each shaft, which thus avoids the critical conditions necessary for the proper functioning of roller reversing motions. Besides being more effective in producing the desired lift, they are not subject to the limitation of having to lift the same number of shafts on each pick. A possible arrangement is shown in Fig. 3.3, in which tension in the spring S tends to rotate the disc D and so raise the shaft through connexions H. The tension in the spring and in the connexions to the shafts will be greatest when the shaft is down and when the spring is doing no useful work. With such an arrangement, the maximum tension in the spring is much greater than is required to raise the shaft. This is illustrated in Fig. 3.4. In Fig. 3.4*A*, the shaft is down, and tension T_1 in the warp sheet produces a force F_1 acting upwards, so that, if W is the weight of the shaft and its associated parts, the force required to raise it at that point will be ($W - T_1$). When the shaft is up, as in Fig. 3.4*B*, the tension T_2 in the warp produces a

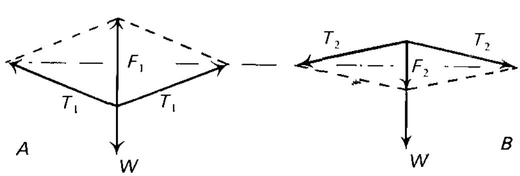


Fig. 3.4 Forces acting on the warp sheet

Surojit Dutta.
College of Textile Technology.

force F_2 acting downwards, and the force required to keep the shaft up will be $(W + T_2)$. Hence, to be efficient, the force exerted by the spring should increase as the shaft rises. It actually decreases owing to the decreasing extension of the spring, and the system is therefore inefficient. This has practical significance because the excessive tension when the shaft is down may cause the heald-shaft frame to bend and thus cause the healds to bind and perhaps to break. It is therefore desirable to seek a more efficient way of using springs.

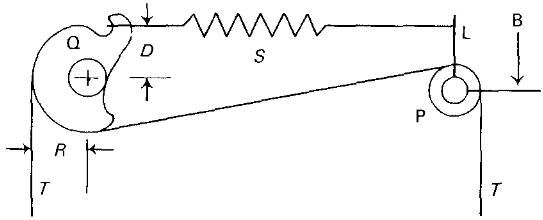


Fig. 3.5 A practical reversing motion

The simple modification shown in Fig. 3.5 achieves this, at least to some extent. If R is the radius of the quadrant Q, S is the tension in the spring, D is the distance perpendicular to the axis of the spring between the axis of the quadrant and the point of attachment of the spring, and T is the tension in each of the connexions to the shaft, then:

$$2TR = SD,$$

or

$$T = \frac{SD}{2R},$$

and, since R is constant:

$$T \propto SD.$$

Now, if Fig. 3.5 represents the position of the parts when the shaft is up, when it is down the quadrant will have turned anti-clockwise and reduced the distance D. Hence, although the tension S in the spring will have increased, the product SD may not, and may even have decreased, which is what is required. This is therefore a more efficient arrangement than the one shown in Fig. 3.3. A similar principle is used in the Kenyon undermotion for negative dobbies (see Fig. 3.14). A refinement in the motion represented in Fig. 3.5 consists of an elbow lever L, which pivots freely on the shaft of the pulley P. The free end of the spring is attached to the vertical arm of the elbow lever, and a screwed bolt B bears on its horizontal arm. This enables the tension in the spring to be adjusted. Such an adjustment is necessary to cover widely different tensions in the warp sheet, which arise with different warps and different numbers of heald shafts.

Nishith Kr. Laha.
CTT (B) 91.

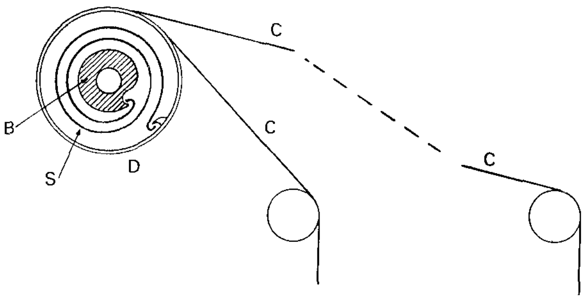


Fig. 3.6 A clockspring reversing motion

A relatively recent development favoured by Draper and Picanol is the use of 'clock-spring' reversing motions. In one type, which is illustrated diagrammatically in Fig. 3.6, a short shaft mounted above heald shaft level at one side of the loom carries a boss B, which is shaped so as to provide an anchor for the inner ends of several strong spiral springs S, one of which is shown. There is a spring for each shaft, and each spring is housed inside a separate drum D, the inner face of which has a projection for anchoring the outer end of the spring. The outer cylindrical face of the drum has two grooves, in which the ends of two cords, or preferably flexible steel wires, C, are fixed. The boss B can be rotated by means of a handle (not shown) to alter the tension in all the springs at the same time. Graduations on the front of the drum indicate the level of tension. When six or eight shafts are to be controlled, drums can be mounted at both sides of the loom, alternate shafts being controlled from opposite sides. This system has the same fundamental defect as the one shown in Fig. 2.3. Nevertheless, it is convenient and easy to keep clean, and, by using modern materials (e.g., plastics-coated flexible steel wire for the cords and high-grade steel for the springs), the tensions, once correctly set, remain so indefinitely.

3.2.4 Shedding with Positive Cams

Positive cams are capable of both raising and lowering the heald shafts without the use of a reversing motion. There are two main types, one of which is shown in Fig. 3.7. In this case, a cam follower or anti-friction roller is constrained to follow a groove machined in the face of the cam. A practical application of this principle is shown in Fig. 3.7, which represents the shedding motion on a Saurer automatic loom. The set of cams on the loom from which the drawing was made was designed for weaving 1/2 twill with two repeats to the round. In this particular case, the cams rotate once every six picks. The main shaft in the Saurer loom rotates at three times the speed of the loom, so that, between the main shaft and the cam shaft, there has to be a reduction of 1:18, which is obtained through bevel-and-spur gearing. In following the cam track, the cam follower moves up and down, and the lower end of the lever, marked 'tappet lever' in the diagram, moves to and fro. The wooden connecting members between the tappet lever and the heald shaft have spring-steel snap-

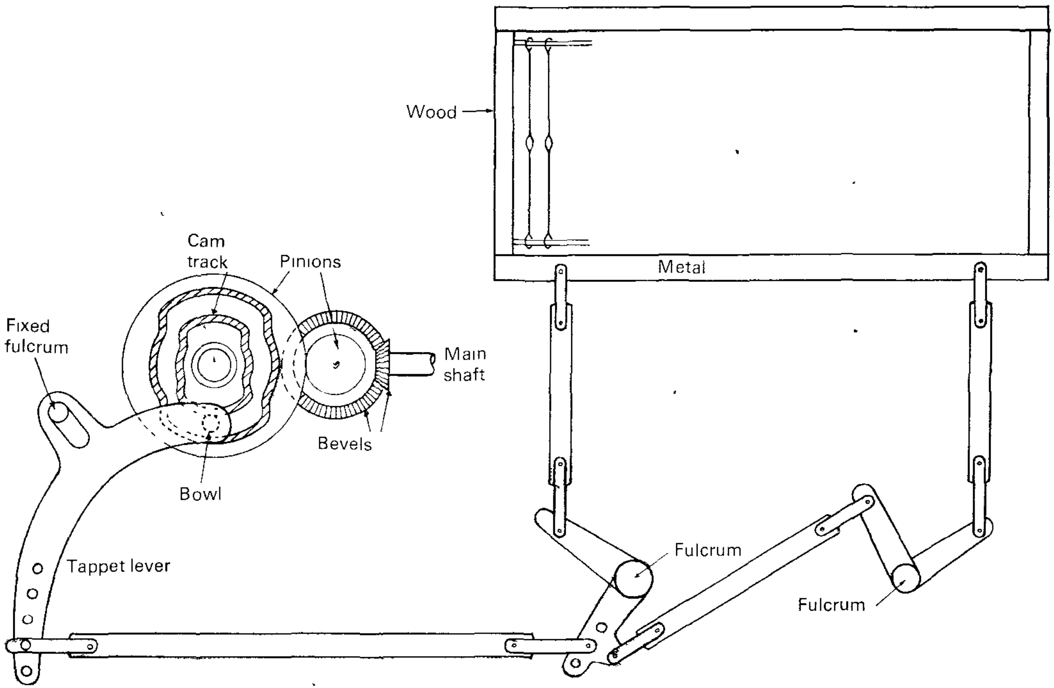


Fig. 3.7 Positive-cam shedding

on clips, which engage holes in the various levers. Alternative holes in the tappet lever and in one of the elbow levers provide means of adjusting the extent of the movement of the individual heald shafts.

The wooden ends of the heald shafts are thicker than the metal top and bottom pieces. As the shafts move up and down, their wooden end pieces are in sliding contact with each other, and the ends of the front and back shafts are supported by guides fixed to the loom frame, which thus ensures that the shafts remain erect. This mechanism is typical of many found on modern automatic looms. The snap-on connexions facilitate the overlooker's work, especially when he is changing one set of shafts for another at the end of the warp. There is no need to disturb any of the settings. There is, however, the disadvantage that any alteration that may from time to time have to be made to the level or lift of the shafts may have to be made underneath the loom in a rather inaccessible position.

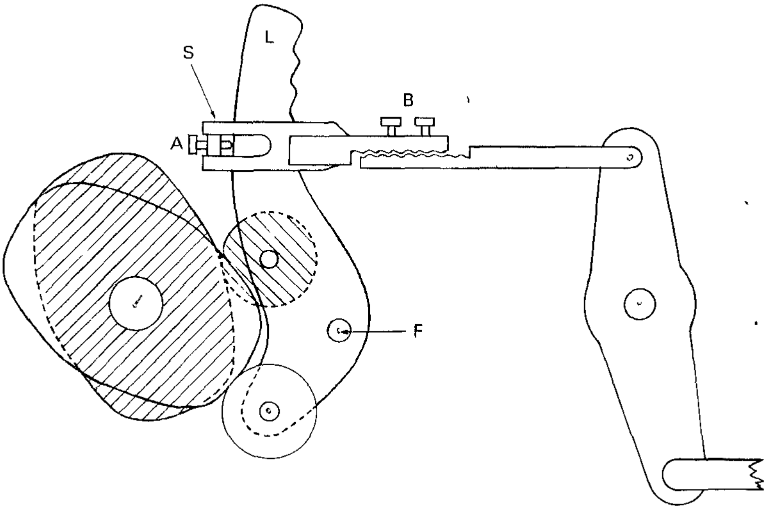


Fig. 3.8 Matched shedding cams

The second type of positive-cam shedding uses a pair of matched cams for each shaft. The cam motion on the Sulzer weaving machine (Fig. 3.8) is of this type. The lever L is oscillated about its fulcrum F by two anti-friction rollers, one of which is in contact with each cam face. Adjustment for the depth of the shed is made at A by moving the stirrup S up or down. This does not affect the height of the shed. The adjustment at B provides for altering the height of the shed. Both adjustments are in an accessible position at the side of the loom.

A system of this kind requires precision engineering and high-grade materials in order to maintain contact between both cams and their anti-friction rollers at all times. This is necessary for smooth operation of a mechanism on a loom such as the Sulzer, which may run at up to 300 picks/min. Wear is reduced to a minimum by enclosing the cam system in an oil bath. These provisions, together with rigid connexions to the heald shafts, provide a smooth and positive action, capable of high-speed operation over a long period.

An incidental advantage of positive cams mounted low at the side of the loom and used in conjunction with rigid connexions to the heald shafts is that the loom is freed from superstructure, which thus improves visibility and lighting in the weaving room and enhances the stability of the loom. The risk of oil from lubricated parts mounted over the warp dripping onto the warp is eliminated. The force of this last point has been reduced by the increased use of self-lubricating bearings, but the advantages of a loom with nothing above heald-shaft level are substantial.

3.2.5 Factors that Limit the Size of Repeat

The most severe limitation to the size of the weave repeat obtainable with cam shedding is imposed by the maximum practicable number of picks to the repeat. Suppose, for example, that we wish to design a set of negative cams for weaving 8-end weft sateen. We shall require eight cams, and each cam will rotate once every eight picks. Two possible contours for one of the cams are shown in Fig. 3.9. The throw is the same in both diagrams because $R_1 - r_1 = R_2 - r_2$, but $r_2 = 2r_1$. In each diagram, one pick occupies one-eighth of a revolution, that is, 45°. If a dwell of one-third of a pick is assumed, the shaft will rise while the cams turn 30° between a and b, will dwell during the next 15° between b and c, and will fall during the next 30° between c and d. The cam contour should be such as to give approximately simple harmonic motion to the cam follower, which is assumed to move in a vertical path on a line passing through the axis of the cam shaft. The slope of the contour will be steepest halfway through the rise and fall when the cam follower is in the position shown.

If friction is neglected, to produce a vertical force F, which can be applied to

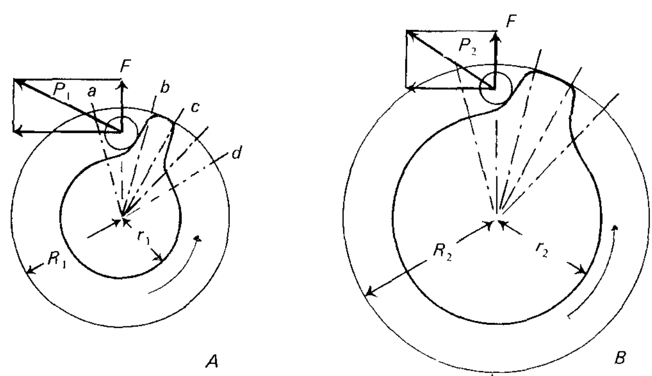


Fig. 3.9 Cam design

lift the heald shaft, the cam must exert a force P_1 or P_2 on the cam follower, this force acting perpendicular to the contour of the cam at the point of contact. It is clear that P_1 is much greater than P_2. Two main points emerge from these considerations:

(*a*) for a given size of cam, the maximum slope of the cam contour and the maximum force acting in the system both increase with the number of picks to the round; for any given cam size, there must be a practical limit to the number of picks to the round; and

(*b*) for a given number of picks to the round, the maximum slope of the cam contour and the maximum force acting in the system both decrease as the cam size increases.

It follows that, in order to obtain the maximum number of picks to the repeat, one should use the largest practicable size of cam. For most looms, the lift required at the heald shaft will be between 12·5 and 20 cm. If a lift of 15 cm and a treadle leverage of 3:1 are assumed, the cam must have a throw of 5 cm. In this case, the outside diameter of the cam in Fig. 3.9*B* would be 30 cm, which is probably too large to accommodate inside the loom frame. Much larger cams are used outside the loom frame for weaving heavy materials such as carpets and belting because the force required to raise the heald shafts is very large. The normal limit is eight, or occasionally ten, picks to the round. For special purposes, very large tappets made by bolting interchangeable segments onto a disc, and with twelve or even sixteen picks to the repeat, have been used on slow-running looms. These are quite exceptional.

If one is limited to eight picks to the repeat, there is not much point as a rule in trying to accommodate more than eight cams, because very few weaves require more shafts than there are picks in the repeat.

3.3 Dobby Shedding

3.3.1 Introduction

The terms 'positive' and 'negative' recur in connexion with dobby shedding, but the distinction is perhaps less subject to argument than it is for cam shedding. In negative-dobby shedding, the shafts are raised by the dobby and lowered by some form of spring undermotion. In positive-dobby shedding, the dobby machine both raises and lowers the shafts. Negative dobbies tend to be simpler, and, because they are satisfactory except for heavy fabrics and high loom speeds, they are commoner than positive dobbies, except in woollen and worsted weaving and for high-speed unconventional looms.

3.3.2 Negative-dobby Shedding

Several types of negative dobby have been used in the past, but nearly all modern negative dobbies, and some positive ones, are derived from the original 'Keighley' dobby. One reason for its success is that its cycle occupies two picks, and hence most of its motions occur at half the loom speed, which enables it to operate at relatively high speeds. In passing through various stages of evolution, it has been gradually refined and developed as a result of experience and improvements in engineering techniques. Its action can be understood from the line diagram in Fig. 3.10, which represents the main parts of the Keighley dobby in one of its early forms.

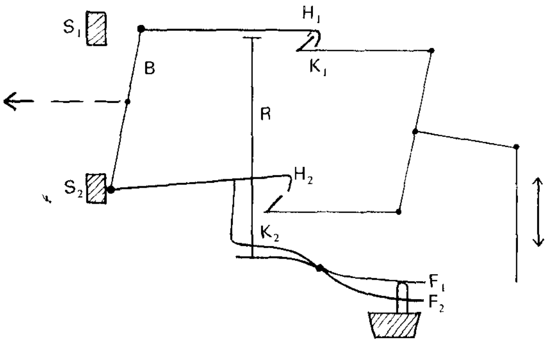


Fig. 3.10 The Keighley dobby

For each heald shaft, there are a baulk B, two hooks H_1 and H_2, and two feelers F_1 and F_2. The stop bars S_1 and S_2 extend the full depth of the dobby, as do the knives K_1 and K_2, which reciprocate in slots that may be horizontal as in Fig. 3.10. In modern versions of the dobby, they are usually inclined as in Fig. 3.15. The means of reciprocating the knives is not shown in detail in Fig. 3.10, but they complete one reciprocation every two picks; and in this case they are driven from a crank on the bottom shaft. The heald shaft is connected indirectly to the centre of the baulk B. In the diagram, the heald shaft has been raised by moving the top end of the baulk B away from its stop bar S_1. This has happened because the top knife K_1 has previously engaged the hook H_1, and has drawn the top end of the baulk B away from its stop bar S_1, this action causing the baulk to pivot about the point of contact between its lower end and the stop bar S_2. The knife K_1 was able to engage the hook H_1 because a peg in the lag forming part of the pattern chain had raised the right-hand end of the feeler F_1, which thus allowed the rod R to lower the hook H_1 onto the knife K_1. In the diagram, there is no peg to support the right-hand end of the feeler F_2, which has therefore fallen, this fall allowing its upturned left-hand end to raise the hook H_2 clear of the knife K_2. As the action continues, the top end of the baulk will be returned to its stop bar, and at the same time the bottom knife will move to the right without disturbing the bottom end of the baulk. The shaft will therefore be lowered and will remain down for the next pick. In the absence of a peg, the shaft is lowered or remains down. A peg will lift the shaft, and a succession of pegs will keep the shaft raised. Some form of spring undermotion, acting through the shaft and its connexions to the baulk, keeps one end of the baulk in contact with its stop bar while the other end is being displaced. Alternatively, it keeps both ends of the baulk in contact with the stop bars when the shaft is not being raised.

The full cycle of operations is illustrated by the simplified line diagrams in Fig. 3.11. In each diagram, the knives are shown at one extreme of their movement. Selection for the next pick necessarily takes place while the hook is in contact with its stop bar because only then can the hook engage or disengage its knife.

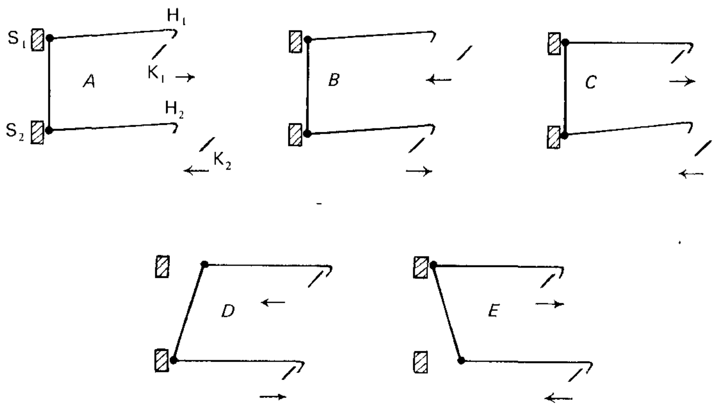


Fig. 3.11 The action of the Keighley dobby

Diagram A The top hook has just been raised because no peg has been presented to its feeler, and, since the bottom hook was not engaged on the previous pick, the shaft is down and will remain down for the next pick.

Diagram B This represents the situation one pick later than diagram *A*. The bottom hook has been raised because again no peg has been presented to its feeler, so the shaft will remain down for the next pick, and it will continue to remain down so long as no peg is presented to either feeler.

Diagram C The shows the situation one pick later, on the assumption that the shaft is to be lifted on the next pick. It is the same as diagram *A* except that the top hook has been lowered through the action of a peg in the appropriate hole in the lag. As the top knife moves to the right, the shaft will be lifted as in diagram *D*.

Diagram D Here a peg in the appropriate hole has allowed the bottom hook to engage its knife so that, as the cycle continues, the bottom knife will move the bottom end of the the baulk away from its stop bar, while at the same time the top knife will allow the top of the baulk to return to its stop bar, as in *diagram E*. While this is happening, the baulk is rotating about its centre, to which the heald shaft is connected: there will be virtually no movement of the shaft, which will remain up as long as pegs continue to be presented to each feeler in turn. There is no wasted movement: a shaft moves only when it is required to do so. This economy of movement can be achieved only if the dobby is 'double-acting', that is, if its cycle extends over two picks. All modern dobbies, whether based on the Keighley principle or not, are double-acting.

The method of pegging the lags for a Keighley dobby is illustrated in Fig. 3.12, in which a 2/2/1/1 twill is taken as an example. Each lag serves for two picks, and the holes in the lags are staggered to correspond with the positions of the feelers. The pattern barrel is turned intermittently by a Geneva wheel or similar motion so as to present a new lag every second pick. In the diagram of the lags, a filled circle represents a peg. Pegging starts at A, which corresponds to A in the weave diagram. In practice, it would be necessary to peg two or more repeats of the weave in order to have a pattern chain sufficiently long to encircle the rotating barrel that presents the lags to the feelers.

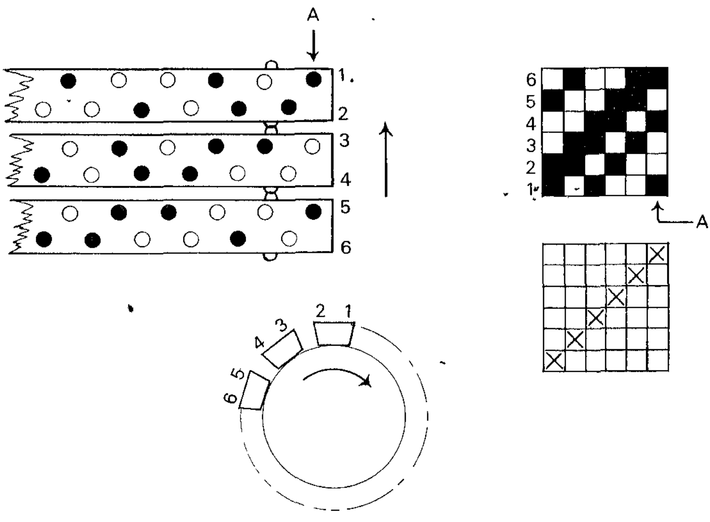


Fig. 3.12 Pegging the lags

The conditions shown in Fig. 3.12 will produce a Z twill in the fabric as required. If, however, the draft had been a straight S one instead of a straight Z one, the twill direction would have been reversed. For any weave that has a directional effect, it is therefore necessary to know what draft has been used before pegging the lags. If an S draft had been used in Fig. 3.12, the required Z twill would result by pegging as before, but by reading the peg plan from left to right.

Dobbies are normally mounted at the left-hand side of the loom, so the lag barrel in Fig. 3.12, seen from the front of the loom, is turning clockwise or inwards, which is usual. If the direction of rotation of the barrel were reversed, the twill direction would also be reversed. This could again be corrected by reading the peg plan from left to right. In practice, this situation would probably never arise, because, if the dobbies in a weaving room did rotate outwards, one would use S drafts to give Z twills with normal pegging and reading.

There are many different methods of connecting the centres of the baulks (see Fig. 3.10) to their heald shafts. Negative dobbies are always designed to raise the shafts, and the connexions are therefore made between the baulks and the tops of the shafts. Four of the many possible methods are illustrated in Fig. 3.13. In each diagram, the arrow represents the link with the centre of the

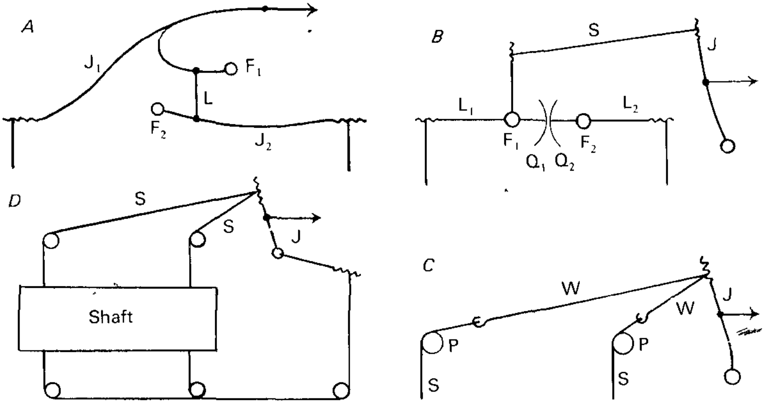


Fig. 3.13 Heald-shaft connexions

baulk. In diagram *A*, the main jack, J_1, has a fixed fulcrum at F_1. It is connected by a link L to a secondary jack, J_2, which is fulcrummed at F_2. The heald shaft is suspended from cords attached to the outer ends of the two jacks. Alternative points of connexion allow the lift of individual shafts to be adjusted. In diagram *B*, an elbow lever, L_1, which is fulcrummed at F_1, carries a toothed quadrant, Q_1, which engages a similar toothed quadrant, Q_2. This is carried on an extension of lever L_2, which is fulcrummed at F_2. The upper end of the lever L_1 is connected by a streamer S to the top of the jack J. This point of attachment provides adjustment for the lift of the shaft.

Both these methods necessitate the positioning of bearings, and of numerous metal parts in rubbing contact with each other, above the warp. It is essential to fix a tray underneath the moving parts to catch oil drips, which are always heavily contaminated with dark-coloured metallic impurities. Stains on the warp produced by dirty oil are very difficult to remove, and, if not completely removed, may cause tendering of the yarn during bleaching. The problem of preventing oil stains is largely overcome, and visibility and lighting are also improved, by using some form of flexible coupling to the heald shafts. A typical method is shown in Fig. 3.13*C*. Straps or cords, S, from the heald shaft are hooked onto a piece of stiff bent wire, W, which is connected by a stirrup to the jack J. The straps or cords pass over free-running pulleys P, which should preferably have self-lubricating bearings. An adaptation of this idea for use with a positive dobby is illustrated in Fig. 3.13*D*. This will be referred to again in Section 3.3.3.

A disadvantage of the systems of connexion shown in Fig. 3.13 *C* and *D* is that the two ends of the heald shaft do not necessarily receive the same lift. The geometry of the system determines which end, if either, moves further. This can be corrected by making the two connexions to different points on the jack and by experimenting with the points of connexion until an even lift is obtained. The point just discussed is not important as a rule in narrow looms, but it becomes necessary to pay attention to it in wider looms.

The negative dobby raises the shafts but cannot lower them, and it will not work without some means of keeping the ends of the baulks in contact with their stop bars when they are not being displaced by the action of the dobby. A spring undermotion normally performs both these functions. The simplest arrangement consists of two coil springs for each shaft, the springs being stretched between the shaft and a horizontal bar fixed to the loom frame near the floor. This, as we have already seen in connexion with cam shedding (Section 3.2.3), is an inefficient arrangement because the tension in the springs is

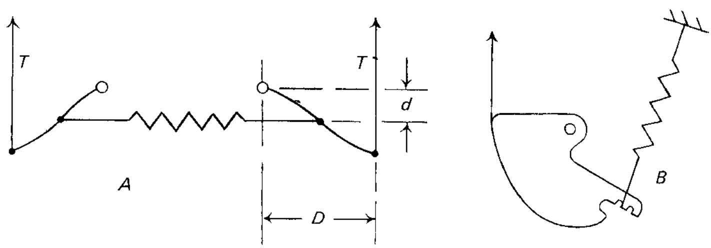


Fig. 3.14 Spring undermotions

least when the shafts are down, which is when the springs should be performing both their functions. Nevertheless, this simple system has been widely used for weaving lightweight fabrics, which can be woven with a relatively low warp tension. As the warp tension increases, stronger springs have to be used to overcome the vertical component of the warp tension. This tends to bend the heald-shaft frames when the shafts are lifted and the springs are fully extended. A widely used device, which is designed to avoid an increase in spring tension when the shafts are raised, is the Kenyon undermotion, which is illustrated diagrammatically in Fig. 3.14*A*. It is similar in principle to the spring overmotion already discussed in connexion with cam shedding (see Fig. 3.5).

If T is the tension in each of the connexions to the heald shafts and S is the tension in the spring, then, by taking moments:

$$TD = Sd,$$

and

$$T = \frac{Sd}{D}.$$

As the shaft rises, the spring will stretch and S will increase, but only slightly.

At the same time, *d* will decrease substantially, and *D* will increase slightly. The slight increase in *D* and the large decrease in *d* as the shaft rises both tend to reduce *T* and will more than compensate for the increased tension in the spring. The tension *T* will therefore tend to decrease as the shaft rises, which is what we require. One variation of a more modern application as used by Rüti, and which uses the same principle, is shown in Fig. 3.14*B*. In this case, there would be a similar unit at each side of the heald shaft, but it is possible to have connexions from each end of the heald shaft to a similar type of spring motion in which only one spring is used.

Until comparatively recently, it was customary to actuate the knives of Keighley dobbies from a crank mounted on the end of the bottom shaft. This produces approximately simple harmonic motion, with no possibility of introducing a planned period of dwell. The modern tendency is to actuate the knives from cams mounted on a shaft in the dobby, this shaft being driven through bevel gears or by chain and sprocket from the crankshaft. The cams can be designed to give any required period of dwell, usually about one-third of a pick. Cam dobbies were originally introduced for weaving continuous-filament yarns, for which a period of dwell is particularly desirable in order to minimize abrasion and filamentation of the warp by the shuttle. Their advantages are being increasingly appreciated in other sections of the industry.

The Rüti dobby illustrated by line diagrams in Fig. 3.15 is an example of a cam-driven dobby. Fig. 3.15*B* shows how the knives K_1 and K_2 are actuated by cams C_1 and C_2 mounted on the camshaft C.S. The cams are negative in action, and the cam followers are kept in contact with the cams by means of springs not shown in the diagram. The knives are pushed and not pulled by the cams, and this, together with the inclined knife tracks, produces a more compact and efficient machine. These features are commonly found on modern Keighley dobbies. The selecting mechanism (pattern chain, feelers, etc.) is not shown in Fig. 3.15, but it is similar to that in Fig. 3.10.

Fig. 3.15*A* and Fig. 3.15*C* show how the motion of the knives is transmitted to the heald shafts through the hooks H_1 and H_2, the baulk B, the jack J, and a chain attached to the top of the slider S. Diagrams *A* and *B* have been separated for clarity, but to get the complete picture they should be superimposed. The arrangement is clearly much more compact than the one shown in Fig. 3.10. To revert to Fig. 3.15*C*, the spring undermotion is connected to the lower ends of the sliders S_1 and S_2. All the connexions are semi-permanent and need not be disturbed when the shafts are changed. When this is necessary, the shafts are released from their sliders and then removed. No adjustment should be needed when the new set of shafts is put in. A simple screw adjustment is provided for altering the height of individual shafts, and their lift can be varied by changing the point of attachment to the jack. In order for it to be possible to lift the shafts by chains attached to one side of the shafts, it is necessary for the links of the undermotion to be rigid and free from play. The loom is relatively free from superstructure, because, although the dobby is above shaft level, there is nothing directly above the warp. The complete elimination of superstructure requires a positive dobby mounted low down at the side of the loom and having rigid connexions to the bottom of the heald shafts.

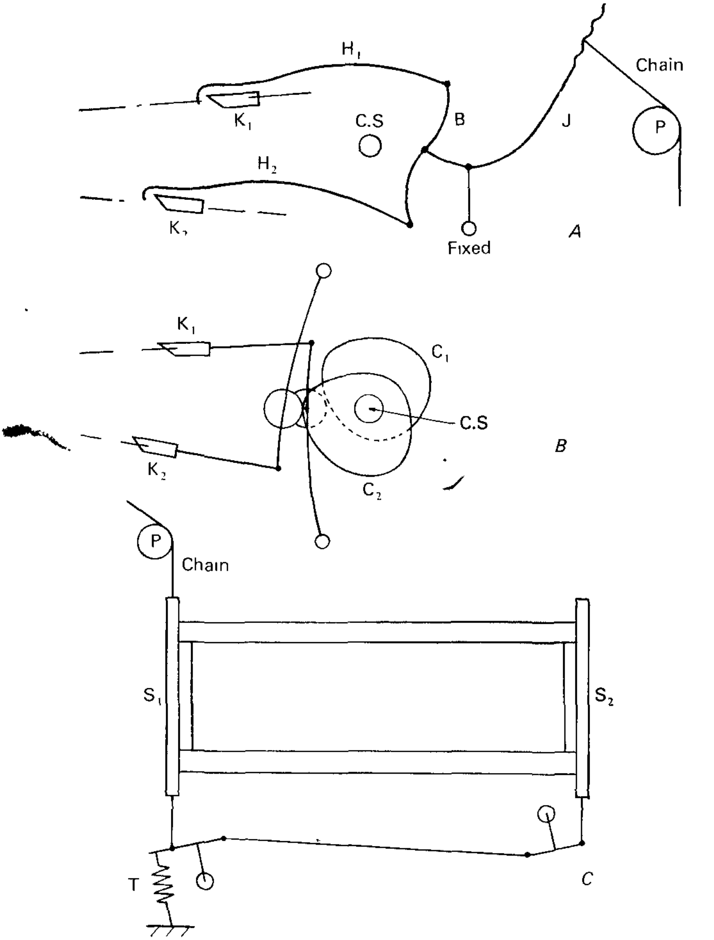


Fig. 3.15 The Rüti cam dobby

In another development of the Keighley dobby, the chain of lags is replaced by a loop of very tough paper or plastics sheet. Holes punched in the paper correspond to pegs in the lags, a hole causing the corresponding shaft to be lifted. Light feelers are used to detect the presence or absence of a hole, but the force required to move the hooks is not supplied by feelers. A typical system of selection for a paper-controlled dobby is illustrated in Fig. 3.16.

The feelers, selection needles n_1 and n_2 in this case, are lowered onto the paper pattern by a cam control, which is not shown in the diagram. If there is a

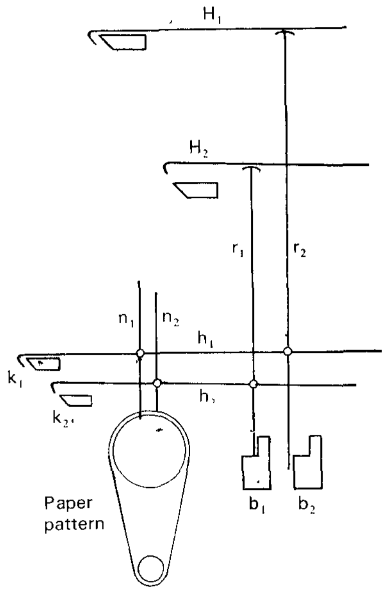


Fig. 3.16 A selection mechanism for paper patterns

hole, the corresponding supplementary hook, h_1 or h_2, is lowered into the path of one of the reciprocating suppplementary knives, k_1 or k_2, which then moves the corresponding vertical rod, r_1 or r_2, out of the path of the vertically reciprocating lifting block, b_1 or b_2. In this case, the main hook, H_1 or H_2, is lowered onto its knife, and the heald shaft is lifted. If there is no hole in the pattern, the corresponding lifting block, b_1 or b_2, lifts the main hook out of the path of its main knife, and the shaft is not lifted. As a result of this indirect action, very little force is exerted by the selection needles on the pattern paper, which consequently has a long life. Pegs tend to wear, break, and fall out, which produces wrong lifts and faults in the fabric. Paper control eliminates these faults. The paper pattern takes less time to prepare—by means of a punching machine designed for the purpose—than the corresponding set of lags unless the repeat is small, in which case several repeats need cutting for the paper-pattern method. The paper pattern is very much lighter and more compact than the corresponding set of lags, and this is particularly important in weaving very long patterns, such as occur in the manufacture of bordered handkerchiefs and table cloths, which normally have several hundred picks to the repeat. In practice, these obvious advantages of paper control have to be weighed against the extra cost and complexity of the machine.

A traditional method of dealing with long patterns on the Keighley dobby is to provide it with two, or sometimes with three, pattern barrels. Suppose, for example, that a terry towel has a ground design on 48 picks repeated twenty times in the length of the towel, with a border at each end consisting of one repeat on 60 picks of a different design. To weave this on a dobby with only one pattern chain would require 540 lags (1080 picks). With a twin-barrel dob by—also known as a cross-border dobby—we simply put one repeat of the ground weave (24 lags) on one barrel and the border design (30 lags) on the other barrel. A third control barrel carries a short set of small lags, pegged so that the required number of repeats of the two patterns are woven alternately. The paper-pattern-controlled dobby is tending to replace the cross-border dobby, especially when looms are not always engaged on weaving very long repeats.

3.3.3 Positive-dobby Shedding

For weaving heavy fabrics, such as fancy woollens and worsteds, it is better to use a positive dobby, which eliminates the need for a spring undermotion. Without an undermotion, however, there is no longer any force tending to return the baulks to their stops after having been displaced by the knives or to hold the baulks in contact with their stops when not being displaced by one of the knives. The vertical component of warp tension tends to return the shafts to their mid-position, so that the baulks tend to float out of contact with their stop bars. In any positive dobby, it is therefore necessary to provide means for returning the ends of the baulks to their stop bars and for holding them there. This can be achieved without radical alteration to the Keighley dobby, as indicated in Fig. 3.17. Push bars B_1 and B_2 are rigidly connected to the knives K_1

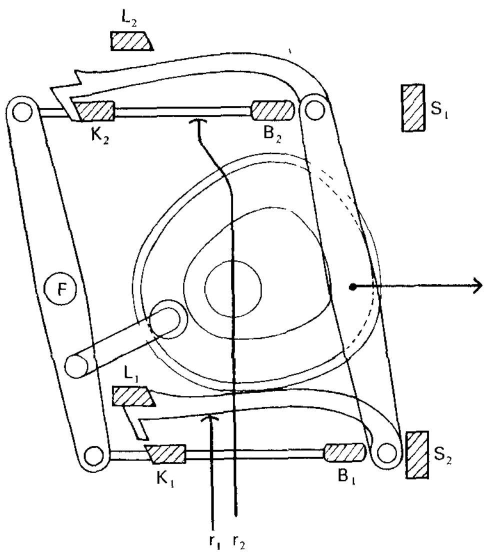


Fig. 3.17 A positive dobby

and K_2 and reciprocate with them. As a knife returns after displacing a hook, its baulk bar pushes the corresponding end of the baulk against its stop bar. The appropriate locking bar, L_1, then engages a notch in the hook because the hook has been pushed up from the selection mechanism. This prevents that end of the baulk from moving until the time arrives for the next selection. In the diagram, the locking bar L_1 will hold the bottom of the baulk against its stop while the knife K_1 and the push bar B_1 make one complete cycle. Note that the locking bars can engage and disengage only at the time of selection, when the corresponding end of the baulk is against its stop bar. The dobby just described happens to be paper-controlled, the control mechanism being similar to that shown in Fig. 3.16.

The Knowles positive dobby, which is used on the British Dobcross and on the American Crompton & Knowles woollen and worsted looms, has an action that is quite different from that of the Keighley dobby. In Fig. 3.18*A*, two cylinders, C_1 and C_2, rotate continuously on fixed centres and make one revolution every pick. Approximately half the circumference of each is machined to form an elongated gear wheel. Toothed discs, B, one for each shaft, are pivoted individually on levers L, which are fulcrummed at F. In Fig. 3.18*A*, the lever L has been raised by a bowl on the pattern chain, and the cylinder C_2 is about to turn the disc B through approximately half a revolution. A missing tooth in B enables it to engage with the teeth on the cylinder C_2. The rotation of B from the position shown in Fig. 3.18*A* moves the jack J in a clockwise direction by means of a link E. This lifts the heald shaft through connexions to the jack similar to those shown in Fig. 3.13*D*. The position is now as in Fig. 3.18*B*. A slot in the disc B, concentric with its axis, is used to limit the extent of its rotation.

If another bowl raises the lever L on the next pick, the cylinder C_2 will be unable to turn the disc B because there is now a blank space, corresponding to three missing teeth, in B at the point of engagement. The shaft will remain up so long as a bowl continues to be presented on successive picks. Suppose, however, that at some stage a sleeve is presented, the sleeve having a smaller diameter than a bowl. Lever L and disc B are now allowed to fall and take up the position shown in Fig. 3.18*C*, in which B is about to be turned in the opposite direction by the cylinder C_1. This will rotate the jack in an anti-clockwise direction and lower the heald shaft as in Fig. 3.18*D*. So long as sleeves continue to be presented on successive picks, the shaft will remain down. As soon as a bowl is presented, we have the situation shown in Fig. 3.18*A*, and the shaft will be lifted.

A locking bar, not shown in the diagram, engages the free ends of the levers L during the time in which the discs B are in mesh with one or other of the two cylinders so as to ensure that they are not forced out of mesh. The locking bar is actuated by a cam on the bottom cylinder shaft. It would be possible to eliminate superstructure by making rigid connexions from a single jack of suitable shape to the underside of the shaft.

Dobross and Crompton & Knowles looms, in common with other looms for weaving fancy woollen and worsted fabrics, have rising and falling shuttle-boxes at one or both sides of the loom. The Knowles dobby incorporates ad

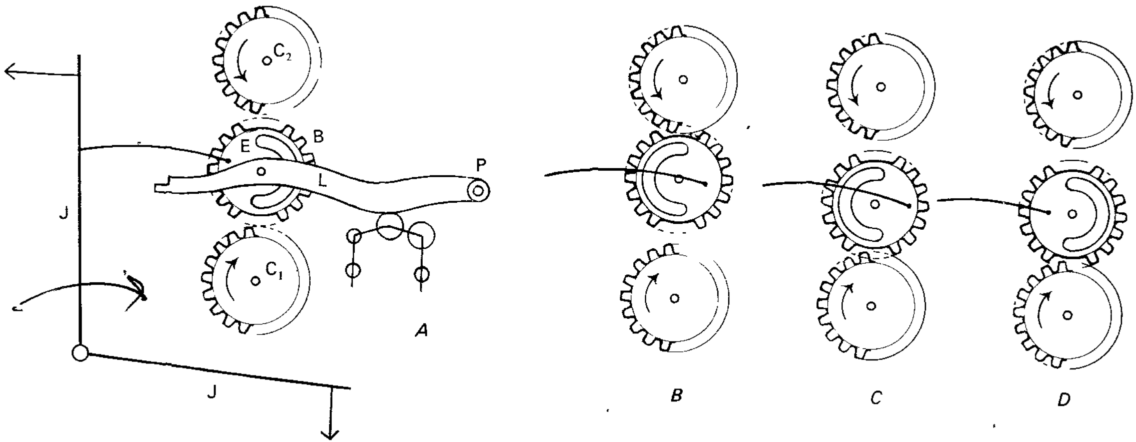


Fig. 3.18 The Knowles positive dobby

ditional levers L and discs B for controlling the boxes from a separate pattern chain. The alternative would be to control the boxes from a mechanism quite separate from the dobby, in which case there would be the possibility that the boxes would get out of step with the weave, especially after unweaving. This possibility does not exist if the boxes are controlled from the dobby.

Until recently, positive dobbies were used only for weaving heavy, wide fabrics on comparatively slow-running looms. Speed was not a factor in the decision to use a positive dobby, nor were positive dobbies designed to run at high speeds. The Knowles dobby is one of the few single-acting dobbies still in

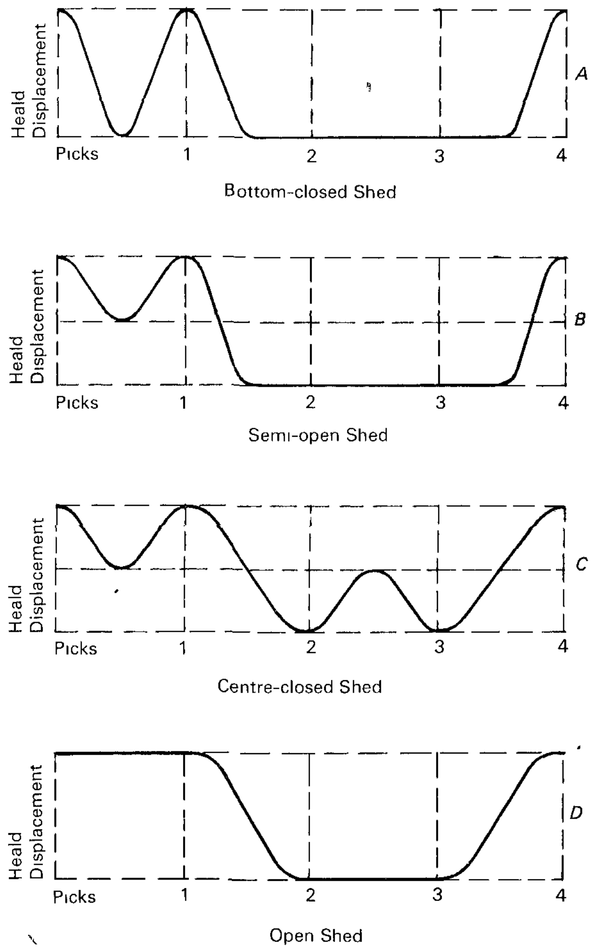


Fig. 3.19 Types of shed

use, but, because most of its motions are rotary, its speed is less limited than was that of other single-acting dobbies that have become obsolete. In recent years, the speed of conventional looms has tended to increase, so that, for example, single-shuttle automatic looms of about 1·5-m reedspace can run at speeds of up to 220 or 270 picks/min. Some unconventional looms run at much higher speeds than this—up to 300 picks/min for a 216-cm Sulzer machine, and over 400 picks/min for an air- or water-jet loom. The gravity-controlled hooks and feelers of conventional design tend to be too sluggish for speeds much in excess of 200 picks/min, and most makers of dobbies now offer a positive dobby capable of running at speeds of up to about 300 picks/min, with some makers having models capable of speeds of up to 500 picks/min.

The examples that have been given above sufficiently illustrate the principles of positive-dobby shedding without exhausting the possibilities. The Leeming dobby, for example, is a widely used alternative to the Knowles dobby in woollen and worsted weaving, and Stäubli enjoy worldwide renown for their range of dobbies. As already mentioned, the principle of paper-pattern control is applied to modern high-speed positive dobbies. Developments in design proceed almost continuously.

3.4 Types of Shed

3.4.1 Introduction

This is a convenient point at which to introduce the type of motion given to the heald shafts by different types of shedding mechanism. We may classify the shedding action or the type of shed produced according to the position the ends assume between successive picks and the nature of the movement given to the ends. The four main types are illustrated in Fig. 3.19, in which the horizontal lines represent the top, middle, and bottom positions of the ends. The spaces between the vertical lines each represent one pick. Let us assume, for example, that a shaft is to be lifted for the first two picks and lowered for the third and fourth picks. The curves then represent the motion of this shaft over the four-pick cycle.

3.4.2 Bottom-closed Shed

Here (Fig. 3.19*A*) all the ends return to the bottom position to form a closed shed after every pick. There is a great deal of wasted movement in this weave, as there would be in any other weave that required some ends to be lifted for two or more picks in succession. Some early types of dobby produced this type of shed, but it is encountered nowadays only in handlooms and in single-lift jacquards (see Section 3.5).

3.4.3 Semi-open Shed

In this case (Fig. 3.19*B*), as in bottom-closed shedding, the rest position is at the bottom, but an end required to stay up for two or more picks in succession drops only to the centre position between picks. There is less wasted movement than in the types of shed shown in Fig. 3.19*A* or *C*. This type of shed is formed by double-lift jacquards (see Section 3.5). The ends do not form a single sheet between successive picks (i.e., the shed is not closed), but neither does it remain fully open, and hence we have the term 'semi-open'.

3.4.4 Centre-closed Shed

Here (Fig. 3.19*C*) all the ends return to the middle position to form a closed shed after every pick. Whatever the weave, every end must be either raised or lowered at every pick, so again there is a great deal of wasted movement. Centre-shed dobbies have been used for weaving some kinds of leno fabrics in which the crossing of the ends was facilitated by the return of all the ends to the centre position after each pick. A simple adaptation of the Keighley dobby, by the addition of a 'shaker' motion, serves the same purpose, and the centre-shed dobby is now practically obsolete. It is still used in some jacquard machines because the movement is balanced and the wasted movement is reduced and distributed uniformly over all the ends, and it is therefore preferable in this respect to bottom closed shedding.

3.4.5 Open Shed

Here (Fig. 3.19*D*) there is no rest position. Ends required to stay up or down for two or more consecutive picks simply stay up or down. There is no wasted movement, and the shed never closes, unless the weave is such that there is a complete interchange in the position of the ends between successive picks. This is an exceptional condition encountered in weaves such as plain, warp-rib, and matt weaves. For most purposes, this is the ideal type of shed. It is attained without difficulty in tappet shedding, especially with positive tappets. Double-acting dobbies give a close approximation to it, with a tendency for the shafts to drop slightly from their top position when up for two or more consecutive picks, particularly on crank-driven dobbies. This is because there must be some clearance between the edge of the griffe and the hook at the time of selection, but even this slight drop is minimized by giving special attention to the cam profile on cam-driven dobbies so that the inner griffe starts to move out before the other one commences its inward movement. The result is a movement that falls between that of Fig. 3.19*C* and that of Fig. 3.19*D*, but it approximates more closely to the latter. For all practical purposes, we can regard double-acting dobbies as producing an open shed. The same applies to the Knowles dobby (Fig. 3.18).

3.4.6 Advantages of the Different Types of Shed

It is much easier for the weaver to repair a broken end if all the ends are in one plane. With bottom-closed and centre-closed sheds, this happens after every pick, but, in general, it does not happen at all with semi-open and open sheds. It is therefore customary to fit a heald-levelling device to double-acting dobbies to enable the weaver to bring all the shafts into the bottom position quickly and easily while broken ends are being repaired.

3.5 Jacquard Shedding

3.5.1 Introduction

It is intended here to introduce only the elementary principles of jacquard shedding, since anything more, to be effective, would require a rather extensive treatment. Many specialized types of jacquard machine have been developed for weaving particular kinds of fabric, such as terry towels, damasks, lenos, moquettes, and carpets. It will not be possible to describe these. Most of the rest—general purpose types—are comparatively easy to classify. The first broad classification is into coarse-pitch (English-pitch) and fine-pitch (French- or Continental-pitch) machines. The term 'pitch' refers to the distance between the needle centres. Within these two main groups, a further classification may be made according to whether the machine is single-acting (single-lift) or double acting (double-lift).

3.5.2 Coarse-pitch (English-pitch) Machines

The simplest of these is the single-lift, single-cylinder jacquard. In this machine, we have one needle and one hook for every end in the repeat. Common sizes have 200, 400, or 600 needles. A 600-needle jacquard, for example, would have twelve horizontal rows of needles with 50 needles in each row, plus a few extra needles, which need not concern us. Each needle is kinked round a vertical hook, which it controls. One short row of needles and hooks is shown in Fig. 3.20. Coil springs press the needles towards the right. There is a lifting knife for each long row of hooks, so that, in a 600-needle machine, there would be twelve knives. The knives are fixed in a frame called a *griffe*, which reciprocates vertically once every pick and is normally driven by crank or by chain and sprocket from the crankshaft. The knives are shown end-on in Fig. 3.20.

The design is punched in pattern cards, one for each pick in the repeat, and the cards are laced together to form a continuous chain. The cards are presented to the needles by a card 'cylinder', which has a square section in the diagram, but which may be pentagonal or hexagonal. After presenting a card to the needles, the cylinder moves away from the needles a distance sufficient to allow it to be turned to present the next card. If there is a hole in the card opposite a particular needle, the needle will enter the hole (the cylinder being perforated to receive it), and the needle spring will cause the hook to engage its knife. This particular hook and the ends it controls will therefore be lifted when the griffe rises. If there is no hole opposite a particular needle, it will be forced to the left as the cylinder moves inwards. The hook it controls will be moved out of the path of its knife, so that the hook and the ends it controls will not be lifted.

The single-lift, single-cylinder (S.L., S.C.) jacquard generally forms a bottom-closed shed (Fig. 3.19*A*). There is therefore much wasted movement,

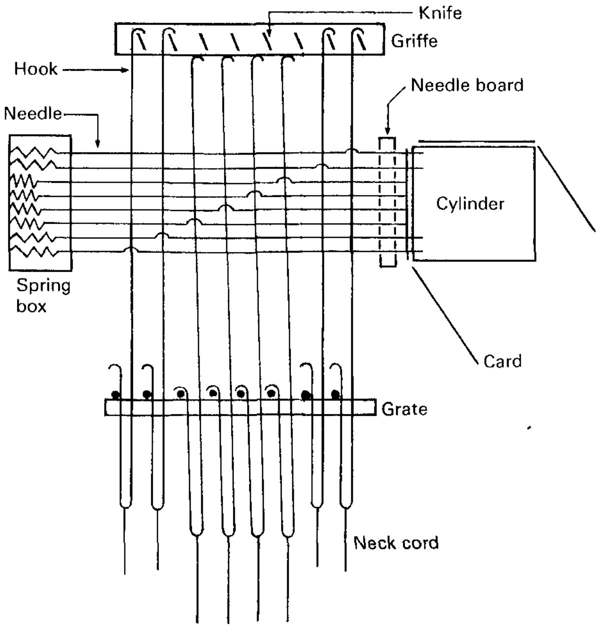


Fig. 3.20 A single-lift, single-cylinder jacquard

together with a tendency for the whole harness to swing. Moreover, the cylinder must reciprocate and turn, and the griffe must rise and fall every pick. These considerations drastically limit the speed at which the loom can be run.

The double-lift, single-cylinder jacquard has two sets of knives, each mounted in a griffe. The two griffes move up and down in opposition over a two-pick cycle. A 600-needle machine has 1200 hooks, and each needle controls two hooks as shown in Fig. 3.21, which represents (on the left) the situation when a harness cord has been lifted by one hook. It might equally well have been lifted by the other hook. The situation when neither hook has lifted its harness cord is shown on the right. The sequence of events when a harness cord is required to remain up for two or more consecutive picks is as follows.

Imagine that the hook D is about to descend and that the hook C is about to rise. The harness cord it controls will be lowered to the centre position as hook D descends. When it reaches the centre position, it will be taken over by the hook C, which will return it to the raised position. The result is that an end required to remain up for two or more consecutive picks is lowered halfway between the picks. This machine produces a semi-open shed (Fig. 3.19*B*). There is less unnecessary movement than with the single-lift, single-cylinder jacquard and less tendency for the harness to swing. The cylinder must still reciprocate and turn every pick, but the rate of reciprocation of the knives is

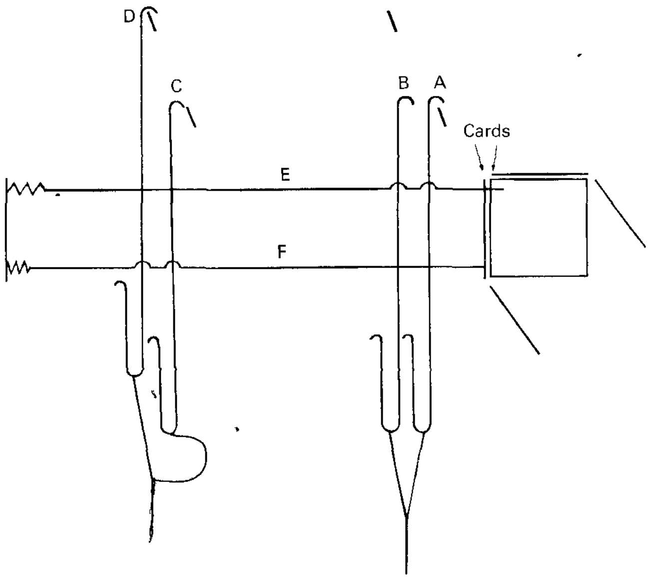


Fig. 3.21 A double-lift, single-cylinder jacquard

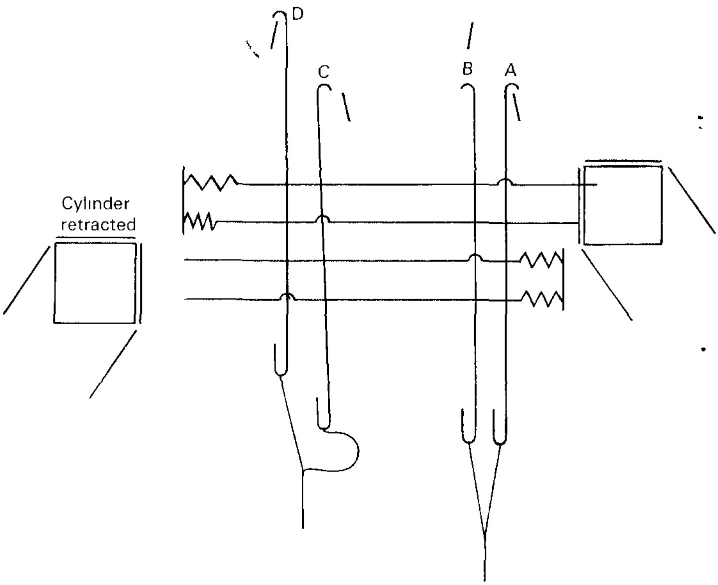


Fig. 3.22 A double-lift, double-cylinder jacquard

halved. The net result is that the machine is capable of somewhat higher speeds than its single-lift counterpart.

The double-lift, double-cylinder machine represents a further development. In this machine, each harness cord and each end in the repeat are controlled by two needles and two hooks, so a 600 machine has 1200 needles and 1200 hooks to control 600 ends in the repeat. There are also two cylinders, one carrying the odd-numbered and the other the even-numbered cards. If it is assumed that the left-hand cylinder in Fig. 3.22 carries the odd numbered cards, it will be presented to its needles on odd picks, the right-hand cylinder being presented on even picks. Apart from this, its action is the same as that of the double-lift, single-cylinder machine, and it forms a semi open shed. It is, however, fully double-acting, and, because the speed of the cylinder is halved compared with the double-lift, single-cylinder machine, it is capable of running at somewhat higher speeds than the latter. For many purposes, it has largely replaced the other two types. The weaver must take care to keep the two sets of cards in step, especially after pickfinding or unweaving.

Jacquard machines that produce an open shed have been introduced recently, and an example of these will be given in the next section.

3.5.4 Special Jacquard Machines

There are two types of fine-pitch jacquard, both having originated in France. The Vincenzi machine uses pattern cards and a direct method of selection, similar to that of the coarse-pitch machines. The Verdol machine uses an endless paper pattern and an indirect method of selection, similar to that used in paper pattern-controlled dobbies. Standard sizes control 880 or 1320 ends per repeat (Vincenzi) or 892 or 1344 ends per repeat (Verdol). Double sizes control twice as many ends per repeat. A 1320 Vincenzi machine is no larger than a 600 coarse-pitch machine, and a 1344 Verdol is smaller. Because of their compactness, fine-pitch machines of both types have been widely used for designs having a large number of ends in the repeat. They are useful for weaving very large designs in relatively coarse constructions, such as might be used for bedspreads or curtains, and they are equally useful for weaving medium sized repeats in very fine constructions, such as might be used in damask and brocade designs for table linen, upholstery fabrics, and evening gowns. Until quite recently, the Vincenzi tended to be preferred to the Verdol in Britain, chiefly because a clean atmosphere with uniform temperature and humidity is necessary for Verdol machines. With the increasing use of air-conditioning plants, these conditions have become available, and the Verdol machine is gaining in popularity.

In double-lift coarse-pitch machines (Figures 3.21 and 3.22), each harness cord is controlled by two separate hooks linked by a neck cord, but in both Vincenzi and Verdol machines these two hooks are replaced by a double hook, the upper ends of which are always tending to spring apart because of spring-loading in the U-loop at the base of the hook unit. This eliminates the need for coil springs to return the needles to their original positions as used in coarse pitch machines. To avoid excessive needle pressure on the card, stop bars are incorporated as illustrated in Figure 3.23 and 3.24. The lower ends of the hook units are kept vertical by steadying or anti-swing bars.

The Vincenzi machine has been made in single cylinder, single-lift; single cylinder, double lift; and double-cylinder, double lift forms, but nowadays the last-named is the common form, and, like the double lift coarse pitch machines, it has two sets of knives, each mounted in a griffe. In Fig. 3.23, the

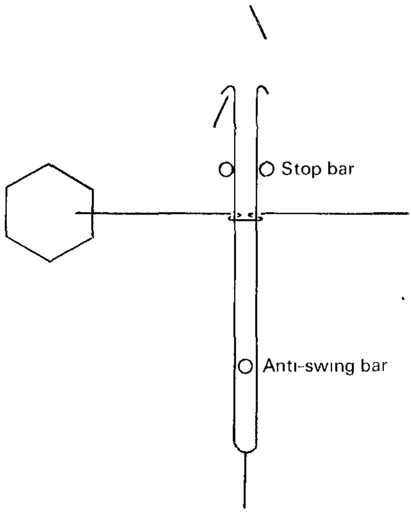


Fig. 3.23 The Vincenzi jacquard

left hand cylinder has just been presented, and a hole in the card has allowed the needle to penetrate the cylinder. Consequently, the left-hand hook of the hook unit has not been displaced, and it is about to be lifted by its knife. One pick later, the right-hand cyclinder will be presented, and the harness cord will be either lifted or left down by the right-hand hook, according to whether there is a hole in the card or not. A semi-open shed is produced, but, instead of the transfer's taking place at the neck cord, it occurs at the hook. This tends to produce a high breakage rate at the hooks instead of at the neck cords. It is a debatable point as to which is the more inconvenient.

The cylinders of the Vincenzi machine are hexagonal to facilitate rapid presentation of the cards. This is achieved because the cylinder will turn through a reduced angle between selections and will not have to move as far away from the needles to allow it turn. It is for this reason that many coarse-pitch machines are now made with pentagonal cylinders.

The Verdol machine is always a single-cylinder machine, which may be single-lift or double-lift. The method of selection, which is common to both machines, is shown in Fig. 3.24*a*. The selecting needles S are lowered onto the paper pattern by the selection board B as it swings to the right in its downward

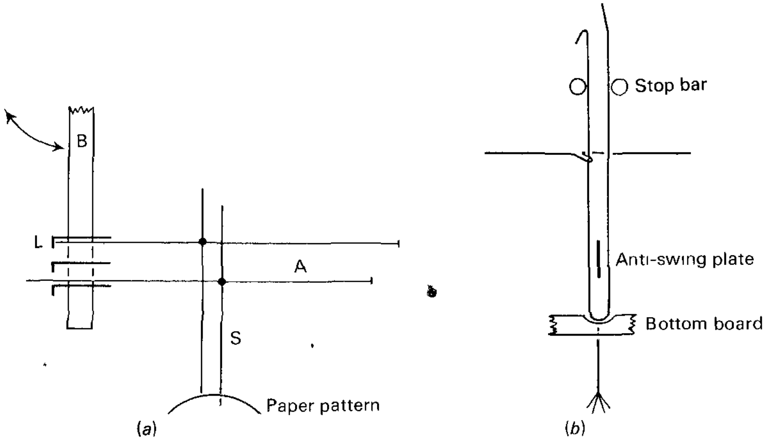


Fig. 3.24 The Verdol jacquard (single-lift)

arc. If a selecting needle enters a hole in the paper pattern, it allows the corresponding auxiliary needle (A) to fall and so escape the action of one of the lips (L) on the selection board. The auxiliary needles (and thus the jacquard needle and hook) are not displaced, and the hook unit is lifted by its knife. If there is no hole in the paper pattern, the auxiliary needle is prevented from falling and is

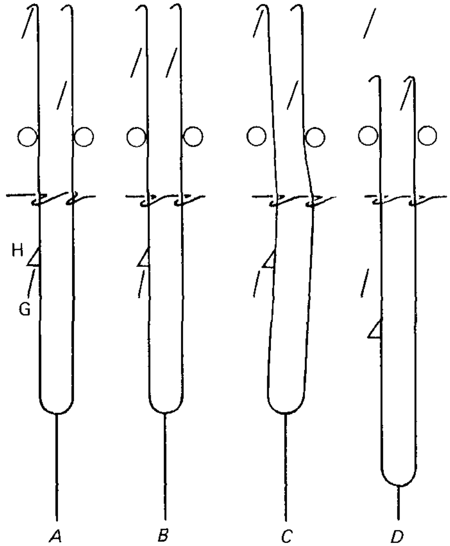


Fig. 3.25 The Verdol jacquard (double-lift)

acted upon by a lip on the selection board. This displaces the needle and corresponding hook unit so that the hook unit will not be lifted.

In the single-lift Verdol machine, the right-hand leg of the hook unit is slightly inclined to the vertical as shown in Fig. 3.24*b*. The selected hooks raise the ends from their centre-shed position, and the bottom board lowers the unselected hooks to the bottom-shed position. Since all the hooks must be returned to the mid-position after each pick, a centre-closed shed is produced.

Both legs of the hook unit of a double-lift machine have hooks. The hook unit can thus be raised by either griffe according to whether or not there is a hole in the paper pattern. The various positions of the hook unit are illustrated in Fig. 3.25. In Fig. 3.25*A*, the hook unit has been raised, and it should be noted that the supplementary hook H is now locked over the fixed griffe G. This position will be maintained, even between picks (Fig. 3.25*B*), as long as successive lifts are required, so that an open shed is produced. Immediately uncut card displaces the needle-and-hook unit, H will be pushed off G (Fig. 3.25*C*) so that, as the jacquard griffe falls, the hook unit will be lowered to the bottom position. In this position, shown in Fig. 3.25*D*, the hook is again ready to be raised.

3.5.4 Special Jacquard Machines

Reference was made in Section 3.5.1 to various special jacquard machines. Several of these have been developed for the economical production of a particular type of fabric, often in conjunction with a special type of harness. In weaving figured terry loop-pile fabrics, for example, there is commonly one row of pile loops (formed by the pile warp) for every three picks. By using heald shafts to control the ground ends and an inverted-hook jacquard to control the pile ends, two material advantages are obtained:

(*a*) if it is assumed that there is one ground end to each pile end, the widthway repeat is doubled, and, if one card can be made to operate for three picks (i.e., one row of pile), the lengthway repeat is trebled for a given number of cards; in this way, quite large repeats are possible with a small jacquard and relatively few cards; and

(*b*) the cost of designing is greatly reduced because it is not necessary to paint up the ground weave, and only the figure need be painted in a solid wash of colour; the cost of card-cutting is also greatly reduced, because only one-third of the number of cards is required.

Similar examples could be given in connexion with the weaving of damask, leno brocade, tapestry, and many other special constructions. The above example illustrates how economies can be achieved if one is prepared to design a machine and harness for one particular purpose, although it must be borne in mind that the set-up will be no use for any other purpose. This is no disadvantage if one is producing only a certain specialized type of fabric, such as terry towelling. It is a very real disadvantage if one is producing, for example, dress or furnishing fabrics in constructions and styles that tend to change according to the dictates of fashion. In such circumstances, the tendency is to use straightforward, fine-pitch jacquard machines, which provide both the required figuring capacity and the necessary versatility. Fortunately, there are condensed methods of designing that can be used for most types of fabric and help to keep down the cost of designing and card-cutting.

In carpet-weaving, special jacquards have been developed for rather different reasons. In Wilton carpet, for instance, the function of the jacquard is to choose which one of up to five colours of pile yarn shall be selected to form a tuft on a particular pick. In the gripper Axminster loom, the jacquard is required to present one of eight pile colours, but the method of presentation is quite different from that used on the Wilton loom.

The original machine invented by Joseph Marie Jacquard was the prototype for a very wide range of machines, used not only in the weaving section of the textile industry, but also in the lace section. The idea of punched cards is basic to modern methods of accounting and record-keeping.

3.5.5 The Jacquard Harness

The jacquard harness is the system of cords, healds, and lingoes that transmit the movement of the hooks to the individual warp threads. A simple form of harness is represented in Fig. 3.26. The jacquard machine is assumed to have 400 hooks, which are represented at the top of the diagram in eight rows of fifty hooks. At some distance, usually about 1 metre, below the bottoms of the hooks is the horizontal comberboard. This is a piece of hardwood, or more usually sections of hardwood assembled in a frame. The comberboard (or its separate sections) has drilled in it as many small holes as there are ends in the warp. In this case, we suppose there to be four repeats of 400 ends, making 1600 ends in all. The holes are arranged in rows of eight corresponding to the rows of eight hooks in the machine. From the first hook, cords are led through the first, 401st, 801st, and 1201st holes (which are at the back left-hand corner of each repeat in the comberboard), since this hook controls the first end in each repeat. From the eighth hook at the front of the machine, cords are led through the eighth, 408th, 808th, and 1208th holes at the front of the comberboard. Similarly, each of the 400 hooks carries four cords, which are led through appropriate holes in the comberboard. The lower end of each cord is attached to a heald eye or mail. This may be a small elliptical-shaped piece of brass with three holes, as shown in the inset in Fig. 3.26. In this case, the mail is attached by its upper hole to the harness cord and by its lower hole to a lingo, which is a thin metal rod of mass about 25 g. The function of the lingo is to overcome the vertical component of warp tension and so keep its end down when it is not being lifted by the machine. More recently, individual elastic rubbers have been introduced for greater effectiveness and a quicker return of the raised hook, so that jacquard shedding is possible at higher speeds. The warp thread is drawn through the larger central hole in the mail. Instead of a brass mail, one may use a twisted-wire heald or a flat-steel heald.

Problems can arise when jacquards are used on wide looms because the lifting of a hook will give the necessary lift to the heald directly beneath it, but there will be a loss of lift at the sides of the jacquard. In the arrangement illustrated in Fig. 3.27, the vertical distance from the bottom end of the central

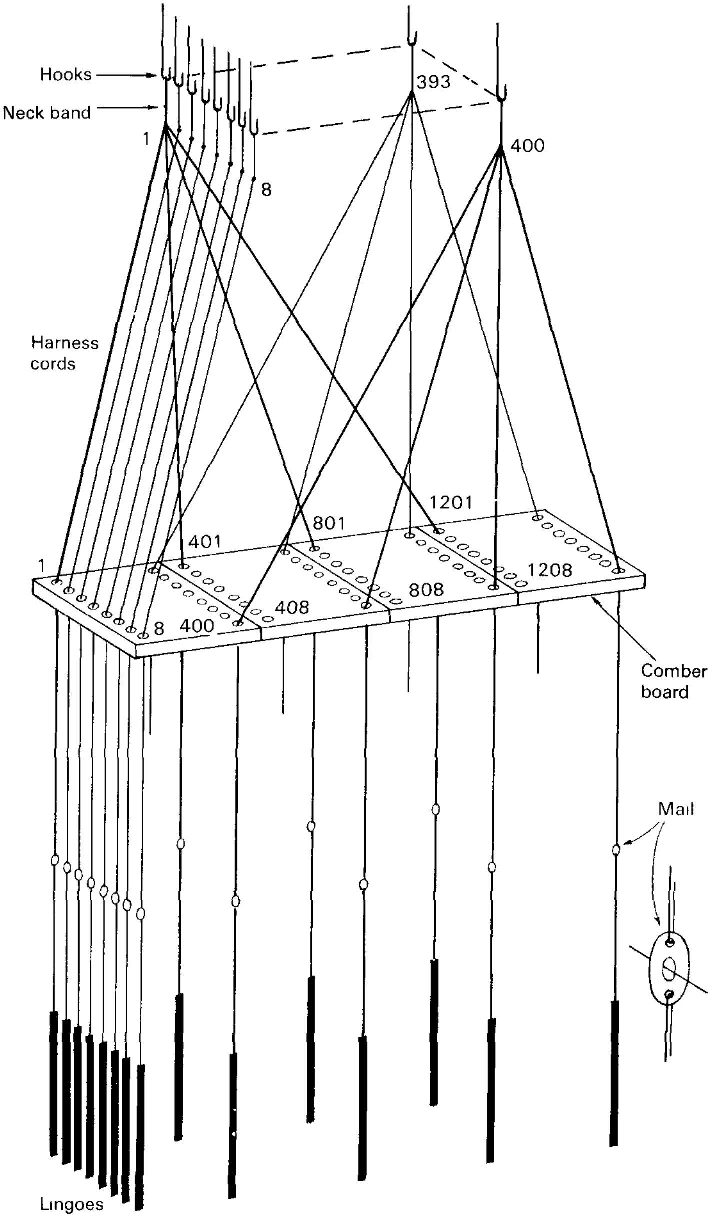


Fig. 3.26 A jacquard harness

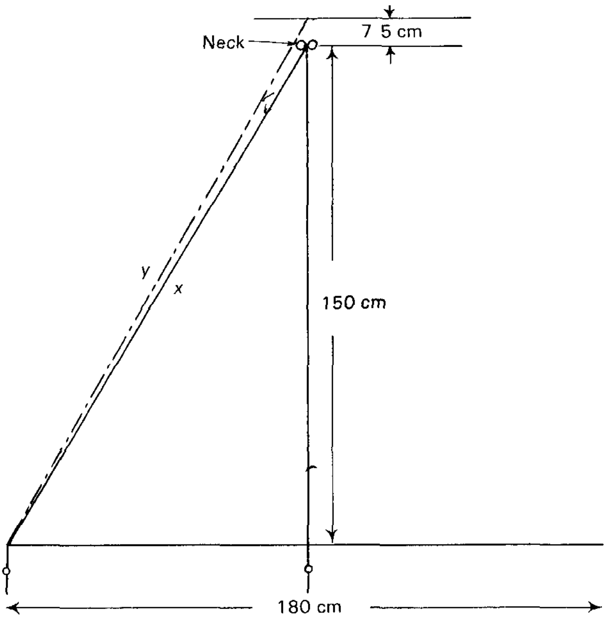


Fig. 3.27 Jacquard lift

hook in the jacquard to the comberboard is 150 cm, which means that the length of the harness cord controlling an end operated by the same hook, but 90 cm from the centre of the machine, would be:

$$x = \sqrt{150^2 + 90^2} - 174\ 9 \text{ cm.}$$

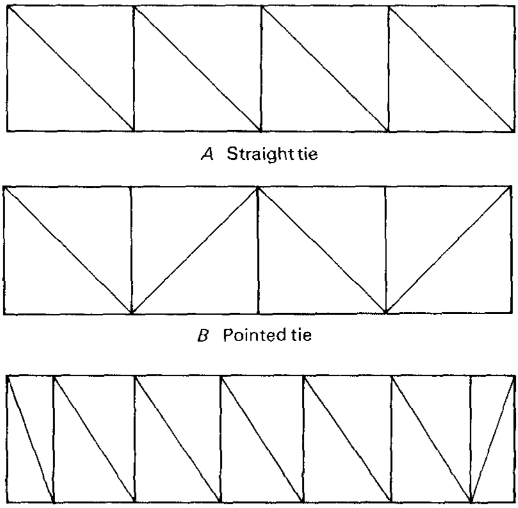
Now, if the shed depth is 7 5 cm, the hook will have to be raised by this amount to give the required lift at the centre of the machine, but the amount of lift at the side will be equal to $y - x$ and:

$$y - \sqrt{157\ 5^2 + 90^2} = 181\ 4 \text{ cm.}$$

Thus the lift at the side would be 181 4 − 174 9 = 6 5 cm or a loss of $13\frac{1}{3}$% in shed depth. The introduction of a heck at the lower end of the neck cord can substantially reduce and virtually eliminate this difference. More recently, the technique of having uniform lengths of harness cords running in flexible plastics tubes has completely eliminated the problem, but the technique has yet to achieve wide acceptance.

3.5.6 Harness Ties

There are many ways of arranging the harness of a jacquard machine. The arrangement shown in Fig. 3.26 is a straight repeating tie, with four repeats in the width. This can be represented as in Fig. 3.28*A*. The number of repeats can be anything from one upwards. Fig. 3.28*B* illustrates a pointed tie in which the



A Straight tie

B Pointed tie

C Border tie

Fig. 3.28 Harness ties

second repeat is a mirror image of the first. This tends to give the impression of doubling the size of the repeat, but the double repeat must be symmetrical about its centre line. Fig. 3.28*C* illustrates how some of the hooks (say, one-third of the total) can be used to weave borders that are mirror images of each other. The rest of the hooks can then be used in several straight repeats, as in Fig. 3.28*A*, or in several reversed repeats, as in Fig. 3.28*B*. There are other possibilities, such as dividing the harness longitudinally to control two warps and to facilitate designing.

In Fig. 3.26, the long rows of needles are parallel to the comberboard and at right angles to the warp. The cards will be at the front or at the back, or at both the front and back of the loom. This produces a neat arrangement of the harness cords, but the cards tend to obstruct light in the working areas at the front and back of the loom. This is known as the *Norwich tie.* If the jacquard is turned through a right angle, the cards will then fall at the sides of the loom, where they will not obstruct light, but the harness will now be twisted. This is called the *London tie.* Both types of tie have their advocates, but, with modern fluorescent lighting, it is easy to secure adequate illumination with the more straightforward Norwich tie, which therefore tends to be preferred.

3.5.7 Casting-out

In tappet and dobby shedding, the number of ends/cm can be varied at will. With jacquard shedding, the maximum number of ends/cm is determined by the harness. There is no possibility of weaving a fabric with more ends/cm than the harness has. It is possible, however, to weave fabrics with fewer ends/cm by casting-out. This consists in leaving selected hooks and the harness cords they control idle. Suppose, for instance, that the harness illustrated in Fig. 3.27 is designed for weaving a fabric with 32 ends/cm. The size of the repeat will be $400/32 = 12 \cdot 5$ cm. If we wish to weave a fabric with only 24 ends/cm, we may do this by leaving every fourth row of mails empty. We shall then have 300 ends in a repeat of 12·5 cm. In casting-out, it is desirable to omit whole rows of hooks if possible in order to simplify designing and card-cutting. There are, however, occasions when it would be more convenient to leave out every fourth end instead of every fourth row. There can be no hard and fast rules about casting-out.

Another device enables the size of the repeat to be doubled. Considering again the harness shown in Fig. 3.27, we may wish to reproduce a design with a widthway repeat of 25 cm, which is twice the normal repeat of the harness. This can be done if the cloth has only half as many ends/cm as the harness. In this case, it would have only 16 ends/cm, which is quite reasonable for a folkweave type of curtain fabric, for example. To achieve this, we may use, for instance, odd-numbered rows of the first and third repeats to produce the first half of the design and even-numbered rows of the second and fourth repeats to produce the second half of the design. Alternatively, but less conveniently, we might use odd-numbered ends of the first and third repeats for the first part of the design and even-numbered ends of the second and fourth repeats for the second part of the design. In either case, we have a design on double the widthway repeat with half the number of ends/cm. Clearly, this idea can be extended to produce designs with twice the widthway repeat and fewer than half the number of ends/cm for which the harness was designed, or with three times the widthway repeat but only one-third of the ends/cm for which the harness was designed.

3.5.8 Card-cutting Instructions

Reference has been made in Section 3.3.2 and Fig. 3.12 to the correct orientation of a directional pattern in dobby weaving. Similar considerations apply in jacquard weaving when one is passing instructions to the card-cutter. In Fig. 3.29, the cylinder is assumed to be at the back of the loom, so we see the front of the card being presented to the cylinder in Fig. 3.29*A*. Fig. 3.29*B* shows the first two cards in the set and that part of the design from which they have been partly cut. Fig. 3.29*C* indicates that the warp has been drawn in from left to right and from back to front—a straight S draft.

The first pick is at the bottom of the design, and the card-cutter has read from left to right. The solidly filled squares in the design correspond to the holes that have been cut in the cards. This will produce the correct orientation of the pattern in the fabric. The same result can be obtained by rotating the design

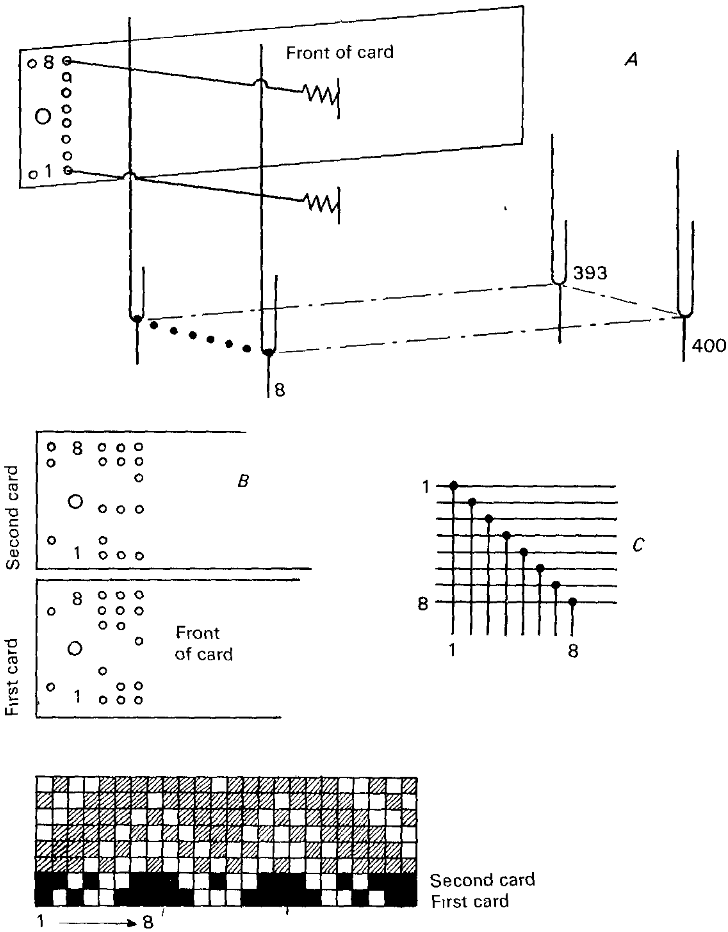


Fig. 3.29 Card-cutting instructions

through 180° so that the square on the design corresponding to the first end and first pick will be in the top-right-hand corner. The card-cutter will then read from right to left and from top to bottom. This is the customary and more convenient method. When this method is used, the design is rotated before the picks are numbered, so that the numbers will be the right way up for the card-cutter to read.

The conditions outlined above would normally be agreed and established once and for all in any jacquard-weaving shed, and the card-cutter would know how to proceed without further instructions.

Reference has been made in Section 3.5.4 to special jacquard and harness arrangements designed for the economical production of a specific structure. Some of these arrangements may require quite complicated card-cutting instructions. The card-cutter would be given standing instructions to cover each such case, and, in sending a design for cutting, it would be necessary only to inform the card-cutter to which structure the design related.

CHAPTER 4

Picking and Checking Mechanisms

4.1 Types of Picking Mechanism

4.1.1 Introduction

Some consideration has been given in Chapter 2 to the general principles of picking and checking (Sections 2.4 and 2.5) and to their relationship with shedding and beating-up (Sections 2.6, 2.7, and 2.8). It was tacitly assumed that the picking mechanism would utilize a cam to impart the required motion to the picking stick and to the picker, and one such mechanism (the cone-underpick) was illustrated by a simplified line diagram in Fig. 2.6. This happens to be the most important picking mechanism for conventional automatic looms, but there are others that use different arrangements to transmit the motion of the cam follower to the picker. Some of these will be described in Section 4.3. These mechanisms are closely related because they all use a cam to generate the picking force precisely when it is required.

A different approach would be to generate mechanical energy before it is required and store it until the time arrives to release it at the start of the pick. This can be done, for example, by compressing or extending a spring, suitably connected via a picking stick to a picker, and by providing some means for releasing the energy stored in the spring at the right time. This system is used in some very wide looms for weaving papermakers' felts and some types of carpet. Instead of a spring, one may use the energy stored in a twisted steel rod. This is the method used in the Sulzer weaving machine for propelling the small weft grippers or projectiles that replace the shuttle.

We have, then, two main types of mechanism for imparting motion to a shuttle or weft carrier. These are:

(*a*) mechanisms that generate energy when it is required by using a cam to displace the picker against the inertial resistance offered by the shuttle, and
(*b*) mechanisms that generate and store energy in a spring or torsion rod and release it suddenly when required.

In type (*a*), the shuttle speed varies with the loom speed. If, for example, the loom is turned over slowly by hand, the shuttle will probably not leave the shuttle-box. If the loom is turned over sharply by hand, the shuttle may just about reach the opposite side of the loom. In type (*b*), because the energy is released suddenly, the shuttle speed is independent of the loom speed, or very nearly so.

As we have seen in Chapter 2.4, in the first group of mechanisms (*a*), the force acting on the shuttle tends to increase to a maximum about halfway through the period of acceleration and then to decrease to zero at the instant at which the shuttle loses contact with the picker. The 'catapult' analogy fits the second half of the period. In the second type of mechanism (*b*), the force exerted by the spring or torsion rod is greatest at the instant of release and then decreases steadily to zero if the spring or torsion rod is allowed to return to its unstressed state at the end of the pick. In this case, the 'catapult' analogy fits the whole of the period of acceleration of the shuttle or projectile. We know from Section 2.4.2 that uniform acceleration is unattainable with mechanisms of type (*a*). It is even less attainable with those of type (*b*) since the force producing acceleration decreases steadily from the moment of release. We can obtain more uniform acceleration with mechanisms of type (*b*) by releasing only part of the stored energy, but this produces other problems, as will be explained in the next section.

4.1.2 Principles of the Spring Pick

The line diagram in Fig. 4.1 represents a picking mechanism in which the energy for accelerating the shuttle is supplied by releasing a stretched spring.

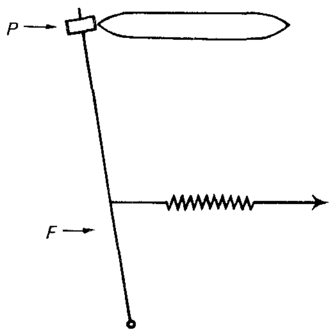


Fig. 4.1 Spring-picking

The method of extending and releasing the spring is not shown, since it is immaterial in the present argument. Suppose, for example, that the mass of the shuttle and its contents is 0·5 kg and that we wish to generate a shuttle speed of 13 m/s. Neglecting the mass of the picking stick, picker, and other associated parts that have to be accelerated by the spring, we may calculate the force required to accelerate the shuttle as follows.

Assume that the spring is attached to the picking stick at a point two-fifths of the distance from its fulcrum to the picker and that the spring contracts 50 mm while the shuttle is being accelerated. The stroke of the picker will therefore be 125 mm. Considering the shuttle, we have:

$$v^2 = 2as,$$

where v is the final shuttle velocity in m/s, a is the average acceleration in m/s^2, and s is the distance in m over which acceleration occurs. Hence:

$$a = \frac{v^2}{2s} = \frac{13 \times 13 \times 1000}{2 \times 125} = 676 \text{ m/s}^2.$$

If, in Fig. 4.1, P is the average force acting on the picker and F is the average tension in the spring, then:

$$P = ma = 0{\cdot}5 \times 676 = 338 \text{ N},$$

and

$$F = 338 \times \frac{5}{2} = 845 \text{ N}.$$

We also have:

$$s = \tfrac{1}{2}at^2,$$

where t is the time in seconds over which acceleration occurs, so that:

$$t^2 = \frac{2s}{a} = \frac{2 \times 125}{1000 \times 676},$$

from which $t = 0{\cdot}019$ s.

Suppose now that the spring is to be fully relaxed at the end of the period of acceleration. It must therefore be stretched 50 mm, and the force to stretch it will increase linearly, according to Hooke's Law, as shown in Fig. 4.2. The

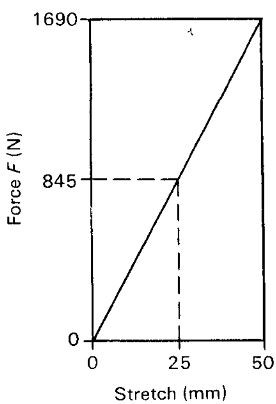


Fig. 4.2 A force–extension diagram for a spring that is not pre-stressed

average force it exerts must be 845 N, and this will be when it is stretched half the total amount, namely, 25 mm. The stiffness of the spring must therefore be:

$$\frac{\dfrac{845}{25}}{1000} \times \frac{1}{1000} = 33{\cdot}8 \text{ kN/m},$$

and the force required to stretch it 50 mm will be:

$$33{\cdot}8 \times \frac{50}{1000} = 1{\cdot}69 \text{ kN}.$$

This implies a very strong spring.

Now assume that the spring is to be pre-stretched 50 mm and stretched a further 50 mm before picking takes place. In this case, its residual tension at the

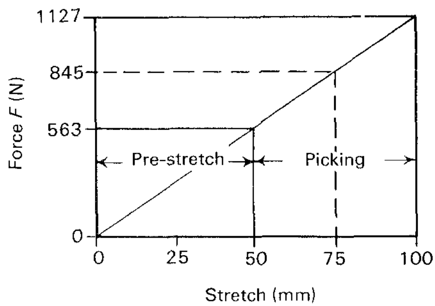


Fig. 4.3 A force–extension diagram for a pre-stressed spring

end of the period of acceleration will be half its initial tension. Its average tension during picking must be 845 N as before, and this must now occur when it is stretched 75 mm as shown in Fig. 4.3. Its required stiffness is

$$\frac{\dfrac{845}{75}}{1000} = 11{\cdot}27 \text{ kN/m}.$$

By pre-stressing, its stiffness has been reduced to one-third and its maximum tension to two-thirds of the values for a spring that is not pre-stressed.

This seems to be a strong argument for using a pre-stressed spring, but, if we do so, we are faced, in the example quoted, with the need to ensure that the picker loses contact with the shuttle after the spring has contracted 50 mm and the shuttle has moved 125 mm. If contact is not broken at this point, the picker will continue to accelerate the shuttle, and the final shuttle speed will be higher than we intend. A means must therefore be found for retarding the picker and eventually bringing it to rest. Retardation must begin as soon as it has travelled 125 mm, when the tension in the spring will be 67 6 kN/m and the picker will still be accelerating. In a conventional picking mechanism of type (*a*), the picking stick is arrested by some kind of buffer (see Fig. 4.12), but in this case the picking stick has already begun to slow down, and the problem is comparatively simple. There is no force corresponding to the residual spring tension to contend with, and all that is required is to dissipate the momentum of the picking stick. This momentum has also to be dissipated in the case of the pre-stressed spring. It is clear that it will not be easy to devise a method of arresting the picking stick. A possible solution might be to incorporate a device in the picking mechanism to release the residual spring tension at the end of the pick. In the example given, this would mean moving the free end of the spring 50 mm to the left at the end of the period of acceleration and then 100 mm to the right, ready for the next pick from that side of the loom. In practice, a compromise has to be sought between the advantages and disadvantages of pre-stressing.

Picking usually accounts for about half the power required to drive a loom, and the idea of using some of the power that is wastefully dissipated when the shuttle is checked by friction to help accelerate it on the next pick is attractive. A possible way of doing this is to connect the opposite ends of a coil spring to the two picking sticks, as indicated in Fig. 4.4, in which the shuttle is about to

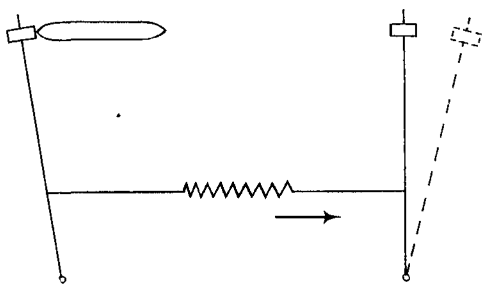


Fig. 4.4 A power-saving device

be picked from the left-hand side, the spring having been stretched as indicated by the arrow. The right-hand picking stick has remained in contact with its buffer at the end of the previous pick. When the shuttle arrives at the right-hand side, it will return the picking stick to the position it occupies at the start of a pick, as shown by dotted lines. While this is happening, the spring will be stretched and will thus store some energy to contribute to the next pick. The spring-loaded picking stick is being used as an energy-storing device. Attempts have been made from time to time to use this idea in a more sophisticated way, but without much success.

4.1.3 Principles of Torsion-picking

This type of picking has been developed successfully in the Sulzer weaving machine. A slightly simplified diagram of the Sulzer picking mechanism, as

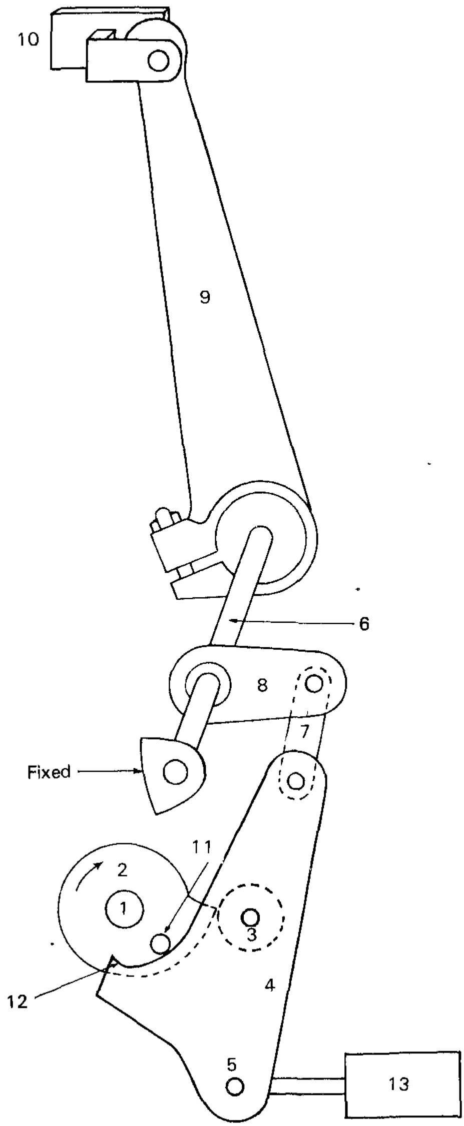


Fig. 4.5 The Sulzer torsion-picking mechanism

seen from the front of the loom, is given in Fig. 4.5. The picking cam shaft 1, which rotates once every pick, has a picking cam 2, which is similar in shape to that used in a cone-overpick but rotates in the opposite direction (clockwise in the diagram). In the position shown in the diagram, it has gradually displaced the roller 3 and turned the toggle lever 4, to which it is attached, in a clockwise direction about its fulcrum 5. In doing so, it has twisted the free end of the torsion rod 6 through the link 7 and the torsion lever 8, which is secured to the torsion rod 6 near its free end. The picking lever 9 is attached to the free end of the torsion rod 6 and carries the picker 10. At this stage, the amount of twist at the free end of the torsion rod is about 30°, and the picking lever has rotated through a similar angle. The mechanism is now fully stressed.

As the cam 2 continues to turn, the roller 3 will lose contact with the nose of the cam, but this will not release the torsion because the axis of the link 7, when produced, passes to the right of the axis of the fulcrum 5. Some time after the cam nose has cleared the roller 3, however, another roller 11, which is fixed to the cam 2, turns the toggle lever 4 slightly anti-clockwise by depressing its curved contour 12. As soon as this movement brings the axis of the link 7, when produced, slightly to the left of the axis of the fulcrum 5 of the toggle lever, the system collapses, and the torsion is practically instantaneously released. The picker 10 then accelerates the projectile (not shown) over a distance of about 65 mm. The picker and the rest of the lever system are then brought to rest over the next 40 mm of picker movement by the oil brake 13, which acts on the hydraulic principle. The speed of the projectile as it leaves the picker is normally about 24 m/s.

Ormerod[11] gives velocity, acceleration, and retardation curves for the picker. Acceleration is far from uniform and reaches a peak value of 6630 m/s^2 when the projectile has travelled nearly 15 mm. The peak deceleration has a value of 9920 m/s^2 and occurs about 12·5 mm before the picker comes to rest. If acceleration were uniform, it would have a value of $v^2/2s$, where v is 24 m/s and s is 65 mm. This gives a value of about 4430 m/s^2, which is about two-thirds of the actual peak acceleration. This is a good deal better than the average conventional picking mechanism achieves. If deceleration of the picker were uniform, it would have a value of about 7400 m/s^2, which is about three-quarters of the actual peak retardation. This represents a reasonably close approach to uniform retardation. It relates, of course, to the picker and not to the projectile when it reaches the other side of loom.

The peak value for acceleration quoted above is much higher than that in conventional shuttle looms because the projectile velocity is nearly twice that of a shuttle in a conventional loom, and it is accelerated over about one-third of the distance. The peak forces acting on the projectile, however, are of the same order as those acting on the shuttle of a conventional loom because the projectile has a mass of less than 40 g. This gives a peak accelerating force of about 262 5 N and a peak retarding force of about 397 N.

The form of the velocity and acceleration curves for the picker shoe are shown in the top part of Fig. 4.6. Acceleration of the shoe and projectile occupies 0·007 s. The lower diagram shows how the velocity of the projectile varies during its acceleration, flight across the loom, and retardation to its rest

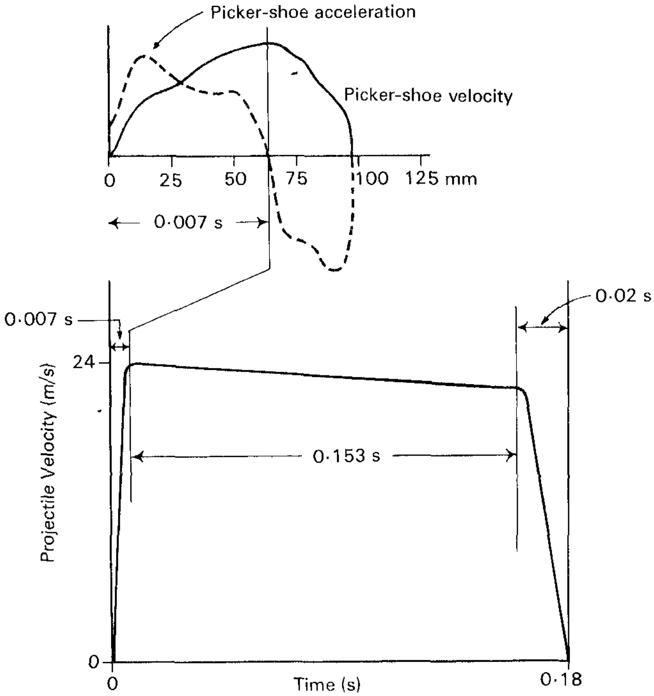


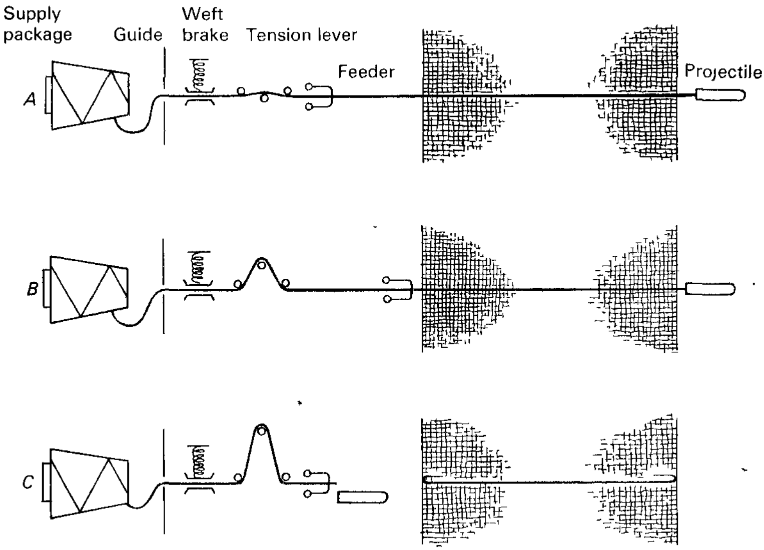
Fig. 4.6 Velocity and acceleration curves for the Sulzer machine

position at the opposite side of the loom. Retardation occurs over a substantially longer time than acceleration and is therefore relatively gentle. The loss in speed of the projectile as it passes through the shed is similar to that of the shuttle in a conventional loom.

4.1.4 Weft Control in the Multiple-gripper Weaving Machine

Reference has already been made (Section 1.3) to the functions of the shuttle-furnishing and the shuttle-eye tensioner associated with the shuttle of a conventional loom. These functions of controlling the weft as it is withdrawn from the pirn and of ensuring that it is satisfactorily tensioned are important. In this case, the weft is fed in a continuous length from the pirn, and there are relatively few problems associated with control of the weft between picks, although weft holders and cutters have to be provided on pirn-changing mechanisms (Sections 9.5.3, and 9.5.4) to control the loose ends of weft produced when the pirn is changed.

Weft control on a multiple-gripper loom such as the Sulzer is more com-



plicated because each pick of weft is repositioned and cut after insertion. Three devices—the weft brake, the weft-tension compensator, and the weft feeder—are used to provide this control. The following description, taken in conjunction with Fig. 4.7, explains how they work.

Weft tension during picking is controlled by the weft brake and the tension compensator. At the start of a pick, the weft brake is released by raising the brake shoe so that the weft passes freely between the shoe and the brake band underneath it. When the gripper is about halfway across the loom, the brake shoe is partly lowered so as to provide sufficient tension to prevent overfeeding of the weft. It is fully lowered when the gripper has completed its traverse.

The arm of the tension compensator is lowered at the start of the pick, and in this position it deflects the weft yarn only sufficiently to remove any snarls (diagram *A*). When the gripper comes to rest, the tension arm is partly raised to take up any slack in the pick and to take up the yarn released when the gripper is returned to a predetermined distance from the right-hand selvedge (diagram *B*). The jaws of the weft feeder, which have been open until now, close on the yarn, which is then cut. The weft feeder moves to the left in preparation for the next pick, and the tension arm moves to its highest position to take up the slack (diagram *C*).

At this stage, the lifter raises the next gripper to the picking position with its jaws held open by the gripper–opener to receive the end of the weft projecting from the weft feeder. When the opener retracts, the weft is gripped by the jaws of the gripper, and a feeder–opener moves in to open the feeder jaws so that weft insertion can take place. The weft must always be trapped by at least one set of jaws at the left-hand side of the loom.

4.2 Conventional Picking Mechanisms

4.2.1 Introduction

The main functions of any picking mechanism are:

(*a*) to deliver the shuttle along the correct flight path; and
(*b*) to project the shuttle at a predetermined velocity.

The first of these functions is achieved by ensuring that the shuttle-box back and the reed are in perfect alignment, as also are the shuttle-box base and the raceboard. If either of these alignments is not true, the shuttle will become chipped, and the resulting damage could cause end-breaks. Furthermore, if the reed or raceboard stands out from the box back or base, there is the possibility that the shuttle may be made to 'fly out', i.e., pass through the upper warp sheet and over the weaving area. Apart from the damage that this may do to the warp or loom, there is always the possibility of injury to the operatives. Correct shuttle flight is also dependent on satisfactory delivery of the shuttle by the picker. Since the sley is moving backwards at the time of shuttle projection, it will also cause the shuttle to move in a downwards arc before rising again to enter the opposite box. Special care is thus needed to ensure that the picker guides the shuttle into the shed and does not in any way encourage 'flying out'.

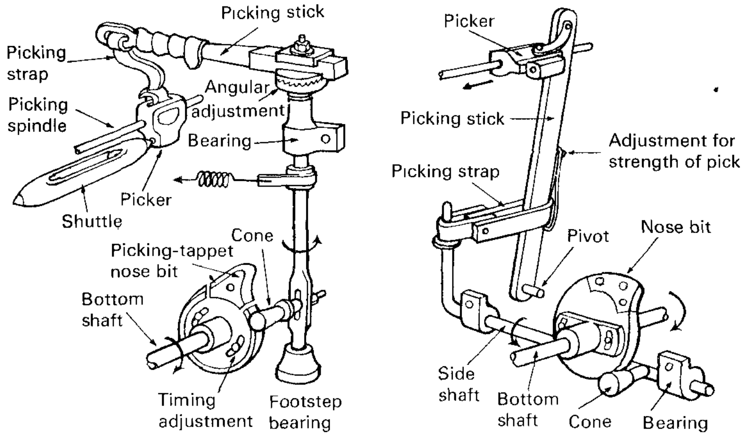
Consideration is now given in rather more detail to the important types of picking mechanism (and their points of adjustment) that are used for propelling weft-carrying shuttles at the desired velocity in conventional automatic and non-automatic looms. Many less important variants, which introduce no new principles, will be omitted or mentioned only briefly.

4.2.2 The Cone-overpick Mechanism

Until recently, the cone-overpick mechanism was used more than any other. It was used on practically all non-automatic cotton looms, whether single- or multiple-shuttle, fast- or loose-reed. It was also used on many flax and jute looms and on some woollen and worsted looms. It was not used on looms specifically designed for weaving continuous-filament yarns, although very large quantities of man-made-fibre continuous-filament yarns used to be woven on Lancashire looms equipped with cone-overpick.

A slightly simplified perspective view of the mechanism is shown in Fig. 4.8*a*. It is robust, easy to adjust and maintain, and comparatively gentle in action, so that shuttles and pickers, as well as parts of the mechanism itself, have a long life. These facts account for its former popularity, which began to wane with the growth of automatic weaving, for which it is not suitable, chiefly because the picking stick and spindle at one side of the loom occupy the position in which the battery or magazine of an automatic loom must be placed.

This spindle is situated over the shuttle-box and is essential to guide the shuttle along the correct path. It is normally set slightly up and slightly towards the front of the loom at its inner end. The back end of the shuttle will thus receive a similar lift at the end of the stroke so that its leading end will receive correct delivery down and into the shed.



(*a*) Cone-overpick mechanism (*b*) Cone-underpick mechanism

Fig. 4.8 Cone-picking mechanisms

The mechanism is also unsuitable for standardized settings, which are being increasingly used for automatic looms, because of the tendency of the flexible leather picking strap to stretch slowly in use and to vary with regard to its elastic properties as atmospheric conditions change. The mechanism incorporates several adjustments for the timing and strength of the pick. The timing of the pick (i.e., the crankshaft position at which the picker begins to move) can be altered by turning the picking tappet on its boss. The common method of altering the strength of the pick (i.e., the shuttle speed) is by altering the length of the picking strap. Shortening it removes some of the slack and increases the distance moved by the picker and hence increases the shuttle speed, but it also advances the timing slightly. Lowering the picking cone in its slot in the picking shaft also increases the shuttle speed by altering the leverage of the system and thus increasing both the extent of the picker's movement and its velocity. In this case, however, an increase in shuttle speed is accompanied by a slight retardation of the timing because the tappet nose will strike the cone slightly later. This adjustment is seldom used. It is also possible to alter the position of the picking stick at the point of attachment to the picking shaft. This has a somewhat unpredictable effect on shuttle speed and timing. Not only has the mechanism a surfeit of points of adjustment, but none of them alters the shuttle speed without also altering the timing to some extent. Standardized settings are virtually impossible in these circumstances.

Large changes in shuttle speed, such as might be required for looms of different width or for weaving widely different types of fabric, can be made by changing the nosebit of the picking tappet for one with a different profile. In this connexion, it is interesting to note that certain makers of high-speed automatic looms offer a range of interchangeable picking cams for their underpick looms for weaving different widths and types of fabric.

4.2.3 The Cone-underpick Mechanism

This is the mechanism that has, in one form or another, to a large extent replaced the cone-overpick, since it is suitable for nearly all types of loom and fabric. An example of this mechanism in a form suitable for a high-speed automatic loom was illustrated in Fig. 2.6 in Section 2.4. The more detailed perspective view given in Fig. 4.8*b* was taken from a woollen and worsted automatic loom. In this version, the picker slides on a spindle, which is positioned in front of the shuttle-box. This allows the mechanism to be used on a bobbin-changing automatic loom and on a multiple-drop-box loom.

The timing of the pick can be varied by turning the picking tappet on its boss, as in the cone-overpick. The normal adjustment for shuttle speed is by raising or lowering the picking strap (commonly called the *lug strap*), which is robust and virtually inextensible. Lowering it increases the distance moved by the picker and thus increases the speed of the shuttle. Here, in contrast to the cone-overpick, we have a mechanism with two simple, independent adjustments, which, because of the absence of parts that will stretch, retains its adjustment for long periods. This is a much more promising proposition for standardized settings.

In cone-underpick looms for single-shuttle high-speed automatic weaving, the picker is usually fixed to the picking stick (as in Fig. 2.6 and also in Figures 4.9 and 4.10), and the picking spindle is eliminated. It is then necessary to vary the height of the fulcrum at the bottom of the picking stick during the pick in order that the picker may follow a straight, horizontal path. (The picker may actually be allowed to rise very slightly towards the end of its movement so as to lift the trailing end of the shuttle, which helps to keep its leading end in contact with the raceboard.) The two common ways of achieving this substantially horizontal movement of the picker are:

(*a*) the parallel pick (Fig. 4.9), and
(*b*) the link pick (Fig. 4.10).

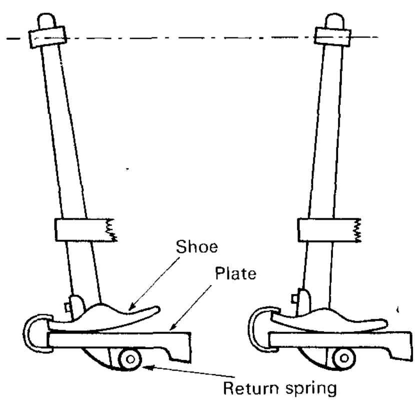


Fig. 4.9 The parallel pick

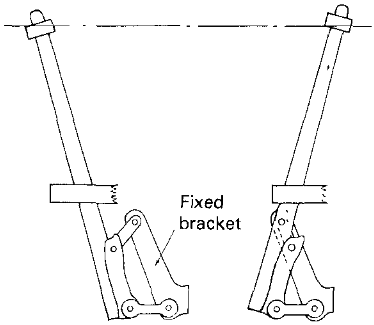


Fig. 4.10 The link pick

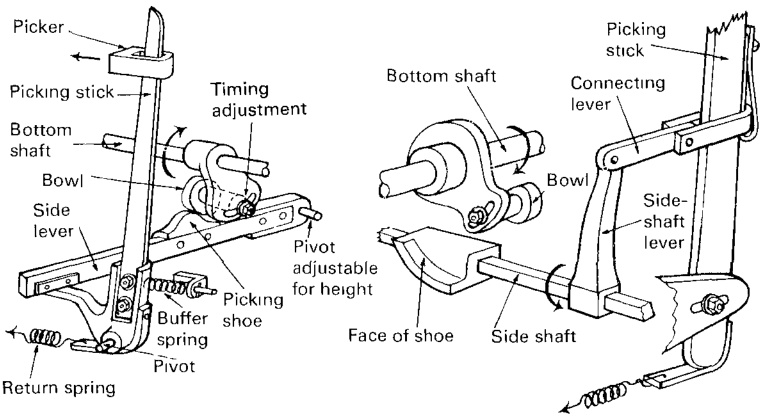
In the parallel-pick method, a curved shoe fixed to the bottom of the picking stick rides on a horizontal plate fixed to the sley sword, the shoe being kept in contact with the plate partly by gravity and partly by spring pressure. It is also necessary to prevent the shoe from sliding along or across the plate. For this purpose, a semi-rigid loop connects the shoe and plate, and the picking stick usually extends through a slot in the plate. A spring attached to this extension may be used to return the picking stick at the end of the pick. These features can be seen in Fig. 4.9.

The parallel-pick method has been successfully used in many automatic looms, but, as loom speeds have increased steadily over the years, it has become increasingly difficult to maintain contact between the shoe and the plate throughout the pick, and, if contact is lost, the desired motion of the picker is not obtained. This is sure to lead to unsatisfactory shuttle flight, excessive wear of the shuttle, and the possibility that the shuttle may fly out. It was to overcome this problem that the link-pick method was developed. In this device, the picking stick is bolted to a casting, which carries short, freely pivoted arms at its upper and lower ends. The opposite ends of these arms are fulcrummed to a bracket fixed to the sley sword. The four pivots may be thought of as being at the corners of an irregular quadrilateral. By suitably designing the shape and size of the quadrilateral in relation to the angular movement and length of the picking stick, the desired motion of the picker can be obtained with sufficient accuracy over the required distance, which is usually between 160 and 200 mm. The link pick gives positive control of the movement of the picker and is satisfactory at speeds considerably in excess of those commonly used.

More recently, shuttles having flat instead of tapered ends have been introduced. They reduce the possibility of a variable shuttle position in the box. This can result in faulty bobbin transfers or shuttle fly-outs as a result of incorrect picker-settings, which may create downward pressure on the tapered end of the shuttle (by the hollow in the picker) at the end of the stroke. Improved shuttle flight is thus claimed because it is independent of the picker condition and, more particularly, the picker-settings. Furthermore, the heavier shuttles that are necessary in modern high-speed looms are more likely to retain contact with the reed and raceboard during flight. Correct entry of the shuttle into the warp shed is ensured by the back wall of the shuttle, which is extended and tapered to a point. This taper extends behind the picker when the shuttle is in the shuttle-box.

4.2.4 Other Conventional Picking Mechanisms

Bowl-and-shoe-underpick mechanisms have been used in various forms, especially in multiple-shuttle looms, such as those used for weaving cotton and rayon jacquard furnishing fabrics or fancy woollen and worsted fabrics. The two main types are the side-lever and the side-shaft mechanisms. A simple example of the first is illustrated in Fig. 4.11*a*, in which all the parts shown are



(*a*) Side-lever underpick (*b*) Side-shaft underpick

Fig. 4.11 Side underpicking

outside of the loom end-frame. In this example, the picker is loose on the picking stick and is guided between the bottom of the shuttle-box and a lip along the top front edge of the box. Other arrangements, such as, for example, the one shown in Fig. 4.11*b*, are possible. The timing of the pick is altered by moving the bowl along a slot in the casting that carries it. The principal means of varying the shuttle speed is by altering the height of the fulcrum of the side lever. Raising it will increase the picker movement and hence the shuttle speed. This is not a very convenient form of adjustment, because a small movement may cause a relatively large change in shuttle speed. The side-lever mechanism tends to give a harsher and noisier pick than one that uses a tappet and cone because of the unavoidable impact when the bowl strikes the picking shoe. Mechanisms similar to the one shown in Fig. 4.11*a* are used on some wide, heavy looms for weaving industrial fabrics. They are not suitable for use at high speeds.

The example of a side-shaft bowl-and-shoe mechanism in Fig. 4.11*b* is used on some woollen and worsted looms. Its action is similar to that of the cone-underpick (Fig. 4.8*b*), including the method of varying the shuttle speed by raising or lowering the lug strap. The timing adjustment is similar to that of the side-shaft mechanism (Fig. 4.11*a*). The picker is loosely attached to the picking stick as in Fig. 4.8*b* or 4.11*a*. Its action tends to be somewhat smoother than that of the side-lever mechanism.

In the forms illustrated, both mechanisms in Fig. 11 are suitable for single-shuttle looms or for multiple-shuttle looms with a box unit at one side of the loom only, which pick alternately from opposite sides and which cannot therefore insert a single pick or an odd number of picks from one shuttle. Both, however, are readily adaptable for pick-at-will looms, which can insert odd picks. To do this, it is necessary for the loom to be able to pick from the same side on two or more successive picks and, conversely, not to pick from the other side on two or more consecutive picks. If, for example, in Fig. 4.11*a*, we provide two diametrically opposite picking bowls and mount the picking shoe so that it can be swivelled into or out of the path of the bowls, the mechanism is then suitable for a pick-at-will loom. In practice, the two picking shoes at opposite sides of the loom are connected together and operated from a spare jack in the dobby or a spare hook in the jacquard. This makes certain that the loom cannot pick simultaneously from both sides of the loom. Similarly, the mechanism shown in Fig. 4.11*b* can be modified by providing extra picking bowls and arranging their mounting so that the casting on which they are carried can be slid along the bottom shaft to allow the appropriate bowl either to engage or miss the face of the shoe. In this case, the two sliding members would be linked together and operated from the dobby or jacquard. The picking sequence must be very carefully controlled in co-ordination with the positioning of the shuttle-boxes. To ensure that there is an empty shuttle box to receive the projected shuttle, a special pick-at-will control system is used (see Section 8.2.4).

We have now considered the four main types of conventional picking mechanism, namely:

(*A*) tappet-and-cone (or cam-and-cone) mechanisms, of which there are two types:
- (i) the cone-overpick, and
- (ii) the cone-underpick; and

(*B*) bowl-and-shoe mechanisms, of which there are also two types:
- (iii) the side-lever mechanism, and
- (iv) the side-shaft mechanism.

Variants of these basic types are encountered. For example, it is possible to modify the side-lever mechanism by replacing the picking shoe with a bowl or roller and the bowl with a picking cam or tappet similar to that of the cone-underpick. The object of this modification would be to obtain a smoother pick. It is not necessary in this treatment to consider other possible variants because the general principles are sufficiently represented by the four basic mechanisms listed above.

4.3 Power Required for Picking

The energy usefully expended in accelerating the shuttle is equal to its kinetic energy when it leaves the picker, so that:

$$\text{energy/pick} = \frac{mv^2}{2}\ \text{J},$$

where m is the mass of the shuttle in kg, and
v is the maximum velocity in m/s,

and:

$$\text{power} = 1\ \text{W} = 1\ \text{J/s}.$$

Thus, if P is the loom speed in picks/min, then:

$$\text{power for picking} = \frac{mv^2}{2} \times \frac{P}{60} \times \frac{1}{1000}\ \text{kW}.$$

This suggests that the power consumed increases at the same rate as the loom speed, but this is not so because the shuttle speed also tends to increase with the loom speed for a given width of loom, so that the power consumed increases more rapidly than the loom speed. Because of this, and in order to include the effect of the loom width, it is more instructive to proceed as follows.

Let:

R be the useful reedspace in cm (i.e., the width of the warp in the reed when the reedspace is being fully utilized),
L be the length of the shuttle in cm, excluding its tapered ends, and
θ be the number of degrees of crankshaft rotation occupied by the passage of the shuttle through the warp shed.

Then the time for the passage of the shuttle is:

$$t = \frac{\theta}{360} \times \frac{60}{P} = \frac{\theta}{6P}\ \text{s},$$

and the distance moved by the shuttle is:

$$d = \frac{R + L}{100}\ \text{m}.$$

If v is now the average speed of the shuttle during its passage through the shed, then:

$$v = \frac{R + L}{100} \times \frac{6P}{\theta} = \frac{6P(R + L) \times 10^{-2}}{\theta}\ \text{m/s} \qquad (4.1)$$

We then have:

$$\text{work done/pick} = \frac{mv^2}{2} = \frac{36mP^2(R+L)^2 \times 10^{-4}}{2\theta^2}$$

$$= \frac{18mP^2(R+L)^2 \times 10^{-4}}{\theta^2}\ \text{J}. \tag{4.2}$$

Hence:

$$\text{power for picking} = \frac{mv^2}{2} \times \frac{P}{60} \times \frac{1}{1000} = \frac{18mP^3\,(R+L)^2 \times 10^{-4}}{\theta^2 \times 6 \times 10^4}\text{kW},$$

$$= \frac{3mP^3(R+L)^2 \times 10^{-8}}{\theta^2}\ \text{kW}. \tag{4.3}$$

This is an approximation because v in Equation (4.1) is the average shuttle speed. Its initial speed would be slightly greater. If, for example, its initial speed were $2\frac{1}{2}$% greater than its average speed, v in Equation (4.1) should be increased by a factor of 1·025, and the work done per pick and the power for picking should be increased by a factor of $(1{\cdot}025)^2$ or approximately 1 05. This is a somewhat academic point, which does not invalidate the general conclusions that may be drawn from Equation (4.3). From this equation, we see that, for a given width of loom, the power usefully employed in picking is proportional to the cube of the loom speed. The power consumed per pick, and therefore the cost of power usefully employed for picking per unit of production, increase only as the square of the loom speed, as indicated by Equation (4.2). It is interesting to consider two widely different numerical examples.

Example 4.1

Consider a cotton or rayon loom of 110-cm reedspace, running at 216 picks/min, with a shuttle of mass 450 g and length 28 cm. Assume the passage of the shuttle to occupy 135°. We then have:

$$\text{work done/pick} = \frac{18 \times 0{\cdot}45 \times 216^2 \times 138^2 \times 10^{-4}}{135^2} = 39\ 51\ \text{J},$$

and

$$\text{power for picking} = \frac{39{\cdot}51 \times 216}{60} \times \frac{1}{1000} = 0{\cdot}142\ \text{kW}.$$

Example 4.2

Consider a heavy blanket loom with a reedspace of 533 cm, running at 65 picks/min, with a shuttle 47 cm long and of mass 900 g. Since the loom has a high sley-eccentricity ratio (0·54), assume the passage of the shuttle to occupy 150°. We then have:

$$\text{work done/pick} = \frac{18 \times 0{\cdot}9 \times 65^2 \times 580^2 \times 10^{-4}}{150^2} = 100{\cdot}2\ \text{J},$$

and:

$$\text{power for picking} = \frac{100{\cdot}2 \times 65}{60} \times \frac{1}{1000} = 0{\cdot}109 \text{ kW}.$$

In order to check that the assumptions made in the above calculations do not imply unrealistic shuttle speeds, we may use Equation (4.1) to calculate the shuttle speeds that would be necessary to satisfy the assumptions. This gives speeds of 13·25 and 15·08 m/s for Examples 4.1 and 4.2, respectively. These are not unreasonable.

The wide, slow-running loom consumes about two and a half times as much power per pick in accelerating the shuttle as does the narrower, fast-running loom, but in doing so it inserts nearly five times the length of weft. The rate at which it consumes power is actually less than that of the narrower loom.

We have been considering only power that is usefully employed in accelerating the shuttle. A good deal is consumed wastefully in accelerating others parts of the mechanism and in overcoming friction in the mechanism and between the shuttle and the shuttle-box. The work done per pick in overcoming friction should be substantially independent of the loom speed, but the work done per pick in accelerating other parts of the mechanism will increase as the square of the loom speed. The net effect will be that the total power consumed per pick by the whole picking mechanism will increase somewhat more slowly than the square of the loom speed. Even so, it is clear that power costs per pick must increase quite rapidly with the loom speed.

Power-consumption tests made on the modern automatic loom to which Example 4.1 above relates showed that the picking mechanism was actually consuming 0·41 kW, which, at 216 picks/min, is equivalent to 110 J/pick. In this case, the percentage of this energy usefully employed in accelerating the shuttle is $39{\cdot}51/110 \times 100 = 36\%$. This value is probably fairly typical. It is interesting to note that, according to Ormerod[11], the efficiency of energy utilization of the Sulzer torsion pick is only about 15%. This may be because the mass of the projectile (37 65 g) is very much smaller in relation to the mass of the moving parts of the mechanism.

4.4 Shuttle-checking Devices

4.4.1 Conventional Shuttle Looms

The principles of shuttle-checking and the functions of the swell as a checking device and also as a means of operating the warp-protector motion have been discussed in Section 2.5. We saw that it was necessary that the shuttle should retain an appreciable fraction of its initial speed when it strikes the picker. Its speed must then be reduced to zero over a short distance, which must not vary appreciably. The inertia of the picker and picking stick is insufficient to produce the necessary retardation, and some additional means of absorbing the shuttle's residual energy must be provided. This can be done by imposing a restraint of some kind on the movement of the picker or the picking stick or both.

If, for example, the picker is fixed to the picking stick, as in modern cone-underpick mechanisms, the restraining device can take the form of a check strap acting on the picking stick. The restraint will be partly inertial and partly frictional, and either effect may dominate according to the design of the device. We may also provide a buffer to cushion the final impact of the picker against the box end. Something as simple as a piece of folded leather may suffice, or a more elaborate buffer may be designed to take over the functions of the check strap. For this purpose, spring-loaded, hydraulic, and pneumatic buffers are used.

In the Lancashire cone-overpick loom, the check strap extended the full width of the loom along the front of the sley underneath the warp. Extensions at each end of it were threaded onto the picker spindles outside the pickers. Its length was adjusted so that, when the shuttle entered one box and carried the picker hard up against the box end, the check strap drew the opposite picker about 5 cm from its box end. Each time the shuttle was checked, the check strap was dragged laterally about 5 cm. The effect was mainly frictional. The final impact was cushioned by a piece of folded leather between the picker and the box end.

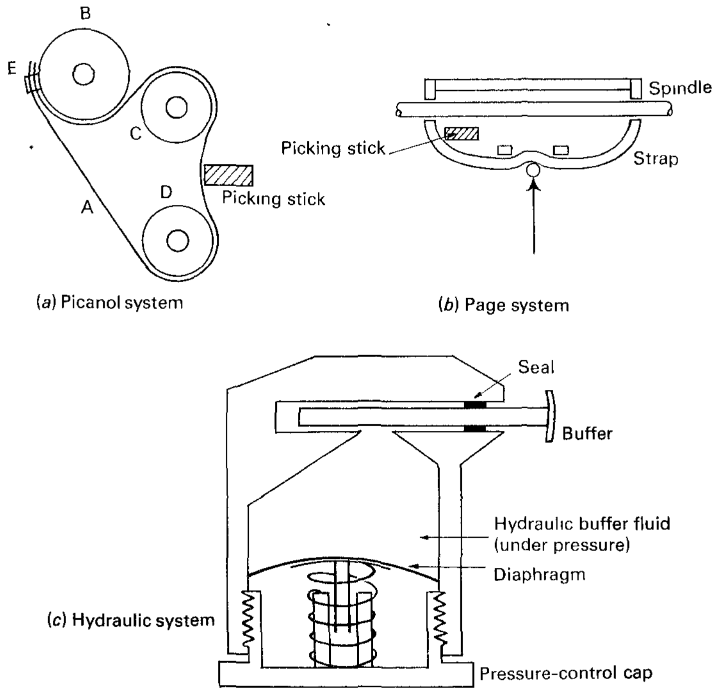


Fig. 4.12 Shuttle-checking devices

A similar arrangement, but with the check strap passing round the picking sticks and connected to the underside of the raceboard, has frequently been used for automatic looms intended to run at not more than about 200 picks/min. For higher speeds than this, it is usual to fit separate checking devices at each side of the loom. Several loom makers use devices similar to the one shown in a simplified plan view in Fig. 4.12*A*, which is taken from the Picanol President loom. A coil spring (not shown) tends to rotate the drum B clockwise and thus tension the leather strap A, which passes over two guide rollers, C and D. These parts are carried in a bracket fixed to the sley. During checking, the picking stick impinges on the strap, which offers a resistance to displacement depending mainly on its stiffness and on the spring tension, which is adjustable.

The essential features of the Page check, which is supplied as an accessory for fitting onto different makes of loom, are shown in Fig. 4.12*B* in plan view. The picking stick, which is shown in section in the diagram, is inside a stiff loop, which is of such a length that it is moved to and fro a short distance along its spindle by the picking stick each time checking occurs. It moves against an adjustable frictional resistance applied at the front, as indicated by the arrow in the diagram. Both the devices illustrated in Fig. 4.12 are effective at the highest speeds currently being used.

Hydraulic and pneumatic devices in which a plunger is displaced by the picking stick are also used. Leader, Zama, and Wildt Mellor Bromley produce such buffers in which the resistance of the plunger to displacement is obtained by pumping air through a valve in a casting that is partly filled with oil (Fig. 4.12*C*). When the plunger is pushed inwards, it displaces some oil against the pressure of the entrapped air. The resistance it encounters is due partly to the viscosity of the oil and partly to the air-pressure.

When the shuttle leaves the picker, the picker and picking stick are moving at their maximum velocity, and, although no longer under the influence of the picking cam, their momentum would carry them further than is convenient if they were allowed to do so. In order to limit their movement, it is usual to allow the picking stick to impinge on some kind of buffer fixed to the loom frame. Some simple arangement incorporating a leather strap in tension or formed into a stiff loop is normally sufficient. Occasionally, something more elaborate may be used, as in the Sulzer loom, which uses an oil-filled buffer.

4.4.2 Multiple-gripper Weaving Machine

In the Sulzer gripper weaving machine, the receiving unit is responsible for more than merely stopping the projectile. Checking is achieved by a long flat brake under the gripper and two smaller individually spring-loaded brakes operating from above.

The extent of the previous working life and the surface condition of the grippers in use on any one loom are likely to vary considerably, so that each gripper will tend to come to rest at a different position in the brake area. The rest position for the gripper travelling the shortest distance into the receiving unit must be far enough advanced to allow a sufficient trail of weft for the tuck-in selvedge and also for ejection to take place. The fastest gripper, however, should not be allowed to strike the returner, because frequent repetition of such an occurence would be likely to cause damage to the leading end of the gripper, and this in turn might cause damage to the guides or an increase in end-breaks.

The arrival of each gripper is checked by a detector, which is lowered onto the gripper at a point between the two upper brakes. After the initial check, the detector is raised and then lowered again 17° later to ensure that the gripper has not rebounded from its rest position.

After checking, the gripper is pushed back to a predetermined distance from the selvedge. The weft is then trapped by the selvedge gripper and the opener lowered to release the weft from the gripper jaws. The upper brakes are raised, and the gripper is ejected into the guide. There are normally three grippers in the guide, and the bottom one is expelled onto the return conveyor chain at the appropriate time in the pick cycle. This time is adjustable and is determined by the position of the receiving unit because the gripper must not be expelled onto one of the blocks in the chain. These blocks are fixed at regular intervals of 254 mm in the chain, and they push the expelled grippers to the picking side of the loom.

It will be realized that one of the major virtues of this system is that the efficiency of the picking mechanism is no longer dependent on a variable rest position of the weft-insertion media.

CHAPTER 5

Rapier, Jet, and Continuous Methods of Weft Insertion

5.1 Comparative Factors

5.1.1 Introduction

In the main, the alternative methods of weft insertion are generally compared for weft velocity, rate of weft insertion, range of weft-patterning scope, and type of selvedge construction.

5.1.2 Weft Velocity

Picking mechanisms of the type discussed in Chapter 4 have two inherent disadvantages, which prevent them from achieving higher rates of weft insertion. In the first instance, they insert weft during only a fraction of the pick cycle, and, secondly, the rate of weft insertion is limited by the mechanical problems and the extent of the dynamic forces involved in picking and checking. These problems have in many instances been responsible for the extensive efforts that have been put into developing alternative methods of weft insertion.

Fig. 5.1 illustrates weft-velocity curves for looms with different methods of weft insertion. It is obvious that, with the exception of the circular and multiphase looms, weft is inserted during only a fraction of the pick cycle, and thus, in order to achieve a high weft-insertion rate, it is necessary to have exceptionally high weft velocities.

The circular loom can achieve a high weft-insertion rate, not only because it is inserting weft continuously throughout the pick cycle, but also because it is possible to have several shuttles inserting weft simultaneously. A similar situation exists on the multiphase loom. Looms using jets or rapiers to insert the weft do not require dynamic forces of anything like the same magnitude as those involved in conventional shuttle and multiple-gripper machines. They do, however, have their own inherent problems, which are highlighted in Fig. 5.1. Fluid jets require a very high initial velocity, which falls away rapidly as the jet traverses the loom, and this limits the machine widths as well as imposing a limit on the ultimate weft-insertion rate. The weft-insertion rate in a rapier loom is very much influenced by the method of weft control. Rapier movement is basically simple harmonic, but in many instances this is modified to achieve a

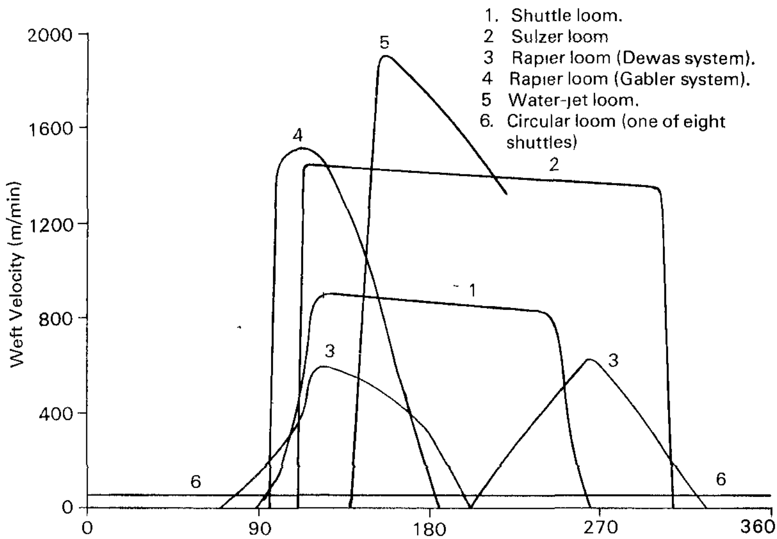


Fig. 5.1 Weft-velocity curves

higher rapier velocity in the early and late parts of the movement and thus a lower maximum velocity halfway through the movement.

Single-rapier looms can insert weft only on alternate rapier traverses (the second traverse, i.e., from left to right in Fig. 5.2*a*), and thus each rapier must be long enough to extend across the full width of the loom. The mass of the reciprocating parts is high because of the rigidity required to ensure that the rapier moves along the desired path. Such looms are therefore very limited in application and popularity, and it is proposed to consider only the velocity of the weft in double-rapier looms. In this case, there are two entirely different systems to consider. The Dewas system (Fig. 5.2*b*) grips the weft outside one of the selvedges and carries it to the centre of the loom, where the yarn is transferred to a gripping point in the other rapier head. This rapier then pulls the weft across the loom to complete the traverse. Weft withdrawal from the supply package thus occurs in two stages. In the alternative (Gabler) system, the weft is not gripped by the rapier head, and thus the possibility of dropped weft on transfer at the rapier heads in the middle of the loom is eliminated. The weft is gripped outside the selvedge, and, as the rapier traverses the loom, a hairpin of weft is formed. At the point of transfer, the yarn extends from the supply package, round the rapier head, and back to the gripping point on the loom frame. When the rapiers start to withdraw, the yarn is released from the gripping point and is straightened out by the second rapier. In this case, the velocity of the weft during the first half of the pick rises rapidly from zero to a

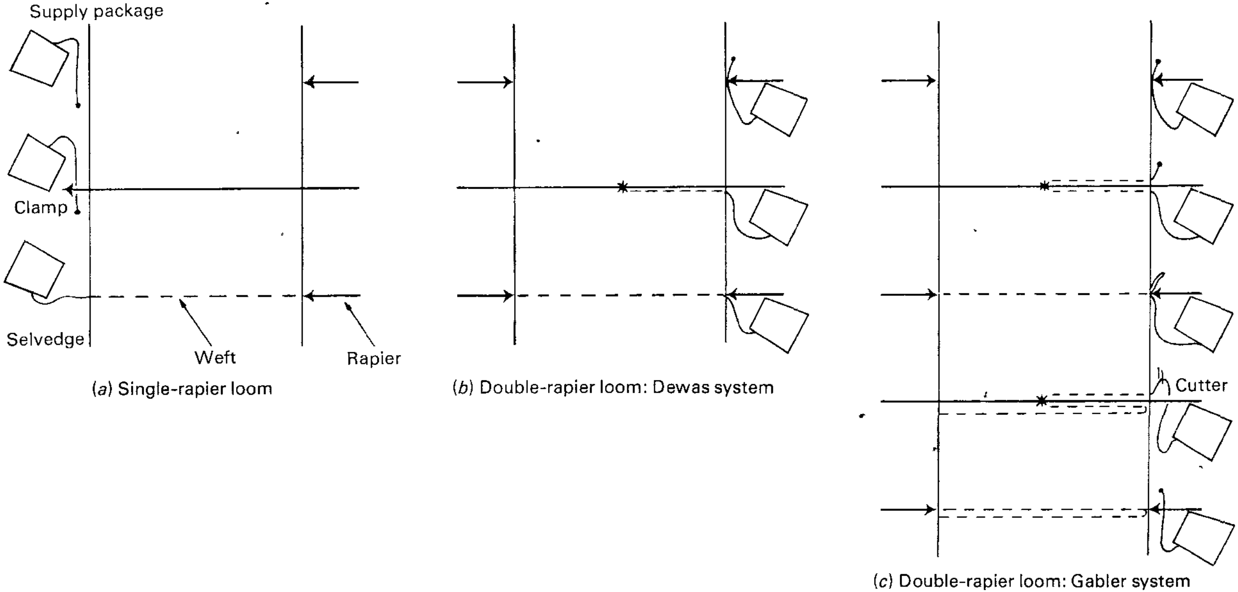


Fig. 5.2 Rapier looms: the sequence of weft insertion

value that is so high that weft breaks are likely to occur more frequently. It is possible to cut the weft after each pick and transfer the cut end of yarn to the gripping point so that each cycle is identical, but an alternative system allows the uncut yarn that extends from the supply package to the selvedge to be transferred to the gripping point. Weft insertion then occurs in exactly the same manner on the second pick, and a traditional selvedge is produced at the supply-package side of the loom.

The possibility of an increased breakage rate arises because the weft is subjected to large variations in velocity (and thus tension) as it is withdrawn from the supply package at intermittent high speeds. Additionally, it is necessary to control the weft by passing it through an effective tension arrangement so that the weaving tension will be more uniform. This again creates a situation in which there is a potential for a weft break. Furthermore, when weft delivery is stopped instantaneously after withdrawal at high speed, extra yarn will be pulled off the supply package. This yarn will tend to hang loosely between the supply package and the tension arrangement and is liable to snarl. Consequently, the snarl may cause a curl on the surface of the fabric if the weft does not break during the insertion of the next pick.

Such a situation may exist with jet looms, some rapier looms, and occasionally the Sulzer weaving machine. It would, of course, be regrettable if the weft-insertion rate had to be limited because of this problem, and for this reason continuous weft withdrawal into a yarn-storage chamber has been developed. The weft is withdrawn from the supply package for the whole (or virtually the whole) of the pick cycle and stored in an accumulator device, from which it can be projected across the loom under virtually no tension. This system also assists the jet, which otherwise would not have the ability to pull the weft intermittently from the supply package without suffering a severe retarding force to its own velocity.

The principles used in weft-storage units, which are sometimes known as *weft accumulators*, are very similar. The yarn is generally withdrawn continuously from the supply package at a rate governed by the measuring drum. To allow for different widths of fabrics, the measuring drum used on a loom may be interchangeable as on the Prince–Nissan water-jet loom (Fig. 5.3*a*). Alternatively, the period of time during which the weft is wound onto the drum may be adjustable. In some cases, the drum stops rotating when it contains sufficient yarn for the next pick (Fig. 5.3*b*). This system is used on the Savi, Safarti, and Sulzer weft accumulators. A second method, which is more popular on looms that have the accumulator designed into the machine, uses a measuring drum that picks up the weft soon after insertion of the previous pick has been completed. The amount of weft wound onto the drum is not sufficient for a full pick length. It does, however, release yarn for the initial insertion under virtually no tension, although the yarn does pass through a brush ring to remove any snarls. In this method, the yarn inserted towards the end of the pick cycle is withdrawn directly from the supply package. It will be realized that weft withdrawal from the supply package in these two methods is not strictly continuous, but, provided that the non-winding time is kept minimal, then the unit will achieve its function.

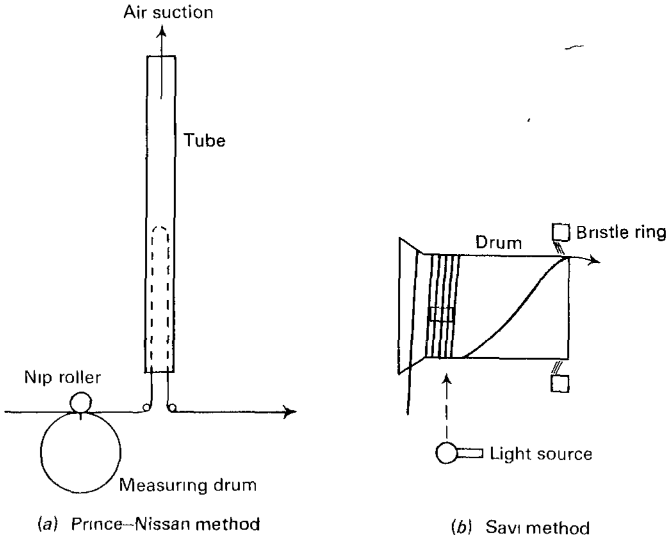


Fig. 5.3 Weft-storage units

When the drum itself acts as a storage unit, there is no problem associated with controlling the weft that has been prepared for the next pick. However, the turns per metre will be affected if the yarn is directed onto the rotating drum by a stationary guide. This is avoided in the Sulzer unit, which uses a stationary drum and a guide revolving in the opposite direction to that in which the yarn will loop off the drum during withdrawal.

If the yarn is not stored on the drum, a second roller is needed to form a nip with the continuously rotating measuring drum to withdraw the yarn from the supply package. Once the yarn has passed between the nip of these rollers, it must be stored before weft insertion takes place. The Prince–Nissan loom (Fig. 5.3*a*) has a storage tube in which air-suction maintains the yarn in a controlled U-loop, which does not tangle. Immediately before insertion, the loop is long, but, on the completion of jetting, it is quite small and ready to be increased again for the next pick.

A final assessment of the various curves in Fig. 5.1 in the light of the foregoing information would infer that circular and multiphase looms are approaching the ideal as far as weft velocity is concerned. They do, however, have their own inherent problems, which demand new concepts in shed formation and beating-up. These are not easily achieved, and a few circular looms weaving speciality fabrics, such as hosepipes and sacks, represent the very limited acceptance of continuous weft insertion to date.

Although various ingenious ideas have received public acclaim for their initial concepts, many of them have never progressed beyond the prototype stages, and it is generally true to say at this stage that only rapier and jet techniques have proved to be successful alternatives to picking as a means of weft insertion.)

5.1.3 Rate of Weft Insertion

Equation (4.1) in Section 4.3 gives:

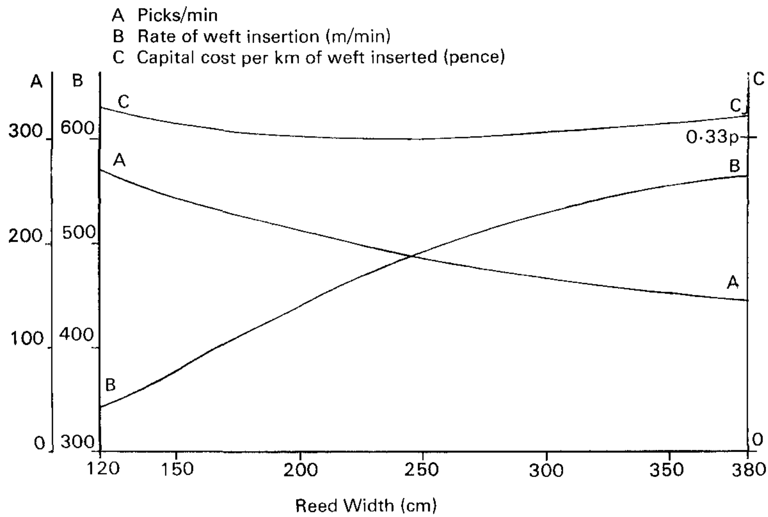
$$v = \frac{6P(R + L)}{\theta} \text{ m/s},$$

and, since R, L, and θ are fixed for any specified loom, then $v \propto P$. It is this fact, coupled with the dynamic problems of designing a suitable weft-insertion mechanism to achieve high rates of weft insertion, that often leads to the question 'At what speed is the loom running?' This question invariably refers to the speed expressed in picks/min and thus overlooks the importance of the width of the loom. If the two examples of looms referred to in Section 4.3 are studied a stage further, it will be seen that the wider loom (B), in addition to having a 13·8% advantage in average shuttle velocity, gains an even greater advantage in the rate of weft inserted per minute of 41·2%, where the rate of weft insertion in m/min is equal to the reed width in metres multiplied by the picks/min.

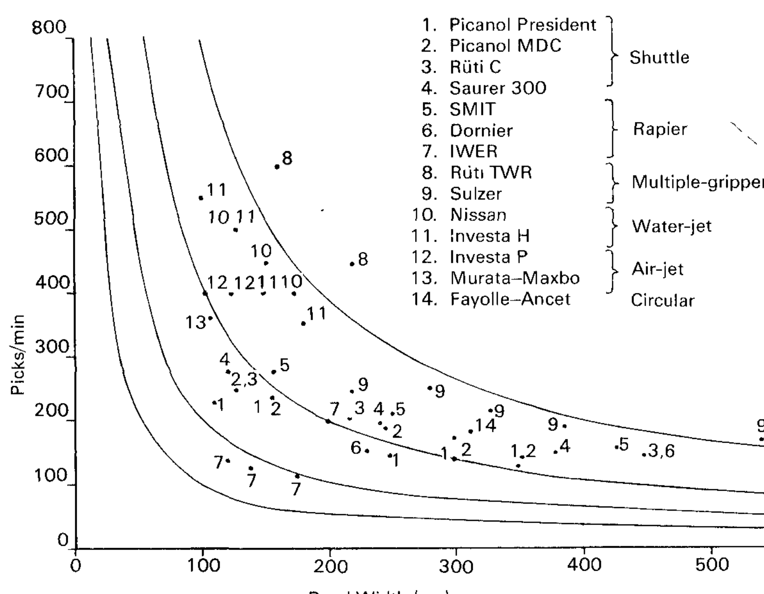
The relationships between the loom speed in picks/min, rate of weft inserted per minute, and reed width has been illustrated by Saurer with reference to the 300 Versaspeed loom, although the graphs produced for this loom will be similar for a wide range of looms. Curve A in Fig. 5.4*a* represents the loom speed plotted against the reed width, and curve B shows the corresponding rates of weft insertion. A further interesting curve, C, relates the capital cost per kilometre of weft inserted. This last curve confirms the findings of Lord and Mohamed[12] that maximum speeds and weft-insertion rates do not necessarily result in minimum weaving costs.

This does not mean that high weft-insertion rates are undesirable, but it does indicate that there is an optimum speed for each type of loom marketed by each manufacturer, which may not only be determined by the mechanical features of the loom but which may also be influenced by the cost of converting yarn to fabric. It is of interest, however, to compare the rates of weft insertion achieved when different methods are used to propel the weft across the loom. Fig. 5.4*b* illustrates a modification to the type of curve used by Dawson[13] and again by Snowden[14].

The reference points near the top of the graph indicate the looms that operate at the highest number of picks/min, and the points that are nearest to the top curve refer to the machines that achieve the higher rates of weft insertion. It will be realized that the Rüti TWR multiphase loom is ahead of the field in both situations, but this loom is not yet commercially available. This leaves the Sulzer multiple-gripper weaving machine and the water-jet looms as the most successful in achieving high rates of weft insertion on narrow and medium-width looms, but only the Sulzer continues to achieve this higher rate on wider looms. Air-jet looms, although running at high speeds, generally only achieve weft-insertion rates comparable with those of a wide range of rapier and



(*a*) Relation between reed width, loom speed, weft-insertion rate, and capital cost



(*b*) Effect of reed width and loom speed on weft-insertion rate

Fig. 5.4 Comparative values of loom speed and reed width conventional shuttle looms. The recent improvements in the design of the conventional shuttle loom, which have mainly been based on modifications to the techniques of warp protection, are illustrated by comparing the values for the Picanol President loom (with fast-reed warp protection) with those for the Picanol MDC and the Rüti C machines. The Picanol MDC uses an electronic method of warp protection and the Rüti C a modified and updated version of the loose-reed warp protector (see Section 7.1).

5.1.4 Scope for Weft-patterning

The acceptance of multiple grippers and jets as alternatives to the conventional shuttle and of circular looms for specific applications having been justified, it now becomes necessary to determine the virtue of rapier looms.

Several different rapier systems are available. Some have gained acceptance because they have been specifically designed to insert two picks simultaneously in the production of double plush (or face-to-face warp-pile fabric). Others have been accepted because they have been successful in weaving difficult types of weft, e.g., glass-fibre yarns. In the main, however, it can be said that any type of rapier loom that has gained any degree of popularity has been capable of the random selection of single picks from a range of different weft-supply packages. This is generally achieved without any loss in loom speed and thus is a major advantage when compared with the conventional shuttle loom. There is also an undisputed price advantage in favour of rapier looms when compared with the multiple-gripper weaving machine.

Single-pick insertion does preclude the production of a selvedge of conventional appearance. This type of selvedge is still demanded by some garment manufacturers, even if it is present at only one of the two selvedges. As previously mentioned, some rapier looms are able to insert two picks in the form of a hairpin in order to produce a conventional selvedge at one side of the fabric, but this technique requires that the two successive picks should be supplied from the same supply package. A wide range of rapier looms offer both options.

5.2 Selvedges

5.2.1 Requirements of Selvedges

The basic function of any selvedge is to lock the outside warp threads of a piece of cloth and so prevent fraying. This is easily achieved if the weft yarn is inserted continuously from one package, such as a pirn on a conventional shuttle loom, because, each time the package-carrying shuttle reverses its direction, a hairpin of weft yarn will form round the last thread in the cloth provided that this end changes its position between picks as in Fig. 5.5*a*.

Other requirements are demanded of a selvedge, however. In the first place, it should be strong enough to withstand the strains of the stenter in the finishing process. This type of machine grips the fabric after a wet-process treatment and pulls the fabric to the desired width during drying. Secondly, the selvedge

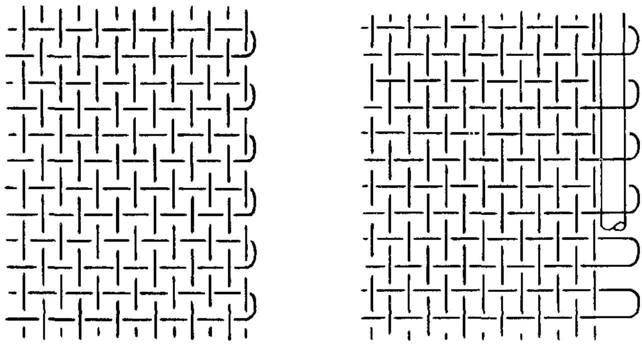


Fig. 5.5 Conventional selvedges

should have a neat and uniform appearance. This is most desirable from an aesthetic point of view. Furthermore, uniformity is also essential to allow the garment manufacturer to line-up one edge of the fabric, layer upon layer, so that many layers can be cut simultaneously and accurately.

5.2.2 Conventional Selvedges

The selvedge to be produced on a conventional shuttle loom, apart from requiring strength for the post-weaving process, must have strength to withstand the rubbing action of the shuttle as it enters and leaves the warp shed during weaving. Strength is also necessary to counteract the pulling of the weft thread as the shuttle traverses the loom, and this function must be performed by the outside ends of the fabric.

Increased strength in the vicinity of the selvedge is usually achieved by:

(*a*) using stronger yarns (often two-fold);
(*b*) increasing the number of ends/cm;
(*c*) modifying the weave, when it is quite common to have two ends weaving as one.

Uniformity of appearance may be a problem if the outside ends in the fabric do not change position every pick. This happens in twill and satin-type weaves. If the appearance of the selvedge is unacceptable when one of these weaves forms the ground structure, then it will be necessary to use some special technique of threading up[15] or to introduce a plain-weave selvedge for a width of up to 1 cm at each side of the fabric. Alternatively, it may be quite satisfactory to make only the outside end in the fabric change position after each pick. An end performing this duty is generally known as a *catch cord* or *catch end*. A catch cord or special plain-weave selvedge is absolutely necessary in weaving warp-rib or matt fabrics, since the ends in the ground fabric do not change position after each pick. An additional catch cord is necessary in weaving extra-weft spot effects. It is necessary because only the ground ends that interweave with the extra weft to form the spot effect will change positions between picks. The outside ends of the outermost spots in the fabric will therefore have to take the strain of the tensioned weft, and, if they do not break under this strain, then it is quite certain that the outside edge of the spot will be deformed. The extra catch cord will take the strain and prevent deformation of the spot. It is situated about 1 cm outside the normal selvedge, and the end needs only to change position when the extra weft is being inserted.

Occasionally, the character of the fabric surface must be absolutely uniform from one side of the cloth to the other. It is thus not possible to vary the construction of the fabric in any way in the selvedge. Under these circumstances, a thread of strong monofilament yarn or a length of flexible wire is used as the catch cord. This thread, which is fastened to a point on the frame at the back of the loom, changes its position after each pick. However, it extends for only 1–2 cm into the fabric. As the cloth is drawn forward, it will slide off the thread to leave a small loop of weft (Fig. 5.5*b*) at the side of the fabric, but there will not be an increase in thickness.

5.2.3 Tuck and Fringe Selvedges

There are several instances in which special selvedge considerations become necessary because the yarn is severed, namely:

(*a*) at the centre of a wide fabric that is cut at the loom so that several narrow fabrics are produced side by side;
(*b*) at both sides of a fabric in which the weft is cut after each pick length; and
(*c*) at one side of the fabric in which the weft is cut after every two picks;

tuck and fringe selvedges are most commonly used under these circumstances.

The tuck selvedge (Fig. 5.6*a*) is probably the more effective in preventing fraying and yet achieving the necessary standards in strength and appearance. A trail of yarn, which extends beyond the last selvedge end, is pulled into the

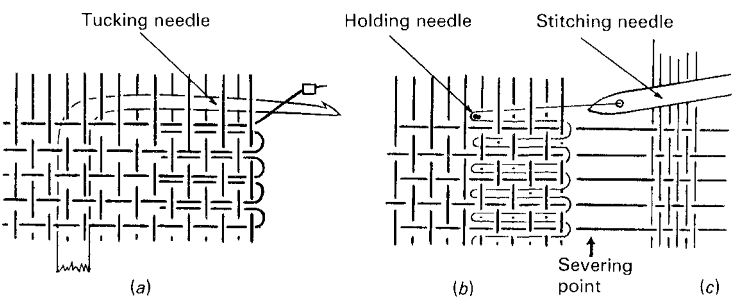


Fig. 5.6 Tuck and stitched (with dummy) selvedges

shed formed for the next pick. Provided that the last end changes position after each pick to catch the loop, a selvedge is produced that is acceptable for a wide range of uses. In many instances, it is necessary to reduce the number of ends/cm in the selvedge to compensate for the increase in weft density.

Probably the most successful tucking mechanism is to be found on the Sulzer machine. The weft, which protrudes from the cloth for a distance of about 1·5 cm, is trapped and held by a selvedge gripper. A double-acting cam unit situated underneath the temple activates a needle, which initially moves towards the back of the loom. Eventually, it makes a sideways sweep to penetrate the bottom warp sheet and pass through the warp shed formed for the next pick until it is in a position in front of the selvedge gripper outside the selvedge as in Fig. 5.6*a*. The selvedge gripper then rises and moves towards the front of the loom to place the thread into the hook that is in the leading tip of the tucking needle. The return movement of the needle next pulls the weft yarn from the selvedge gripper and places it in the warp shed. The needle finally returns to its starting position underneath the temple so that it will be clear of the sley during beating-up.

All the other types of selvedge form a fringe, which can be produced by making a leno structure (Fig. 5.7*a*) with the outside ends: by twisting the outside ends around one another (Fig. 5.8*a*) or by locking the outside ends with an additional fine monofilament thread, which is stitched into the edge of the fabric (Fig. 5.6*b*).

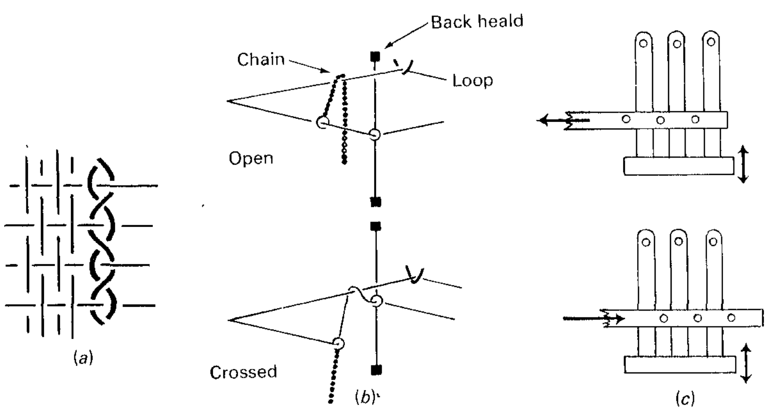


Fig. 5.7 Leno selvedges

The leno method (Fig. 5.7*a*) is most successful in preventing fraying and providing strength, but, as successive picks are beaten-up, the weft tails are made to point up and down alternately by the crossing end so that the fringe does not appear neat.

Many patents have been taken out over the years to cover the formation of a selvedge by the leno technique. One of the simplest types, which requires only a

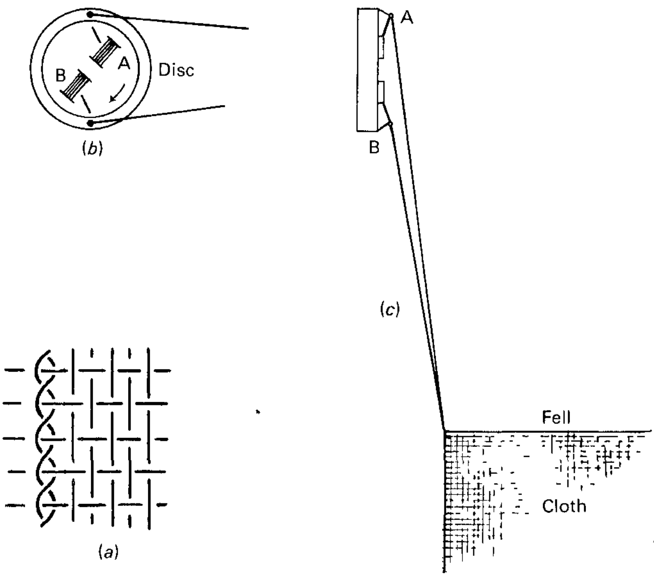


Fig. 5.8 A twist selvedge

chain, is illustrated in Fig. 5.7*b*. The standard end is held in a fixed raised position by a loop, and the crossing end passes through the eye of a heald on the back shaft and also through a ring mounted on the end of the chain. When the back heald is down, the chain is pulled round the standard end, but, on the other pick, when the heald is raised, the chain pulls the crossing end to the opposite side of the standard end.

Another method, which requires an additional mechanism, uses wires with holes in their upper end (Fig. 5.7*c*). These control the crossing ends that are in the raised position when the weft is inserted. Between picks, they fall and then rise again in the same vertical plane by means of cam operation. In the meantime, the standard ends that are permanently in the down position are given a side traverse by a second cam when the crossing ends are in their lowest position. The direction of the side traverse is reversed after each pick.

Twisting the outside ends around each other is done by using a disc that carries two bobbins (Fig. 5.8*b*). As the disc rotates, the threads from the bob bins are made to twist around one another between picks. The protruding trail of weft is always pushed in the same direction by the outside end so that the fringe is given a much neater appearance than that formed by the leno structure. Half a revolution of the disc between picks will give a half-twist selvedge. This is the most common type of twist selvedge, and it is quite satisfactory for a wide range of fabrics. With a fabric that has a high weft cover factor, it is satisfactory to make the half-twist only on alternate picks. However, a full revolution of the disc between picks will give greater locking of the threads if the fabric has a very low weft cover factor.

The formation of the twist selvedge by means of a rotating disc is based on the fact that, when the bobbins on the disc are in the top (A) and bottom (B) positions as in Fig. 5.8*b*, they form a clear shed for the passage of the weft. Between picks, the disc rotates, and the bobbin that is furthest away from the fell of the cloth (A in Fig. 5.8*c*) will supply the end that is nearest to the body of the cloth. When the next pick has been inserted with A in the raised position, the bobbins will continue to rotate, and B will then be nearest to the body of the cloth. The ends must therefore always twist round one another between picks, with the end moving down from the top position forcing the weft to point downwards.

The technique of stitching a fine monofilament thread into the edge of the cloth by one needle (Fig. 5.6*b*) so that the loop can be trapped by a second needle also requires an additional mechanism. Since the selvedge appearance is comparable only with the leno- and twist-type structures, and since the mechanism is relatively complicated, the system has not gained any degree of popularity. The sewing needle must oscillate once every one or two picks depending on the openness of the weft sett. The locking needle oscillates vertically with the same frequency from a second cam to catch and hold the sewing thread in its loop.

The fringe selvedge is basically untidy, and a neater appearance is generally essential. Such an improvement can be achieved if all the protruding weft threads are cut off to the same length and as near to the body of the fabric as possible. To achieve this, the weft threads must be caught and held under control at a predetermined distance from the selvedge so that they can be fed positively to a point at which they will be severed by either cutting or fusing. For this purpose, a group of catcher threads is made to weave in plain order (see Fig. 5.6*c*) at a distance from the selvedge. Such a selvedge, which is generally known as a *dummy selvedge*, may consist of four, six, eight, or nine threads weaving plain. The severing point is between the main selvedge and the dummy selvedge and as near to the main selvedge as possible. The fusing method is simple and very convenient when thermoplastic yarns are being woven, and it also needs fewer catcher threads in the selvedge. When a cutting action is used to sever the weft, more catcher threads become necessary because the blade movement tends to pull the yarn to a greater degree before cutting is finally complete, especially when the blade is not very sharp.

5.2.4 Weft Waste

Although a dummy selvedge is normally required at only one side of the loom, it is natural to expect that there will be more weft waste on a loom needing this type of selvedge than on a loom producing a conventional selvedge.

The unavoidable waste on a conventional shuttle loom is determined by the linear density of the yarn and the reedspace of the fabric being woven. The length of weft per pirn in metres is equal to:

$$\frac{1000}{\text{linear density (tex)}} \times \text{mass of weft on pirn (g)}.$$

Now, if the bunch length is equal to four pick lengths (see Section 9.2.2), then the average waste will be equal to two pick lengths. This is not exactly true in practice, but it is an approximation that is near enough for the purpose of this exercise. The percentage weft waste per pirn is thus:

$$\frac{2 \times \text{reedspace}}{\text{length of weft per pirn}} \times 100,$$

or:

$$2 \times \text{reedspace} \times \frac{\text{linear density (tex)}}{1000 \times \text{mass of weft on pirn (g)}} \times 100.$$

For example, if a pirn of 25-tex yarn contains 20 g of weft, then the length of weft per pirn will be:

$$\frac{1000}{25} \times 20 = 800 \text{ m},$$

and the weft waste in weaving a fabric 1 m wide in the reed will be:

$$\frac{2 \times 1}{800} \times 100 = 0{\cdot}25\%.$$

The weft waste for fabrics 2 and 3 m wide in the reed will be 0 5 and 0 75%, respectively. When a coarser yarn is used, the length of yarn per pirn will decrease if the weight remains constant so that, for a 100-tex yarn, the weft waste on a 1-m-wide fabric will be:

$$(2 \times 1) \times \frac{100}{1000 \times 20} \times 100 = 1\%,$$

and, for fabrics 2 and 3 m wide in the reed, the weft waste will be 2 and 3%, respectively. These values ignore avoidable waste that is created by excessively long bunch lengths, soiled or damaged pirns, and faulty supply packages. It is quite common to find that total weft-waste values on conventional looms are at least double the calculated values, and they may be even trebled or quardupled.

The calculated results show that unavoidable weft waste increases with coarser wefts and wider fabrics.

The introduction of a dummy selvedge creates a situation in which there will be waste on each pick. The amount of waste will be greater on jet looms because the weft is not positively controlled during insertion, but it will be only slightly greater than when the more positive control achieved in rapier insertion is used. If it is accepted that 5–7 cm of weft will be unavoidable waste on each pick, then, if we take an average of 6 cm and consider fabrics 1, 2, and 3 m wide in the reed, the percentage weft waste will be:

$$\frac{6}{100} \times 100 = 6\%,$$

$$\frac{6}{200} \times 100 = 3\%,$$

and

$$\frac{6}{300} \times 100 = 2\%,$$

respectively. Avoidable waste is also possible as a result of damaged supply packages, but it is likely to be less than that on a conventional loom because there will be no pirn waste. Further avoidable waste will be produced if the weft waste per pick is not kept minimal as the result of an incorrectly set weft-measuring or weft-feeding device.

It will be realized that the weft linear density does not affect the amount of weft waste when a dummy selvedge is used, but the percentage of waste will now be less when wider fabrics are produced.

Under ideal circumstances, and if it is accepted that an amount of avoidable weft waste is inevitable, then the amount of weft waste produced in weaving yarn of an average weft linear density (say, 30 tex) will depend on the width of the fabric being woven. If the fabric is wide, there will be very little difference between the amount of weft waste produced on the two systems, but, if a typical fabric having a reedspace of 180–200 cm is being woven, then the unconventional loom will produce 60–100% additional waste, and, on a narrower, say, 100-cm-wide, fabric, the amount of weft waste may be two, three, or even four times as great as that on the conventional loom.

From a weft-waste point of view, the unconventional methods of weft insertion requiring a dummy selvedge become advantageous in weaving coarse wefts on the widest fabrics.

5.3 Rapier-weaving

A general classification of rapier looms was made in Section 1.6. This could be extended over a much broader detailed range, since the number of manufacturers who have attempted to produce a machine of this type is unbelievably high. It is therefore realistic to consider only the principles involved. In many cases, they are similar for many different machines.

The limitations of the single rapier have been stated in Section 5.1. However, they do eliminate the need to transfer the weft at the centre of the loom. This is a virtue that hardly makes their justification acceptable except in weaving wefts that are difficult to control. The single rapier is invariably rigid, but, when two rapiers are used, they may be rigid or flexible. The double-rapier system is accepted as existing when two rapiers enter the shed from the opposite sides. This is distinct from a twin-rapier system, which gains better recognition as two rapiers moving together from the same driving source, as is necessary in face-to-face (i.e., double-plush) weaving.

When a rigid rapier is used, the space required by the loom must be at least double that of the reedspace to allow the rapier to be withdrawn from the shed. This is a very apparent space waster with the single-rapier loom, but it is less obvious with the double rapiers, especially when the reedspace is relatively small. For instance, the space required for a loom having a reedspace of 130 cm may be no greater than that needed for a conventional shuttle loom of similar width. This is because the latter type of loom requires space for the shuttle-boxes outside the normal weaving width. Rapier looms are usually wider than this, and thus flexible rapiers must receive preference if space is at a premium.

Looms that use rigid rapiers eliminate entirely the need to assist the rapier head through the warp shed. This is an undoubted advantage compared with several flexible-rapier looms, which use a rapier head specially shaped on its underside so that it will be directed along the correct flight path by guides in the sley. The guides pass under the cloth fell when the sley approaches its most forward position in a similar manner to that illustrated in Fig. 2.4, which is the method used in the Sulzer machine, in which the problem is identical.

Not all flexible rapiers require guides mounted in the sley. More and more rapier-loom makers are directing the flexible-rapier band through a fixed housing on the loom frame just outside the reed. As the rapier head traverses the loom, it is allowed to rest on the raceboard, which is much reduced in dimensions from that found on a shuttle loom. It is essential that the warp threads should not be damaged by the rapiers and that the rapier heads should meet cleanly for weft transfer. By eliminating guides, there is the advantage of reducing end-breaks due to knot failure, and the weaving of fancy yarns becomes much easier because the obstructions are removed from the warp shed.

The technique of weft transfer is dependent on the principle of operation, but it is essential in both the Gabler and the Dewas systems that one rapier head should enter the other cleanly in order to make the transfer. Any non-sliding transfer contact between the heads is likely to cause a burr on some part of the rapier, and this may then catch the warp or weft during weaving to cause a yarn break.

Fig. 5.9*a* illustrates rapier heads that are suitable for use with the Gabler technique as found on the Versamat loom. The weft is never gripped. It is placed opposite the cut-out A in the right-hand rapier head, and, as the rapier advances towards the centre of the loom, the yarn passes from its clamped position, round the rapier head, to the supply package in the form of a hairpin. When the two rapier heads meet at the centre of the loom, the smaller left-hand rapier head enters the yarn-carrying right-hand head. The thread at A is passed under the spring-loaded cover guide at B, and, as the left-hand rapier is withdrawn, it repositions the weft at C. The yarn can then slide through the left-hand rapier head as it is withdrawn so that the hairpin is straightened out.

In the Dewas system, shown in Fig. 5.9*b*, the yarn is gripped in each rapier head. The gripping unit usually consists of a fixed point against which a spring-

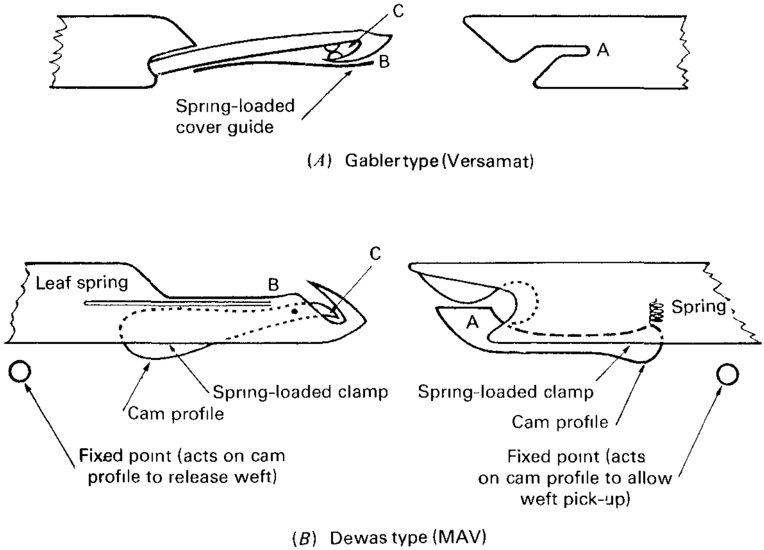


Fig. 5.9 Rapier heads

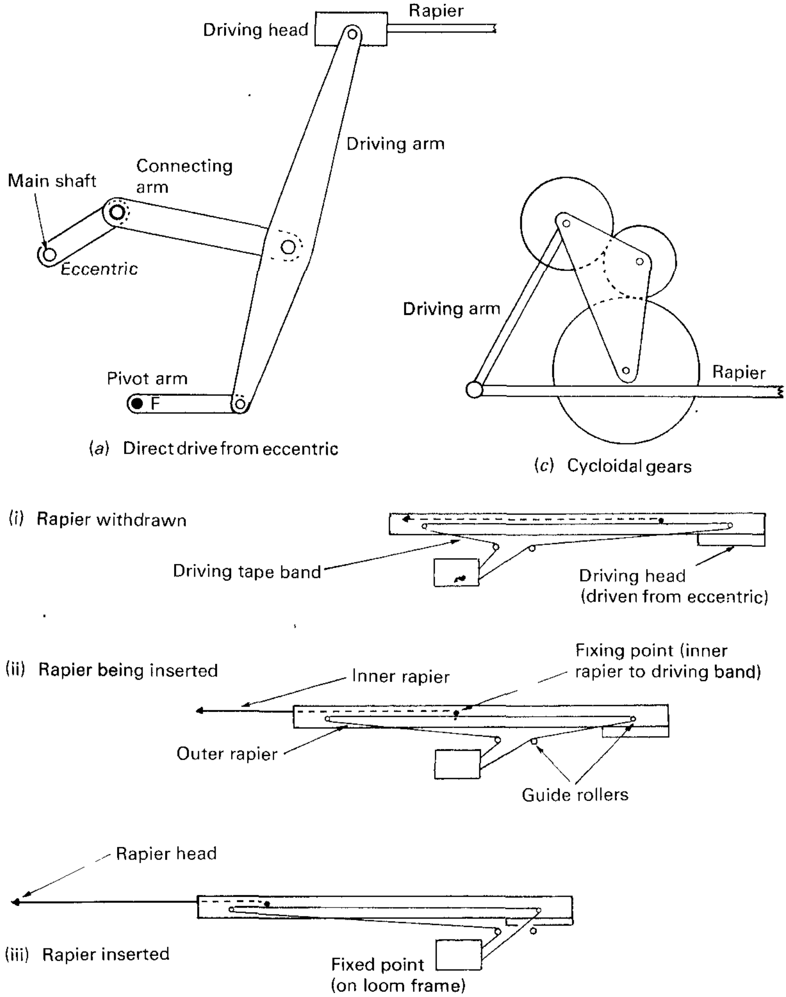
loaded clamp presses to trap the weft. In the MAV loom illustrated, and in many other looms, the spring-loaded clamp has a cam profile that meets a fixed point on the loom or sley-mounting. These points open the clamps when the weft is to be picked up or released outside the selvedge. The right-hand head thus traps the weft at A and pulls it through the shed until the rapiers meet. The thread is then guided round point B, and, as the left-hand head withdraws, the thread is trapped at C and pulled across the loom to complete insertion. In some looms, the spring in the delivery head is opened at the time of transfer. This is done by the specially shaped profile of the left-hand rapier head, and it obviates the need to pull the weft out of the springclamp in the delivery head. Thus the possibility of weft breaks at transfer is minimized. Furthermore, there is no chance of the weft's being pulled out of the receiving spring.

The back walls of the rapiers in a rigid-rapier loom and the back walls of the rapier head in a flexible-rapier loom are tapered to a point at their leading ends to ensure that the rapier correctly enters and passes through the warp shed. Weft insertion may be from the left or right sides of the loom as determined by the loom maker.

In both systems, it is essential that the weft is trapped by the closing shed when the rapier releases the weft. Thus it is necessary for the rapier head to leave a warp shed that is virtually closed (i.e., the rapier-interference factor approaches zero) in order to ensure that the outside ends trap the weft and prevent a tension release that would create a slack pick. The design of the rapier head is therefore critical if an increase in the end-breakage rate is to be avoided.

It is possible to minimize rapier interference around the rapier head that is not delivering the weft by allowing it to leave the shed before the other rapier.

The main principles used in driving rapiers are illustrated in Figures 5.10–13. The direct eccentric arrangement as illustrated in Fig. 5.10*a* is used mainly for rigid rapiers for which the rapier path is usually straight. The mechanism operates vertically in double-rapier machines, but there is in-



(*a*) Direct drive from eccentric

(*c*) Cycloidal gears

(*b*) Telescopic rapier drive

Fig. 5.10 A rigid-rapier drive from an eccentric sufficient space from the floor to the weaving height when a full-width single rapier is used. The eccentric is thus made to operate horizontally under the warp sheet, with the driving arm extending from the back of the loom to the rapier at the front. In all instances, the driving head of the rapier must follow a straight line, and it is necessary for the fulcrum to oscillate on a pivot arm as indicated. Space can be saved on this type of rapier drive if a compound rapier operating on the principle of telescopic expansion is used as illustrated in Fig. 5.10*b*. This system is used on the Versamat loom. The main outer body of the rapier is driven by the eccentric through the connecting and driving arms, but the inner body is fastened to a tape at its outer end. The tape is attached to a fixed point on the loom, and it passes round four pulleys, two of which are mounted in each end of the outer-rapier body. The result is that, when the rapier is out of the shed, the inner rapier is withdrawn inside the outer rapier, but, as the outer rapier is driven towards the centre of the loom, the tape is made to slide round the rollers so that the inner rapier moves in the same direction at an even faster rate.

Two problems exist when movement is originated from an eccentric arrangement. In the first place, the mass of the moving parts is high, and wear at the fulcrum points may occur. This can be reduced by using a system of cycloidal gears similar to that used in the Gusken loom, as illustrated in Fig. 5.10*c*. The rapier is connected directly to the cycloidal-gear driving arm. This arm is attached to the driving gear, which revolves round the main gear, which in turn is driven from the main shaft of the loom. The need to use an intermediate driving arm, a pivot arm, and an oscillating fulcrum is eliminated. The second problem exists because a crank drive will give the rapier a movement approaching simple harmonic. The disadvantage with that type of motion in relation to weft velocity is that the maximum velocity is $\pi/2$ times the average velocity. Furthermore, simple harmonic motion can be considered undesirable because the rapiers only need to pause momentarily at the centre of the loom for weft transfer, but they must be outside the limits of the selvedge for a longer period of time. However, the rapier must move well clear of the sley to allow beat-up and weft selection to take place, and, since it is possible for the rapier to enter a relatively small shed, it is quite feasible for the rapier to enter the shed sooner and leave later than in weaving with a conventional shuttle.

A rapier loom having a reedspace of 180 cm and running at 225 picks/min will not have excessive velocities if the rapier can be made to enter the shed at 60° and leave at 300°. These figures are typical for a range of looms, and the average velocity will be:

$$\text{average } v = \frac{\dfrac{180}{100}}{\dfrac{60}{225} \times \left(\dfrac{300-60}{360}\right)} = \frac{180}{100} \times \frac{225}{60} \times \frac{360}{240} = 10\cdot125 \text{ m/s.}$$

The maximum velocity will thus be $10\cdot125 \times \pi/2 = 15\cdot91$ m/s. This value represents the maximum weft velocity when the Dewas system is used, but, with the Gabler system, it will be doubled, i.e., to $31\cdot82$ m/s.

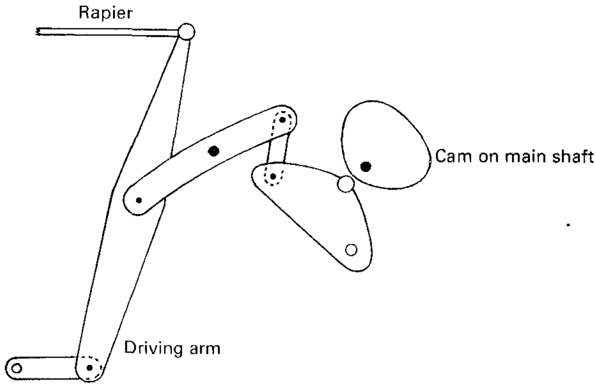


Fig. 5.11 A rigid-rapier drive from a cam

The system embodying the use of a cam instead of an eccentric is illustrated in Fig. 5.11. This system is used in the MAV loom. Since the time of rapier movement is now controlled, the maximum velocity can be kept lower, although the rate of acceleration will be greater. It is, however, possible to allow the rapier to enter later and go into an open shed.

Rigid rapiers can also be driven by an oscillating sprocket, but the system is much more popular with flexible rapiers. When rigid rapiers are used, the sprocket engages with a rack on the underside of the rapier, but, with flexible rapiers, the teeth of the sprocket pass through holes in a driving band. The reciprocating movement may be given to the sprocket by a crank arrangement in which the eccentric is driven from the main loom shaft as shown in Fig. 5.12*a*. The eccentric causes a large intermediate pulley to oscillate, and this movement is transferred to a second pulley to which the sprocket is attached. Snoeck use another method to oscillate a sprocket. A positive grooved cam reciprocates the lower end of a fulcrummed quadrant arm (Fig. 5.12*b*). The upper end thus oscillates a small intermediate gear to which the sprocket is attached. Alternatively, as illustrated in Fig. 5.13, an eccentric may be used to reciprocate a vertical rack. This rack in turn will drive an intermediate gear, which will oscillate and drive a gear mounted on the centre of the rapier-driving drum. In the last-mentioned case, the whole of the drum must oscillate, but in the previous examples the drum is stationary and merely acts as housing for the withdrawn rapier.

Reference has already been made to the relatively early time of rapier entry (60–80°) and late time of leaving (260–300°). When it is acknowledged that the high mass of the rapier-driving mechanism makes it more convenient to mount this mechanism rigidly on the loom frame, then it will be realized that the time available for insertion is also governed by how soon the sley can be made to recede from its most forward position. Thus a high degree of eccentricity or even a period of sley dwell in the back position is desirable. It is not impossible

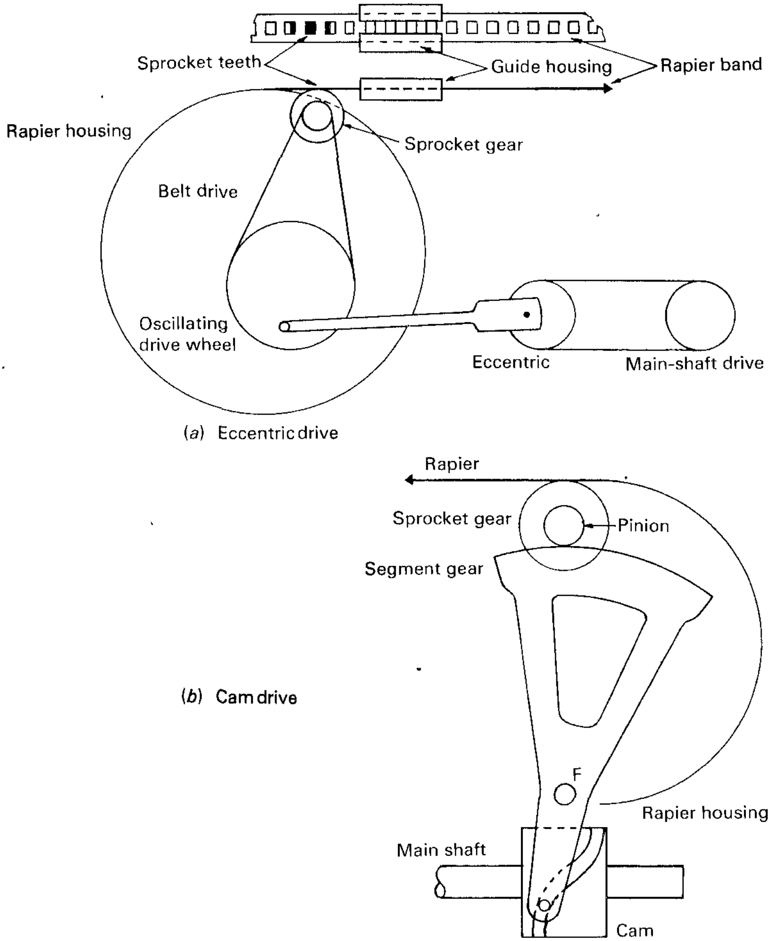


Fig. 5.12 Rapier drives from sprockets

to mount the rapier drive on the sley, but, where this is done, the power to drive the sley is particularly high.

The amount of wasted rapier movement outside the confines of the reed generally can be restricted when a fabric that does not utilize the full available reed width is being woven. This restriction can be achieved by adjusting the point of connexion between the eccentric and the driving arm to make small modifications, but this is only possible within certain limits. When larger modifications are required, it will be necessary to change the cam or adjust the amount of movement given by the eccentric in the first place.

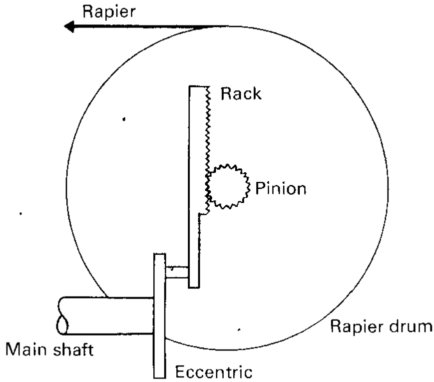


Fig. 5.13 A rapier drive from a rack and pinion

5.4 Jet-weaving

The fluid used in jet-weaving may be air or water. Because the weft-insertion medium does not require support, it is possible for the passage of the warp through the machine to be modified, so that a substantial saving in floor space will be achieved. A cross-section through the Elitex air-jet loom is illustrated in Fig. 5.14. This modified line has been used in the Maxbo (later the

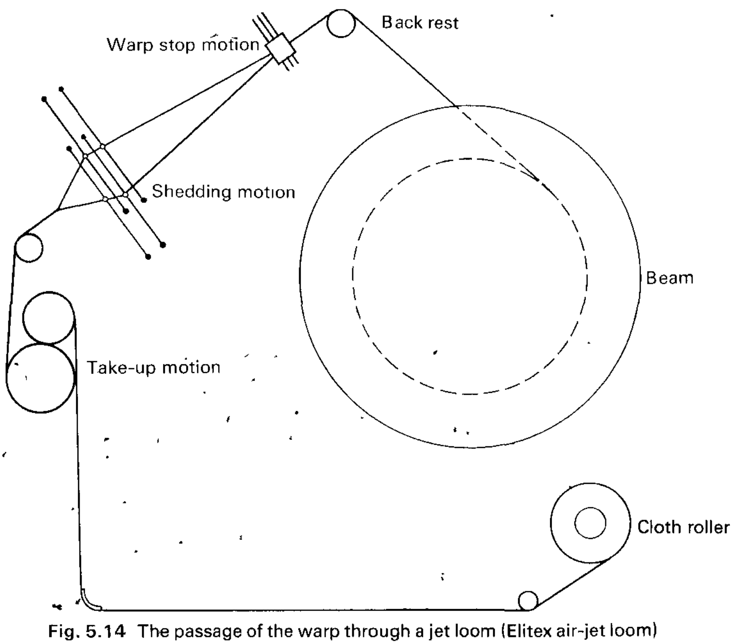


Fig. 5.14 The passage of the warp through a jet loom (Elitex air-jet loom)

Murata–Maxbo) air-jet loom and the Elitex air- and water-jet looms. The angle of incline of the centre warp-shed line to the vertical varies between 15 and 55°. In later models of the Elitex loom, the cloth roller is situated under the warp at the back of the loom as illustrated. This makes it possible to carry out all the servicing of the warp, weft, and cloth from the back alley. The manufacturers of the Prince–Nissan, Rüti, te Strake, and ATPR (Russian pneumatic-rapier) looms have retained the more conventional warp line, which in most instances is approximately horizontal. With the exception of the last-named loom, there is still a notable saving in floor space because there is no need for an extensive amount of machinery outside the loom frame.

All the looms in this group jet from one side of the loom with the exception of the te Strake machine, which jets from both sides alternately.

The passage of the weft is similar in all looms, but there are several varying principles used to achieve the necessary control. The weft, after leaving the supply package, initially passes through a tension device to a unit that is responsible for withdrawing the weft from the supply package for a period of time before insertion. The withdrawal of the yarn from the supply package may even be continuous, and the techniques available for each of these systems have been described in Section 5.1.

Jets basically operate on two principles, the first of which allows the fluid to

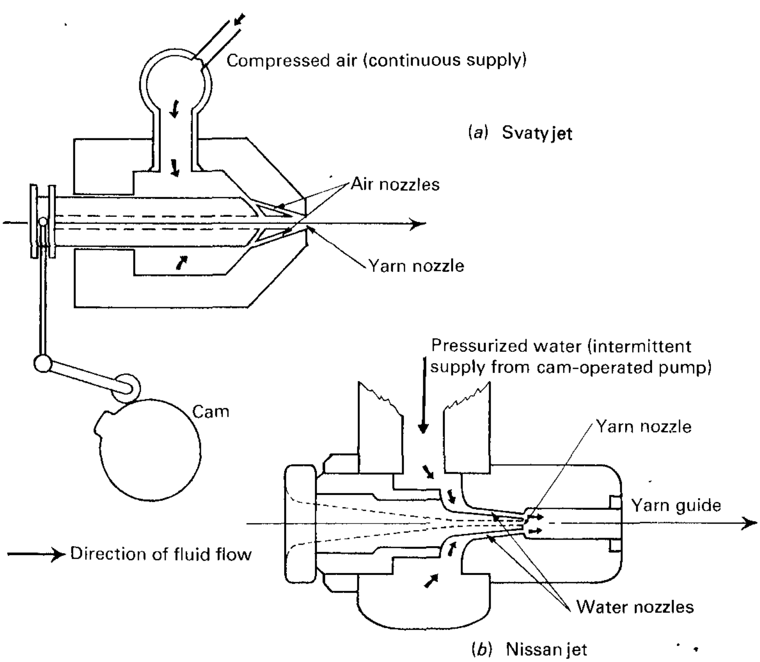
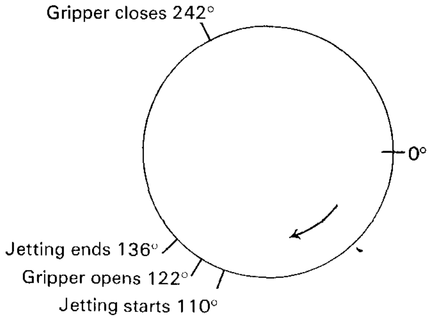
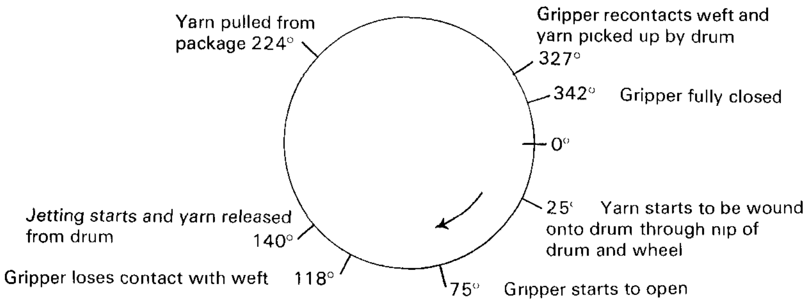


Fig. 5.15 Jet nozzles



(*a*) Nissan water-jet loom



(*b*) Elitex air-jet loom

Fig. 5.16 Weft-control timings (jet looms)

be pumped continuously under pressure to the jet nozzle. The centre barrel of the nozzle is then made to move outwards by a cam system, and this creates a gap through which the pressurized fluid will be forced. An early-type Svaty jet that uses this principle is illustrated in Fig. 5.15*a*. It is more usual nowadays to find the cam control operating on the pump, which thus allows the pressure to be released only when required. Fig. 5.15*b* illustrates a pump that uses this principle and can be found on a Nissan loom. Dirt particles and other foreign matter can seriously impede the effectiveness of jetting systems, and filters are essential in the liquid-feed pipes in order to make high efficiencies possible without producing a series of short picks.

A water jet is capable of maintaining its concentration and thus its force over a greater distance than an air jet, but there are no means readily available to assist or supplement the jet as it traverses the loom. Once it has made its initial spurt, the water jet has sufficient force to carry the leading tip of the weft across the loom. The timing diagram in Fig. 5.16*a* illustrates that, on the Nissan loom, jetting occurs from 110 to 136°. The grippers do not open until 122°, so that the loop of weft hanging out of the nozzle, after being severed from the previous pick, will be straightened out by the initial jet force. The jetting that then occurs for the purpose of carrying the weft across the loom acts for only about 14° after the clamp on the weft thread has been released. The short jetting period is verified in Fig. 5.17[16].

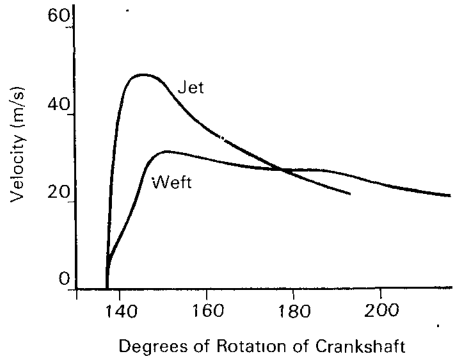


Fig. 5.17 Jet and weft velocities (Nissan water-jet loom)

Many techniques have been tried in an attempt to keep the initial blast of air concentrated in an air-jet loom. The Maxbo loom used baffle plates. These were flat metal plates, situated above and below the top and bottom warp-shed lines and attached to the sley. Together with the reed, they formed a triangular passage within which the jet was confined, but the jet still tended to become dispersed. The Elitex loom uses a series of thin metal plates mounted in the sley adjacent to the lower reed baulk and at right angles to the weft line. The plates are situated at 12.5-mm intervals, and each one has a hole through which the yarn is jetted. The hole is slightly wider at the side nearer to the jet so that air on the outer fringes of the jet will be deflected back into the main jet stream. The plates are known as *constrictors*, and they have a gap through which the weft can slip out as the sley advances to front centre, when they pass under the fell to allow beat-up. Te Strake used a series of supplementary jets at intervals across the loom to maintain the pressure in the main jet stream. This technique is still used in the updated version of that loom, the Rüti L5000. The force acting on the moving thread is given as being proportional to:

$$\rho \times \mu \times D \times L \times (V_a - V_t),$$

where ρ = air density, μ = coefficient of friction of the weft-thread surface, D = diameter of the weft thread, L = length of the weft thread, V_a = velocity of the air stream, and V_t = velocity of the weft thread. It will be realized that, if all other factors remain constant, then L must increase. This will therefore result in a decrease in the value of $(V_a - V_t)$, since V_a cannot be increased. In fact, in order to keep V_t as high as possible, the introduction of the supplementary jets keeps the effective length of the weft thread, L, small and virtually constant, as well as maintaining the value of V_a. The APTR loom inserts a hollow rapier from each side of the loom, and the jet is directed through the rapier. The rapiers are driven by cycloidal gears. It is quite common to have a suction arrangement at the opposite side of the loom to the air jet. This assists in receiving and straightening out the weft.

Problems arise in relation to weft control when the fluid jet that is dragging the weft across the loom loses pressure and also meets an undisturbed wall of air. This latter barrier tends to cause the leading tip of yarn to pile up, and both problems combine to cause the yarn to buckle or cockle at a point quite near to the jet. The associated timing of the grippers and jetting can substantially minimize these problems. Buckling occurs because the leading weft in any pick length will naturally lose velocity as it traverses the loom, but the weft passing through the jet in the later stages of the pick cycle will continue to be fed at the initial speed. Partial closing of the grippers and a reduction in the jet force are the two most common techniques to use to eliminate buckling, since either method will result in a slowing down in the initial velocity of the weft in the later stages of insertion. Reapplication of the grippers to grip the weft fully at a time at which the weft has just reached the opposite side of the loom will cause the pile-up to straighten out by virtue of the momentum still retained in the yarn. This situation has been likened to that of a whiplash, which, when the arm throwing the whip is suddenly retarded, or stopped completely, allows the leading tip to straighten out and crack.

In all instances, it is necessary to cut the weft, or to sever it by fusing, as close to the selvedge as possible. This avoids a long unsightly fringe, but it will cause the length of weft hanging from the jet to be longer than is desirable for satisfactory jetting. The continued action of the gripper in the Elitex loom, even after a firm grip has been taken on the weft, will cause this trail of yarn to be pulled into the nozzle entrance. It has already been mentioned that, in the Nissan loom, the jet operates before the clamp is released for the purpose of positioning the thread in the stream.

5.5 Continuous Weft Insertion

5.5.1 Introduction

Continuous weft insertion occurs in circular and multiphase looms, as was mentioned in Section 5.1.2. Both systems require an ever-changing warp shed, which is sometimes called wave- (or ripple-)shedding. There are usually four but occasionally six sets of sectional harnesses for each shuttle, and each set requires a cam unit to drive the harnesses. In the circular loom, each cam unit is mounted on its own short shaft. Since each shaft must be connected to its adjacent shafts by bevel gears, the machine constitutes a large number of small parts, which need careful setting and accurate timing. When weaving occurs in a flat plane, only one shaft is necessary, the cam units being positioned at intervals along the shaft under their respective sectional harnesses. If four sets of harnesses are used for each pick cycle (i.e., per shuttle), then the timing of the cams must be advanced 90° in the direction of shuttle movement. This advancement would be 60°, of course, if there were six sets of sectional harnesses per shuttle. A set of sectional harnesses will consist of a minimum of two harnesses for plain weave, with an increasing number depending on the weave; for example, four sets of harnesses will be required for 2/2 twill.

5.5.2 Circular Weaving

Fig. 5.18 shows a cross-section through the Fayolle–Ancet loom. Two beams are used to supply the warp, which passes round the back rests and is then spread into circular form by a series of closed reeds. The warp shed is formed from this point. An arrangement showing eight shuttles equally spaced round the loom is illustrated in Fig. 5.19*a*. The appropriate positions of the 32 pairs of harnesses, which will be required to weave plain, are indicated. The continuous lines represent the harnesses that control one set of alternate ends (say, the odd ends) and the dotted lines show the position of the other (even) ends.

A series of electromagnetic blocks is rotated on a motor-driven main centre shaft. The electromagnets attract the metal shuttles, which thus follow the magnets in a circular path. Attraction of the shuttles into physical contact with the electromagnets is prevented by the intervening warp shed, which is tensioned and thus supports the shuttle. The shuttle is also supported by the fixed reed that holds the warp ends at the desired spacing. Beat-up is achieved by a spiked wheel mounted on an antenna trailing behind the shuttle. The spikes penetrate the warp sheet as shown in Fig. 5.19*c*, and this causes the wheel that is mounted on the antenna arm to turn so that the spikes pick up the trailing weft and finally press it into the cloth fell.

Automatic weft replenishment is not possible, but the shuttles are capable of carrying weft packages that are much larger than the pirns normally encountered in shuttle-weaving. A warp stop-motion is incorporated in the heald eyes of the sectional harnesses.

The work of the temple is done by a fixed ring, which prevents early contraction of the cloth. The take-up motion is mounted in a high position, and the loom thus requires a lofty room or space on two floors, one above the other.

The number of shuttles per loom is not fixed at eight, and, in general, it is the diameter of the loom that determines the space available in the warp shed and thus the number of shuttles per loom. Two shuttles per loom are frequently encountered, especially in weaving hosepipes or small-diameter sacking.

An alternative possibility is found in the Fairbairn Lawson loom. In this case, the loom is mounted on a gantry. The warp is supplied by beams or from a creel underneath the loom, and the cloth is passed down the centre of the loom to be wound up at ground level. With this technique, an alternative method of shuttle drive is necessary. The harnesses operate vertically, and, in front of the reed, a series of rotating pinions is situated round the loom. These pinions engage with a rack on the underside of the shuttle, which is thus traversed round the loom. The teeth in the pinions are so shaped that the ends in the bottom warp sheet easily rest in the clearance area between the teeth and

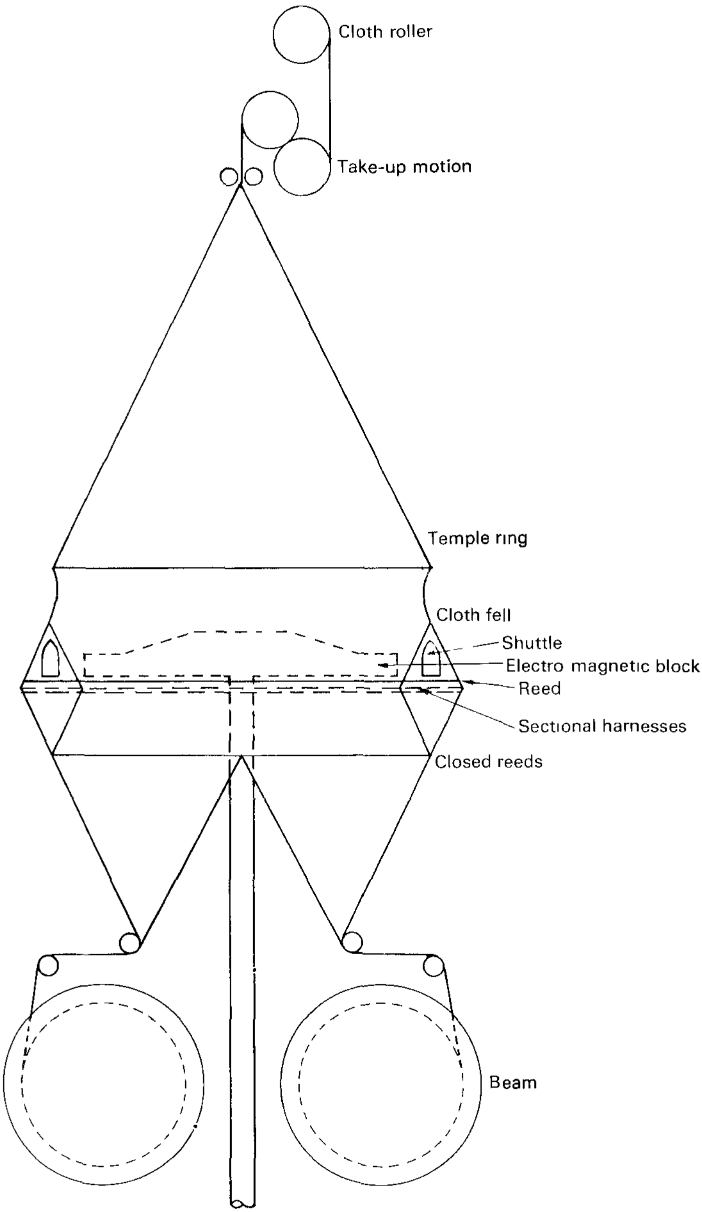
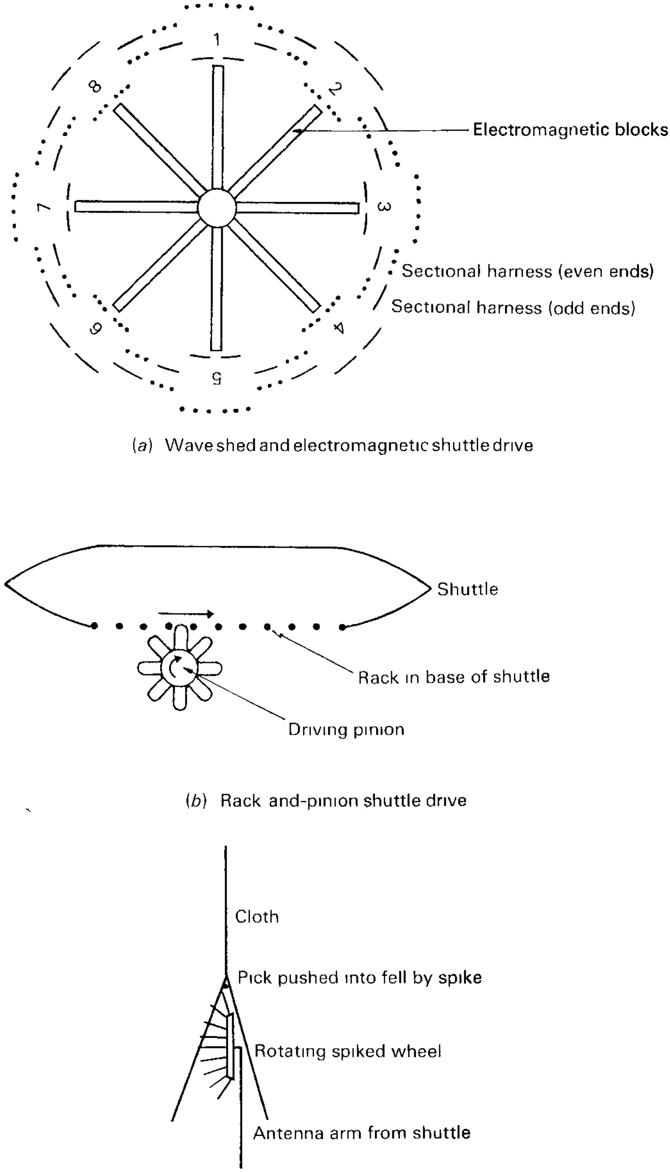


Fig. 5.18 The passage of the warp through a circular loom (Fayolle–Ancet)



(*a*) Wave shed and electromagnetic shuttle drive

(*b*) Rack and-pinion shuttle drive

(*c*) Beat up

Fig. 5.19 Shed formation, shuttle drive, and beat-up (circular loom) the pegs in the shuttle rack. The system is quite satisfactory for the harsher bast fibres and also copes successfully with cotton and even split-film polypropylene. It is less suitable for weaving the more conventional continuous-filament yarns.

5.5.3 Flat Multiphase Looms

These looms are also unsuitable for the weaving of continuous-filament yarns, and they have the added problem of stopping the loom in the event of a weft break, a technique that has been mastered on the circular looms.

In tne Rüti TWR loom, the warp is placed low down in the loom, and cloth formation occurs as the yarn rises in the loom, so that the cloth roller is mounted in a high position towards the back of the loom.

The weft carrier in this loom must be returned to the insertion side of the loom after it has inserted a pick. Since the carrier does not contain a supply package, and since several carriers are traversing the loom simultaneously, it is necessary to pre-measure each pick length on a unit known as a *turbo-winder*. The yarn is drawn continuously from a weft-supply package at the right-hand side of the loom by the turbo-winder, which has a bell-shaped rotating guide. This guide places the yarn on the inclined shoulder of a fixed blade. The yarn slides down the incline and along the blade. As the returned carrier is moved

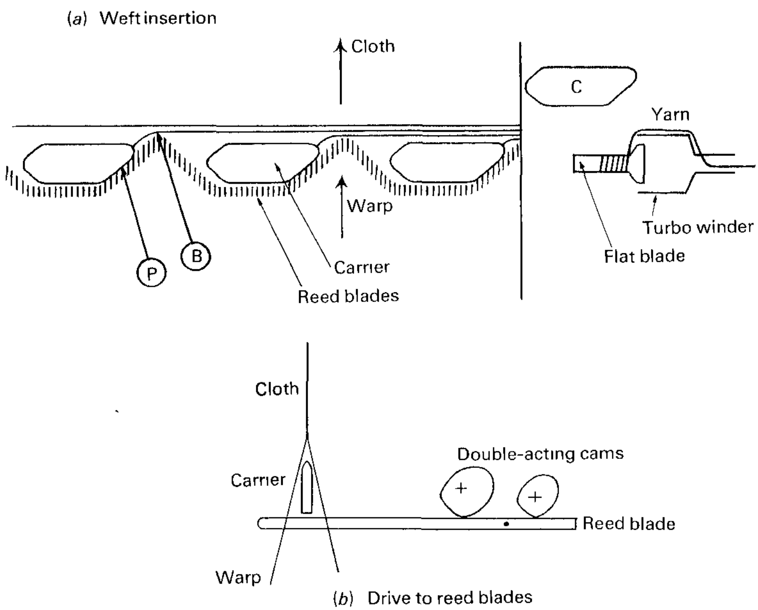


Fig. 5.20 The Rüti TWR multiphase loom

into position for weft insertion, a slot in its edge passes over the fixed blade. The pre-measured pick length is then trapped by springs within the carrier so that the yarn is pulled off the blade.

The shape of the carrier, which is approximately 6 cm long, is illustrated in Fig. 5.20*a*. A series of oscillating blades then acts on the tapered rear end of the carrier. Each blade is made to oscillate by a pair of cams, which produces a positive cam-and-counter-cam action. Both cams extend the length of the machine, and the profile is spiralled round each cam with a spiral repeating after each pick cycle, i.e., every 10 cm of reed width. As the front spiral cam changes from a rise to a hollow, then the back cam will change from a hollow to a rise, and, because of the fulcrum F (Fig. 5.20*b*), the blade will move upwards. It is this upward movement that acts on the back incline of the shuttle (P) so that the shuttle will be pushed along. When the blade reaches its highest position (B), it pushes the weft into the fell of the cloth to perform beat-up. The weft trailing from the carrier is clamped at the supply side of the loom and is thus released from within the carrier as it traverses the loom.

More recently, further looms based on similar principles have been introduced. The IWER ONA loom inserts the weft on both the front and back sides of the loom as the carriers traverse in opposite directions alternately. The Investa Kontis loom reloads the weft carriers while they are being returned down the back of the machine to the starting position.

Continuous weft insertion approaches the ideal in many respects, but it has several shortcomings and has still therefore to gain commercial acceptance.

CHAPTER 6
Warp and Cloth Control

6.1 Cloth Formation

6.1.1 Introduction

The take-up motion controls the rate at which the cloth is drawn forward, and its function may be to secure either uniform pick-spacing (i.e., the same number of picks/cm everywhere along the cloth) or uniform density of weft (i.e., the same bulk or weight per unit length of weft everywhere along the cloth). In most cases, uniform pick-spacing is the objective, and we use a 'positive' take up motion. If the weft yarns are very irregular (e.g., some woollen yarns) and if the cloth is to be heavily milled and raised, it may be preferable to try to achieve uniform weft density, and for this we use a 'negative' take-up motion. Neither objective will be achieved if the warp tension is allowed to vary. It is the function of the let-off motion to control the warp tension, so, to achieve the required result, both the take-up and the let-off motions must be functioning correctly.

Uniformity of pick-spacing is important in most cloths, and especially in those made from uniform yarns, because the more uniform the yarns, the more likely are variations in pick-spacing to produce visible faults in the fabric. Yarn irregularities tend to mask variations in pick-spacing. Uniformity of pick-spacing is most important in weaving man-made-fibre continuous-filament yarns, which are the most uniform of all. In the rest of Section 6.1, we assume that uniform pick-spacing is the objective. Uniform weft density will be referred to again in Section 6.2.

6.1.2 From the Fell to the Relaxed Cloth

We consider first what happens during the weaving of a square plain cloth and remember that in such a cloth the warp and weft linear densities are similar, as are the ends/cm and picks/cm. The warp occupies a certain width in the reed (RW in Fig. 6.1*A*), and each pick of weft has this length as it is beaten-up. The warp threads close round the pick just inserted as the shed changes for the next pick and attempt to crimp it. If the pick is to crimp, either it must stretch or the cloth must contract widthway, or both must happen. If the cloth is free to contract, it will do so as in Fig. 6.1*A*, which shows the reed in contact with the cloth fell. At this instant, the reed is preventing the cloth at the fell from contracting, and, because the warp is trying to crimp the weft, tension in the weft at and near the fell must be high. As the reed moves away from the fell, the

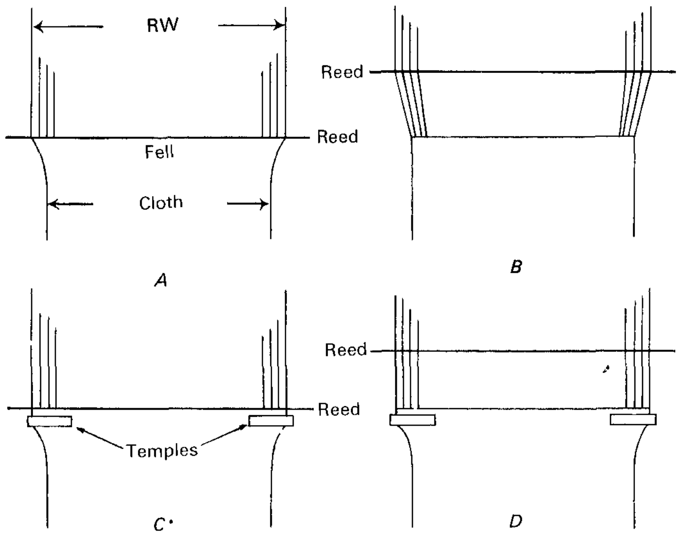


Fig. 6.1 Cloth formation

cloth at the fell will contract widthway under the influence of the tension in the weft produced by the effort of the warp to crimp it. This is represented in Fig. 6.1*B*. When the reed comes forward again, it will stretch the cloth at the fell. This will result in severe rubbing of the ends near the selvedges by the wires of the reed. Weaving cannot proceed under these conditions. We therefore use temples to keep the cloth near the fell stretched to reed width, or as near to it as possible.

With efficient templing, we have the condition shown in Fig. 6.1*C* and *D* as the reed moves to and fro. Contraction at the fell is prevented, and rubbing of the ends near the selvedges is greatly reduced. The picks are now under high tension until they are clear of the temples, and the weft crimp near the fell is very low. It has been shown by Townsend[17] that, for a square plain worsted cloth with a fractional cover* of 0·5 (corresponding to a firmly set cloth), the weft tension may increase from about 5 gf (49 mN) as the shuttle lays it in the shed to as much as 180 gf (1765 mN) when the pick is incorporated into the cloth.

Since effective templing prevents the weft at the fell from crimping, it follows that the warp crimp at the fell must be correspondingly high. The balance between the warp and weft crimps near the fell is therefore artificial. Some redistribution of crimp will occur when the cloth escapes from the influence of the temples. It will then be under warpway but not weftway tension, and it may be that the weft crimp will then exceed the warp crimp. A further redistribution of crimp, tending towards an equalization of warp and weft crimps, will occur

* Calculated as d/p, where d is the yarn diameter and p the thread-spacing

when the cloth is allowed to relax off the loom. Crimp redistribution will be complete when the fabric is wet out and allowed to relax freely. Subsequent finishing processes, especially drying under tension, may again produce an artificial balance of crimp in the finished fabric. If so, the cloth will have a tendency to be dimensionally unstable in use, especially when it is washed.

An interchange of crimp implies some relative movement of the yarns, and, since any such movement is resisted by friction between the yarns and by the resistance of the yarns to deformation, warp and weft crimps are unlikely to be equal in an exactly square, wet-relaxed cloth. The exact balance of crimp will depend on the history of the cloth—on the forces to which the yarns have been subjected before, during, and after weaving. In general, the warpway tension tends to predominate, and we may expect the weft crimp to be rather greater than the warp crimp in the relaxed cloth.

The unbalanced-crimp conditions in a square plain cloth near the fell will be exaggerated in a weft-faced plain cloth such as a limbric, in which the fractional weft cover is substantially greater than the fractional warp cover. Such cloths will exhibit greater weftway contraction between the reed and the loomstate cloth and will therefore require very effective templing. They will also develop still higher weft tensions near the fell.

In warp-faced cloth, such as poplin and poult, the weft crimp is low in the relaxed fabric, usually not more than 2 or 3%. It follows that, in weaving such cloths, high weft tensions will not develop at the fell, and there will be little tendency for weftway contraction as the cloth moves away from the fell. It may therefore be possible to weave them without temples.

A practical consequence of the high weft tensions developed near the fell in weaving square and weft-faced plain cloths is the fault known as *weft-cutting*. When this happens, the weft is broken by a combination of high weft tension and the scissors-like action of the ends trying to crimp the picks. It usually happens in or near the selvedges for two reasons. Firstly, the ends in the selvedges are often more closely spaced than those in the body of the cloth, and this exaggerates the effect. Secondly, templing is seldom fully effective in holding out the cloth to reed width, and this leads to extra tension in the weft near the selvedges when the reed comes forward. Weft-cutting can be minimized by delaying the crossing of the healds until the shuttle is at rest, but this may sometimes result in slack weft or weft loops at the selvedges.

Townsend[17] has treated what he calls the 'physics of cloth formation' in a more formal and quantitative way by making use of Peirce's equations[18], which define the geometry of plain-weave cloth, and by introducing other simple relations involving the forces acting on the yarns in the fabric, to define which we need to know the elastic moduli of the yarns and fabrics.

6.1.3 The Cloth-fell Position

As the reed moves the newly inserted pick up to the cloth fell to incorporate it into the cloth, it at first encounters only a slight resistance due to friction between the warp and weft and then meets some further resistance due to crimping the warp round the weft. The reed encounters no other resistance in the weaving of cloths with very low weft cover factors. It merely pushes the pick to its correct position and leaves it there. There is no movement of the cloth fell during beat-up. In weaving fabrics with the normal range of weft cover factors, such as are necessary to produce a firm cloth, the required pick-spacing cannot be achieved unless the reed exerts some substantial pressure on the fell at beat-up. This is called the *beat-up force*. Now, the reed can exert pressure on the fell only if the fell offers resistance to displacement. This is called the *weaving resistance*. The beat-up force and weaving resistance are equal and opposite. The fell resists displacement by virtue of tension in the warp and cloth. In Fig. 6.2, T_1 and T_2 represent the tensions in the upper and lower warp sheets at any

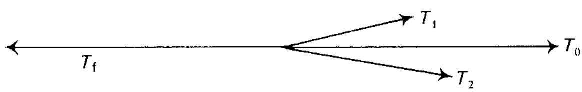


Fig. 6.2 Warp and cloth tensions

time when the fell is not being displaced by the reed—say, just before the reed strikes the fell. We call the resultant of T_1 and T_2 the *basic warp tension*, T_0. If T_f represents the tension in the fabric, $T_0 = T_f$, provided that the fell is not being displaced by the reed. When the reed strikes the fell and displaces it to the left as shown in Fig. 6.2, the warp will stretch and the fabric will contract: T_0 will increase and T_f will decrease. At any instant during beat-up, the pressure exerted on the fell by the reed (the beat-up force) and the resistance offered by the fell to displacement (the weaving resistance) must be equal to the difference between T_0 and T_f.

Suppose that we progressively increase the picks/cm while we are weaving a plain fabric, all other conditions remaining the same. It is clear that the force required to produce the desired pick-spacing will increase with the picks/cm. In other words, the weaving resistance and beat-up force will increase. The extra force required to achieve the closer pick-spacing can only come from an increase in the difference between T_0 and T_f, but this difference will increase if the displacement of the fell by the reed at beat-up increases. We thus conclude that, for a given set of conditions, the displacement of the fell at beat-up must increase as the weft cover factor increases.

The above argument is, of course, purely theoretical and in very general terms. Greenwood and his co-workers[19–22] at the British Rayon Research Association (later incorporated into the Cotton Silk and Man-made Fibres Research Association, the Shirley Institute) have investigated the problem thoroughly. We have seen that the normal position of the cloth fell just before beat-up is behind (i.e., further from the weaver than) the most forward position of the reed. The cloth-fell position (C.F.P.) is stated quantitatively (in mm, for example) in relation to the most forward position of the reed. According to the convention adopted by Greenwood *et al.* for the purpose of their mathematical argument, the cloth-fell position has a negative sign when it is, as it normally is, behind the most forward position of the reed. A cloth-fell position of −7 5 mm, for example, implies that, when the loom is running normally, the fell, just

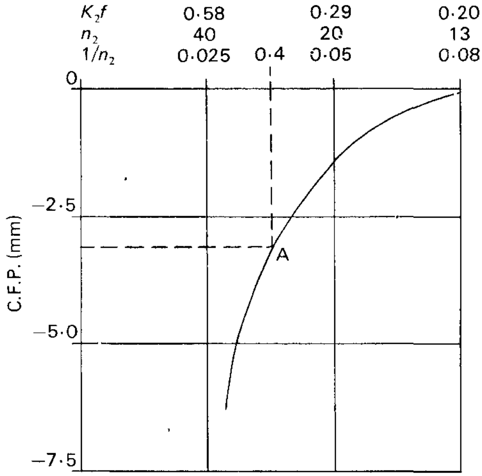


Fig. 6.3 The cloth-fell position

before the reed strikes it, will be 7·5 mm behind the most forward position of the reed. It follows that the fell would be displaced 7 5 mm at beat-up. The negative sign is important only in the mathematical treatment.

The curve in Fig. 6.3 was plotted from data Greenwood and Cowhig[19] obtained by observing the cloth-fell position in a loom weaving a 16-tex (140-den) cellulose acetate warp with 34 ends/cm and a 16-tex (140-den) cellulose acetate weft with different numbers of picks/cm in plain weave. For the particular conditions of this experiment, the cloth-fell position is zero for a fractional weft cover of about 0·2 or less. In other words, the fell is not displaced by the reed at beat-up if the fractional weft cover is less than about 0·2. This corresponds to a very open cloth.

In an approximately square plain cloth made from low-twist continuous-filament yarns, a fractional weft cover of about 0·4 is sufficient to give a firm cloth. This corresponds approximately to point A on the curve. Beyond this point, the cloth-fell position increases rapidly with increases in fractional weft cover, which seems to approach a limiting value of about 0·58. This indicates that, for the conditions in which this experiment was carried out, it would be impossible to exceed a fractional weft cover of about 0·58. Similar curves would be obtained for other yarns, weaves, and weaving conditions, but the limiting values would differ. The curve in Fig. 6.3 provides experimental verification for the theoretical argument outlined above that the cloth fell is displaced by the reed at beat-up in weaving cloths with the normal range of fractional weft covers and that the displacement increases with the fractional weft cover. It also shows that the cloth-fell position (or the displacement of the fell at beat-up) increases very rapidly as the fractional weft cover approaches the limit for a particular set of conditions.

We have seen that the weaving resistance (or beat-up force) is equal to the difference between the warp and fabric tensions ($T_0 - T_f$), and that this difference increases with the displacement of the reed at beat-up (i.e., with the cloth-fell position), but we do not yet know exactly how they are related. According to the excess-tension theory, there is a direct proportionality.

Let:

R = the instantaneous weaving resistance (i.e., the force the reed exerts on the fell at any instant during beat-up);

Z = the instantaneous displacement of the cloth from its basic position;

L_w and L_f = the free lengths of the warp and fabric, respectively (see Fig. 6.4);

E_w and E_f = the elastic moduli of the warp and fabric;

T_w and T_f = the instantaneous warp and fabric tensions, respectively, at any instant during beat-up; and

T_0 = the basic warp and fabric tension just before beat-up.

It is assumed that the warp and fabric are free to stretch and contract over their respective free lengths, but the following argument is not invalidated if their free lengths are not exactly as indicated in Fig. 6.4.

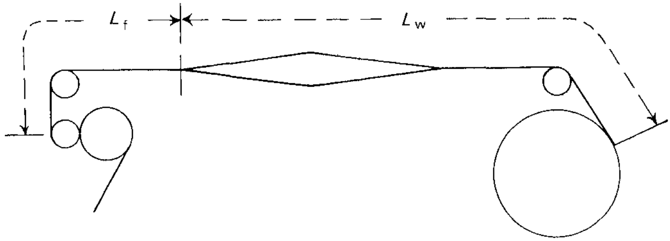


Fig. 6.4 A free length of warp and cloth

Immediately before beat-up:

$$T_0 = T_w = T_f.$$

At any instant during beat-up:

$$T_w = T_0 + \frac{E_w Z}{L_w}$$

and

$$T_f = T_0 - \frac{E_f Z}{L_f}.$$

But:

$$R = T_w - T_f = (T_0 + E_w Z/L_w) - (T_0 - E_f Z/L_f).$$

Hence:

$$R = Z(E_w/L_w + E_f/L_f). \tag{6.1}$$

This shows that the weaving resistance, which is equal and opposite to the beat-up force, is proportional to the displacement of the fell from its basic position. The peak beat-up force is therefore proportional to the maximum fell displacement, which is equal to the cloth-fell position. Moreover, since no term involving the basic warp tension, T_0, appears in Equation (6.1), we conclude that the beat-up force is independent of the basic warp tension, from which it follows that we cannot increase the beat-up force by increasing the basic warp tension. We shall see in the next section that this does not apply to bumping conditions.

It may help to clarify what has been said about the cloth fell position if we take a simple practical example. Suppose a loom to be weaving with a moderate fractional weft cover, say, 0·4. This is enough, as we know from Fig. 6.3, to require an appreciable displacement of the fell at beat-up. The cloth-fell position will have a certain finite value, which will remain unchanged as long as weaving proceeds normally without disturbance. Now suppose that we alter the rate of take-up so as to give more picks/cm and a higher fractional weft cover, say, 0·5. The cloth-fell position previously established at the lower fractional weft cover will no longer be sufficient to provide the required beat-up force, and the required pick-spacing will not immediately be achieved. We shall be producing cloth more rapidly than it is being taken up, and consequently the cloth fell will begin to move further from the weaver; the value of the cloth-fell position will increase. As it does so, the weaving resistance and beat-up force will, according to the excess-tension theory, increase proportionally, until, after perhaps 20 or 30 picks have been woven, a new cloth-fell position, appropriate to the closer pick-spacing, will have become established. This new cloth-fell position will then remain unchanged until something happens to disturb it.

6.1.4 Bumping Conditions

If the displacement of the cloth fell by the reed at beat-up is sufficiently large, the cloth tension at beat-up will be reduced to zero, and the cloth will be momentarily quite slack. This condition, which is known as 'bumping', is most likely to occur in weaving near the limiting value of the fractional weft cover. It is easily recognized by the noise the cloth makes when it suddenly becomes taut again as the reed recedes. It can also be detected by placing the fingers on the cloth near the fell.

It is probably self-evident that the cure for bumping is to increase the basic warp tension, and hence the basic cloth tension, so that the cloth can sustain a larger displacement of the fell without becoming slack. Greenwood and Cowhig[19] explain this by reference to warp- and cloth-tension traces, recorded electronically while the loom is running. Two such idealized traces are shown in Fig. 6.5. Except when the reed is displacing the fell, the warp and cloth tensions must be substantially equal, as shown in Fig. 6.2.

The traces show that the warp and cloth tensions gradually increase as the shed opens, remain at a high level during the period of heald-shaft dwell, and fall gradually as the shed closes. At beat-up, there is a sharp momentary in-

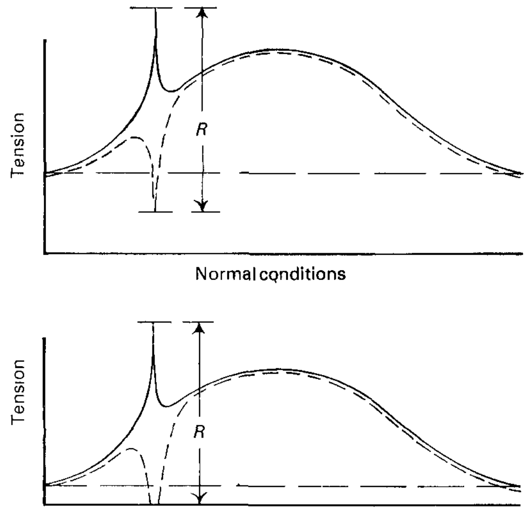


Fig. 6.5 Warp- and cloth-tension traces

crease in warp tension owing to the warp's being stretched, and there is a corresponding decrease in cloth tension. The weaving resistance, *R*, is proportional to the vertical distance between the peaks. Normal weaving conditions, under which the cloth tension does not fall to zero, are represented by Fig. 6.5*A*.

Under bumping conditions (Fig. 6.5*B*), the tip of the cloth-tension peak is cut off, and the weaving resistance is reduced. In other words, bumping reduces the effectiveness of beat-up. It is clear that normal weaving conditions can be restored by increasing the basic warp tension until the tip of the cloth-tension peak is no longer cut off. Note, however, that it is only when the loom is bumping that an increase in the basic warp tension increases the effectiveness of beat-up. It is clear that, for any particular fabric and weaving conditions, there will be a certain value of warp tension that will be sufficient to prevent bumping. This is the minimum warp tension that can be used in those circumstances. There is no point in applying a much higher warp tension than this, with the probability of an increase in the warp-breakage rate, unless other considerations require it. In weaving very hairy yarns in cloths with low fractional weft covers, for example, the minimum warp tension necessary to form a clear shed may be more than is required to prevent bumping. In general, however, it is the onset of bumping that determines the minimum practicable warp tension.

6.1.5 Disturbed Weaving Conditions

Under stable weaving conditions, the position of the cloth fell just before beat-up does not change from pick to pick. The introduction of a new pick merely restores the fell to the position that it occupied before take-up occurred. It does not matter at what stage in the loom cycle take-up occurs, provided that it always occurs at the same stage. When the position of the cloth fell changes owing to some accidental cause, we have disturbed weaving conditions. Fortunately, the action of a positive take-up motion is self-correcting, although not instantaneously so, and, if the disturbance is temporary, the fell will soon resume its normal position. In the meantime, however, a fault is likely to have appeared in the fabric, because any change in the cloth-fell position will produce a change in pick-spacing.

If the fell moves towards the weaver, the fell displacement, the beat up force and the picks/cm all decrease. Conversely, if the fell moves away from the weaver, the fell displacement, the beat-up force, and the picks/cm all increase. The disturbance may be either sudden or gradual. If it is sudden, the result is either a thin place across the fabric (too few picks/cm) if the fell moves towards the weaver, or a thick place (too many picks/cm) if the fell moves away from the weaver. Either of these may produce a visible fault taking the form of a sudden change in apparent shade, followed by a gradual and therefore imperceptible return to the normal shade. This apparent change in shade is due to the change in pick-spacing and the consequent change in warp crimp affecting the light reflecting properties of the fabric. The severity of the fault will depend on the type of yarn being woven, the cloth construction, and the magnitude of the disturbance. Smooth, uniform yarns, such as continuous-filament yarns and combed cotton yarns, especially if gassed and mercerized, show up variations in pick-spacing that would be masked by the irregularities and fibrous surface of spun yarns. Fabrics with high warp crimp, such as poplin and poult, tend to be sensitive to variations in pick-spacing because a given change in pick-spacing produces a relatively large change in warp crimp. For the same reason, plain-weave fabrics are more sensitive than those with looser weaves. Fabrics with low fractional weft covers also show up variations in pick spacing for the reasons to be explained in the next paragraph.

It is clear from the shape of the curve in Fig. 6.3 that a given change in pick-spacing produces a much larger change in cloth-fell position at high than at low fractional weft covers. Conversely, a given change in cloth-fell position has a much larger effect on pick-spacing at low than at high fractional weft covers. This explains why cloths with low fractional covers are particularly liable to show thick and thin places and other faults associated with variations in pick-spacing. For this reason, cheap, impoverished fabrics are often more difficult to weave relatively fault-free than more soundly constructed ones.

A gradual disturbance of the cloth-fell position will produce a gradual change in pick-spacing. This may not be noticed in the fabric unless it repeats at regular intervals. As we shall see in the next section, there are several potential causes of periodic variations in pick-spacing.

6.1.6 Causes of Variations in Pick-spacing

We have seen that any disturbance of the cloth-fell position will produce a variation in pick-spacing, which, if severe enough, will produce a fault in the fabric. Most, but not all, variations in pick-spacing are caused by displacement of the cloth fell. There are three main sources of variations in pick-spacing:

(*a*) the take-up motion;
(*b*) the let-off motion; and
(*c*) loom stoppages.

It is obvious that any variation in the rate of take-up must cause a change in cloth-fell position and therefore a change in pick-spacing. Variations of this type arise mainly from mechanical imperfections, such as eccentric or eccentrically mounted rollers or gear wheels, and from worn or incorrectly shaped teeth. These causes will be discussed more fully in Section 6.2.

Variations in the rate of let-off result in variations in the warp tension, which

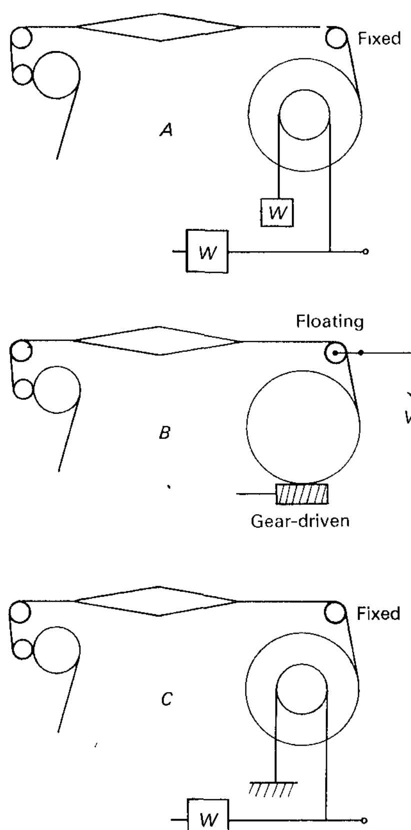


Fig. 6.6 Warp-tensioning systems

in turn cause variations in the cloth-fell position and in pick-spacing. These will be discussed more fully in Section 6.3.

Prolonged loom stoppages result in temporary variations in pick-spacing when the loom is restarted. The resulting cloth faults are called *starting places* or *setting-on places*. These faults may be due to a fall in warp tension or to a change in the position of the fell, or more usually to a combination of both. Fig. 6.6*A* represents a simple negative friction let-off. With this arrangement, if the beam is assumed to rotate freely in its bearings, any stretching of the warp and cloth that occurs during a loom stoppage will be taken up by a clockwise rotation of the beam under the influence of the weighting system. There will be no fall in warp tension during a stoppage, but the fell will move to the right (away from the weaver), and a thick place will be produced on starting up. That same situation exists with an automatic positive let-off motion that has a floating backrest, as shown in Fig. 6.6*B*, provided the friction does not prevent the backrest from moving to take up the slack. The same also applies to a positive let-off motion such as that on the Saurer loom (Fig. 6.23), which has a fixed backrest and a floating beam. In all these cases, we may therefore expect a thick place on starting up after a prolonged loom stoppage.

If neither the backrest nor the beam floats, the combined free length of the warp and fabric is fixed. This situation exists with a negative friction let-off in which the slack sides of the ropes or chains are attached to the loom as shown in Fig. 6.6*C*. During a prolonged loom stoppage, the warp and fabric tension will tend to fall owing to creep, and there may also be a movement of the cloth fell. The direction in which it will move will depend on the relative changes in the elastic moduli of the warp and fabric during the loom stoppage. The system can be represented by two springs in series, as shown in Fig. 6.7. Both springs are extended, their tension, T, is the same, and their combined length is fixed. Their junction represents the cloth fell. If their elastic moduli are, respectively, E_f and E_w and their extensions are δ_f and δ_w, then:

$$T = E_w \delta_w / L_w = E_f \delta_f / L_f.$$

Since any movement of their junction (i.e., the cloth fell) will be small compared with their length, we may regard L_w and L_f as constant, in which case:

$$E_w \delta_w = E_f \delta_f. \tag{6.2}$$

If their tension, T, decreases during a stoppage, this can only be because their elastic moduli have decreased. If both elastic moduli decrease by the same fraction, Equation (6.2) will be satisfied without a change in δ_w and δ_f, and there will be no movement of the fell. Should the modulus of the warp, E_w, decrease by a greater fraction than that of the fabric, E_f, δ_w must increase and δ_f decrease to satisfy Equation (6.2). The cloth fell will move to the left in Fig. 6.7 (i.e., nearer the weaver), and this will tend to produce a thin place on starting up. This tendency will be reinforced by the reduction in warp tension as a result of the stoppage. Conversely, should the modulus of the fabric, E_f, decrease by a greater fraction than that of the warp, E_w, δ_f must increase and δ_w decrease to

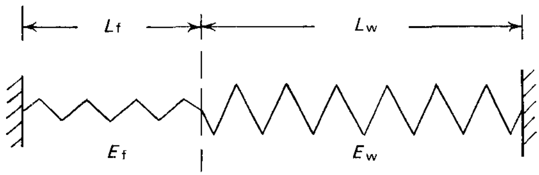


Fig. 6.7 The spring model

satisfy Equation (6.2). The cloth fell will move to the right in Fig. 6.7 (i.e., away from the weaver), and this will tend to produce a thick place on starting up. But the reduction in warp tension during the stoppage will tend, as before, to produce a thin place, and we have insufficient information to decide which of these two effects will predominate. We cannot therefore predict whether to expect a thick or a thin place, but, because of the opposing effects, we can be fairly sure that it is unlikely to be a severe fault.

The above argument suggests that an arrangement similar to that of Fig. 6.6*C* may be preferable to either of the others, but this is not necessarily so. Friction in the beam bearings (Fig. 6.6*A*) or in the bearings of the levers associated with the floating backrest (Fig. 6.6*B*), may cause these systems to behave in practice like that shown in Fig. 6.6*C*. If friction can be eliminated or reduced to the point at which it does not interefere with the floating movement of the beam or backrest in the system of Fig. 6.6*A* and *B*, we know to expect a thick place after a stoppage, and this may enable the weaver to take corrective action by adjusting the take-up motion before restarting the loom. With the system of Fig. 6.6*C*, the weaver has to discover by experience whether to expect a thick or a thin place, and it is quite possible that a short stoppage may produce one of these and a longer stoppage the other.

The problem we have been discussing is very complex, and it is not possible to make recommendations that would lead to the elimination of setting-on places. It is obvious that any means of reducing warp and cloth tensions during a stoppage would be helpful, but, if this is done by interfering with the let-off motion, it may be difficult or impossible to restore the correct tension on starting up, in which case the fault produced may be worse than if nothing had been done. Warp tension is greatest when the shed is open, and it is worth while to leave the loom in the 'healds-level' position during a stoppage. This is easy to do with plain weave, which is the most susceptible to setting-on places.

6.2 The Take-up Motion

6.2.1 Introduction

Reference has been made in Section 6.1.1 to negative and positive take-up. There is a wide variety of types of positive take-up, and these will be discussed at some length in Section 6.2.3. The applications of negative take-up are very much more limited, and one example will be sufficient to illustrate the general principles.

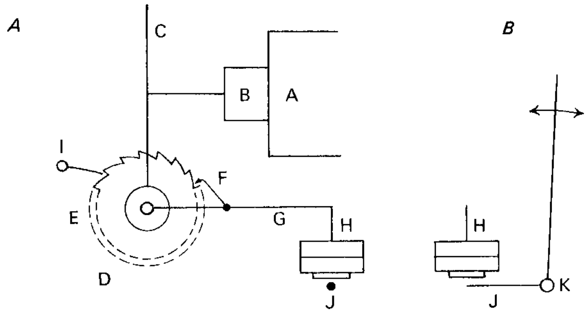


Fig. 6.8 Negative take-up

6.2.2 *Negative Take-up*

The principles of negative take-up are illustrated in Fig. 6.8, in which Fig. 6.8*A* is a front view and Fig. 6.8*B* a side view of part of the system, which cannot be conveniently shown in the front view. In this example, the cloth is wound directly onto a roll A on the cloth roller B. A worm wheel C on the cloth-roller shaft is driven by a worm D, which is on the same short shaft as the ratchet wheel E. This is turned intermittently by a pawl F carried on a lever G, which is pivoted loosely on the ratchet-wheel shaft. Weights H suspended from lever G tend to wind the cloth onto the roller against the resistance offered by the tension applied by the let-off motion. A holding pawl I prevents the ratchet wheel from turning in the reverse direction, except when intentionally released by the weaver for letting back after pick-finding or unweaving. A lever J attached to the rocking shaft K rises and falls as the sley reciprocates.

The weights are raised as the sley moves towards the back of the loom and remain suspended out of contact with the lever J as the sley moves forward. At beat-up, the weights are therefore floating, since they are not heavy enough to turn the ratchet wheel. As the reed displaces the cloth fell, the cloth tension falls, and the weights are now able to turn the ratchet wheel by an amount that will depend on the thickness of the pick being beaten-up. If it is excessively thick, for example, there will be more fell movement than usual, the cloth tension will fall more than usual, and the weights will be able to turn the ratchet to a greater extent. The thicker the pick the greater is the take-up, which is what is required to insert a uniform density of weft.

The average density of the weft is controlled on a trial-and-error basis by varying the warp tension and weights H. It is clear that increasing the weights will increase the average rate of take-up and that decreasing the warp tension will have the same effect. For a given fabric, the beat-up force required to produce the required weft density will determine the minimum warp tension needed to prevent bumping. When this has been determined by trial, the weights are adjusted to obtain the necessary floating action. Some further adjustment of the warp tension may then be required. There is a large element of uncertainty in the behaviour and adjustment of a negative-take-up motion.

If the cloth is wound directly onto the driven roller, as in Fig. 6.8, as the diameter of the roll increases, it will be necessary to increase the weights periodically to provide the increased force needed to turn the ratchet wheel. This introduces another element of uncertainty, which can be avoided by using the driven roller as a take-up roller and winding the cloth onto a separate roller, as shown, for example, in Fig. 6.9*A*.

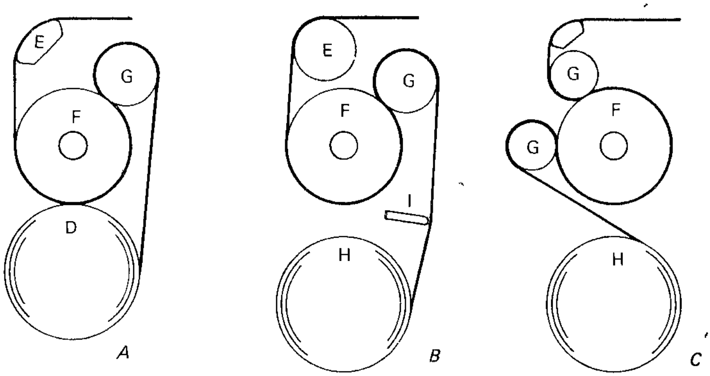


Fig. 6.9 Cloth-wind-up systems

6.2.3 Positive Take-up: Cloth Control

Positive-take-up motions are sometimes classified as continuous (worm-and-worm-wheel drive) or intermittent (pawl-and-ratchet-wheel drive). This is not a very fundamental difference, although the distinction may be convenient, because wide, heavy looms tend to use the former and narrower, lighter looms the latter. Continuous take-up is common, for example, in woollen and worsted weaving but is much less popular in weaving cotton and continuous-filament yarns.

In either case, the cloth is usually drawn forward by frictional contact with a take-up roller, which is driven through gearing at the required uniform rate, the cloth being wound up onto a separate roller, as in Fig. 6.9*A*, *B*, and *C*. This is called *indirect take-up*. Alternatively, the cloth may be wound directly onto the driven roller. This is called *direct take-up*. In this case, the rate of rotation of the driven roller has to be reduced progressively as the cloth builds up so as to maintain a constant linear rate of take-up. Direct take-up is relatively uncommon and is normally used in conjunction with intermittent drive, particularly in some looms designed specifically for weaving continuous-filament yarns.

With the above points in mind, we may classify positive-take-up motions in the following way:

(i) continuous, indirect;
(ii) intermittent: (*a*) indirect;
(*b*) direct.

As already mentioned, indirect take-up relies on friction between the cloth and the surface of the take-up roller to draw the cloth forward. Slippage would produce severe variations in pick-spacing, which might take the form of random thick places or periodic but more gradual variations. To prevent slippage, the take-up roller is always covered with a material having a high coefficient of friction. Perforated steel fillet is often used for spun yarns, which are unlikely to be damaged by the rough, outward-projecting edges of the perforations. For more delicate yarns, cork or relatively hard rubber with rough surfaces is used. Slippage is further discouraged by arranging for the cloth to be held in contact with a large part of the circumference of the take-up roller. Three typical arrangements are shown in Fig. 6.9*A*, *B*, and *C*.

Fig. 6.9*A* shows a method commonly used for weaving spun yarns. The breast beam E takes various forms. It may be a smooth metal bar, as in Fig. 6.9*A*, or a non-rotating roller, as in Fig. 6.9*B*. In the latter case, spiral grooves in the form of a screw-thread are often cut in the surface of the roller near the selvedges. The directions of the spiral at the two ends are opposite and thus tend to stretch the cloth widthway so as to prevent creasing.

In Fig. 6.9*A* and *B*, F is the driven take-up roller, and G is a smaller roller, usually felt-covered, which is forced against the take-up roller by springs and acts as a nip roller. The roll of cloth D in Fig. 6.9*A* is rotated by frictional contact with the cloth on the take-up roller. A system of levers and weights or levers and springs maintains pressure contact but allows the cloth roller to fall as the roll builds up. In the arrangement shown in Fig. 6.9*B*, the cloth roller H is driven from the take-up roller, usually by a chain and sprockets, and a slipping clutch on the cloth-roller shaft allows for the slowing down in the rate of rotation of the cloth roller as the roll builds up and at the same time keeps the cloth under tension so as to discourage slipping and creasing. With this system, creasing is further discouraged by a slightly curved anti-crease bar I. The arrangement shown in Fig. 6.9*C* is similar to that in Fig. 6.9*B*, except that the take-up roller rotates in the opposite direction. It is used in some looms for weaving continuous-filament yarns and also in the Sulzer machine, in which the top pressure roller is replaced by a steel plate.

It is desirable for the weaver to be able to inspect both sides of the fabric. Warp satin, for example, is usually woven face down (i.e., weft side up), and it is often difficult to detect a warp fault, such as a missing or faulty end or a wrong dent, by inspecting the weft side. With all three arrangements shown in Fig. 6.9, the warp side would be on the outside on the cloth roll, where the weaver could inspect it from time to time.

Provision has to be made for removing the cloth at intervals. In the system of Fig. 6.9*A*, this involves some means of lowering the cloth roller out of contact with the take-up roller and locking it there while the cloth is rewound onto another roller, mounted in a transportable trolley, which is brought to the loom when required. Alternatively, provision may be made for removing the cloth and its roller and replacing it with an empty roller. In the systems of Fig. 6.9*B* and *C*, provision must be made either for disengaging the clutch drive to the cloth roller to allow the cloth to be rewound at the loom or for removing the cloth on its roller.

With high-speed automatic looms, cloth removal constitutes a significant element in the work of the weaver or of the ancillary worker responsible for this task. This, together with the advantages to the finisher of long lengths of fabric, has prompted loom makers to redesign their take-up motions to accommodate very large rolls of cloth and to allow the cloth to be removed on its roller while the loom is running. This latter requirement demands a very firm grip of the cloth by the take-up and pressure rollers.

6.2.4 Positive Indirect Take-up: Method of Driving

This section includes calculations of the possible periodicities in pick-spacing attributable to imperfections in the take-up mechanism. These calculations are based on established mechanisms, most of which were designed to give the same number of picks/in. as the number of teeth in the change wheel. This condition would be upset if values expressed in centimetres and picks/cm were substituted for those expressed in inches and picks/in. In this section and Section 6.2.6 only, the more traditional imperial units have therefore been retained.

All types of positive-take-up motion make use of spur gears and also sometimes of a worm and worm wheel. If any wheel in the train of gearing is eccentric or eccentrically mounted, or if the teeth are worn or badly designed, periodic variations in pick-spacing will result. Whether these variations are likely to produce serious faults in the fabric depends on the type of yarns and the fabric being woven, the severity of the mechanical fault, and the wavelength of the faults in the fabric. According to the experience of workers at the Shirley Institute, periodic variations in pick-spacing that have a wavelength of less than about $\frac{1}{8}$ in. or more than about 10 in. are unlikely to produce serious faults. One should therefore try to design the take-up motion so that, even if it develops mechanical imperfections as a result of wear, it will not produce faults with a wavelength within the range of $\frac{1}{8}$–10 in. We shall see that this is possible, and indeed quite easy, to achieve but that very few take-up mechanisms have been designed with this in mind.

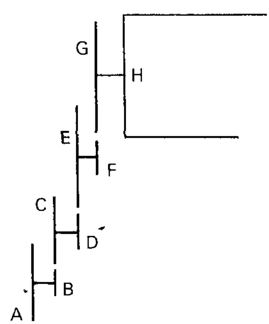


Fig. 6.10 The seven-wheel take-up motion

If a cloth shows periodic variations in pick-spacing, it is usually possible to determine with certainty whether or not the take-up motion is responsible for them and, if so, to identify the offending part of the mechanism. This can be done by calculating the periods of the various faults that the particular mechanism is capable of producing. If one of these is equal to the period in the cloth, the cause is identified: if not, we have to look elsewhere for the cause.

Consider, for example, a typical seven-wheel take-up motion, such as is commonly used on cotton looms. The gearing of such a motion is represented in Fig. 6.10, and typical sizes for the gears would be as follows:

(A)	ratchet wheel, one tooth per pick	24 teeth
(B)	ratchet pinion	36 teeth
(C)	change wheel	
(D)	change pinion	24 teeth
(E)	compound wheel	89 teeth
(F)	compound pinion	15 teeth
(G)	beam wheel	90 teeth
(H)	take-up-roller circumference	15 05 in.

In this example, the ratchet wheel is turned by one tooth for each pick, and the amount of cloth taken up for each pick may be calculated as follows:

$$\text{pick-spacing} = \frac{1 \times 36 \times 24 \times 15 \times 15\ 05 \text{ in.}}{24 \times \text{CW} \times 89 \times 90} = \frac{1\ 015 \text{ in.}}{\text{CW}},$$

where CW denotes the number of teeth in the change wheel,

and thus:

$$\text{picks/in.} = \frac{\text{teeth in change wheel}}{1\ 015}.$$

The picks/in. in the cloth on the loom will therefore be slightly less than the number of teeth in the change wheel. If the cloth contracts $1\frac{1}{2}$% lengthways when it is taken off the loom, the picks/in. in the loomstate cloth will be equal to the number of teeth in the change wheel. Take-up motions for cotton weaving usually allow for some contraction, which is commonly of the order of $1\frac{1}{2}$% but may be as high as 3% in some cases.

Consider now the periodicities that may result from eccentricity. It is obvious that eccentricity of the take-up roller H or the beam wheel G will produce a period having a wavelength equal to the circumference of the take-up roller, which is 15 05 in. in this case. If either the compound wheel E or the compound pinion F is eccentric, the wavelength of the period will be 15/90 × 15 05 = 2 51 in. These and other possible periods are summarized below.

Periods due to eccentricity are:

G or H		15 05 in.
E or F	15/90 × 15 05	2 51 in.
C or D	24/89 × 2 51	0 68 in.
A or B	36/CW × 0 68	24 35/CW in.

The wavelenth of the period produced by eccentricity of wheels A or B will be 0 61 in. for a cloth with 40 picks/in. and 0 24 in. for a cloth with 100 picks/in. We see that eccentricity of any one of the wheels A–F inclusive will produce a period within the prescribed limits of $\frac{1}{8}$–10 in.

Consider now the effect produced if all the teeth on any one of the gear wheels are faulty, owing to faulty design, inaccurate construction, or wear. This will result in a continuous succession of periodicities, each of which will have a wavelength corresponding to one tooth of the wheel.

If either of the gears F or G has faulty teeth, the wavelength of the period will be 15 05/90 = 0 17 in. If either of the gears D or E has faulty teeth, the wavelength of the period will be 0 17 × 24/89 = 0 046 in. The various periods due to faulty teeth are summarized below.

Periods if all teeth on one wheel are faulty are: .17 × 15/89 = .028

F or G	15 05/90	0 17 in.	
D or E	15/89 × 0 17	0 029 in.	
B or C	24/CW × 0 046 .028	1 104/CW in.	.6 76/CW
A	36/24 × 24/CW × 0 046	1 506/CW in.	1.01/CW

Finally, if only one tooth in a gear were faulty, the variation in pick-spacing would extend over a distance corresponding to one tooth of that wheel, and it would recur with a frequency corresponding to one revolution of that wheel. For example, if one tooth in F were faulty, the variation in pick spacing would extend over a distance of 0 17 in., and it would recur every 2 55 in.

We can see from the above calculations that gears A–F, inclusive, are liable to produce dangerous periodicities if eccentric or eccentrically mounted and that faulty teeth in F or G are liable to produce periodicities just within the prescribed limits. The mechanism is clearly not designed to avoid periodicities, but eccentricity is a much more likely cause of faults than faulty teeth.

Take-up motions for weaving continuous-filament yarns usually incorporate a worm and worm wheel and a multiple-pawl ratchet wheel, the intention being to secure a more precise rate of take-up and to permit the weaver to

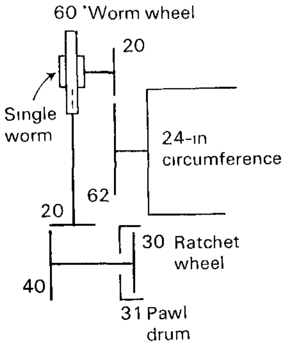


Fig. 6.11 The rayon take-up motion

adjust the cloth-fell position accurately to a fraction of a pick after pick-finding or after a prolonged loom stoppage. The gearing of a typical mechanism of this kind is shown in Fig. 6.11. In this, and in other similar mechanisms, there is no change wheel. The rate of take-up is altered by varying the amplitude of the oscillation of the drum that encircles the ratchet wheel and carries the multiple pawls. The periods that this mechanism is capable of producing are summarized below.

Periods due to eccentricity are:

take-up roller or 62 wheel		24 in.
worm wheel or 20 pinion	20/62 × 24	7 74 in.
worm or 20 bevel wheel	1/60 × 7 74	0 13 in.
ratchet wheel or 40 bevel wheel	40/20 × 0 13	0 26 in.

Periods due to faulty teeth are:

beam wheel or 20 pinion	24/62	0 39 in.
worm wheel	20/60 × 0 39	0 13 in.
20 or 40 bevel wheel	1/20 × 0 13	0 0065 in.
ratchet wheel	40/30 × 0 0065	0 0086 in.

Here again, there are several potentially dangerous periods, and the design of the mechaism cannot be considered satisfactory. Nevertheless, similar mechanisms have been widely used on looms designed specifically for weaving continuous-filament yarns, which demand the highest standards of uniformity of pick-spacing. They have been successful because of the high standard of engineering maintained in their manufacture. Such standards are expensive and could probably have been relaxed if more attention had been given to finding a design incapable of dangerous periodicities. This approach was taken in designing the take-up motion for the Shirley loom.

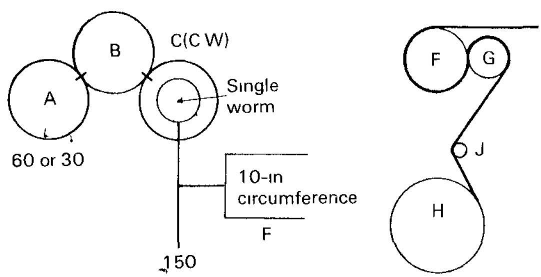


Fig. 6.12 The Shirley take-up motion

The gearing and cloth-control arrangements for this mechaism are shown in Fig. 6.12. The take-up roller is smaller than usual and replaces the breast beam. The pinion A normally has 60 teeth and is driven continuously by chain and sprockets at one-quarter of the loom speed. It drives the change wheel C through a carrier wheel B. A single worm on the same shaft as the change wheel drives a 150-tooth worm wheel on the take-up-roller shaft. The take-up roller has a circumference of 10 in. The periods that may arise from the mechanism are summarized below.

Periods due to eccentricity are:

take-up roller or worm wheel		10 in.
worm or change wheel	1/150 × 10 in.	0 067 in.
wheel A	60/CW × 0 067	4 0/CW in.

Periods due to faulty teeth are:

worm or worm wheel	1/150 × 10	0 067 in.
change wheel, A or B	1/CW × 0 067	0·067/CW in.

There are no potential periods between $\frac{1}{15}$ and 10 in., so the risk of dangerous periodicities is eliminated. The necessary conditions are quite simple. The circumference of the take-up roller must not be less than 10 in., and there must be a large step-down in speed between the wheel mounted on the take-up-roller shaft and the wheel it engages. This is provided by a worm and worm wheel, but it is not sufficient to incorporate these anywhere in the train of gears. They must be in the right place.

When the pinion A in Fig. 6.12 has 60 teeth, the picks/in. in the loom are equal to the number of teeth in the change wheel:

$$\text{picks/in.} = \frac{4 \times \text{CW} \times 150}{1 \times 60 \times 10} = \text{CW}.$$

The range of picks/in. can be extended without having to use very large change wheels by replacing pinion A with one with 30 teeth. The picks/in. in the loom will then be twice the number of teeth on the change wheel. Note that no allowance is made for lengthway contraction of the cloth when it is taken off the loom. It is unusual to make such an allowance in take-up motions designed for weaving continuous-filament yarns.

6.2.5 Positive Direct Take-up

In direct take-up, the cloth is wound onto the drive roller, and it is necessary to reduce the speed of rotation of the driven roller as the diameter of the cloth roll increases. To maintain constant pick-spacing, the angular speed of the driven roller must be inversely proportional to the diameter of the cloth roll. This type of take-up motion was once quite common in silk and rayon looms, but it is now obsolescent and need not be considered in detail. The roller was driven through gearing from a multiple-pawl drum and ratchet wheel, and the movement of a feeler roller in contact with the cloth roll was used to reduce the angular movement of the pawl drum as the cloth roll built up.

6.2.6 Changing the Number of Picks/in.

In many take-up motions, the number of picks/in. is determined by the number of teeth in the change wheel, and it has already been stated that there are advantages to be gained by being able to change more than one wheel in the system. Not only can a reduction be made to the range of change-wheel dimensions, but it is also possible to reduce the extensive stock of spare wheels required for any one loom.

Some loom makers have as many as four interchangeable change wheels per loom, so that a stock of only twelve wheels used in predetermined positions will give an extensive range of picks/in. or picks/cm. The Sulzer weaving machine uses this technique. It is necessary to read off the recommended wheels for each of the four positions from a table supplied with the instruction manual in order to produce the required number of picks in the fabric. All these wheels are mounted on precision-engineered splines to eliminate the possibility of eccentric mounting and thus the occurrence of weft bars. Such a fault is most likely to occur in association with change wheels because they are the most frequently adjusted wheels in the motion.

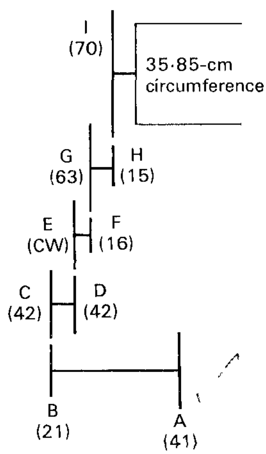


Fig. 6.13 The Picanol take-up motion

Picanol also use both these principles. Three of the interchangeable wheels (B, C, and D in Fig. 6.13) are pre-positioned to give a limited range of picks/cm or picks/in., which is directly related to the number of teeth in the change wheel as a whole, half, or quarter number. They can be rearranged to give another range of picks, and each of the basic positions is given in the instruction manual, but it is necessary to change the beam wheel I if it is desired that the teeth in the change wheel should have a relationship to the number of picks/in. With the arrangement shown in Fig. 6.13, the amount of take-up-roller movement per pick is:

$$\frac{1}{41} \times \frac{21}{42} \times \frac{42}{CW} \times \frac{16}{63} \times \frac{15}{70} \times 35{\cdot}85 = \frac{1}{CW} \text{ cm.}$$

Thus, in this case, the number of picks/cm is equal to the number of teeth in the change wheel.

When a multiple-pawl drum is used, as shown in Fig. 6.11, for example, it is possible to control the picks/in. by altering the angular movement of the pawl drum. In the arrangement shown in Fig. 6.14, the pawl drum has, say, 31 pawls

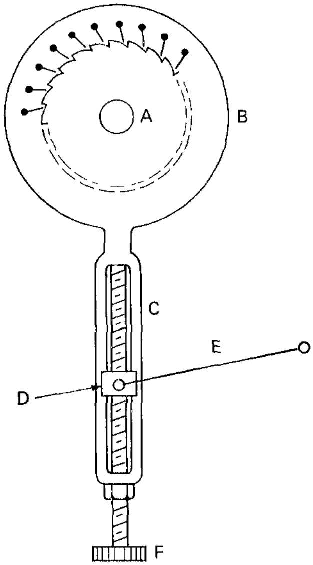


Fig. 6.14 The method of changing pick-spacing

acting on a ratchet wheel A with 30 teeth. At any one time, only one pawl is operative, and the minimum movement of the ratchet wheel is 1/31 of a tooth, or 1/930 of a revolution. According to the angular movement of the pawl drum, the ratchet may be turned by any multiple of this. The slotted lever C, which is attached to the pawl drum, has a screwed rod carrying a nut D, which is oscillated with a constant amplitude by a rod E attached to the sley sword. The nut D can be moved up and down in the slot by turning a knurled disc F. This controls the angular movement of the pawl drum, and the picks/in. can be engraved on the slotted lever and read against a mark on the nut D. This system has been used extensively in Hattersley and Saurer 100W looms, although in the latter case an alternative method of adjusting the picks/in. is available.

More recently, Rüti have introduced a take-up motion that includes a variable throw from the sley to a multiple-pawl-and-ratchet arrangement. This gives a limited range of picks/cm or picks/in. with a specific change wheel, so that the number of change wheels required for any one loom is low. Since the range in the variability of the throw is more limited, then the setting of the throw point is somewhat less critical than that shown in Fig. 6.11 because the range of throw adjustments in the latter case is responsible for the whole range of possible pick-settings that the loom is capable of weaving.

6.3 The Let-off Motion

6.3.1 The Negative Friction-type Let-off Motion

The negative friction-type let-off motion, as illustrated in a simple form in Fig. 6.6*A* and *C*, has been very widely used in one form or another for non-automatic weaving. Its chief virtue lies in its mechanical simplicity and low initial cost, but this is offset by disadvantages that become crucial under modern conditions of automatic weaving. It is liable to cause short-, medium-, and long-period variations in warp tension, which may result in corresponding varitions in pick-spacing.

Short-period variations in tension extending over a few picks may result from the 'stick–slip' action that is an inherent feature of the system, which causes the movement of the beam to be intermittent and sometimes irregular. Static friction is greater than dynamic friction: it requires more force to start a body sliding than to keep it sliding once it has begun to move. Consequently, let-off occurs when the action of the take-up motion, assisted by the reed at beat-up, has increased the warp tension sufficiently to overcome the static friction resisting the rotation of the beam. The beam begins to move and continues to move until the warp tension, reduced by the beam's movement and the withdrawal of the reed from contact with the fell, is no longer sufficient to overcome dynamic friction. The beam stops until the warp tension again increases sufficiently to overcome dynamic friction. How frequently this cycle recurs will depend on the difference between the coefficients of dynamic and static friction, the elastic modulus of the warp, and the rate of take-up. Except in weaving very coarse wefts with few picks/in., it is impossible to secure let-off every pick. If the mechanism is in good order, it is usually possible to ensure that it occurs regularly every two or three picks. Provided that it occurs regularly and not too infrequently, the fact that it occurs intermittently has no material effect on the pick-spacing. If let-off is both irregular and rather infrequent, the consequent tension variations may be sufficient to cause visible faults consisting of a thick place, formed as tension builds up, followed by a thin one, or even a crack, when let-off occurs.

It is instructive to consider a little more fully the part played by beat-up in the above cycle. While the beam is stationary and the warp tension is building up, the cloth fell will tend to move further away from the weaver because the cloth will stretch more and the fell will therefore move forward at take-up less than the average pick-spacing. The weaving resistance will therefore increase, which will cause a greater increase in the basic warp tension at beat-up. Hence, as the cycle proceeds, both the basic warp tension and the excess tension due to the cloth-fell displacement at beat-up increase. Their combined effect helps to ensure that the cycle recurs frequently.

Medium-period variations in warp tension occur as a result of changes in the coefficient of friction between the faces of the ruffles and the ropes, chains, or brake shoes. Patches of dirt, oil or rust, or rough or excessively smooth places on the friction surfaces will cause the coefficient of friction to vary. Suppose, for example, that the ruffle in Fig. 6.6*C* has a rough place across its width at one part of its circumference and that the rope makes one and a half turns round the ruffle. The rough patch will be continuously in contact with one turn of the rope, but it will be out of contact with the other half-turn for approximately half of each revolution of the beam. Thus, for approximately half the beam's revolution, the warp tension will be higher than that during the other half. The beginning of the high-tension period will be marked by a thick place and its end by a thin place. We therefore have two changes in pick-spacing within a length of cloth corresponding to one revolution of the beam. There may, however, be several rough patches at irregular intervals round the circumference of the ruffle. The changes in pick-spacing would be unlikely to be sudden, since they would extend over a length corresponding to the width of the rough patch, and the general effect would be one of shadiness, that is, of bands of irregular width of darker and lighter shade, imperceptibly merging into each other.

When the beam is full, the period over which the pattern will repeat will be about 2 m or slightly more. As the beam weaves down, this period will gradually decrease until it is about 50 cm or a little more when the beam is nearly empty. This reduction in wavelength usually tends to make the fault more noticeable, and, since the actual variations in tension tend to be magnified as the beam weaves down, it is towards the end of the beam that faults of this kind are most often noticed. This is also true of some automatic let-off motions, and it is one of the considerations that determine the minimum practicable empty-beam diameter.

Long-term tension variations from the start to the finish of the beam are inherent in the system. If in Fig. 6.6*C*, for example, the weights remain unchanged, the warp tension will increase steadily as the beam weaves down. If the full- and empty-beam diameters were 60 and 15 cm, respectively, there would be a 4:1 increase in tension from the full to the empty beam. The weaver is therefore expected to reduce the tension in the tight sides of the ropes or chains from time to time by removing weights or by sliding the weights nearer the fulcrum of the weight lever, or both. With a 4:1 ratio as above, the weaver would need to alter the weights by the correct amount fifteen times at regular intervals during the weaving of the warp in order to avoid tension variations of more than 10%. In practice, the weaver has to guess how much to alter the weights and is very unlikely to alter them at regular intervals or as often as fifteen times. Each time the weights are reduced, a thin place will result unless the weaver takes corrective action, and, however carefully this may be done, it will be impossible to avoid a fault in fabrics that are senstive to variations in pick-spacing. A gradual increase in warp tension, such as would result from not altering the weights over a considerable period, would not produce a visible fault, but the increase in tension would tend to reduce the warp crimp and hence to increase the weft crimp and produce a narrower fabric. After dyeing, there would almost certainly be a difference in shade between the beginning and end of the piece. It is therefore important to avoid large changes in warp tension even though they may be very gradual.

Many different types of let-off motion, all of them more complicated than the negative friction type, have been designed with a view to overcoming as far as possible the potential variations discussed above. Nearly all of them are automatic in the sense that, once they have been adjusted at the start of the warp, they require no attention from the weaver as the beam weaves down. They maintain, or seek to maintain, a constant tension and rate of let-off from the beginning to the end of the warp. A few are automatic but negative, the warp being pulled off against the frictional resistance, which is automatically reduced as the beam weaves down so as to maintain a constant warp tension. The majority are automatic and positive, the warp being delivered at a controlled rate by gearing that turns either the beam or the nip rollers through which the warp sheet passes. The examples that follow have been selected from many to illustrate the general principles.

6.3.2 *Controlled Negative Automatic Let-off*

The let-off motion developed many years ago at the Shirley Institute, and incorporated more recently into the Shirley loom, is one of the few examples of this type. It is 'controlled' because the rate of let-off is governed by the position of a floating backrest. It is 'negative' because the beam is turned by tension in the warp sheet acting against a frictional resistance. It is 'automatic' because its maintains constant warp tension from the start to the finish of the beam.

The essential features of the system are shown in Fig. 6.15*A* and *B* for the full and empty beam, respectively. The braking force is applied to a phosphor-bronze ruffle in the form of a V-pulley on one end of the beam by a steel-cored rope. Phosphor-bronze is used to eliminate rusting, and a steel core is necessary in this mechanism because it is important to prevent the rope from

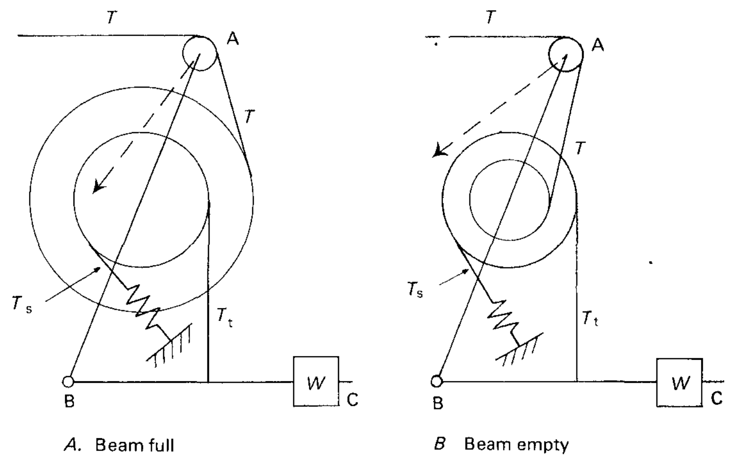


Fig. 6.15 The Shirley let-off motion

stretching. The slack side of the rope is attached to the loom frame though a strong coil spring. Its tight side is fixed to the weight lever BC. In the actual mechanism, the force exerted by the weight lever is transmitted indirectly to the shaft B, but the mechanism behaves as though the weight lever is keyed to the shaft B, which extends the full width of the loom. Two long levers AB, one at each side of the loom, are keyed to the shaft B and have bearings at their upper ends to carry the freely rotating back roller. If friction at the back roller is neglected, the tensions T on each side of the back roller are equal, and their resultant bisects the angle TAT. This resultant force R is tending to turn the lever AB anti-clockwise and therefore to raise the weight lever and reduce the tension in the tight side of the rope. When the system is in equilibrium, the tendency of R to produce anti-clockwise rotation and that of the weight to produce clockwise rotation are in balance.

If, now, the warp tension were to increase, R would also increase, and the levers AB would turn slightly anticlockwise, which would thus reduce the tension in the tight side of the rope and hence the braking force. Conversely, if the warp tension were to decrease, the lever AB would move slightly clockwise, which would increase the braking force. An increase or decrease in the warp tension is therefore immediately corrected.

When the beam is nearly empty, as in Fig. 6.15*B*, the resultant R now acts in a different direction and is more effective in rotating AB. As the beam weaves down, the tension in the tight side of the rope will therefore be progressively reduced. This is what is required to maintain the warp tension constant as the beam weaves down.

We have seen that the system is self-compensating for both short- and long-term variations in warp tension, but it does not follow that the compensation will be correct in magnitude. It may overcompensate or undercompensate for either or both effects. To ensure that compensation is correct in magnitude, it is necessary to study the geometry and mechanics of the system, from which the necessary conditions can be derived. If the slack side of the rope were connected directly to the loom frame, the system would be too sensitive. A small change in the position of the lever AB would result in a large change in warp tension. Its sensitivity is reduced to the required value by interposing a spring having suitable characteristics.

6.3.3 *Positive Let-off*

The principles and action of automatic let-off motions have been examined in some detail by Foster[23] in a Wira publication, 'Positive Let-off Motions', which is recommended for further study. Foster defines a positive let-off motion as one 'in which the beam is turned at a rate which tends to maintain a constant length of warp sheet between the fell of the cloth and the beam, the means of applying warp tension being separate from the beam-driving mechanism'. What is required may be summarized as follows:

(*a*) a means of applying tension to the warp sheet and for keeping the tension constant as the warp weaves down;

(*b*) a means of detecting small changes in the length of the warp sheet between the fell and the beam; and
(*c*) a method of utilizing these changes to vary the rate at which the beam is turned.

It is helpful to keep these requirements in mind in examining the action of a positive let-off motion and to look for the method used to achieve each of them.

A convenient means of sensing changes in the warp length is to use a floating backrest, which, if pressed upwards against the warp sheet, will move up and down as the warp length alters. By pressing the backrest against the warp sheet, we apply tension to the latter, so that a floating backrest can be used both to detect changes in the length of warp between the fell and the beam and to apply the required amount of tension to the warp sheet. For this reason, many automatic let-off motions use a floating backrest. A simple arrangement is

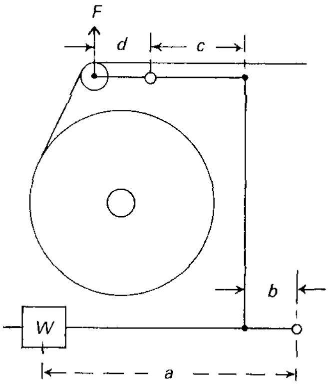


Fig. 6.16 Dead-weight tensioning

represented in Fig. 6.16, in which the force F exerted by the backrest on the warp sheet is given by:

$$F = \left(W \times \frac{a}{b} \times \frac{c}{d}\right) + f,$$

where f is a constant force due to the weight of the levers.

With such an arrangement, it can be shown that the warp tension will vary very little from the start to the finish of the warp (see Appendix). If, however, the slightly different system shown in Fig. 6.17 is used, the force F now acts at approximately 45° to the vertical, and it can be shown that the warp tension will decrease by 20% or more from the start to the finish of the beam. This is opposite to what occurs with a negative friction let-off. The reason for it can be understood from Fig. 6.17, in which A and B represent the forces acting at the back rest when the beam is nearly empty. The back roller is assumed to rotate freely, and the argument is simplified by assuming the forces to act at the axis of the back roller. This does not invalidate the argument.

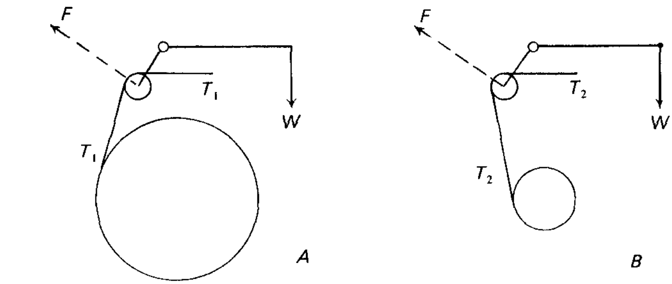


Fig. 6.17 The effect of beam diameter

In Fig. 6.18*A*, the resultant of the warp tensions T_1 at each side of the back roller is R_1, which is balanced by an equal and opposite force R_2. The magnitude of R_2 is determined by the force F acting at right angles to the short lever from which the back roller is suspended. In Fig. 6.18*B*, the magnitude and direction of the force F are unchanged, but, owing to the change in the angle of wrap of the warp sheet round the back roller as the beam weaves down, R_3 and R_4, although equal to each other, are different from R_1 and R_2 in both magnitude and direction. The net effect of these changes is that T_2 is clearly less than T_1. If, in Fig. 6.18, the short arms that carry the back roller were vertical, the force F would then act horizontally, and the decrease in warp tension as the beam weaves down would be greater, i.e., approximately 40% (see Appendix, Fig. 6.26).

The decrease in warp tension as the beam weaves down in the cases considered above is due entirely to the changing angle of wrap of the warp around the back roller. This decrease will not occur if the angle of wrap can be kept constant. One method of achieving this is to use a supplementary roller, as indicated in Fig. 6.19. Tension will remain constant in whatever direction the force F acts. This device is used, for example, on the Crompton & Knowles let-off motion for woollen and worsted looms.

6.3.4 The Beam-driving Mechanism

The movement of the back roller as it senses changes in the length of warp

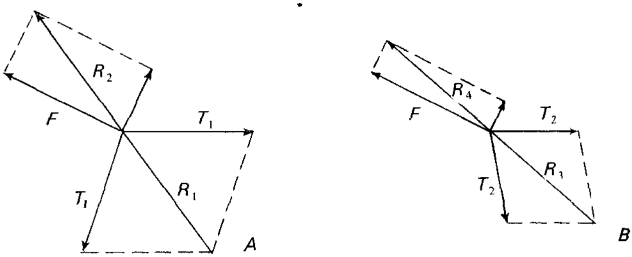


Fig. 6.18 Forces acting in Fig. 6.16

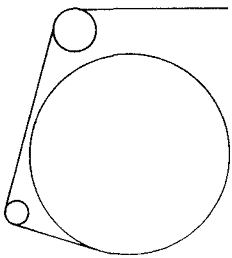


Fig. 6.19 The use of a supplementary roller

between the feil and the beam has to be converted into changes in the rate of rotation of the beam so as to restore the warp length to its original value. There are many ways of doing this. One of the simplest is illustrated in Fig. 6.20, which represents a principle rather than a particular mechanism. The beam is driven by a ratchet wheel R through gearing not shown in the diagram. The ratchet wheel is driven by a pawl, which receives an oscillating movement of constant magnitude at each pick, usually from the sley sword. A shield S en circles part of the ratchet wheel, and the amount the ratchet wheel is turned depends on the position of the shield. If the shield is moved anti-clockwise, the ratchet wheel is turned less, because the shield prevents the pawl from engaging the teeth of the ratchet wheel for a greater part of the pawl's movement.

Suppose, for example, that the warp length decreases. The weight lever A will be turned anti-clockwise, and the short lever B will turn the shield clockwise. The movement of the ratchet wheel will be increased, which will speed up the rate of let-off. Conversely, if the warp length increases, the shield will be turned anti-clockwise, which will slow down the rate of let-off. The

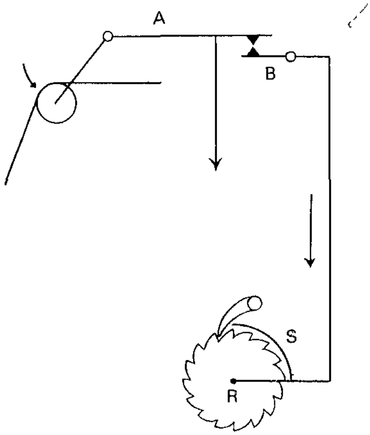


Fig. 6.20 A beam-driving mechanism

system automatically compensates for changes in the warp length, but the degree of compensation will be correct only if the mechanism is correctly designed and adjusted. Note that the weighting system maintains a constant warp tension while the rate of let-off is varying. The floating back roller senses changes in warp length and not changes in warp tension. This point is brought out in Foster's definition[23], which was quoted at the beginning of Section 6.3.3.

A mechanical system in which the movement of the back roller is required to supply the energy to effect an alteration in the rate of rotation of the beam can never be perfect. A measurable, and perhaps a quite substantial, force is required to move the levers connecting the back roller to the regulating device, against frictional resistance in their bearings and at their points of contact. Because the back roller is used to apply the warp tension, it and its associated system of levers must be robust. Such a system cannot be very sensitive, and we have already seen in Section 6.1.6 that its lack of sensitivity may modify the incidence of setting-on places. Because force is required to move the back roller and its associated levers, a decrease in warp length must inevitably cause an increase in warp tension before the system can respond. In other words, there will be fluctuations in warp tension due to the normal functioning of the system. These fluctuations can be minimized by reducing friction at the bearings of the levers and at their points of contact. In this connexion, it is interesting to note that Wira has suggested a way of eliminating the bearings altogether by using short strips of heavy cotton canvas to suspend the levers.

In any type of let-off motion in which the warp sheet passes over a back roller, the tension in the horizontal part of the warp sheet will be greater than that in the warp sheet between the back roller and the beam, because:

(*a*) if the back roller rotates, force is required to overcome friction in its bearings, and
(*b*) if the back roller does not rotate, there will be coil friction at its surface.

The difference between the tensions on either side of the back roller is of no consequence if the difference is constant. If the back roller rotates, faulty bearings, lack of lubrication, or a damaged roller may cause it to offer more resistance to rotation at one position. In other words, it may tend to stick once every revolution. When this happens, the warp tension will increase, and the pick-spacing will be closer than normal until the roller frees itself. Since there is usually some slippage between the warp sheet and the surface of the back roller, the wavelength of the fault produced in the fabric is likely to be rather greater than the circumference of the back roller.

When the back roller does not rotate, the tension in the horizontal part of the warp sheet will increase as the angle of wrap of the warp sheet round the back roller increases as the warp weaves down. The increase may be quite substantial (see Appendix, Fig. 6.28). This source of variation is avoided by the use of a supplementary roller (see Fig. 6.19). Foster[23] considers this to be the ideal arrangement and prefers a non-rotating back roller even without a supplementary roller on the grounds that tension variations associated with a non-rotating back roller are very gradual, whereas the tension variations associated with a faulty rotating back roller recur at regular, relatively short intervals. That they do indeed produce serious faults in the fabric has been amply demonstrated.

6.3.5 *Tensioning by Springs*

So far, we have considered only the weight-and-lever method of applying tension to the warp sheet. It is often more convenient to use coil springs at one or both sides of the loom. The spring(s) may be in tension, as shown in Fig. 6.21*A*, or in compression, as shown in Fig. 6.21*B*.

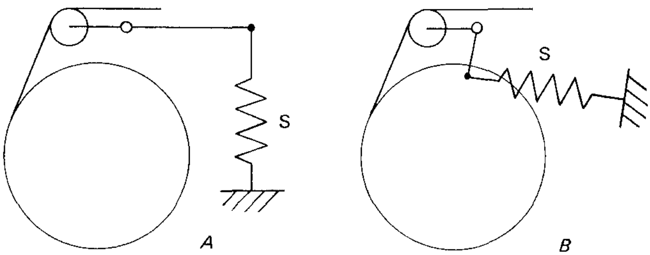


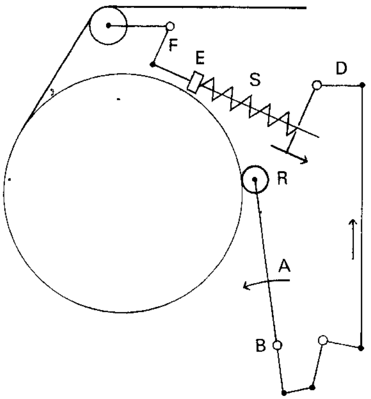
Fig. 6.21 Spring-tensioning

The beam-driving mechanism of any positive let-off motion must be so designed that the rate of rotation of the beam increases as the beam weaves down in order to maintain the same linear rate of let-off. If the ratio of the full to the empty diameter of the beam is 4:1, the beam must rotate four times as fast when it is nearly empty as when it is full. With the arrangement shown in Fig. 6.20, for example, this increased rate of rotation requires that the shield should gradually turn clockwise as the beam weaves down. Lever A must therefore turn anti-clockwise, and the back roller must fall gradually as the beam weaves down. The changing angle of wrap of the warp sheet around the back roller ensures that this will happen.

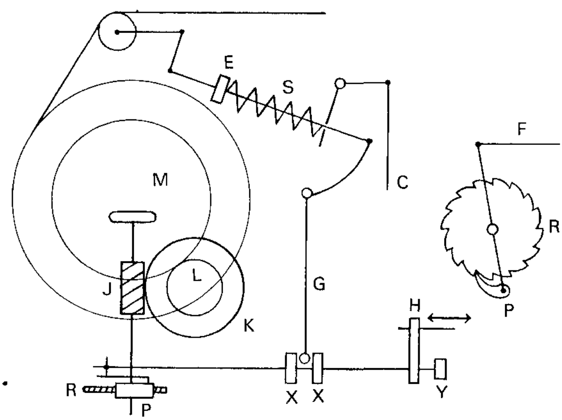
With a weight-and-lever tensioning system, this gradual lowering of the back roller does not result in any appreciable change in the warp tension, but, with spring tensioning, the downward movement of the back roller will increase the tension or compression of the spring and hence also the warp tension. The simple arrangements in Fig. 6.21 would therefore result in a steady increase in warp tension as the beam weaves down. Some method of compensating for this is clearly required. This is usually achieved by a beam feeler that maintains contact with the surface of the yarn on the beam. The movement of the feeler is transmitted to one end of the spring in such a way as to prevent an increase in stretch or compression as the beam weaves down. The practical application of this idea is illustrated in the two examples that follow.

6.3.6 *The Bartlett Let-off Motion*

The Bartlett let-off motion and the Roper let-off motion, which is similar in



(*A*) Tensioning mechanism



(*B*) Beam drive

Fig. 6.22 The Bartlett let-off motion

principle but different in detail, are commonly used on Northrop and other makes of loom for weaving cotton and rayon cloths. A simplified diagram of the tensioning arrangements is shown in Fig. 6.22*A*. Tension is applied to the warp through a floating back roller by means of springs S, one at each side of the loom. The short levers that carry the back roller are substantially horizontal, so there is no need to compensate for variations in tension due to the changing angle of wrap of the warp sheet around the back roller as the beam weaves down (see Section 6.2.3. and Appendix). It is, however, necessary to compensate for the compression of the springs that occurs as a result of the downward movement of the back roller.

At the side of the loom not shown in Fig. 6.21, a similar spring S is compressed between a collar E and a fixed stop on the loom frame. On that side of the loom, the spring will be compressed more as the beam weaves down. It is therefore necessary to reduce the compression of the spring on the side of the loom shown in Fig. 6.22*A* by an equal amount, so that the sum of the forces exerted by the two springs will remain constant. This is achieved as follows.

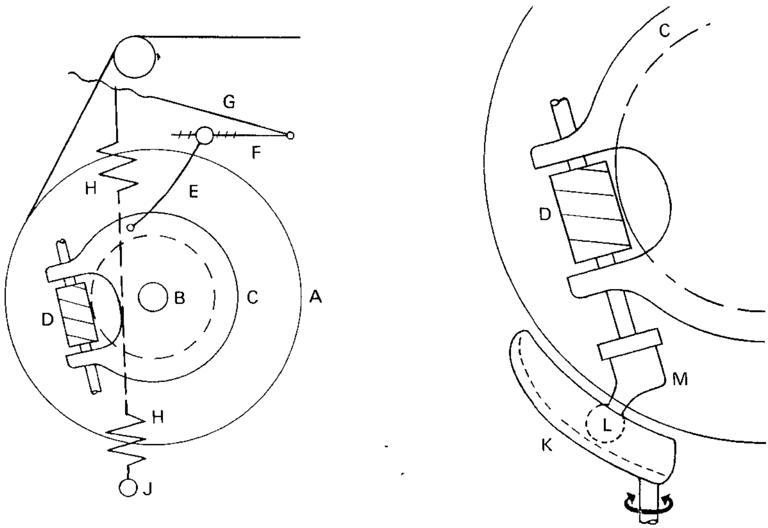
The screwed collar E allows the tension of the springs to be adjusted to give the required tension at the start of the warp. As the beam weaves down, lever A turns anti-clockwise on its pivot B. This moves the rod C upwards through a series of links and turns the elbow lever D anti-clockwise. The lower end of D is slotted over the rod that carries the spring, which is compressed between the collar E and the slotted lever D. The effect is to lengthen the spring as the beam weaves down by the same amount as the uncompensated spring on the other side of the loom is shortened.

The beam-turning mechanism is shown in Fig. 6.22*B*. The beam is driven by a ratchet wheel R on a short vertical shaft, which also carries the worm J, which drives the worm wheel K. A spur gear L on the same shaft as the worm wheel drives the large beam wheel M, which is fixed to one of the beam flanges. The ratchet wheel R is turned by a pawl operated by the rod F. This is seen more clearly in the inset plan view in Fig. 6.22*B*. The rod F is moved to the right each time the sley comes forward by a projection H, which receives a reciprocating motion of constant amplitude from the sley sword. The position of the rod F when it is not being acted upon by the projection H is governed by the position of the back roller through collars X, X and the rod G.

As the back roller moves downwards as the warp weaves down, the rod carrying the collar E will move to the right, and the rod G will move the rod F to the left. This will bring the collar Y closer to H, and the rod F will receive more movement, turning the ratchet wheel more. Superimposed on the gradual, steady movement of the rod F to the left, there will also be small, frequent changes in its position to compensate for short-period variations in the length of the warp. The mechanism is therefore fully automatic, but the compensation it provides will be correct only if the mechanism is correctly designed and adjusted. In the actual mechanism, several points of adjustment not shown in Fig. 6.22*A* and *B* are provided. These are necessary to cover a wide range of warp tensions and rates of take-up. Foster[23] discusses the setting of the mechanism, and the conditions under which it may fail to provide adequate compensation, in some detail.

6.3.7 The Saurer Let-off Motion

This is an example of a positive automatic let-off motion that does not use a floating back roller. The back roller is carried in fixed bearings and may rotate or not as required. Changes in warp length are sensed and corrected by oscillation of the beam. A simplified diagram of the tensioning arrangements is shown in Fig. 6.23*A*. The whole of the mechanism is at one side of the loom. The worm wheel B is fixed to the beam flange A and is inside a hollow casting C, which is free to turn on the beam shaft. A worm D drives the beam through the worm



(*A*) Tensioning system (*B*) Beam drive

Fig. 6.23 The Saurer let-off motion

wheel B. The position of the casting C is governed by the link E and the levers F and G. The upper end of a coil spring H is attached to the free end of the lever G in such a way that its distance from the fulcrum of lever G can be reduced as the beam weaves down. The other end of the spring H is attached to the loom frame. The point of attachment of the link E to the lever F is adjustable. This provides a fine adjustment for the level of warp tension established at the start of weaving. Two alternative points of attachment of the lower end of the spring to the loom frame at J provide a coarse adjustment. This is an example of the use of a spring in tension.

The tension in the warp sheet is derived from the tension in the spring H, which endeavours to turn the beam anti-clockwise, through levers G and F, the link E, the casting C, the worm D, and the worm wheel B, which is fixed to the beam. Variations in warp length will therefore result in slight oscillations of the beam. Rotation of the beam independently of the casting C is obtained by turning the worm D as shown in Fig. 6.23*B*.

The lower end of the worm shaft carries a multiple-pawl-and-ratchet-wheel assembly housed in a casting M. A projection on M carries a ball L, which is in a channel in the casting K. This casting receives an oscillating movement of constant amplitude from the sley sword. This motion is transmitted to the ratchet assembly M by the ball L, and the worm D is turned so as to rotate the beam. The amount by which the worm turns for each pick depends on the position of the ball L in the channel of the casting K. Suppose, for example, that the warp length decreases. This will cause the whole beam assembly to turn clockwise, the ball L to move to the left, and the movement it receives from the casting K to increase. This will increase the rate of let-off and tend to restore the warp length to its original value.

As the beam weaves down, the take-up motion will endeavour to turn the beam more rapidly, and the whole assembly will turn gradually clockwise, steadily move the ball L further from the pivot of the casting K, and increase the rate of rotation of the beam to compensate for its decreasing diameter. The system is therefore fully automatic with regard to the maintenance of a constant warp length. If, however, the point of attachment of the top of the spring H to the lever G were to remain fixed, the warp tension would increase steadily as the beam wove down. There are two reasons for this. Firstly, the casting C turns clockwise as the beam weaves down. This raises the lever G and stretches the spring more. Secondly, even if the spring were not stretched more, the warp tension would increase for the same reason as in a negative friction-type let-off. To maintain the warp tension constant, it is convenient to move the point of attachment of the spring H gradually nearer the fulcrum of the lever G as the beam weaves down. This is achieved by means of a beam-feeler mechanism not shown in the diagrams.

6.4 Temples

6.4.1 Introduction

The function of a temple unit is to grip the cloth by passing it round rollers so that it is held at the fell to the same width as the warp in the reed. Normally, after a shed change and the subsequent beat-up, the weft will crimp, which will cause the fabric to contract, but, if this occurs too soon at the fell, then the ends at the side of the fabric will be pulled inwards and cause a sideways pressure on the reed wires, which may result in end-breaks or damaged reeds. This point was considered in more detail in Section 6.1.2.

Temples may be mounted on a spring-loaded bar extending across the width of the loom, on the loom frame, or on a fixed bar extending across the width of the loom. In the second and third of these cases, each temple unit is individually spring-loaded, the spring-loading being necessary in the event of a shuttle trap in the vicinity of the temple if damage to the reed, shuttle, and the temple is to be avoided. The position of the temple is also critical if the maximum gripping potential is to be achieved. The underside of the temple should clear the top edge of the raceboard by a maximum distance of 2 mm (with the loom at 90 or 270°), and the gap between the reed and the temple with the sley in its most forward position (0°) should be similar. The sideways position of the temple is governed by the fact that the outside edge of the selvedge should be just on, or slightly overlapping, the last gripping point on the temple.

The amount of weftway contraction and the type of material being woven will govern the type of temple being used and also whether it is necessary to have a second temple about 50 mm behind the front one, although this latter requirement is usually only necessary for very wide looms weaving fabrics with a high percentage weft crimp (e.g., high-quality sheetings).

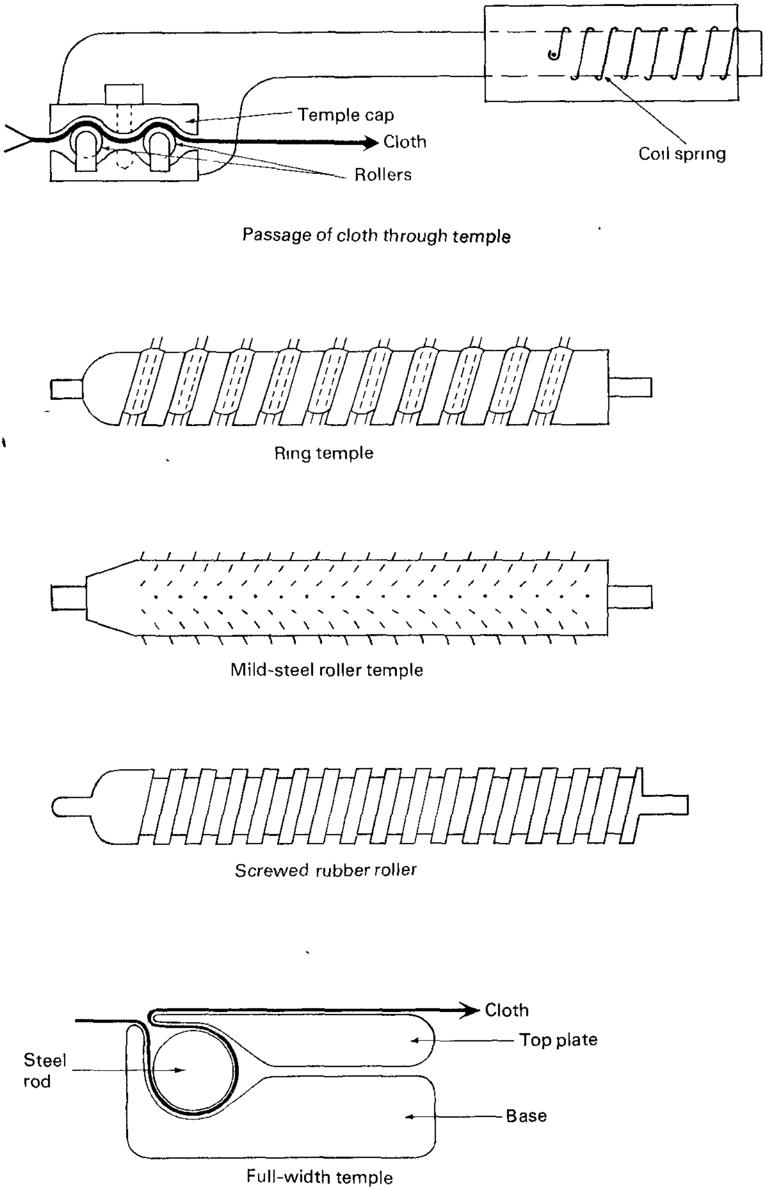


Fig. 6.24 Temples

There are three main types of temple: ring, roller, and full-width (see Fig. 6.24).

6.4.2 Ring Temples

Ring temples are the most efficient for holding the cloth to the required width. They consist of a shaft on which are mounted several spiked rings spaced with washers, which may be made of metal or (preferably) nylon. Each ring will have about 30 pins, and each temple may have between one and 30 rings, depending on the amount of weftway contraction and the width of the fabric. A good guide is one ring for every 10 cm of cloth width.

Although ring temples can be used with a wide range of light, medium, and heavy fabrics, it must be borne in mind that they are always liable to damage the fabric if they are not maintained to a satisfactory standard. The pins are long and easily damaged through misuse, and dirt or waste yarn between the rings and washers may prevent a ring from rotating freely. In either case, the fabric will be damaged as it is dragged over the faulty temple. Furthermore, the pins tend to be rather severe in their treatment of continuous-filament yarns during weaving, and for this type of fabric the roller temple is usually preferred.

6.4.3 Roller Temples

There are usually two rollers in the basic temple, and these may be made of either mild steel with raised teeth or boxwood with inserted steel pins. The number of teeth or pins in the temple, the relative diameter of the pins, and the arrangement of the teeth in the roller (i.e., diamond formation, straight-line, or spiral-line) is determined by the fabric being woven. This type of temple is suitable for light- and medium-weight fabrics woven from spun staple-fibre yarns and having relatively little weftway contraction.

In weaving fabrics from continuous filament yarns of similar weight, the same type of temple can be used, provided that the fabric is not damaged in any way by the pins, but, since these fabrics are damaged easily, it is more usual to use rubber-covered rollers, whose surface may be plain, grooved, or screwed. For fabrics woven from continuous-filament yarns and having a high weftway contraction, the gripping power of the temple may be improved by using a rubber-covered roller with one temple ring at the outside edge. This ring will act on the selvedge so that any resulting marks will not be detrimental to the final appearance. Such a ring is also useful even if the contraction is not particularly high, since it will allow the temple to retain a hold on the fabric even when the temple cover is removed.

6.4.4 Full-width temples

This type of temple is used for fabrics that must be free from temple marks or temple strain (e.g., nylon parachute fabric). The temple controls the fabric close to the fell and so reduces the amount of fell movement at beat up, and thus there is less increase in fabric tension and a corresponding reduction in end-breaks.

The temple consists of a trough-shaped base, a steel rod, and a top plate that is attached to the bottom plate by studs, whose position is adjustable in angled slots. The top or cover plate is set to almost cover the trough portion, and the fabric passes over the lip of the base plate, round the roller, and out over the top plate. The rod cannot pass through the slot, but it is pulled upwards as the reed recedes to trap the cloth. At beat-up, the cloth becomes slack owing to pressure of the reed on the cloth fell, and the rod falls so that take-up can occur before the cloth becomes trapped again.

APPENDIX

Forces Acting at a Floating Back Roller

6.A.1 *The Case of a Freely Rotating Back Roller*

Assume that the force F in Fig. 6.25 does not vary in direction or magnitude as the beam weaves down. It will not vary in magnitude if the beam-feeler device compensates correctly, but it will vary slightly in direction owing to the

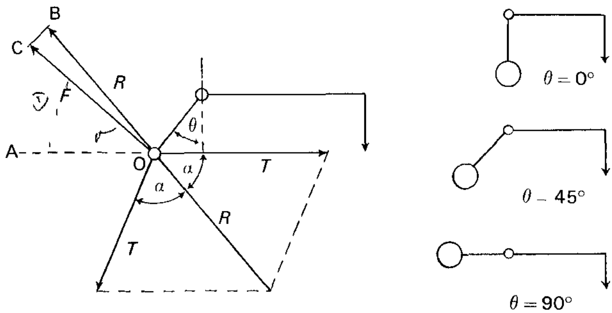


Fig. 6.25 A freely rotating backrest

gradual fall of the back roller as the beam weaves down. This is neglected in what follows.

The tension in the warp sheet is given by:

$$T = \frac{R}{2\cos\alpha}. \tag{6.3}$$

We also have:

$$\widehat{AOB} = \alpha \qquad \text{and} \qquad \widehat{AOC} = \theta,$$

so that:

$$\widehat{BOC} = (\alpha - \theta)$$

and:

$$R = \frac{F}{\cos(\alpha - \theta)}. \tag{6.4}$$

Combining Equations (6.3) and (6.4), we have:

$$T = \frac{F}{2\cos\alpha\cos(\alpha - \theta)}. \tag{6.5}$$

Special cases occur when $\theta = 0$ and $90°$, when Equation (6.5) simplifies as follows.

When $\theta = 0°$:

$$T = \frac{F}{2\cos^2 \alpha}. \tag{6.6}$$

When $\theta = 90°$:

$$T = \frac{F}{2\sin\alpha\cos\alpha} = \frac{F}{\sin 2\alpha}. \tag{6.7}$$

By plotting T/F for selected values of θ for a range of values of α, we can see how the warp tension varies as the beam weaves down. This has been done in Fig. 6.26 for $\theta = 0$, 45, and 90°. Foster[23] gives the same curves without

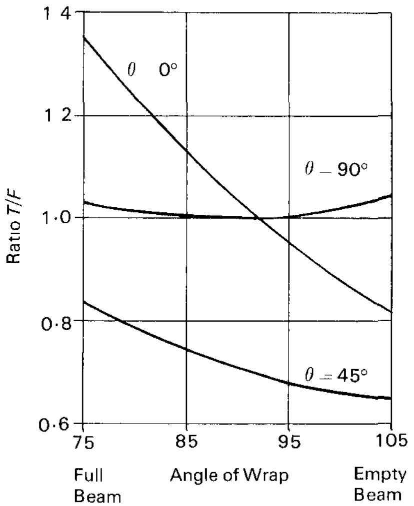


Fig. 6.26 The effect of a freely rotating backrest

explaining precisely how they were obtained. As stated in the text (Section 6.3.2), a value of $\theta = 90°$ as in Figures 6.15, 6.20, 6.21, and 6.22 produces very little variation in tension as the beam weaves down. This is clearly the best arrangement for a freely rotating back roller.

6.A.2 The Case of Friction at the Back Roller

This is represented in Fig. 6.27, in which $T_2 > T_1$ and $\beta < \alpha$. As in Fig. 6.25:

$$R = \frac{F}{\cos(\alpha - \theta)}.$$

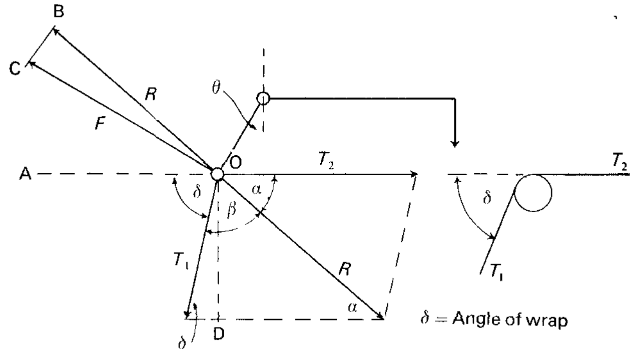


Fig. 6.27 Friction at the backrest

We also have:

$$OD = R \sin \alpha = T_1 \sin \delta.$$

Hence:

$$T_1 = \frac{R \sin \alpha}{\sin \delta} = \frac{F \sin \alpha}{\cos (\alpha - \theta) \sin \delta}. \tag{6.8}$$

But, since $(180 - \delta) = (\alpha + \beta)$, and because $\sin (180 - \delta) = \sin \delta$, we may write Equation (6.8) as follows:

$$T_1 = \frac{F \sin \alpha}{\cos (\alpha - \theta) \sin (\alpha + \beta)}. \tag{6.9}$$

In any triangle, the length of a side is proportional to the sine of the opposite angle, so that:

$$\frac{T_1}{\sin \alpha} = \frac{T_2}{\sin \beta},$$

and

$$T_2 = \frac{T_1 \sin \beta}{\sin \alpha},$$

from which:

$$T_2 = \frac{F \sin \alpha}{\cos (\alpha - \theta) \sin (\alpha + \beta)} \times \frac{\sin \beta}{\sin \alpha},$$

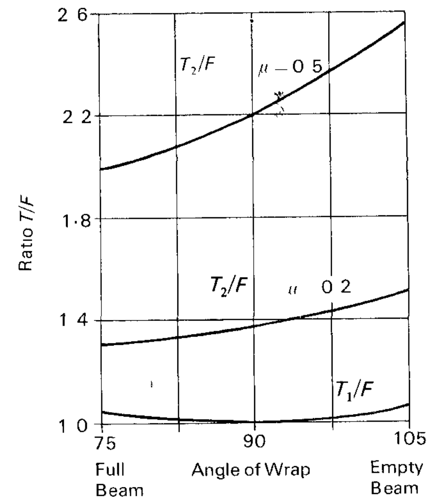
i.e.:

$$T_2 = \frac{F \sin \beta}{\cos (\alpha - \theta) \sin (\alpha + \beta)}. \tag{6.10}$$

In order to use Equations (6.9) and (6.10) to plot curves corresponding to those in Fig. 6.26, we need to make some assumptions about the frictional forces acting at the back roller. If it rotates against a constant frictional resistance due to its bearings, then T_2 will be greater than T_1 by an amount that could be determined by measuring the force needed to rotate the roller. This force should be small compared with the tension in the warp sheet and should have little effect on the shape of the curves. If, however, we use a non rotating back roller as Foster[23] recommends, we have the effect of coil friction, and:

$$\frac{T_2}{T_1} = \frac{\sin\beta}{\sin\alpha} = e^{\mu\delta}.$$

We may select the special case $\theta = 90°$, which we know gives the minimum tension variation with a freely rotating back roller. If we then select certain values between which the coefficient of friction μ is likely to lie, we can calculate the ratio T_2/F for a range of values of δ. The results of these calculations are plotted in Fig. 6.28, together with the curve for T_1/F, which is the same as that for $\theta = 90°$ in Fig. 6.26.



$\theta = 90°$ (all)

Fig. 6.28 The effect of a non-rotating back roller

The effect of a coefficient of friction of 0 2 is negligible, but that of a value of 0 5 is considerable. Values approaching this may be encountered in weaving synthetic-fibre yarns, for which, however, a rotating back roller is normally recommended. Since the trend of the curves in Fig. 6.28 is opposite to that in Fig. 6.26, this suggests that, with a non-rotating back roller, the variation in tension as the beam weaves down could be minimized by using a value of θ between 45 and 90°, the exact value depending on the expected coefficient of friction and the angles of wrap.

CHAPTER 7

Stop-motions

7.1 Warp Protection

7.1.1 *Introduction*

It is possible that the shuttle may fail to reach the opposite box as a result of a faulty pick or some obstruction in the warp shed. In this case, many ends may be broken if the sley comes forward to the beat-up position with the shuttle lying between the top and bottom shed lines and resting against the reed. It is desirable to prevent this from occurring.

The most popular motion available for this purpose is the fast-reed warp protector, although a method known as the loose-reed warp protector was very popular for many years, particularly on Lancashire looms.

7.1.2 *Fast-reed Warp Protector*

This mechanism utilizes the fact that the swell, acting as a brake, is displaced by the shuttle in bringing it to rest. The swell will normally attain its maximum displacement, but, if the shuttle fails to reach its correct rest position or if it rebounds owing to insufficient checking or too much picking force, then the swell will not receive its maximum displacement, so that the finger shown in Fig. 7.1*a* will not be pushed back as far as it should be. This will not let the stop-rod rotate sufficiently to allow the dagger to clear the frog steel.

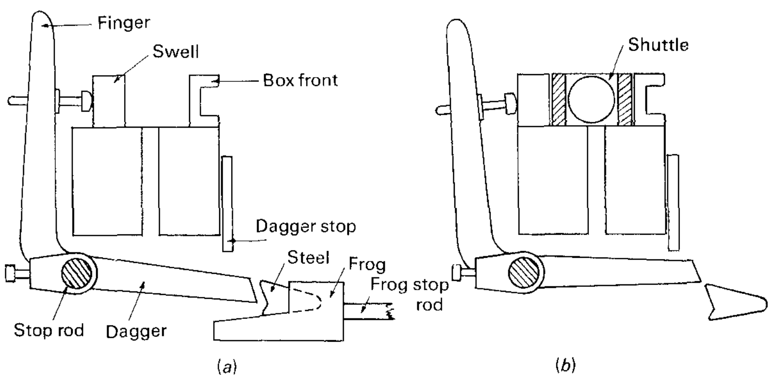


Fig. 7.1 A fast-reed warp-protector motion

The sudden impact of the dagger on the steel will bring the loom to an almost instantaneous halt, which is commonly known as the *bang-off* because of the resulting noise. The steel is directly connected to the starting handle, so that the loom brake is immediately applied to disengage the clutch or switch off the motor depending on the method of drive, and thus overheating of these parts will be prevented.

Absorbing the shock of the sudden stop presents a much greater problem, which is overcome in older Northrop and Rüti looms by the use of a pair of heavy flat springs at each side of the loom. These are fixed to the bottom of the front of the loom frame, and the frog stop-rod is in contact with them near to their top end. In other looms, such as the Picanol, Draper, and Crompton & Knowles looms, the heavy coil springs are built into the crankarms.

If the shuttle is correctly boxed at the end of its flight, then the swell and thus the finger will be pushed back, the rod will rotate, and the dagger will be raised clear of the steel (Fig. 7.1*b*), so that the loom will continue running.

A dagger stop is introduced to limit the amount of upward movement of the dagger to ensure that the swell will not be forced out of contact with the shuttle for too long owing to the sudden force of impact.

Bang-off actually occurs when the sley is travelling at its maximum velocity, and, although the brake is applied very quickly and the shock is absorbed, problems can arise if the sley is allowed to dwell in its back position for a longer period of time than is usual (as for example, when the sley is cam-driven), or if the speed of the loom is increased without any improvement in the performance of the shock absorbers. The force of bang-off will increase considerably because the sley will have to reach a higher maximum velocity, and this may cause the parts that are taking the full force of the shock to fracture.

7.1.3 Loose-reed Warp Protector

It has already been stated (Section 2.6) that the shuttle must come to rest in the box of a fast-reed loom by 270° if a trap is to be avoided, and for this purpose the shuttle must make its initial contact with the swell by 250°. This presents a very serious limitation to the amount of time available for shuttle flight, which was not present to the same extent with the loose-reed warp protector.

With this mechanism, the reed was supported at beat-up by the dagger passing under a heater fixed to the loom frame. In the back position, a light spring was sufficient to support the reed while the shuttle passed from one side of the loom to the other. If a shuttle became trapped, however, the bottom baulk of the reed was forced out of its support, and beat-up was no longer possible. The time of arrival of the shuttle in the opposite box was no longer dependent on the checking mechanism, and up to another 20° became available for shuttle flight, which thus made higher loom speeds possible or alternatively allowed lower shuttle speeds to be used for the same loom speed.

Although this method has the obvious disadvantage of straining the threads in the area in which the shuttle is trapped because of the presence of the shuttle and because the loom does not stop instantly but continues to turn over for a few picks until the weft fork stops it) there are certain attractions that have been incorporated by Rüti in their BANL and C looms. The loose reed hinged at its top baulk is still used to detect the presence of the shuttle as the reed comes forward to beat-up, but, if the reed receives the slightest pressure from a trapped shuttle (Fig. 7.2*a*), then the dagger will be raised to strike the heater, which is part of the starting-handle unit, and the loom will be knocked off instantly. The reed is again supported during shuttle flight by a spring on the

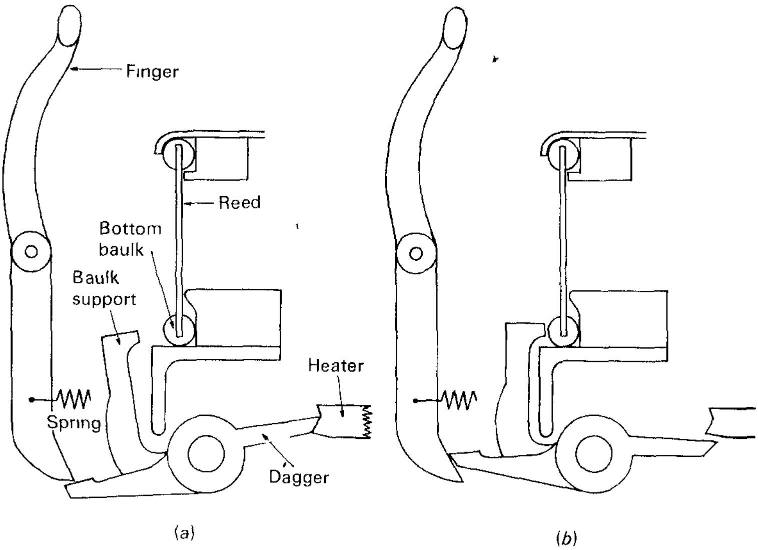


Fig 7.2 A loose-reed warp-protector motion

finger that supports the back arm of the dagger, and at the front position the dagger passes under the heater (Fig. 7.2*b*), so that the reed is supported to withstand the full force of the beat-up. The extra time for shuttle flight is therefore gained, and it is possible to have a softer or gentler pick, which will reduce wear, or higher loom speeds, or a combination of both.

7.1.4 Electromagnetic Warp Protection

The relatively short time available for stopping the loom after detection will always provide a problem, and, in an effort to find the answer, Crompton & Knowles developed in their C 7 loom a system that used a magnet to energize a coil in the sley at a point at which the shuttle should pass at the same time on each pick, irrespective of its direction of flight. More recently, the idea has regained favour, first with Northrop in the Sensamatic loom and then with Picanol (MDC), Saurer (Versaspeed), and Rüti (C 1001), using variations of the principle.

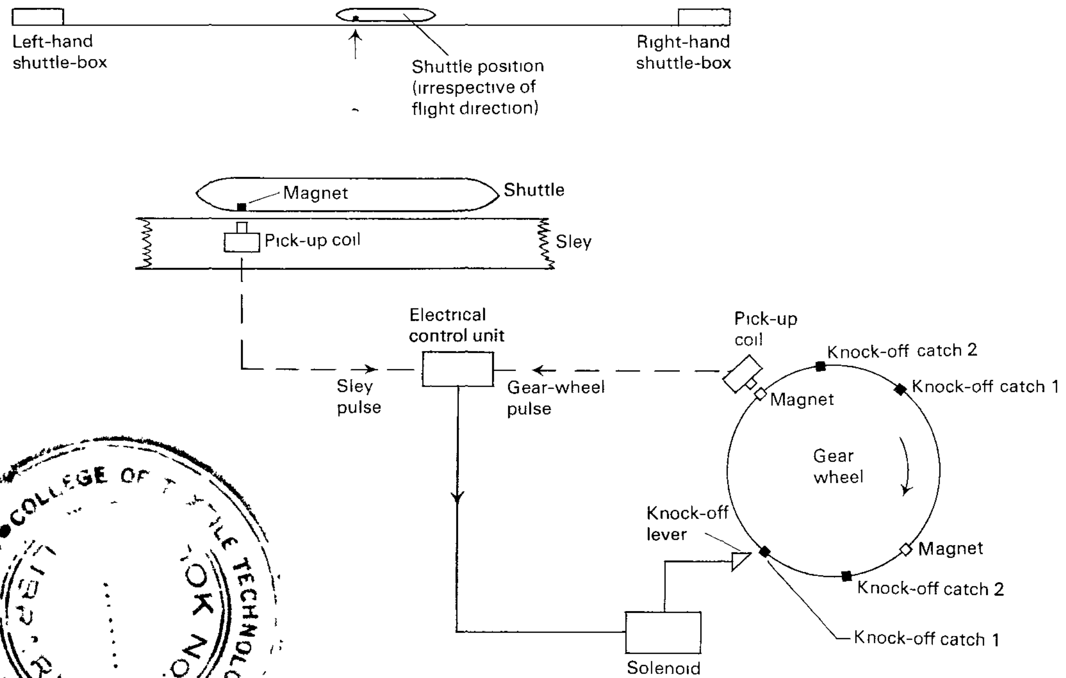


Fig. 7.3 An electromagnetic warp-protector motion

The position of the coil in the sley must be offset from the centre of the sley because it is only possible to carry the magnet in the end of the shuttle opposite to the shuttle eye. In the Sensamatic loom, the passage of the shuttle over the coil causes a pulse to be fed to the electrical control unit (Fig. 7.3). This pulse must alternate with a second pulse, generated by a magnet mounted on the loom-shaft gear wheel and thus occurring at a fixed time in each loom cycle. Any break in the sequence of these pulses caused by a late passage or non passage of the shuttle will activate the solenoid, so that the knock-off lever will be positioned in the path of knock-off catch 1, and the loom will be brought to rest. Other electrical feeds to the solenoid will cause the same catch to bring the loom to rest in the same position for a warp break or as a result of operating the manual push-button. A second knock-off catch (2) is used for the weft stop motion.

The knock-off-catch positions are adjustable and dependent on the type of loom and the loom-timings, which govern the position at which it is desirable to bring the loom to rest. Since the gear wheel used is mounted on the bottom shaft of the loom, it is necessary to have two magnets and four knock-off catches, which are diametrically opposed to each other in pairs.

The efficiency of the warp-protector motion is now no longer dependent on the shuttle's being in a specific position at a specific time to operate the mechanism instantly, so that, not only is the bang-off shock eliminated, but there is also more time available for shuttle flight. The governing factor regarding the time of shuttle arrival in the box is now the amount of space available for the shuttle as it leaves the shed. Furthermore, in the fast- and loose reed methods of warp protection, there is always the possibility of some damage to the cloth fell when the shuttle is trapped. This is completely avoided with electromagnetic warp protection because the loom is stopped before trapping can occur.

7.2 Warp Stop-motions

7.2.1 General Operation

The purpose of warp stop-motions is to stop the loom when a warp end breaks so that an unrepairable fault is avoided. It is a fact that a slight fault is left in the cloth after a warp break because the broken end is not pieced to the thread that has already been woven into the cloth, and this thread is therefore not continuous in the fabric. In high-quality worsted fabrics, such faults must be repaired after weaving.

If the breaking of the warp thread is not detected immediately, then the loose thread will tend to become entangled round adjacent threads, which will cause more threads to break and possibly create a fault known as a *float* in the fabric.

Warp stop-motions may be mechanically or electrically operated, but, whichever system is chosen, it is necessary to support a thin strip of metal, known as a *drop wire, dropper,* or *pin,* on each end so that, when the thread breaks, the drop wire will fall and operate the mechanism.

The drop wire may vary in design depending on two entirely different sets of

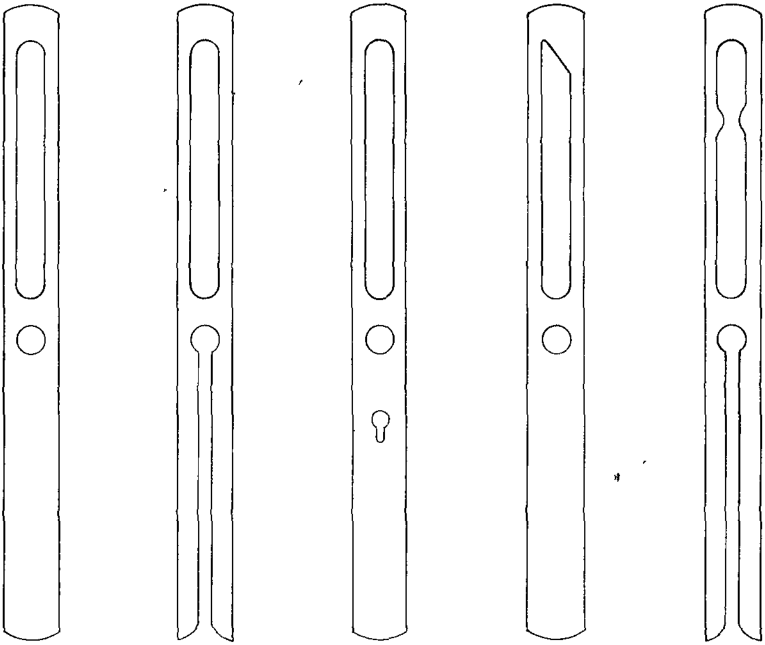


Fig. 7.4 Types of drop wire

circumstances. Fig. 7.4*a* shows a drop wire suitable for use when the thread is drawn through the wire in the drawing-in department, whereas the drop wire in Fig. 7.4*b* is used when the wires are dropped onto the ends on a special pinning machine after the ends have been drawn through the healds and reed or after the warp has been gaited in the loom. These open-ended droppers are more quickly placed on the threads, but they may tend to spread over more than one thread, so that they will not fall if only one of the threads breaks. This circumstance will only occur as a result of operative carelessness, but it is nevertheless an obvious disadvantage.

If the drop wire is to be threaded on a Barber–Colman drawing-in machine, then it is necessary for a key-way to be cut into the drop wire, as shown in Fig. 7.4*c*.

When drop wires are to be used in association with an electrical warp stop-motion, a modified shape of cut-out is essential so that contact between the two parts of the electrode is ensured. Two common shapes are illustrated in Fig. 7.4*d* and *e*, but the shape of the cut-out is dependent on the design of the electrode. The shape of this top cut-out is entirely independent of whether the drop wire is of the closed or open-end type, and a key-way may again be necessary if the wire is to be used on warps to be drawn on the Barber–Colman machine.

The size of a drop wire can vary quite considerably depending on the type of loom, some wires for terry looms being quite long. The weight of a wire should be such that it will fall quickly but should not be so great that it will damage the thread.

7.2.2 Mechanical Warp Stop-motion

The bar unit of the stop-motion passes through the large cut-out at the top of the drop wire. The bar is in three parts, the top of each being castellated. The two outside ones are fixed, and the inside one is made to reciprocate. The amount of reciprocation should be equal to one complete castellation, and, in order to ensure that a wire will fall into the lowest cut-out point of both bars, it is necessary for the centre moving bar to be slightly higher than the two outside ones.

When the wire falls into this lowest position, it limits the movement of the centre bar and thus the parts of the mechanism that have been causing the reciprocation, which, on a Northrop loom, originate from an eccentric on the bottom shaft, whereas Saurer use a cam method of drive.

In the Northrop method, shown in Fig. 7.5, the centre bar is reciprocated by the fulcrum lever, which is held in the bottom of the rocking diamond-shaped

Wrong Picture.

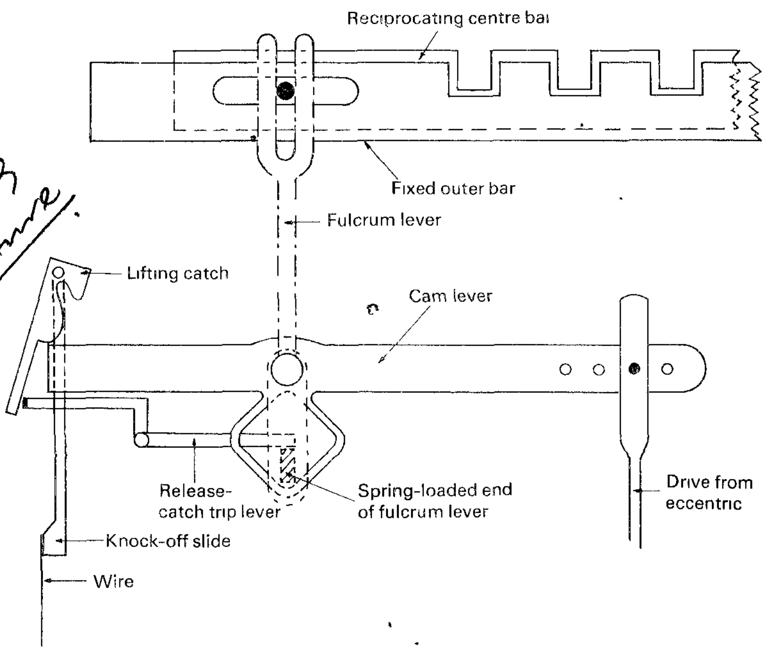


Fig 7.5 A mechanical warp stop-motion

cut-out of the cam lever by spring-loading. When the bar is trapped, this lever is pushed up against its spring-loading because it cannot rock sideways, and this raises the release-catch trip lever so that the outside end of this lever falls and allows the lifting catch to slip over the outside end of the cam lever. As this end of the cam lever rises, it raises the lifting catch and thus the knock-off slide, and through a wire a further catch attached to the sley at the front of the loom is raised into line with a projection of the starting-handle unit for the loom to be knocked off.

If the movement of the bar is not interrupted, then it will not be limited, and the lifting catch will not be allowed to slip over the end of the cam lever, so that the slide bracket will be undisturbed and the loom will continue running.

Probably the major disadvantage of this mechanism is that, if the wire falls onto a raised castellation of the centre bar, it may take a complete two-pick cycle for the adjacent supported wires to push it into a sunken portion and then a further two-pick cycle before the wire becomes trapped and the loom is thus knocked off. No manufacturer is therefore able to give the desirable assurance that his mechnical warp stop-motion will be able to stop the loom in fewer than four picks, and this means that the hole that is created in the cloth could be quite a problem, especially in quality fabrics.

7.2.3 Electrical Warp Stop-motion

The inset in Fig. 7.6 shows one possible construction of an electrode. If a normal rounded-shaped wire were to fall onto the top of the copper strip, it

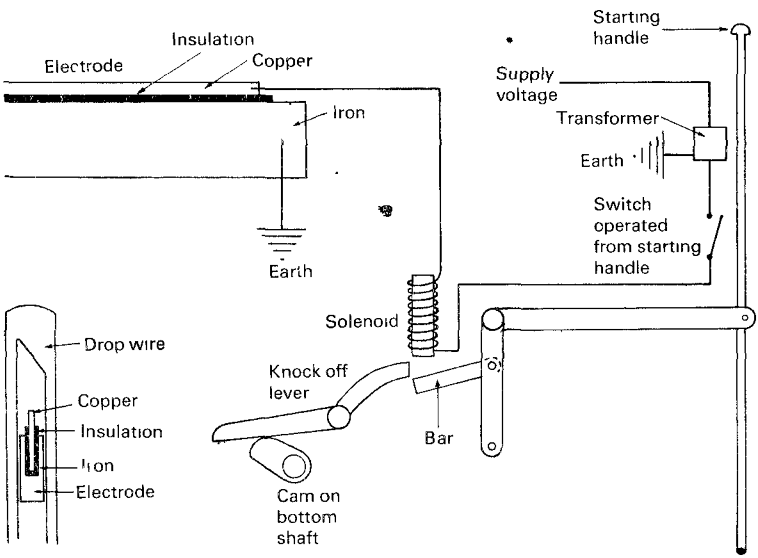


Fig. 7.6 *An electrical warp stop-motion*

might be supported without making immediate contact with the iron. The special inclined shape, however, will cause the wire to tilt to the right and ensure contact.

When this contact is made, the electric circuit is completed, and the core of the solenoid (Fig. 7.6) will be magnetized so that the bar will be attracted upwards into the path of the knock-off lever.

This lever is activated by a cam on the bottom shaft, which may be single- or double-acting. The latter type is obviously quicker, since it operates on each pick, but it may cause the loom to be stopped with the shuttle in the inconvenient right-hand box, in which case it may be necessary to turn the loom over for one pick before it is possible to start repairing the broken end.

When the knock-off lever pushes the bar, the starting handle will be pushed off and the loom stopped, and a switch in the electric circuit will then open to cut off the current. Such a switch is desirable, since, although the current has been reduced by a transformer from the mains voltage to the much safer and more practical level of 12, 14, or 24V and there is no danger to operatives, sparks still tend to occur, and these can easily set fire to the dust and fluff that accumulate around the drop wires as a result of the threads rubbing against the wires.

This fire hazard is probably the biggest disadvantage of an electrical warp stop-motion, and it makes it totally unsuitable for use on looms that are intended for the weaving of soft-spun fibrous warp yarns. Its quick action makes it ideal, however, for looms weaving continuous-filament yarns and relatively fine (probably combed) cotton and worsted yarns, which are generally used in quality fabrics.

The mechanical warp stop-motion of the Rüti C loom breaks a completed electric circuit if the movement of the centre reciprocating bar is limited by a fallen wire. This circuit is very similar to the one used with the electric warp stop-motion, and in both instances it results in the demagnetization of a solenoid, which allows a catch to fall into the path of an oscillating knock off lever.

The particular virtue of this mechanism arises because it depends on the breaking of a circuit, as distinct from the more usual method of making a circuit to stop a loom. Under these circumstances, the loom will knock-off instantly by the same action if there is a power failure for any reason. This is obviously preferable to allowing the loom to run at a reducing speed until a bang-off occurs because, if this happens, the shuttle may become trapped in the shed following a weak pick.

7.3 Weft Stop-motions

7.3.1 Introduction

The importance of stopping a loom immediately after a weft break is probably greater than is the case after a warp break, since it is necessary before restarting the loom to adjust the position of the cloth fell if the shuttle has traversed the loom without inserting a pick.

To restart the loom after a weft break without leaving some indication of the starting place involves one of the primary skills of weaving, and, although it is relatively easily acquired on coarse hairy yarns, for which slight variations in pick-spacing are not so easily discernible, it is one of the most difficult tasks to perform on light, open-structure fabrics produced from continuous-filament or fine spun yarns, such as voiles and super-combed yarns.

Weft stop-motions use the principle of a feeler (commonly called a fork) to detect the presence of the yarn. This fork is held clear of a knock-off mechanism only if it is supported by a thread.

7.3.2 Side-weft-fork Motion

In order to be able to operate this mechanism, it is necessary to have a grid set into the back of the sley adjacent to the reed and opposite to the fork. The raceboard should also have a groove opposite the forks so that the prongs can be operated below raceboard level and it is impossible for the weft thread to pass underneath them.

The side weft fork is situated at the starting-handle side of the loom, which is invariably the left-hand side on modern automatic looms, and, under these circumstances, the weft fork will operate as the sley comes forward to beat-up the pick after the shuttle has entered the left-hand box.

If a trail of weft extends from the selvedge to the shuttle in the box, it will be supported against the grid, and the fork will be tilted about its fulcrum so that the loop of the fork is clear of the knock-off mechanism as shown in Fig. 7.7.

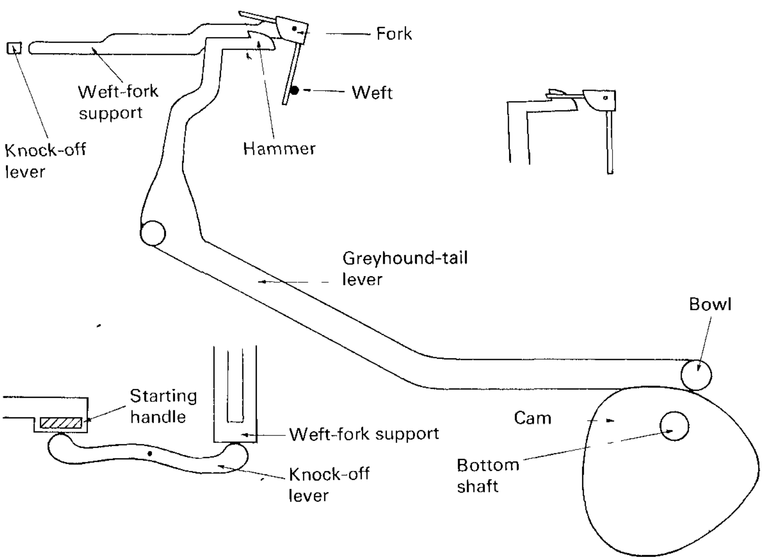


Fig. 7.7 A side-weft-fork mechanism

When the weft breaks, the fork passes through the gaps in the grid and remains undisturbed.

A cam on the bottom shaft raises a greyhound-tail lever every two picks, and it is timed so that it causes the weft-fork hammer to rock towards the front of the loom just after the fork reaches the grid. If the fork is undisturbed, it will be pulled by the weft-fork hammer (top inset in Fig. 7.7), which will thus cause the whole of the weft-fork support to slide towards the front of the loom. In doing so, the knock-off lever will be pushed and will displace the starting handle (bottom inset in Fig. 7.7) for the loom to be stopped.

The whole of this action delays the stopping of the loom by at least one pick, and, since the fork operates at only one side of the loom, the shuttle may have made two traverses without inserting any weft before the fork detects the absence of yarn.

This relatively simple and robust mechanism therefore demands adjustment of the cloth fell, which always provides the likelihood of a bad setting-on place. In addition to the undesirability of setting-on places in quality fabrics, there is the possibility that a weft break may occur if the weft runs out as the shuttle is travelling to the right-hand box on the last pick before a bobbin change. Under these circumstances, the loom will continue running, but only a part-pick will have been inserted, so that a broken pattern results. A box change may occur on a multi-box loom under similar circumstances, and the loom will not stop until the shuttle with the broken weft is in use again.

7.3.3 Centre-weft-fork Motion

All these circumstances make it desirable to be able to stop the loom on the pick in which the weft break occurs. This is made possible by housing the weft fork in a cut-out in the raceboard, somewhere near the centre of the loom, so that, if it is operated in conjunction with a good brake, the loom can be brought to a stop before the broken pick is beaten-up into the fell of the cloth.

Centre-weft-fork mechanisms are much more delicate to set, and access is difficult, so that, whereas they are essential for certain types of loom, they are not universally desirable. They are, in fact, used when the shortcomings of the side-weft-fork mechanism are unacceptable.

The weft fork should be as near as possible to the side of the loom at which the faults are most likely to be created by a change of the pirn being used owing to a box change or bobbin change, but, if a multibox loom is being used, a major problem arises because these two mechanisms are at opposite sides. The actual position should be as near to the chosen side as possible but not so close that it is impossible to knock off the loom before beat-up. The desire to avoid broken picks, especially in wider looms weaving quality (e.g., woollen and worsted) fabrics or in very wide looms weaving fabrics from continuous-filament yarns, has resulted in the incorporation of two individual or coupled centre-weft-fork units situated as near as possible to each side of the loom.

The fork in the cut-out must be raised for the shuttle to pass underneath it on each pick, and the most convenient method is to allow an attachment from the fork prongs to rest on a flat cam attached to the face of the raceboard. The movement of the sley is commonly used to reciprocate this cam by having the knock-off rod fixed to a point on the loom frame, which is often somewhere on the breast beam. On Draper looms, however, the driving rod is often attached to the inside of the loom frame near its base, so that, as the sley reciprocates, the distance from the fulcrum varies, which causes the rod to be driven to the left in the arrangement shown in Fig. 7.8, the return being achieved by the spring.

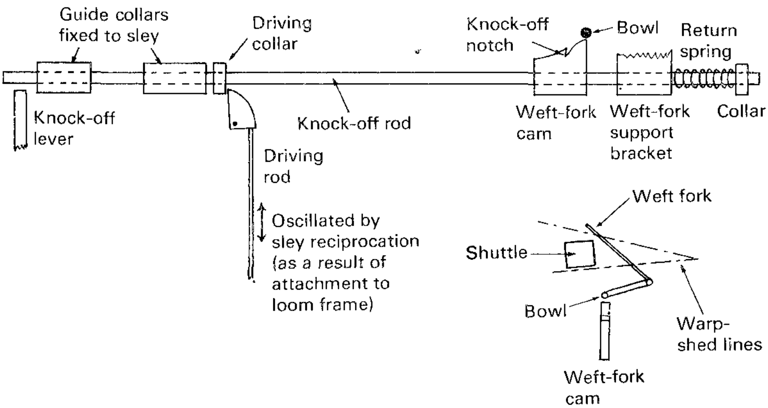


Fig. 7.8 A centre-weft-fork mechanism

As the cam moves to the left, the bowl of the fork rides up the cam surface and causes the prongs to lift. After the shuttle has passed and as the sley starts to move forward, the cam moves to the right, and, if there is no yarn supporting the prongs, then the bowl will be trapped in the knock-off notch, which thus limits the distance that the knock-off rod can move to the right. A small length of rod is now protruding from the left-hand guide collar as the sley comes forward to beat-up. This portion of rod will strike a projection from the starting handle and knock-off the loom.

When the fork is supported by weft, the bowl is carried over the notch, and the rod makes a complete movement to the right, so that there is no protrusion to knock-off the starting handle.

The need to pick-find after a weft break is always liable to cause a setting-on place if the sley has passed its most forward position when the loom comes to rest. It will be necessary to turn the loom over at least twice (for plain weave) and probably four times (after turning the dobby or jacquard cylinder backwards) without inserting weft. A setting-on place will then occur if the weaver fails to judge accurately the required amount of cloth let-back, especially in weaving continuous-filament yarns. Since the correct amount of let-back is very difficult to assess, it is desirable to bring the loom to rest before the sley reaches its most forward position, i.e., before the broken pick is locked in the fell. The fork mechanism can be made quick-acting by introducing a system of electric contacts at the fork itself, but a stop-on-pick motion is also essential. This motion improves the efficiency of the loom's braking system by using a wider and longer brake-band. It is also common practice to introduce a simple foot- or hand-controlled reverse drive so that the sley can be returned to its back position without undue effort on the part of the weaver.

When the loom is restarted, it may again knock-off if there is no weft under the fork. To avoid this annoying occurrence, most centre-weft-fork motions are fitted with an attachment to prevent first-pick knock-off.

7.3.4 Other Weft Stop-motions

The need to detect a weft break still exists on non-shuttle looms, and the technique adopted must be related to the method of weft insertion. In most instances, the basic principle of the presence of weft limiting the movement of a single fork prong is used. The absence of weft will allow a further movement of the prong, which will result in a mechanical or electrical contact's being made to bring the loom to rest.

One unusual method is used on the Nissan (formerly Prince) water-jet loom. Two prongs are mounted on, but are insulated from, the reed between the leno threads and the dummy selvedge. As the reed beats up, the weft makes contact across the prongs, and the presence of water on the weft allows the circuit to be completed. If there is no weft and thus no water, the circuit will not be completed, and the loom will be brought to rest.

CHAPTER 8

Weft-patterning

8.1 Comparison of Systems Available

Weft-patterning in this context is intended to refer to the different yarns that it is possible to introduce across the fabric.

The introduction of yarn of different colour, linear density, or character into the warp of a fabric is relatively easy to achieve by pre-planning the position of the bobbins in the creel at the warping stage. Large differences in yarn linear density or crimp will generally require that the warp yarn should be supplied from more than one beam. If more than three beams are required simultaneously on one loom, problems arise with regard to accommodation and accessibility, and it may even be necessary in some cases to supply the yarn from individual packages in a creel.

Each variation in the weft being supplied to a loom does, however, require a separate source of supply and selection. With shuttle looms, one shuttle is required for each weft and there must always be one empty shuttle-box in the loom, so that a loom with a two-shuttle-box unit at one side of the loom and a single-shuttle-box unit at the other side (generally described as a 2 × 1 box loom) will be capable of inserting two different wefts, 4 × 1 and 6 × 1 box looms being capable of inserting four and six colours, respectively. The most severe limitation of box motions of this type is that the picks must always be inserted in multiples of two, since the shuttle must travel from the multi-box-unit side of the loom and back again before a change of weft is possible. The insertion of a single pick or odd-numbered groups of picks only becomes possible in looms having two, four, or maybe six boxes at each side. Such a 4 × 4 box loom may still only be capable of inserting four different colours if the box units are controlled from the same source (i.e., the box units would rise and fall together), but, if the box units are individually controlled, then up to seven colours become possible (i.e., there must always be an empty box waiting to receive the shuttle).

Single-pick weft insertion is more easily achieved in looms with gripper shuttles or rapiers, since it is then only necessary to bring the desired weft to a position at which it can be picked up by the insertion medium. Individual tension and control units are required for each weft, however, and, if weft accumulators are necessary, then the weft-insertion unit can still be quite bulky. Although eight and twelve colours are possible on some looms, most makers are content to offer up to six.

Multiple-box units for shuttles have operated on a revolving (circular-box) basis or on a rising and falling (drop-box) system (see Fig. 8.1). The box unit

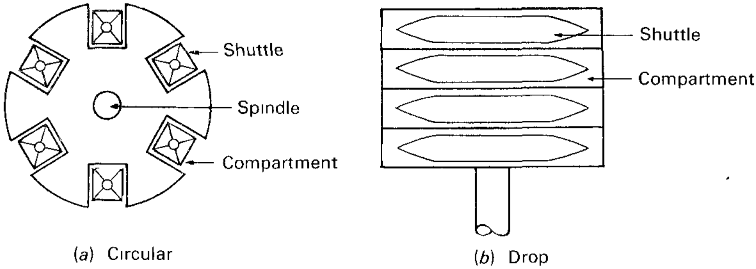


Fig. 8.1 Box arrangements

revolving on a spindle commonly had six compartments for six different shuttles, although five and seven compartments have been known in rare instances, and drop-box units may have two, four, or six compartments.

Circular-box motions are gradually becoming obsolete, mainly because they must operate with an overpick and a loose-reed motion. These mechanisms are generally accepted as being inferior to other mechanisms available for the same purpose, i.e., cone-underpick (Section 4.2.3) and fast-reed warp protection (Section 7.1.3). With a circular-box motion, it is generally only possible to move from the box in use to one of the adjacent boxes when a change is required, although a motion was developed that made it possible to skip boxes and thus select from any box to achieve random selection. This motion was not generally accepted because of the speed reduction necessary. With the limitation of one box movement, it is possible that the number of colours could be limited to three if they were being woven in sequence as compared with a maximum of six. In such a case, or in any other circumstance that requires the box unit to rotate in one direction only, the weft wraps round the spindle, the result being that there are long broken threads hanging from the selvedge and yarn accumulations on the spindle that make it difficult for the unit to rotate freely. Furthermore, the size of the shuttle-box is fixed, so that it is difficult to adjust for worn shuttles and impossible to modify for larger shuttles because each compartment is limited dimensionally and must be kept rather small for reasons of space and mass. The main reasons for the decline of the circular-box motion are, however, the difficulty of applying automatic weft replenishment to the motion in the presence of an overpick stick and the difficulty of verifying which box is in use at any one time. None of the disadvantages of the circular-box motion that have been described are insurmountable in themselves, but in the aggregate they create problems that have resulted in the preference for drop-box motions.

Drop-box units generally require that a loom should run about 10% slower than its single-shuttle counterpart, with an even greater reduction in loom speed if the movement of the unit is greater than two boxes, even though the drive to the mechanism is usually positive. Special care is needed in arranging the order of the wefts in the boxes if lashing-in is to be avoided or possibly only minimized, but this system is much more easily adapted to pick-at-will mechanisms. Such a motion requires a box unit at each side of the loom so that any sequence of picks can be inserted as required within the limits of the pattern sequence and capacity of the mechanism, and it may be necessary to arrange to pick from the same side of the loom on successive picks.

2 × 1 Drop-box motions have been specifically designed to operate a box change after every second pick. These were known as *weft-mixing motions* because they were inserting similar weft from both shuttles with the object of reducing the possibility of weft bars when the weft package was changed. This fault was quite common during the weaving of single-colour fabrics. The need for such a motion has diminished considerably with the improvements that have been made in the uniformity of colour in pressure-dyed packages and also with the greater preference for piece-dyeing rather than colour-weaving of a single-colour fabric. The recommended technique was to have a full bobbin in one of the shuttles and a half-used bobbin in the other shuttle, since at this stage the weft cops themselves were actually pressure-dyed. When the second shuttle needed a new weft package, the first one would be half-used, so that dye variations from the outside to the inside of the package were spread throughout the fabric rather than appearing abruptly at each weft-package change. This system is far from obsolete, since it is used in weaving yarns that have been colour-blended in the spinning process, but 1 × 1 insertion is even more desirable, and it can easily be achieved on gripper and rapier looms.

Weft-mixing was also extremely desirable in the early days of weaving fabrics from continuous-filament yarn because the character of the yarn was most likely to vary, not only from package to package but also within a package. Sequence weaving, as will be described later in Section 9.1, was introduced to avoid the need to weft-mix and yet, at the same time, prevent the faults that were likely to appear in the fabric, but, since the quality of continuous-filament yarns has improved and the loom-winder has gained wide acceptance, the need for weft-mixing motions has diminished considerably. The weaving of certain plain-weave crêpe fabrics requires two picks of Z-twist weft and two picks of S-twist weft to be inserted alternately. A weft-mixing motion is quite satisfactory for this purpose. Such fabrics are very much affected by fashion trends, and, since the loom would be limited in its suitability for other fabrics, it is now more usual to have machines available that would have greater flexibility.

The 4 × 1 drop-box loom is undoubtedly the most popular type of shuttle loom available for weft-patterning, although it does in many instances limit the scope of the designer. With this in mind, 6 × 1 drop-box motions are available, if required, for the weaving of certain types of check fabrics (e.g., tartans), and, although these looms will be required to run even more slowly, it is possible to have automatic weft replenishment.

8.2 Box Motions

8.2.1 Introduction

For conventional shuttle looms, it is intended to deal only with drop-box motions because of the pending obsolescence of the circular-box motion.

8.2.2 *Weft-mixing Motion*

A two-position eccentric, shown in Fig. 8.2*a*, governs the position of a fulcrummed lever. If the connexion of the crank lever to the eccentric is in its highest position, then the box unit will be in its highest position, with the bottom box level with the raceboard, but, if the eccentric is turned through half a revolution, the lever will be lowered, and the box unit will be lowered to position the top box level with the raceboard.

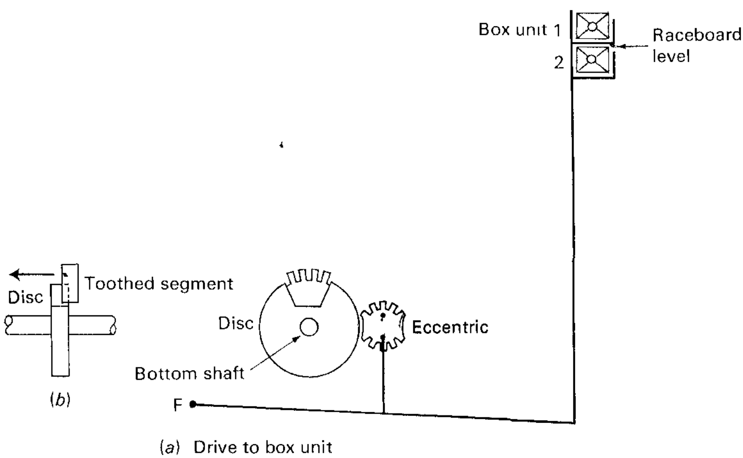
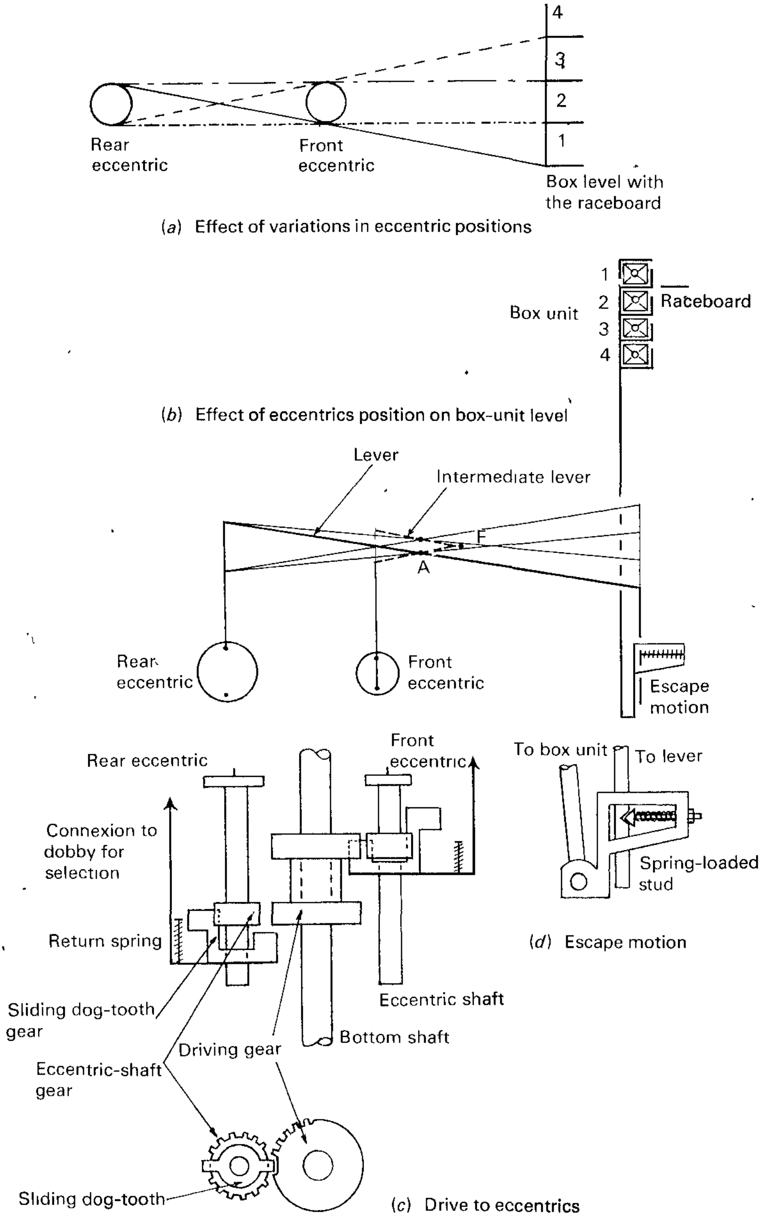


Fig. 8.2 A weft-mixing motion

The motion is driven from the bottom shaft of the loom so that the eccentric will be driven through half a revolution once every two picks to give the 2 × 2 weft-insertion sequence. On the bottom shaft, there is a disc, into which is set a series of teeth. The smooth surface of the disc will normally rotate in contact with a smooth surface on the eccentric, but, when the teeth in the disc reach the eccentric, they engage with teeth in the eccentric. The number of teeth involved is just sufficient to turn the eccentric through half a revolution before the bottom-shaft disc meets the opposite smooth surface of the eccentric. Two picks later, the procedure is repeated to return the eccentric to its original position. The motion is timed so that the teeth engage and thus the box change will take place when both shuttles are in their respective boxes in the two-box unit.

The toothed sector in the disc can be made into a sliding segment. In these circumstances, the disc gear will normally miss the segment gear (insert *b* in Fig. 8.2), but on selection the segment will slide sideways in the direction indicated to engage with the eccentric gear, which will be turned through half a revolution to create a box change. This will allow any two-weft pattern sequence to be woven and thus gives the mechanism greater flexibility. The system is found on Northrop looms.



(*a*) Effect of variations in eccentric positions

(*b*) Effect of eccentrics position on box-unit level

(*c*) Drive to eccentrics

(*d*) Escape motion

Fig. 8.3 A 4 × 1 box motion

8.2.3 Four-box Motion

In order to create four different levels of a box unit so that each box in turn may be positioned level with the raceboard, it is necessary to have two eccentrics as illustrated in Fig. 8.3*a*. By turning the rear eccentric through half a revolution, a one-box movement will be achieved, and, by turning the front eccentric a similar amount, a two-box movement will result. This situation depends on the distance between the two eccentrics being equal to the distance from the front eccentric to the box-unit connexion, but this is not possible on the loom because the eccentrics must be adjacent to the bottom shaft, from which they receive their drive. The arrangement of cranks and levers as used on a Crompton & Knowles C5 loom is shown in Fig. 8.3*b*. The throw of the front eccentric is reduced to the amount of displacement received by the fulcrum point where the intermediate lever and the box-drive lever are joined (i.e., A). Under these circumstances, the front eccentric becomes responsible for the one-box movement and the rear eccentric for the two-box movement, which is contrary to the principle shown in Fig. 8.3*a*.

When the two eccentrics are in their starting positions, i.e., the back eccentric is in its highest position and the front eccentric in its lowest position, then the top box will be level with the raceboard. Subsequently, if the front eccentric is driven through half a revolution, the connexion to the rear eccentric will act as a fulcrum and a one-box movement will be achieved, but, if the rear eccentric moves half a revolution, then the front crank will act as a fulcrum and a two-box movement will result. Furthermore, if both cranks move simultaneously in opposite directions, then a three-box movement will be achieved, but, if they move in the same direction, a one-box movement only will occur. The box level with the raceboard is entirely dependent on the position of the eccentric connexions (see Table 8.1).

Table 8.1

Rear-eccentric Connexion	Front-eccentric Connexion	Box Level with Raceboard
Up (starting position)	Down (starting position)	1
Up	Up	2
Down	Down	3
Down	Up	4

The eccentrics are mounted on their respective shafts on each side of the bottom shaft so that they may be driven through half a revolution if a box change is desired. This should be required only once every two picks on a 4 × 1 box loom (hence the suitability of the bottom shaft), but modified gearing to make changes possible after every pick on a pick-at-will loom is quite possible. The intermittent drive to the eccentric shaft is dependent on selection and may be controlled by either a sliding dog-tooth gear or a paddle-controlled gear. The turning of an eccentric through half a revolution depends on the selection mechanism positioning the eccentric-shaft gear opposite to the bottom-shaft gear. The dog-tooth-gear system illustrated in Fig. 8.3*c* slides in a cut-out in the eccentric-shaft gear so that, when the driving gear engages the dog tooth, which is pulled into position by the selection mechanism, the additional teeth in the driving gear will then turn the dog-tooth and eccentric-shaft gears through half a revolution. Driving is now no longer possible because the other dog tooth on the wheel is not in the same vertical plane and cannot mesh until the selection changes again, which thus allows the other dog tooth to slide back to a driving position. When the sliding gear does not adjust its position, there is a gap opposite the teeth in the driving gear, so that nothing can happen.

If, in any box motion, a shuttle fails to box correctly, then the full box-change movement must be prevented if damage to the shuttle and shuttle-box parts is to be avoided. This is achieved by the spring-loaded stud (Fig. 8.2*d*), which is part of the box-drive mechanism. During normal running, it is engaged with a cut-out on the box-unit support shaft to drive the boxes up and down, but, if a shuttle is projecting from the box so that full movement of the box unit is not possible, then the catch will be forced out of the cut-out so that the drive mechanism will make its full movement, but the box movement will be limited and damage avoided.

It is also necessary to use a pick-stick-easing strap to give a slight inwards movement of the picker on completion of checking so that the tip of the shuttle will not be held in the cut-out of the picker, since movement of the boxes may then be difficult if a box change is scheduled to occur.

Selection for a box change may be made from a separate pattern-control cylinder or direct from the dobby or jacquard. A separate cylinder is essential for tappet looms, and it may on some occasions reduce the pattern-assembly time for dobby or jacquard looms if the sizes of the weft pattern repeat and weave repeat are extensively different, e.g., a 2/2 twill weaving a tartan check repeating on 600 picks or a 1×1 (or maybe 2×2) weft-insertion sequence for a double cloth figured by interchange and repeating on 2000 picks. In most instances, it is generally preferable to control the weft-pattern selection from the same source as the shedding selection if a dobby or a jacquard is being used because pick-finding is much simplified and the two patterns cannot become out of sequence. The major disadvantage of such a system is that the figuring capacity of the dobby may be reduced, although many dobby makers now provide additional jacks for such a purpose, just as there is an extra (spare) row of hooks in a jacquard.

The order of the wefts in the boxes is important if the possibility of the undesirable 1–4 box movement is to be minimized, and, to achieve this, it is generally advisable to aim at placing the most popular colour in one of the middle boxes. Since lashing-in is also a real possibility with a box motion, especially if heavy or lively wefts are being used, it is worth while to plan for these wefts to be placed in the bottom box if at all possible because from this position there is the least chance of their catching the other wefts. One disadvantage of not having the most popular weft in the top box is that, in battery-filling, it is more difficult to see when the stack containing the popular weft is almost empty.

Not infrequently, box looms are required to weave single weft fabrics for various reasons, and in this case the top box would be used because selection would then be unnecessary. To distribute wear on the parts of the mechanism, it is therefore common practice to place the weft that is in least demand in the top box in weaving a multiweft pattern.

With some types of box motion, it is only necessary to indicate to the mechanism when a box change is necessary, but, for the sliding-dog-tooth mechanism already referred to in Fig. 8.3, the sliding gear must be held in the selected position until a further change is required.

A suitable recommendation for the order of the wefts in the boxes of a 4 × 1 box loom, together with an appropriate pattern chain for the pattern given in Table 8.2, is illustrated in Fig. 8.4. The boxing order complies with the preceding

Table 8.2

Weft Pattern
2 picks white
4 picks red
2 picks white
8 picks blue
4 picks yellow
10 picks red
4 picks yellow
8 picks blue
42 picks per repeat

recommendations. There are 42 picks per repeat, but, with a box change possible only once every two picks, then only 21 plates or lags are required in the complete chain. A filled-in hole indicates that selection will take place.

In most looms, it is usual to find that box selection must be made when the pick preceding the desired change is being inserted. The position of the driving gear on the bottom shaft will govern the actual time of the box change.

8.2.4 *Pick-at-will*

On looms having a four-box unit at each side, it is necessary to ensure that the shuttle will be projected into an empty box irrespective of whether the box units rise and fall simultaneously (from one common source of control) or independently of each other. It is necessary because the looms must be so designed that they can pick from either side of the loom in any sequence depending on the weft pattern required, as referred to in Section 4.2.4. The looms are therefore capable of picking accidentally from both sides on the same pick because each picking cam has two picking noses, which are diametrically opposed. This mechanical problem cannot exist on a loom that picks from each side alternately.

In order to allow any one pick to be inserted, it is necessary to position the shuttle-box, with the next pick of weft for insertion, level with the raceboard at one side of the loom and an empty box level with the raceboard at the other side of the loom, and, although picking may occur from either side of the loom, it must be ensured that it occurs only from the side of the loom at which there is a shuttle in the box at raceboard level.

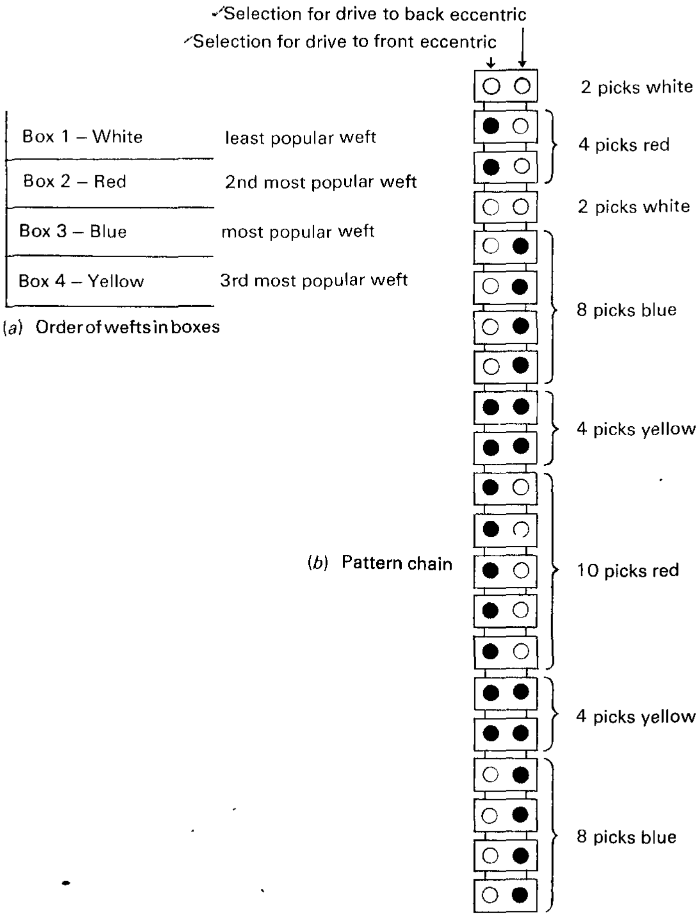


Fig. 8.4 A box plan and pattern chain for the weft pattern

To achieve this control, a series of levers is connected from the swell finger at each side of the loom, and they extend across the loom in such a way that they positively control a picking-catch support at the opposite side of the loom. In Fig. 8.5, a side-lever picking motion is illustrated. If there is a shuttle at the opposite side of the loom, then the catch support shown in the diagram will be pushed to the left and will thus raise the catch so that the downward movement of the picking lever cannot be transferred into a movement of the picking stick. The absence of a shuttle in the opposite box will allow the catch to be down, so that the picking stick will be pushed forward when the picking lever presses down and the shuttle will be projected across the loom.

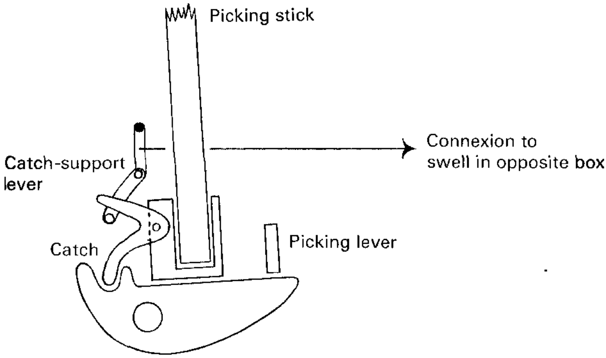


Fig. 8.5 A pick-at-will control motion

8.3 Weft-patterning on Unconventional Looms

8.3.1 Introduction

Weft-patterning on conventional looms complicates the loom equipment and also reduces the loom speed. In general, weft-patterning on unconventional looms complicates the loom equipment to a less degree and requires little or no reduction in loom speed. Provided that it is no more costly to produce single-weft fabrics on unconventional than on conventional looms, then multiweft-patterning on unconventional looms will be economically attractive. It can be expected that this proviso will be met, since one of the attractions of single-weft unconventional looms is that they show a cost advantage compared with single-shuttle conventional looms for many types of fabric.

It is expected that weft-patterning on jet looms may soon become feasible, but weft-patterning on unconventional looms has only been achieved successfully from the fabric manufacturer's viewpoint with the rapier and gripper-shuttle methods of weft insertion.

In these machines, each weft is supplied from a large stationary package situated at the side of the loom, and each weft is controlled by its own individual feeder unit or guide arm. This device then presents the selected weft to the insertion medium just before weft insertion takes place. A complete selection system consists of a series of units, and each unit consists of a package holder, tension unit, tension compensator and/or accumulator, and guide arm. There are normally two, four, six, or maybe eight wefts available in a unit.

In the reciprocating-gripper-projectile method of weft insertion, which has been referred to previously, but which has not gained any degree of industrial acceptance, a selection system would be required at each side of the loom. Thus, if four variations were possible at each side of the loom, then eight selection units would be required on each loom. Although this would make eight variations theoretically possible for some patterns, it is quite likely that the same four wefts would have to be available at each side of the loom in order to weave certain pattern sequences. Although this is restrictive, it would be sufficient for a large number of patterns.

8.3.2 Sulzer Weft-patterning Systems

The weft feeder needed for each weft in a multicolour weaving machine must be modified in design from that used in a single-colour machine because at all times the feeder must be controlled by the feeder housing and yet still be capable of sliding inwards from its selection position to a predetermined distance from the selvedge. This is necessary in order to keep the weft under control in exactly the same manner as that described in Sections 4.1.3 and 4.1.4. The feeder is much longer because it has to move this greater distance away from the selvedge. It is also thinner, so that more feeders can be accommodated in the limited space available.

The feeder housing is moved by a bevel gear, which in turn is driven from a segment gear. As the segment gear is moved up or down, the feeder housing is caused to move in the opposite direction. The amount of movement determines the feeder that is positioned opposite the weft gripper for insertion.

In the original four-colour Sulzer weaving machine, the heights of the links in a pattern chain determine the position of the segment gear. The weight of the pattern chain and the storage space it requires on the loom are the main factors that limit the length of the pattern chain to 200 links. The pattern chain is driven by a rotating star wheel turning a pin wheel in which there are four pins. The weft-repeat size is limited to 200 (single) picks if all the pins are present. If two opposing pins are removed, then the maximum repeat size is increased to 400 picks, but two picks will be inserted for each selection. When a further pin is removed, the maximum repeat size is increased to 800, four picks being inserted for each link in the pattern chain.

This mechanism allows a wide range of fabrics to be produced, but it also precludes fabrics having large weft-pattern repeats, for example, large checks, including handkerchiefs. More recently, a paper-pattern-control mechanism has been introduced that removes all limitations with respect to the size of the weft-pattern repeat. This unit may incorporate its own paper-pattern-selection unit, or it may be controlled from the dobby or jacquard.

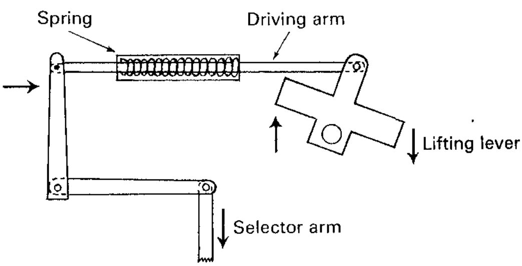


Fig. 8.6 The Sulzer paper-pattern-controlled weft-pattern-selector mechanism

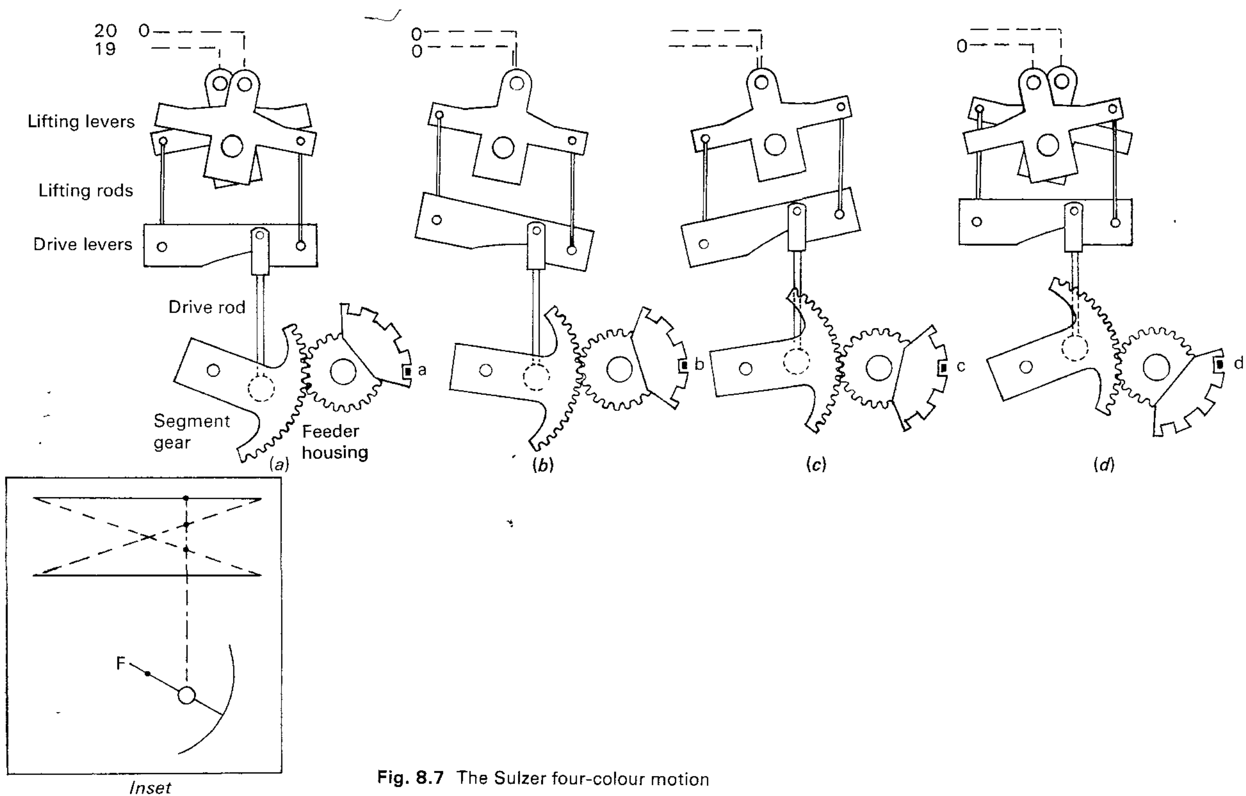


Fig. 8.7 The Sulzer four-colour motion

A hole in the pattern card or a selection from the shedding mechanism will cause the selector arm shown in Fig. 8.6 to move down, and the spring-loaded driving arm will push the lifting lever. The right-hand end of this lever will then be inclined downwards. Two selection and driving arms are needed to control the two lifting levers, and the units are returned by the spring-loading in the driving arm.

A lifting rod is suspended from one end of one of the lifting levers and a second rod from the other end of the other lever, as illustrated in Fig. 8.7. The drive lever can be moved by displacing either lifting rod. The drive rod is suspended from a point on the drive lever that is one-third of the distance from the connexion with the right-hand lifting rod. In the starting position (shown in Fig. 8.7*a*), the feeder housing has positioned the lowest feeder opposite the gripper, but, when both levers are selected to be tilted down to the right, as in Fig. 8.7*b*, then the drive rod will be raised to allow the next feeder to be level with the gripper. The driving rod will be higher still if neither lifting lever is selected because they will both tilt down to the left (Fig. 8.7*c*), and the connexion to the drive rod will be even higher because it is further from the fulcrum. Finally, when both lifting rods are raised as in Fig. 8.7*d*, the drive rod will be in its highest position with the top feeder supplying the gripper with weft for insertion. The principle of using two lifting points is very similar to that used in the four-box motion described in Section 8.2.3 and illustrated in Fig. 8.3*a*. The technique of achieving the four positions from two selection points as used in the Sulzer system is illustrated in the simple line-diagram inset in Fig. 8.7.

The six-colour machine incorporates a similar system, but in this case three control points, lifting levers, and lifting rods are needed to control two drive levers and rods and thus create six different feeder-housing levels as illustrated in Fig. 8.8.

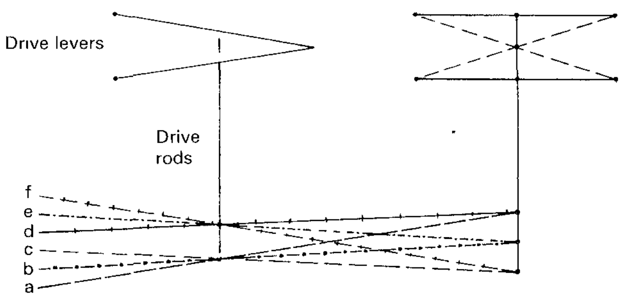


Fig. 8.8 The Sulzer six-colour motion

It is usual in all these mechanisms to use a cam-operated locking bar to secure the feeder housing in the correct position for each pick. This ensures that the feeder is correctly positioned for the gripper projectile, so that there will be no contact between the feeder and the gripper when the weft is transferred to the gripper. The feeder should also slide correctly and freely in the picking-unit housing.

8.3.2 Rapier Systems

Each weft thread must pass through the eye of a guide mounted in a selector arm. This guide is responsible for placing the selected thread in position for pick-up by the rapier head. Single-colour rapier looms are not unknown, but most rapier looms are capable of inserting different wefts on successive picks, and a guide is needed for each weft in use. The guide carrying the selected thread usually swings into the appropriate position at a time between angular positions of 320 and 350°. A hook mounted on the sley just outside the selvedge, which may be fixed or cam-controlled, then picks up the weft and ensures that the thread is positioned across the rapier path as the sley starts to move back. Weft insertion commences at about 60°, depending on the loom. The unselected threads from the other guides stretch to the selvedge but cannot be picked up by the rapier head because they have not been lowered (or possibly raised, depending on the machine) to be picked up by the hook.

If selection is to take place on a tappet loom, then a separate selector mechanism is required, and the virtues of such a motion are exactly the same as those explained in Section 8.2.3. It is not impossible to use such a motion on a loom equipped with a dobby or a jacquard. Probably the most popular method of selecting from a separate mechanism is to use a mini-dobby, which is situated on the front loom frame adjacent to the selvedge at the weft-insertion side of the loom. This system is very popular on flexible-rapier looms, particularly those using the Dewas method (Fig. 8.9*a*). A hole in the pattern card allows a hook to fall over a reciprocating knife. As the knife moves towards the front of the loom, it pulls the hook and the lever to which it is connected. The guide arm is thus lowered, and the appropriate weft is picked up by the hook and positioned across the path of the rapier head. When the knife returns, a spring action lifts the hook, and the selecting needles are raised clear of the pattern barrel by the returning rising knife. The pattern barrel can then turn before the knife starts its forward movement again.

Direct selection from the dobby or jacquard requires only a very simple system of levers or flexible cord to activate the selector arm. One such system, as used in the Dornier loom, merely relies on the upward movement of a guide, jack, or hook to lower the guide (Fig. 8.9*b*). Selection must be carefully timed because the guide must be in position at a time when the knives or griffes of the shedding mechanism are actually changing from one position to another. However, since the dobby jack (or jacquard hook) has normally started to rise before the sley starts to move back, the selected weft can be displaced sufficiently to be picked up by the hook on the sley so that it will be positioned opposite the rapier head.

An alternative method, which overcomes this timing problem, allows the dobby to make an electrical contact that will energize a solenoid. The solenoid will then raise the drive arm into the path of a reciprocating griffe as illustrated in Fig. 8.9*c*. This system is used on the Versamat loom. As the griffe moves forward, the selection arm will be made to swing back and place the weft across the rapier path. The selection arm is returned with the return movement of the drive arm.

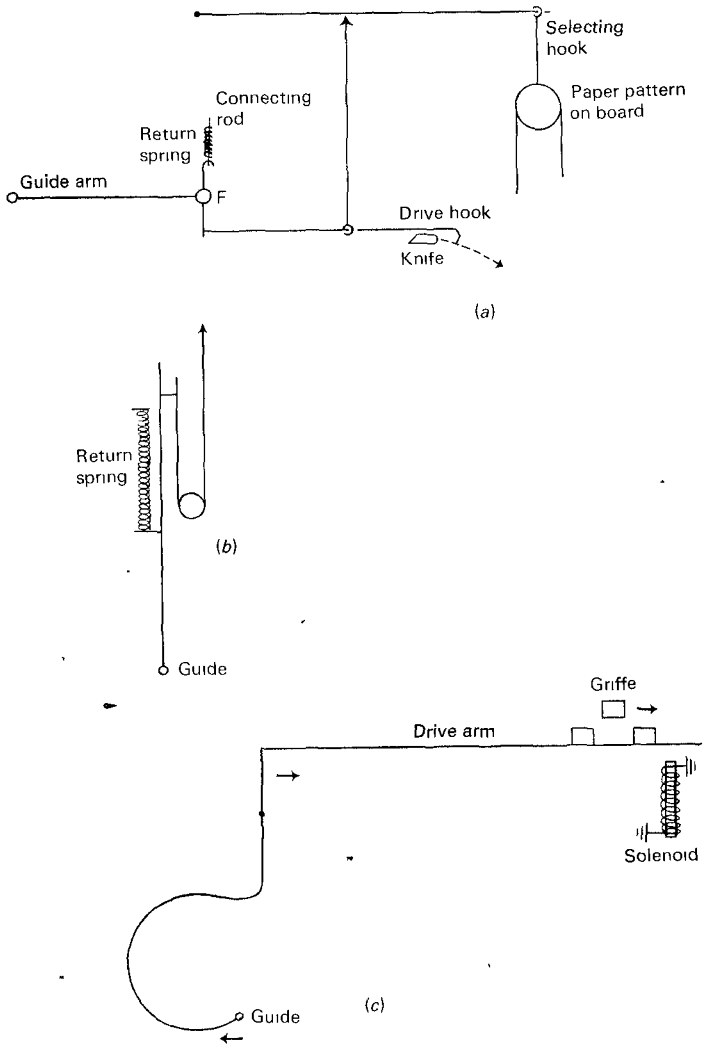


Fig. 8.9 Weft-selector systems for rapier looms

The weft on a rapier loom can be cut by allowing it to pass over a fixed edge just after the rapier has started to enter the shed when the Dewas system is used. This is a rather negative cutting technique, and better results are usually obtained if an oscillating blade is used. The oscillation may be achieved from sley reciprocation, but it is more usual to use a cam drive that can be individually timed. This method is often preferred when the Gabler system is used, since the weft must be gripped on one pick and cut half-way through the insertion sequence on the next pick when transfer is taking place at the centre of the loom. The cam-operated cutting unit may be separate from all other mechanisms or it may be incorporated into the hook on the sley as on the MAV loom.

CHAPTER 9

Weft Replenishment

9.1 The Work of the Weaver

When the duties of the weaver involved weft- and cloth-carrying, oiling, and greasing, as well as the major duties of weaving, it was obvious that a great deal of time was being wasted on unskilled jobs and that a reassessment of duties would have to be made to obtain more effective use of labour.

After extensive redeployment, the main work of the weaver on non automatic looms involved (*a*) general supervision and inspection of the yarn, cloth, and machine, (*b*) repairing warp and weft breaks, and (*c*) stopping the loom to replace the weft package when the yarn on the pirn was almost used up.

It is not feasible to envisage any automation of the first two of these duties, but the time involved in watching for a pirn weaving down to the last few coils of yarn and then stopping the loom, removing the shuttle, replacing the shuttle in the loom, restarting the loom, and placing a full pirn in the spare shuttle in readiness for the next change severely limited the number of looms to which a weaver could attend while maintaining a reasonable loom efficiency. In addition to this major drawback, if the loom was stopped too soon, it would have to be restarted and then stopped again within a few picks, or alternatively the pirn would have to be removed with an excess of yarn that could only become waste. On the other hand, if the weaver did not stop the loom at the correct time, then the weft would be completely used up, and the weft fork would operate so that, on restarting, pick-finding would be necessary and a setting on place possible.

These circumstances indicate an obvious need to replace the weft package automatically, and two main methods, (i) the shuttle change and (ii) the bobbin change, were eventually developed.

The technique of replacing one shuttle containing an almost empty pirn with another shuttle containing a full pirn involved stopping the loom for a few seconds on each occasion, and, although eventual improvements overcame this problem, there was still a limitation in machine speed.

Replacing the bobbin in the shuttle while the loom is running has always been a much more attractive proposition, since there are fewer moving parts, and the transfer can be completed in a relatively short time. The time required for a transfer is limited by the fact that the sley can move only a limited distance while the transfer is taking place. If a loom is running at 216 picks/min and has

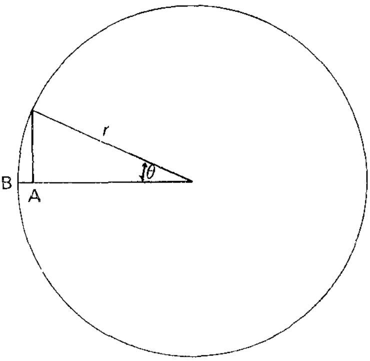


Fig. 9.1 The time for transfer

a crank radius of 140 mm and the distance AB in Fig. 9.1 is 1·3 mm, then:

$$\cos\theta = \frac{140 - 1\cdot 3}{140} = 0\cdot 9907.$$

Hence $\theta = 7°\,48'$, and transfer must therefore be complete in $2 \times 7°\,48' = 15°\,36'$, which represents:

$$\frac{15\cdot 6}{360} \times \frac{60}{216} = 0\cdot 012\ \text{s}.$$

This is less than the time involved in accelerating the shuttle, but it seems reasonable in view of the fact that smaller masses and distances are involved.

There were many initial limitations to the bobbin-change method of transfer, and these were mainly associated with the weaving of delicate continuous-filament yarns and lively yarns, but gradual improvements have now made the bobbin-change mechanism universally acceptable and the shuttle-change method obsolescent.

With the bobbin-change mechanism, the bobbins or pirns for the loom are wound on pirn-winding machines in the preparation department and are then transported to the loom by a weft carrier. Each pirn is placed individually in the magazine of the rotary battery, where it is held by spring-loading at its tip, and the weft from the tip of the pirn is anchored by wrapping it around a boss on the magazine so that it will not be dragged into the cloth when the first pick is inserted after a bobbin transfer. These operations are called battery-filling, and, for medium and coarse weft yarns, it may be necessary to have as many battery fillers as weavers. Although the rate of pay for the personnel who usually do the job will be somewhat less than that of the weaver, the cost of battery-filling is an appreciable fraction of the weaver's wage. It is therefore worth while to try to eliminate battery-filling, and this can be achieved by either the bobbin loader or the loom-winder.

Instead of a rotary battery holding a maximum of about 24 bobbins, with the bobbin loader it is possible to use a metal box with a slot at one end of the base. The box may hold between 72 and 190 bobbins depending on the box size and pirn diameter. The bobbins are then fed through the slot to the weft-replenishing mechanism as required. Each loom has two boxes, one in use and the other in reserve, so that, when the box in use becomes empty, the weaver can replace it with a full one, the empty one being left to be filled up or replaced by the weft carrier when convenient. This requires much less time than battery-filling, and thus a saving in labour costs can be expected. This saving must be balanced against the extra cost of the bobbin loader.

Now the cost per loom per hour of battery-filling is determined by the number of bobbins a battery filler can deal with per hour. The coarser the weft and the higher the weft-insertion rate (i.e., the wider the loom and the higher the loom speed), the more bobbins the loom will require and the higher will be the cost of battery-filling. The potential savings by using the bobbin loader are therefore greater with relatively coarse yarns and wide fabrics (e.g., sheetings), than with finer yarns and narrower fabrics (e.g., shirtings).

The automatic loom-winder (Unifil) has a higher capital cost than the bobbin loader, but it eliminates pirn-winding as a separate operation, and, when other savings are taken into account, it may show an even greater net saving than the bobbin loader. The bobbins are wound at the loom by an automatic pirn-winder from large cones, or, for continuous-filament yarns, from the supplier's packages. Two packages are tied tip to tail as in magazine-creeling for warping so that, when one is exhausted, winding can commence from the other automatically and the empty package can be replaced at a convenient opportunity. There is therefore a very similar saving with battery-filling to that with bobbin-loading.

It is usually considered necessary to have in circulation (at the loom, in the pirn-winding department, in the weft store, in stripping, and in reserve) between 200 and 300 bobbins for each battery loom. About the same number or possibly slightly more are likely to be required for each bobbin loader. With the Unifil (Fig. 9.11), each loom operates with only twelve or fourteen bobbins, which never leave the loom and of which only half contain yarn. There is therefore a considerable saving on the outlay on bobbins and also in replacement costs, since the bobbins are less likely to be damaged. Again, the loom-winder strips the ejected bobbin of the short length of yarn left on it at the change and thus eliminates bobbin-stripping as a separate operation and also the need to transport empty pirns. Servicing and maintenance of the units are done by specially trained mechanics who can look after several hundred units, and there is no need for a separate pirn-winding department with all its ancillary requirements (e.g., heating, lighting). All the pirns woven on the loom are wound under identical conditions of tension and bunch length, which thus eliminates the possibility of mixed weft, irregularity in fabric appearance as a result of weft-tension variation, and excessive waste. When all these considerations are taken into account, the unit will pay for itself under favourable conditions inside three years. This is the chief reason for its notable success.

The unit is claimed to be suitable for spun staple-fibre yarns ranging from 5 to 150 tex and for continuous-filament yarns ranging from 3 to 55 tex, and, for the same reasons as for the bobbin loader, it will justify itself economically, mainly with coarse and medium yarns, but it is widely used for weaving continuous-filament yarns of the order of 7–17 tex, for which the economic advantages are not so obvious. One reason for this is that it ensures that the bobbins will be woven in the same order as that in which they are wound from the supply package. This is very desirable in order to minimize weft bars caused by weaving, in succession, packages wound either from different parts of the same package or from different packages.

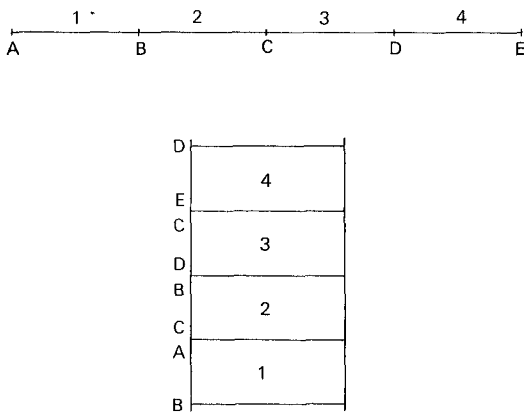


Fig. 9.2 The order of weaving pirns

It is perhaps worth mentioning that, although the bobbins are woven in sequence, the weft in the fabric is not arranged precisely as it was on the supply package. Consider the length of yarn AE in Fig. 9.2, which is wound onto four pirns. The first yarn, A, is wound onto the base of the first pirn, and the yarn wound onto the tip of the pirn is from point B. The remaining pirns wind yarn from B to C, from C to D, and from D to E. During weaving, however, the first yarn to be wound off the pirn will be from the tip, so that the length of fabric will have the last yarn wound off the first pirn (A) next to the first yarn wound off the second pirn (C). In order to weave the yarn in perfect sequential order, it is necessary to weave the pirns in the reverse order to that in which they were wound, i.e., 4, 3, 2, and 1, but this is not possible with Unifil or with the bobbin loader, where one cannot guarantee any precise order. It is only possible if the bobbins are placed manually or automatically on a pin tray in a particular sequential order as they come from the pirn-winder. The weaver or battery filler must then exactly reverse this order as the pirns are fed to the loom. There are circumstances in which this elaborate procedure may be justified.

9.2 Feelers

9.2.1 Introduction

It is necessary to have some means of indicating when the yarn has been almost used up on a pirn, and until recently this was done by a mechanism making contact with the pirn by a probing action to detect if yarn was present or not. The action of probing gave these mechanisms the obvious name of *feelers*, and this name has even been retained for the latest types, which do not actually make a physical contact with the pirn but use a beam of light.

The principle behind any feeler action is that, once every two picks, the presence of yarn on the stem of the pirn about 4 cm from the pirn butt is investigated. If yarn is present, the feeler will allow the loom to continue running normally, but, if there is no yarn at this point, then the feeler will activate the start of a change procedure.

On a single-shuttle automatic loom, it is usual to have the feeler mounted at the left hand side of the loom, whereas the automatic-change mechanism is normally at the right-hand side.

9.2.2 Weft Reserve

The bobbin change may take place approximately one or even two complete pick cycles after the feeler has indicated the need for a change, and, since it is obviously undesirable that the shuttle should be traversing the loom without leaving a trail of weft, then it is necessary to have a special reserve store of yarn on the pirn that may be used at this time. This reserve store is known as a 'bunch' (see Fig. 1.5), and it is wound onto the stem of the pirn adjacent to the butt. During winding, it is given a very limited traverse so that it will not spread and interfere with the feeler action.

The length of yarn on the bunch is dependent on whether the change will take place on the first or the second pick after feeler detection.

Let it be assumed, however, that, when a feeler makes contact with a pirn, it is prevented from activating a change by the very last coil of yarn on the pirn before the bunch. By the time the shuttle has traversed about 30–45 cm across the sley, this coil of yarn will have been used, and yarn will be being withdrawn from the bunch for the whole of the remainder of that pick and for the whole of the next pick, at the end of which the feeler will indicate the need for a change. Another pick has to be withdrawn from the bunch as the shuttle traverses to the change side, and there must still be sufficient yarn for the thread to reach from the selvedge to the shuttle eye and onto the pirn so that it will be held under tension and thus in position to be held and cut by a part of the mechanism. If this total length of yarn is added up, it will be realized that something between three and four pick lengths will be needed in a bunch, and, if the change is to take place two picks after detection, then the bunch length will have to be from four to five times as long as the cloth width in the reed.

These figures are quite independent of the width of the loom and the linear density of the yarn, and the waste percentage will be considerably higher in wide looms or when coarse yarns are used. The average amount of waste is equivalent to $1\frac{1}{2}$ pick lengths provided that the minimum amount of yarn is wound onto the bunch without allowing any possibility of the weft's running out before the change.

9.3 Feeler Position

Any feeler that has to make contact with the pirn for the purpose of detection must either (*a*) let the pirn advance to it or (*b*) rise or fall to make contact with the pirn.

The first instance is probably the more common, since the natural reciprocation of the sley can be effectively utilized. As the sley comes forward to front centre, the feeler that is mounted on the loom frame is allowed to penetrate holes in the shuttle-box front and the shuttle wall to make contact with the pirn at the appropriate point. When detection is made by this method, it has to be held for rather a long time before the change operates, and it may be necessary to modify the design of the drive to the change mechanism for this purpose.

This explains the advantage of the second method, which carries the feeler unit on the sley and in which the feeler is raised and lowered by cam operation to make contact at the most appropriate time in the pick cycle.

The latter method requires more moving parts and is quite unsuitable when the feeler has to be situated at the change side of the loom, as in multiple-box looms, but it does not require any weakening of the box front or shuttle wall.

9.4 Types of Feeler

9.4.1 Classification

Feelers may be classified as follows:

- (I) mechanical:
  - (*a*) side-sweep;
  - (*b*) depth:
    - (i) diameter gauge;
    - (ii) penetration;
- (II) electrical:
  - (*a*) two-prong;
  - (*b*) photo-electric;
- (III) mechanical–electrical.

Mechanical feelers are generally more robust and cheaper, but the yarn is given a rather rough treatment because the contact that the feeler blade must make with the weft package must be strong enough to give sufficient movement to the feeler in order to instigate a change. This type of feeler is quite unsuitable for the more delicate continuous-filament yarns, for which the lightly sprung two-prong feeler is favoured. Even this light contact may be undesirable with certain delicate monofilament yarns, and a photo-electric feeler, which does not need to make any contact, may then be necessary.

9.4.2 The Side-sweep Mechanical Feeler

This mechanism is also known as the 'midget feeler'. The feeler blade makes contact with yarn on the pirn when the loom is nearing its most forward position by penetrating holes in the shuttle-box front and the shuttle wall. It will be pushed in the direction of arrow A (Fig. 9.3) because the serrations in the tip of the blade will have become embedded in the yarn. As the sley recedes, the blade will be returned to its original position by the spring.

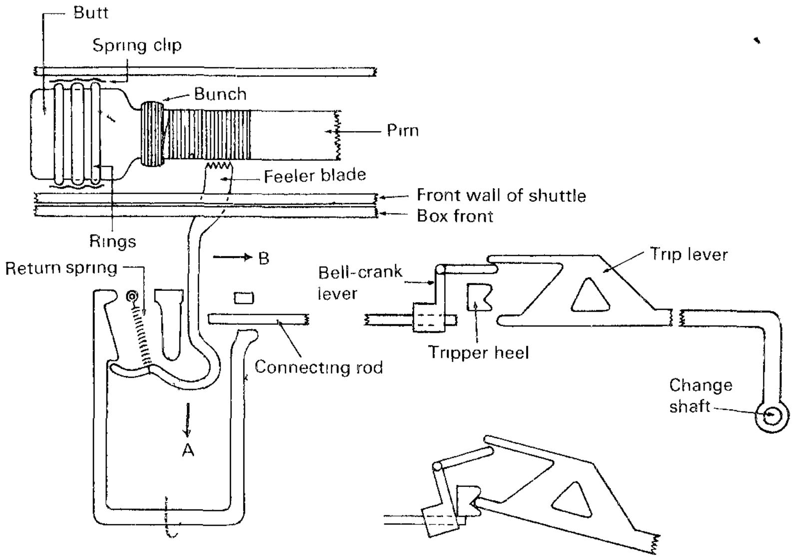


Fig. 9.3 A side-sweep feeler

When yarn is being withdrawn from the bunch, the feeler blade will slide to the right (arrow B) on the smooth polished surface of the pirn as the sley approaches front centre. This will cause the connecting rod to push the bell-crank lever, whose upper arm will then raise the trip lever.

The tripper heel, which is attached to, and therefore oscillates with, the weft fork hammer, will now push the lower arm of the trip lever (see inset in Fig. 9.3) and thus cause a sufficient rotation of the change shaft to effect a bobbin change when the shuttle arrives in the box on the opposite side of the loom.

The feeler blade is returned to its normal position by its spring, the other parts being returned by a spring on the change mechanism.

It is important to note at this stage that the shape of the weft-fork cam is very important in executing a bobbin change because it must not only be capable of oscillating the weft-fork hammer but must also hold the change mechanism's parts in position at the opposite side of the loom until they are locked in readiness for a change almost a pick later.

9.4.3 Depth Feelers

For many years, a feeler that is lowered onto the pirn by a cam action has been used on Saurer 100W cotton looms.

If there is yarn on the pirn stem adjacent to the bunch, then the feeler rod's downward movement is limited, as shown in Fig. 9.4, but, as soon as the yarn has been used, the feeler can make a complete downward movement, and the projection from the feeler rod will strike the trip lever to cause a partial rotation of the change shaft and thus ensure a bobbin change when the shuttle arrives in the opposite box.

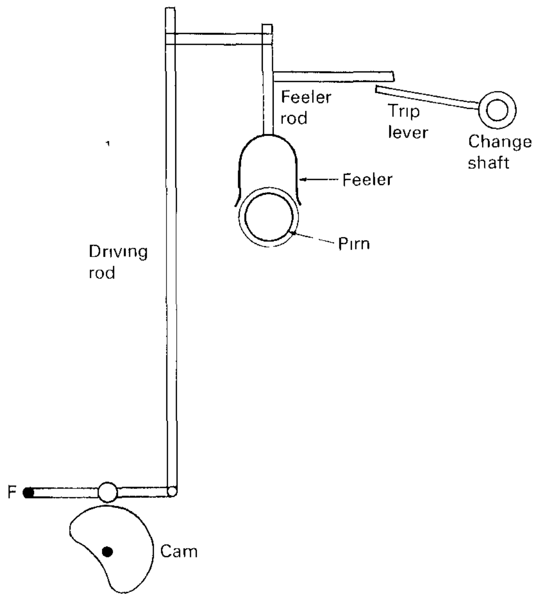


Fig. 9.4 A depth feeler

An alternative type of depth feeler uses a blade that makes contact with a pirn opposite a hole in the pirn stem.

The presence of yarn limits the feeler movement, but, if sufficient yarn has been used to reveal the hole, then the feeler will penetrate the hole and sub sequently bring the change mechanism into operation.

The main weakness of this mechanism is that the position of the hole in the pirn must be guaranted by ensuring that a cut-out in the butt of the pirn fits on a projection in the shuttle. This obviously precludes the bobbin change, but this type of feeler has been used sucessfully on shuttle-change and semi-automatic looms.

9.4.4 The Two-prong Electric Feeler

The current supply is reduced from mains voltage by a transformer, which is similar to the one described in Section 7.2.3. It could even be the same transformer.

The circuit is incomplete until contact is made across the feeler prongs. This contact is made when the sley is approaching its most forward position and a metal band set into the stem of the pirn adjacent to the butt has been uncovered (i.e., yarn is being withdrawn from the bunch). The completed circuit energizes the solenoid so that the magnetized core attracts the trip lever into a position opposite to a suitable driving point, which, in Fig. 9.5*A*, is an oscillating tripper heel.

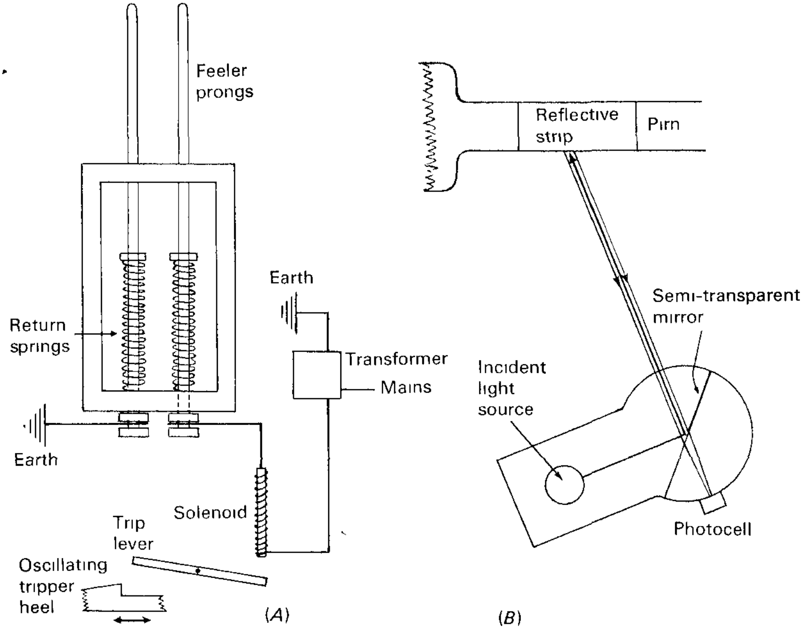


Fig. 9.5 Electric feelers

Since yarn does not conduct electricity, the circuit is not completed when the metal band is covered with yarn.

9.4.5 The Photo-electric Feeler

When this type of feeler is used, it is necessary to cover the pirn stem adjacent to the butt with a reflective strip.

Incident light is redirected onto a point that is in line with the path of the shuttle. Since this system can operate from a momentary reflexion of a light ray, it is possible to inspect the pirn by the feeler when the shuttle is in flight across the loom.

When the light ray is thrown back to the feeler head by the uncovered reflective strip on the pirn (Fig. 9.5*B*), it passes through a semi-transparent mirror to the photocell, where it produces a photo-electric impulse that is amplified by a fully transistorized electronic unit, which in turn activates an electrical circuit and solenoid very similar in principle to those used in the two-prong electric feeler.

9.5 Single-shuttle Automatic Bobbin Change

9.5.1 The Bobbin-change Mechanism

A very high percentage of single-shuttle automatic looms are equipped with the feeler unit at the left-hand starting-handle side of the loom, so that the weft-fork hammer movement can be utilized to assist the bobbin change that takes place one pick later at the right-hand side of the loom.

The partial rotation of the change shaft that results from the tripper heel's pushing on the trip lever causes the swan neck and thus the shuttle-protector lever to swing in an anti-clockwise direction from their normal position, and this lifts the pillar and protector arm. The depresser thus releases its hold on the latch, which then rises by virtue of its coil-spring-loading into a horizontal position.

As the sley approaches front centre, the shuttle having arrived in the right-hand box, the bunter pushes the latch, which now forms a rigid connexion with the hammer. As the hammer moves in a downward direction, it forces the bottom bobbin out of its spring-held position in the magazine onto the top of the bobbin in the shuttle. The hammer pressure continues until the almost-empty pirn is forced out of the shuttle (Fig. 9.6*A*) and through holes in the bottom of the shuttle and the box base, and the new full pirn is positioned in the shuttle.

The weft-fork cam, which has been designed to give a full throw for a period of just over 180° of crankshaft rotation (many more degrees than are required to stop the loom for a weft break), allows the tripper heel to release its pressure on the trip lever as the sley recedes, and this permits the return spring to pull the protector arm and thus the latch back into their normal positions (Fig. 9.6*B*).

When the hammer makes its downward movement, the feed pawl slips down one tooth on the ratchet, so that, when the other parts are being returned and the hammer coil spring raises the hammer to its original position, the magazine will be pushed round one section to leave the next bobbin in position for the next change.

Complete efficiency of the bobbin-change mechanism can only be guaranteed if:

(i) the shuttle is in the correct position for a bobbin transfer;
(ii) the tension in the picks immediately before and after a change is satisfactorily maintained;

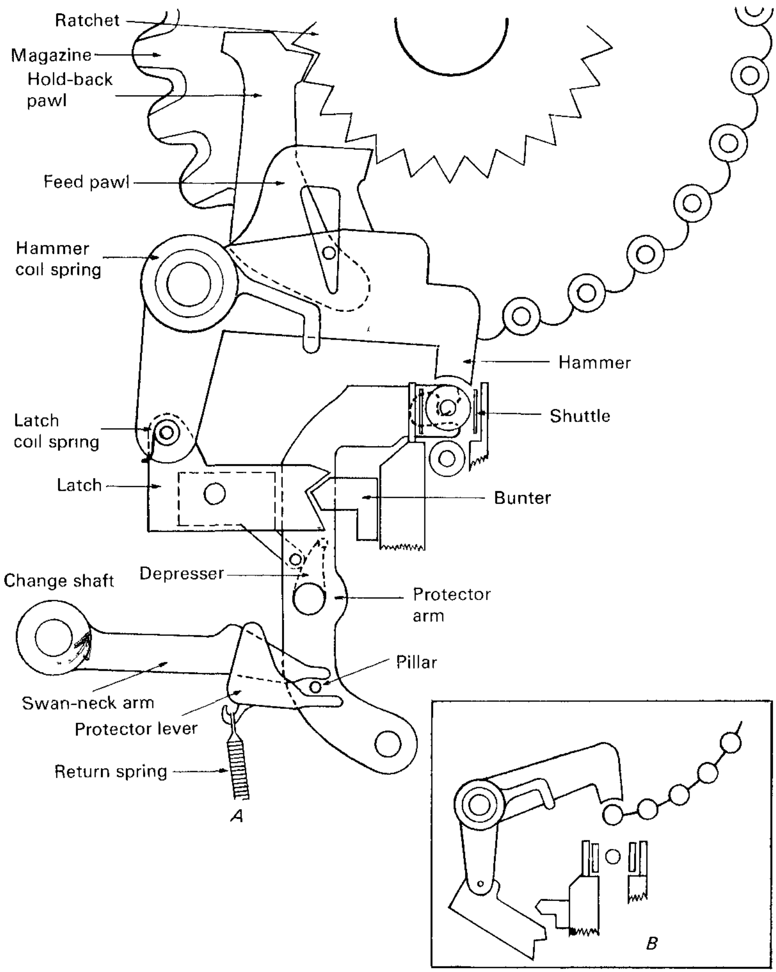


Fig. 9.6 A bobbin-change mechanism

(iii) the ejected bobbin is guided into a storage can; and
(iv) the weft yarn from the new pirn is guided into the shuttle eye.

9.5.2 Shuttle Protection

The position of the shuttle in the box may vary depending on several factors (Section 2.5.2), but, if it is too far into the box, the new bobbin will be forced onto an inclined plate in the shuttle that holds the bobbin clip in position, so that the incline will allow the bobbin to slide into position. The possibility of damage under these circumstances is therefore very slight, but, if the shuttle fails to travel far enough into the box, the new pirn may be forced onto the shuttle eye, straight through the shuttle, or into a position where it is supported by one ring only. Each of these circumstances presents its own problems: the first one will certainly cause damage to the pirn and probably the shuttle eye, and the second will probably result only in a loom stop due to the absence of weft, although the pirn may be left hanging in a vertical position through the shuttle and box base, which will cause damage when picking takes place. A pirn held with one ring is also undesirable, since it may move up and down in the shuttle and cause weft breaks, or it is possible that the pirn may be pointing upwards out of the shuttle, so that it will attempt to pass over the top shed while the shuttle is passing under it in picking from the left, so that many ends are caused to break out.

These problems exist because a uniform rest position of the shuttle cannot be guaranteed and because of the limited time available to perform the actual transfer (Section 9.1). With this in mind, Rüti, who have run their model C loom at 360 picks/min at exhibitions, introduced a shuttle in which the back wall springs back (Fig. 9.7) at the time of actual transfer and then returns under

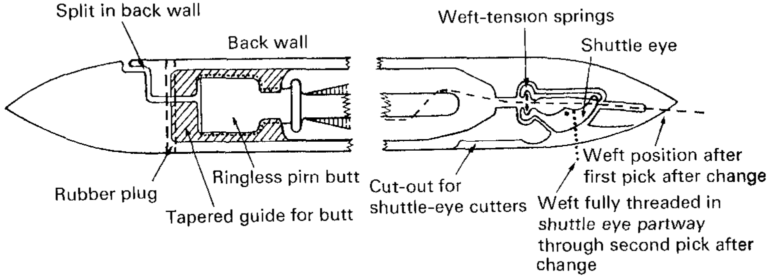


Fig. 9.7 A split shuttle

spring-loading to trap a pirn that does not have rings around the butt. The shuttle is tapered inwards at the point of butt entry to allow an increased degree of latitude in the shuttle-rest position. The pirn is thus held in a constant position in the shuttle and also in a much more rigid manner than is the case when rings are used.

It is not possible to avoid a change if the shuttle is too far into the box, but regular examination of the shuttle and picker will indicate whether this is happening.

When the protector arm is raised, it takes up a position that allows it to be opposite the entrance to the shuttle-box when the sley approaches its front-centre position. If the shuttle is correctly boxed, there will be a gap of approximately 3 mm between the shuttle tip and the tip of the protector arm, but, if the shuttle is protruding from the box, it will make contact with the protector arm, which will be pushed towards the front of the loom and cause the depresser to lower the latch out of the path of the approaching bunter so that a change will be avoided. The loom will then be stopped either by a 'bang-off' or by the weft's running out.

9.5.3 Weft-tension Control

The yarn from each pirn loaded into the magazine is wrapped round the mandrelle of the magazine so that the first pick inserted from that pirn will be at a tension that is approximately the same as that of all the other picks. Securing the weft in this way is, of course, essential in the first place to ensure that a trail of yarn is left behind the shuttle.

The last pick of yarn from the old pirn presents a more difficult problem, which has really only been overcome with the development of shuttle-eye cutters. This unit is mounted on a shaft that oscillates as a result of the protector arm's rocking backwards and forwards.

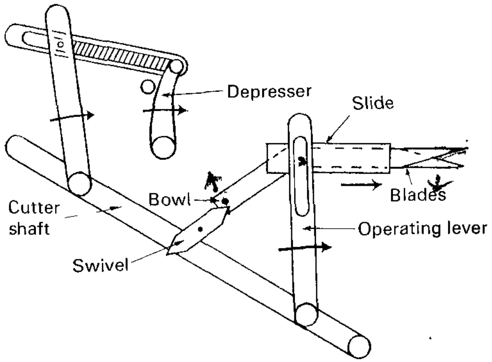


Fig. 9.8 Shuttle-eye cutters

As the depresser swings clockwise, it causes the cutter shaft to rock forwards towards the box, so that the cutter unit, which consists of four blades, moves forward in its slide bracket (Fig. 9.8), and, as the bowl passes over the swivel, the blades open so that they are round the weft behind the shuttle eye just before the change takes place. When the sley reaches its most forward position, the pair of blades nearest to the selvedge closes and traps the yarn just before the other pair of blades closes and cuts it. The slide bracket is inclined to the selvedge so that the distance from the holding blades to the cloth fell at the selvedge is always constant and the thread will not break because of any increase in tension. The two outside blades of the unit of four are fixed, and the opening and closing of the two centre blades are achieved either by a bowl passing over a T-lever or, in more recent looms, by a bowl passing round a spring-loaded swivel plate as in Fig. 9.8.

9.5.4 Temple-cutting

As the first picks of weft are inserted after a bobbin change, there are two threads stretching from the selvedge, one to the magazine mandrelle and the other to the shuttle-eye cutters. It is undesirable that these threads should be allowed to break, since such a break will take place at the weakest point along the length of yarn, and, if a long length of thread is left trailing from the selvedge, it can easily be dragged into the cloth to cause a fault known as *lashing-in* (or *backlashing*). This is a fault that can be successfully repaired in several instances, but the introduction of a cutting unit attached to the temple unit can easily obviate this need.

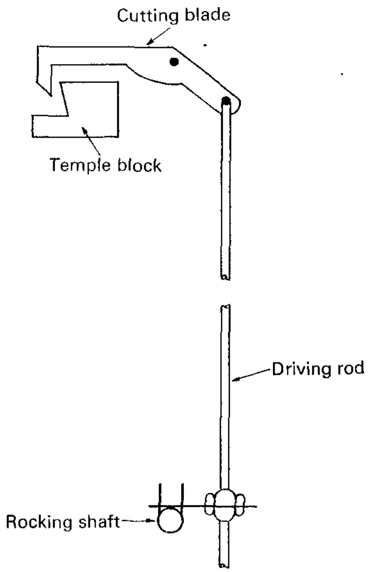


Fig. 9.9 A temple cutter

The threads are easily guided into the space between the blades of the temple cutter as the cloth is drawn forward. A cutting blade is made to oscillate from the movement of the rocking rail (Fig. 9.9) or from the reciprocation of the sley, and this will cut the threads and thus leave them to hang from the mandrelle and the shuttle-eye cutter.

9.5.5 Ejected-bobbin Control

The ejected bobbin can be guided into the can by two inclined plates, attached to the loom frame in such a position that they are directly under the shuttle-box at the time of the bobbin change. Alternatively, a swivel plate is rocked by a bowl operated from the oscillation of the cutter shaft. The swivel plate is attached to the sley and must be moved out of position quickly after a change so that it is out of the path of the picking stick when it swings forward.

Ejected bobbins that are not guided into the can provide a dangerous hazard on the weaving-shed floor in addition to the possibility that they may become chipped or soiled, the result being faulty cloth at a later stage of processing.

9.5.6 Self-threading Shuttle

The bobbin change is only eventually completed when the new weft thread is lying in the shuttle eye. This is achieved by using shuttles having self-threading eyes (Fig. 9.7). When the shuttle is projected from the right-hand box, the weft thread is held onto the top of the eye by the top shed line, so that it will easily slip into a specially shaped groove (indicated by dashes), where it will remain when the shuttle reaches the left-hand box, but, after the shuttle has been picked from this box and the length of trailing slack yarn pulled tight, the weft thread is pulled into the shuttle eye along a second groove (a dotted line in Fig. 9.7).

9.6 Bobbin Loaders

The bobbin loader is gradually becoming obsolete, mainly because it has only a limited advantage over the rotary magazine as compared with the loom-winder. Several machinery makers produced bobbin-loader units, but the Fischer ALV system appeared to enjoy the widest popularity. It replaced the battery of the conventional automatic loom, but it can operate only if the following conditions exist.

- (i) The pirn boxes have a special sliding base. This base covers the slot in the bottom of the box during transportation, but, when the box is placed in the operating position, a handle is raised to slide the base back so that the pirns can fall through the slot and roll down the pirn guide.
- (ii) The empty pirn is completely smooth for about 2 cm at its tip, and the wound pirn has a bunch of yarn near to its tip.
- (iii) The pirn-winding machines are equipped with a nose-bunching attachment.
- (iv) There is a supply of compressed air to operate the unit.

When the transfer hammer is depressed to operate a bobbin change in the normal manner, it releases the pneumatic control to the bobbin loader (Fig. 9.10) so that the compressed air can activate the various parts of the unit and thus prepare the next pirn for transfer. Four valves are used for this purpose:

valve 1: (*a*) lowers the clamp so that the hole in the plate is in line with the tube;
(*b*) opens the air valve to the vacuum pump to produce air-suction;
(*c*) causes the pair of stripping jaws to move to the left and close onto the pirn behind the nose bunch;

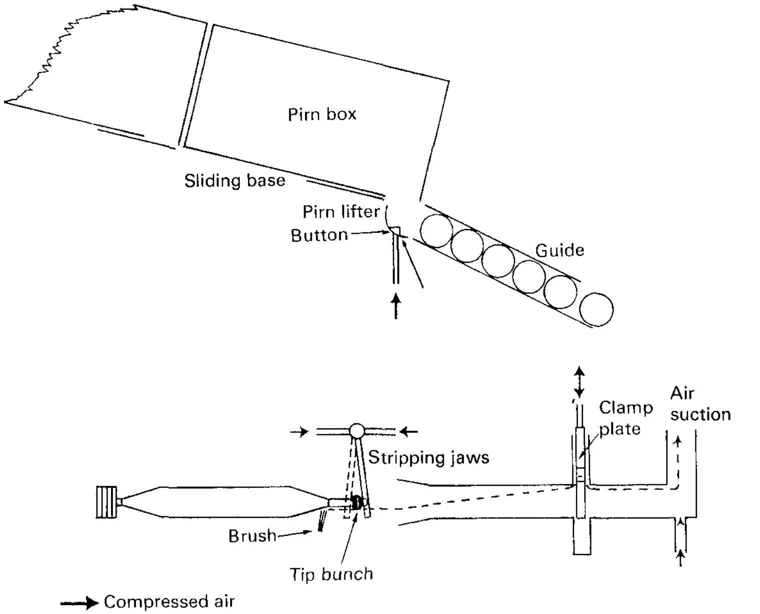


Fig. 9.10 A bobbin loader

valve 2: activates the pirn lifter if the button on the lifter plate is not depressed; this will happen if the pirns do not fall freely from the box into the pirn guide, i.e., its function is to release any blockage that is causing the pirns to jam in the box and thus preventing them from falling freely;

valve 3: returns the closed stripping jaws to their original position, which thus removes the nose bunch from the pirn; the yarn from this bunch is sucked down the tube, but excessive yarn cannot be sucked to waste because of the presence of a brush that is resting on the chase of the pirn; and

valve 4: raises the clamp plate to trap the weft in readiness for the next pirn transfer.

The bobbin loader is relatively simple to maintain, but it takes time to prepare a bobbin, and a second transfer cannot take place within 3–5 s. Furthermore, it is not advisable to refeed part-used pirns since they may roll sideways in the pirn guide and cause a jam. For a successful transfer, the bobbin loader relies on the other bobbins in the pirn guide to hold the bobbin to be transferred steady at the time of transfer, and, if for some reason there are no other bobbins in the guide at the actual time of transfer, then it may jump and be broken if the hammer comes down to strike it in the wrong position.

9.7 Loom-winders

The magazine-creeling of the supply cones has already been referred to (Section 9.1), and the yarn is tensioned for winding by a series of disc tensioners mounted on a plate, which is suspended from the loom frame on springs so that vibration from the loom will not cause the yarn to be shaken out of the tension unit.

Pirn-winding is done on a spindleless, feelerless machine. The absence of a spindle makes the pirn-transfer operation much simpler, and the gear-driven reciprocating-traverse guide avoids the need to use a feeler to determine the pirn diameter, so that any possibility of damage to the pirn from abrasion between the feeler wheel and yarn is eliminated. An adjustment to the gear-driving mechanism is needed every time there is a change of weft, but any variation in weft tension or linear density during normal running will immediately show up as a change in pirn diameter.

When a bobbin transfer takes place, the bobbins in the loom magazine fall, and, if there is a full pirn waiting at the winder, it will be positively transferred to the vertical stack magazine. This can only happen if there is at least one empty pirn in the winding magazine.

The weft yarn trailing between the tension unit and the loom magazine is picked up by two gripper plates. The lower plate moves from left to right, and, after cutting the yarn, carries the end of weft from the wound bobbin to the yarn-ends drum (see Fig. 9.11), where it holds the yarn until the rotating drum picks up the yarn, traps it, and holds it parallel to the pirn throughout its time in the magazine. The upper plate, moving from right to left, carries the other end of the yarn to a point opposite the butt of the new pirn in the winding position. Wire loops on the shoulder of the pirn (known as *cleats*) then pick up the yarn so that winding of the next pirn can commence.

It is necessary to arrange for the winding rate to be in excess of the weaving rate so that any stoppages in winding will not result in the loom's being stopped while it is waiting for weft. This excess, or reserve, winding capacity is usually in the range of 10–20%. There will thus be a time when the loom magazine is full, and this is detected by the bobbin-support plate, which assists the wound bobbin from the winding position to the loom magazine. If this plate is held flat against the back of the loom magazine by the top bobbin in the stack, then the transfer of the next fully wound bobbin cannot take place until the plate rises again, i.e., after the next pirn transfer.

When such a transfer is signalled, the clamps trap the weft that has been lightly held under control by the yarn-ends drum while the pirn has been in the magazine. The transfer can then take place in the usual manner. When the temple cutter has severed the two lengths of weft extending from the selvedge, the threads are picked up by a cut-out on a reciprocating nylon rod and fed into a suction tube to eliminate any possibility of backlashing.

The butt of the ejected bobbin will pass over a non-slip surface just below the box base, which will cause the pirn to spin and release yarn as it falls down the chute and through a horizontally rotating brush into the stripping unit. It is essential that the trailing yarn should extend through this brush if it is to be

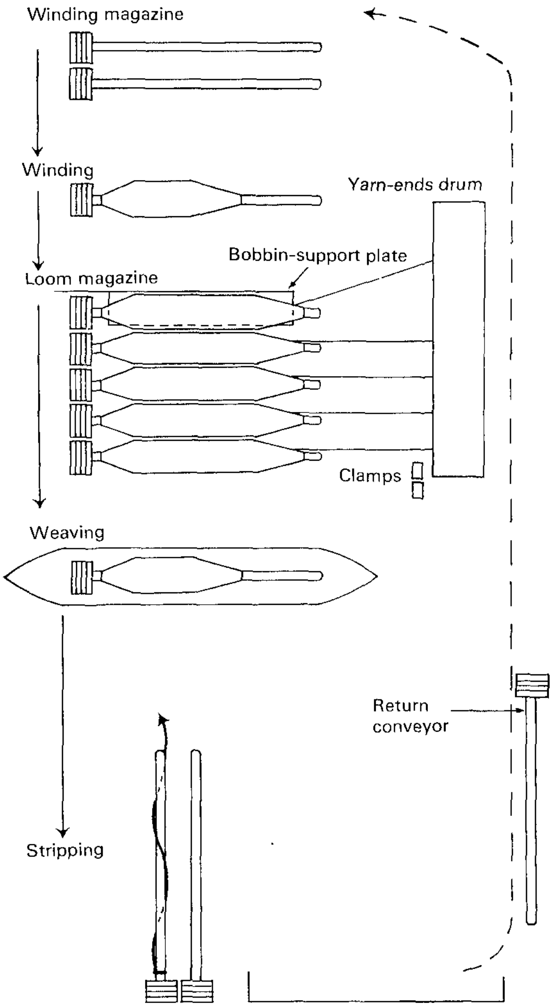


Fig. 9.11 A loom-winder

picked up for successful stripping. When the pirn reaches the stripping unit, it must first be pushed sideways into the stripping position. By this time, the rotating brush should be transferring the yarn to a revolving cone with a brushed-nylon surface, and it is this cone which then unwinds the yarn from the bobbin. The cone has three toothed reciprocating arms set into its surface to give the unit a self-cleaning action.

There are usually two pirns in the stripping unit at any one time, i.e., the one actually being stripped and the one that has already been stripped. This second bobbin helps to hold the new one steady during stripping so that there is less chance of the yarn's breaking during unwinding and thus leaving an unstripped bobbin. When the next pirn enters the stripping unit, the bobbins are pushed sideways, and the right-hand bobbin will fall into a tray from which it will be picked up by a magnet (acting on the rings on the butt) and carried back to the winding-unit magazine on a conveyor chain. If for any reason there is still some yarn on the extreme right-hand pirn in the stripping unit, it will be supported by a brush until after the next change, when it will be pushed even further to the right before being allowed to fall into a reject tray. It is necessary, and possible, to refeed these pirns to the stripping unit by hand.

The factors that are most likely to interfere with the efficiency of the loom-winder are an excess of unstripped bobbins and bobbin jams in the stripper chute, resulting from a bobbin's being ejected tip first. The efficiency of the clamps has also caused problems at times, and there is the further possibility that bobbins may break at the time of transfer if there is not more than one bobbin in the loom magazine.

It is a simple matter to calculate how long the winder will be stopped between the winding of successive pirns if both units are running at 100% efficiency. For example, a loom running at 192 picks/min, weaving a fabric 130 cm wide in the reed, is equipped with a loom-winder with a spindle speed of 6000 rev/min. During winding, 50 cm of yarn are wound onto a five-wind pirn chase for each double traverse. Each pirn contains 25 g of 40-tex yarn.

We have:

$$\text{yarn per pirn} = \frac{1000}{40} \times 25 = 625 \text{ m},$$

yarn wound per minute (100% efficiency assumed)

$$= \frac{50}{5 \times 2} \times \frac{1}{100} \times 6000 = 300 \text{ m},$$

$$\text{time to wind pirn} = \frac{625}{300} = 2{\cdot}083 \text{ min},$$

$$\text{yarn woven per minute (100\% efficiency assumed)} = 192 \times \frac{130}{100} = 249{\cdot}6 \text{ m},$$

and

$$\text{time to weave pirn} = \frac{625}{249{\cdot}6} = 2{\cdot}504 \text{ min},$$

so that:

$$\text{time for which winder is stopped} = 2{\cdot}504 - 2{\cdot}083 = 0{\cdot}421 \text{ min}.$$

This represents a reserve winding capacity of:

$$\frac{0{\cdot}421}{2{\cdot}083} \times 100 = 20{\cdot}21\%,$$

It is more reasonable to use this type of calculation to determine the maximum spindle speed of a loom-winder for a particular loom. For example, a loom designed to run at 240 picks/min with a maximum reed space of 120 cm winds 56 cm of yarn onto the chase while making 14 revolutions in one double traverse. If there is a reserve winding capacity of 20%, it is possible to determine the maximum spindle speed required.

We have in this case:

yarn woven per minute at 100% efficiency $= 240 \times 1{\cdot}2 = 288$ m,

yarn wound per revolution with 20% reserve winding capacity

$$= \frac{56}{14} \times \frac{1}{100} = 0{\cdot}04 \text{ m},$$

and:

yarn to be wound per minute for a reserve winding capacity of 20%

$$= 288 \times \frac{120}{100} = 345{\cdot}6 \text{ m},$$

so that:

$$\text{required number of revolutions per minute} = \frac{345{\cdot}6}{0{\cdot}04} = 8640 \text{ rev/min.}$$

It would be more usual to supply such a machine with a standard spindle speed of, say, 9000 rev/min, and it will be appreciated that such a speed is relatively low (even for a loom running at high speed) when compared with the normal spindle speed on a modern pirn-winder, where the highest possible speeds are desirable to minimize the number of spindles required to keep a shed in full production.

9.8 Multishuttle Weft Replenishment

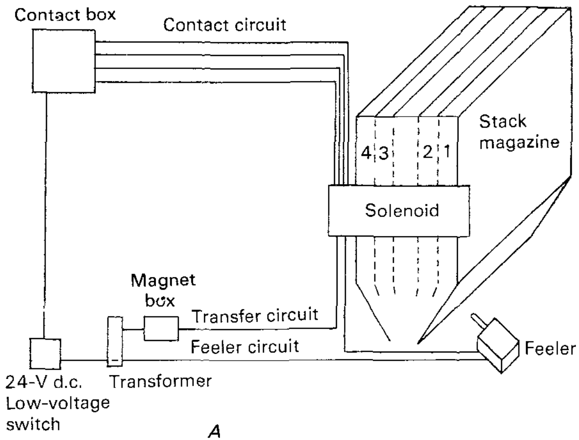
The implementation of automatic weft replenishment is extremely difficult on a loom with a multibox unit at each side, and only Crompton & Knowles in the PAPA 4 × 3 box loom have really claimed commercial success in this field.

Bobbin transfer in 2 × 1 and 4 × 1 drop-box looms is more easily achieved at the single-box side of the loom, but, since this is also the more convenient side for the feeler unit, problems arise because it is not possible to detect the need for a change and complete the transfer in the limited time for which the sley is in its most forward position. Thus it is necessary that detection should be made on one pick and the transfer executed two picks later, always provided that a box change has not taken place in the meantime. If a box change does take place while the shuttle is in the left-hand box, then the transfer mechanism must be capable of memorizing the need for a change in that particular shuttle until the shuttle is again brought into use. The memorizing of the need for a transfer may be done mechanically or electrically. Northrop have successfully used the electrical method for many years with both circular and stack magazines. Circular magazines have the bobbins placed in sequence round the magazine, depending on their position in the box unit, and it is necessary to ensure that the correct bobbin is spun round to the change position before the hammer descends for the transfer. Stack magazines are more popular, since less mechanical movement is required between detection and transfer and it is also possible to have more bobbins of each colour in the magazine at any one time, which thus reduces the walking time of the battery filler.

Electric power for the change unit is supplied by a 24-V d.c. transformer and low-voltage switch when the loom's starting handle is in the 'on' position. Whenever a box change takes place, a circuit (Fig. 9.12*a*) between the appropriate contact in the contact box (determined by the shuttle-box level with the raceboard), the corresponding solenoid in the solenoid box, and the feeler is capable of being completed as the result of the feeler's detecting the need for a change. When this happens, the energized solenoid activates its plunger, which causes the trigger (Fig. 9.12*b*) to move down, and the lower end of the rocker unit will swing to the left into the tilted position so that the contact will be touching the top and not the bottom carbon, i.e., the transfer circuit is now completed through the appropriate contact circuit. The magnet box can now cause the trip lever to be raised opposite the tripper heel, but by this time the shuttle has returned to the left-hand side of the loom. If there is no box change, the transfer can be effected when the shuttle is returned to the right-hand side of the loom, but a box change will de-energize the circuit as a result of the break in the appropriate contact of the contact box, and only when the shuttle-box in question is returned to the weaving position will the contact be remade and the circuit re-energized for a transfer. It is possible for a transfer to be detected and effected in one shuttle while other circuits are de-energized after the initial detection.

When the circuit is complete for the change, the trigger lever is pushed down so that it can be caught by a boss on the rocker shaft, which oscillates once every two picks from a grooved cam that is gear-driven from the crankshaft. This force on the trigger lever causes the bobbin-arm holder to rock and release the bottom bobbin only from the appropriate stack of the magazine. The bobbin will then roll down to the change position in readiness for the transfer, which will then be effected in exactly the same manner as that described in Section 9.5. When the rocker shaft returns, the spring will return the trigger lever, plunger, and trigger, and the eye rod will push the friction plate to return the rocker unit.

The technique used for selecting, holding (if required), and transferring the



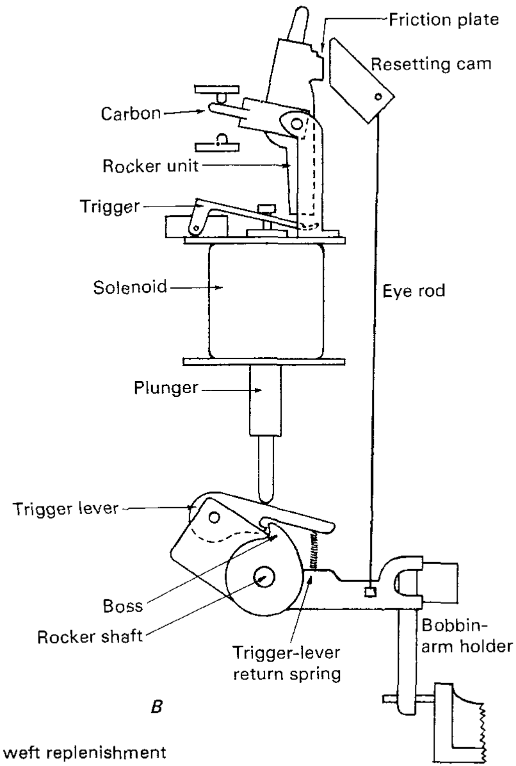


Fig. 9.12 Four-colour weft replenishment

correct bobbin is unaffected when a bobbin loader or loom-winder is used on a box loom.

The Fischer ALV bobbin loader has been applied to a 4 × 1 box loom by loading the back two stacks of the magazine from two individual bobbin loaders, situated one above the other, with a longer pirn guide from the top unit feeding pirns to the stack for box 3. The front stacks are loaded by hand, and it is recommended that the least popular colours should be stored in the front two stacks, i.e., boxes 1 and 2, but this is contrary to normal procedure, whereby the least popular colours are usually placed in boxes 1 and 4. Since the saving in battery-filling is not great and it may be more difficult for the designer to avoid a 1–4 box change, this technique has not proved popular. The Rüti bobbin loader transfers the bobbins from the box to the stack by a positive mechanical lifting action so that, in four-colour work, the problem of design limitation does not exist, but, even so, bobbin-loading is generally unattractive.

Unifil have developed their loom-winder for use on a 4 × 1 box loom, and, although the principles are very similar to those described in Section 9.7, two notable modifications have been necessary. In the first instance, the bases of the pirn butts are drilled to a predetermined depth depending on the box in which the pirn is intended for use. The cut-out for the top box is shallowest and that for the bottom one is deepest. The pirns are fed from the winding magazine as soon as the winder is available, and a feeler from the winder box makes contact with the base of the pirn butt. From this contact:

(*a*) the weft guide-plate positions itself so that the yarn from one supply package, which is for use in a specific shuttle-box, is placed opposite the pirn in readiness to be picked up by the cleats on the butt;
(*b*) the gears in the winder box are adjusted so that a pirn of the correct diameter is produced (this being determined by the weft-yarn diameter); and
(*c*) the delivery of the full pirn after winding is into the correct stack of the loom magazine.

A second modification is then required at the stripping unit, where it is possible that several pirns may be ejected from different shuttles in a very short time. The stripping unit is therefore equipped with a receiving box, from which the pirns are delivered to the actual stripping area by a conveyor chain when the yarn from the preceding pirn is no longer being unwound.

The four-colour Unifil (Multifil) appears to be most effective, but its relatively high cost may account for the rather slow acceptance of the unit in the industry. It is probably more realistic to consider a loom using some form of random weft selection from magazine-creeled packages and thus at the same time eliminate the design limitations of double picks and limited box-change movements.

CHAPTER 10

Loom Drives

10.1 Methods of Driving the Loom

Since the Industrial Revolution, the tendency has been to house a substantial number of machines in one building and to drive them from one source of power. In the early stages, this was often a water wheel, and a few worsted-spinning mills were still being operated by water power as late as the 1950s. By this time, however, the common source of power was usually a steam engine. In the early part of the twentieth century, the looms in most weaving sheds were driven through overhead shafting from a steam engine. A large steam engine is more economical than a small one, and this led, naturally, to the 'room-and-power' arrangement, which was very common in Lancashire. Some en trepreneur would build a weaving shed large enough to accommodate from 400 to 800 looms and provide it with an engine housed in a separate engine house. The space in the weaving shed would then be let to a weaving firm, or more usually to several firms, each of which rented sufficient space for its needs. They paid for the proportion of the total space and power that they used, and hence the term 'room and power' came into use. This was an important system because it allowed, for example, an enterprising overlooker to install a few looms, say, twenty, with finance provided by a trusting banker. Sometimes the bank's confidence was vindicated, and a substantial weaving firm developed from small beginnings. Sometimes the venture ended in failure, usually for commercial rather than technical reasons. One of the more successful competitors would then take over the looms, or the entrepreneur who originally provided the room and power would acquire the looms of his tenants and would himself establish a weaving firm with a substantial capacity.

It was therefore common to find a steam engine supplying power to several hundred looms. The engine drove a main shaft, which in turn transmitted power to a series of line shafts, suspended from the roof and covering the whole of the weaving shed. On these shafts was a pulley for each loom. When the loom was stopped, the belt connecting this line-shaft pulley to the loom was positioned on a loose pulley on the loom shaft, but, when the starting handle was moved into the running position, the belt was transferred to the pulley fixed to the loom shaft, and the loom began to run. The pulley on the line shaft necessarily had a width at least equal to the combined widths of the fast and loose pulleys on the loom shaft, and starting or stopping the loom involved a sideways shift of the whole belt. A degree of feel and skill was required to start looms by this process in order to ensure that sufficient momentum was gained

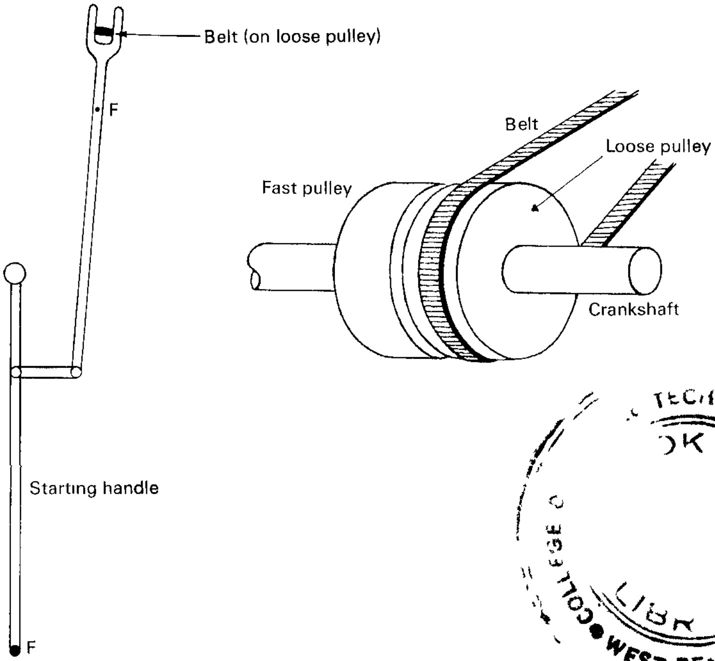


Fig. 10.1 A fast-and-loose pulley

quickly enough for a successful and complete traverse of the shuttle across the loom on the first pick. Similarly, only by correct manipulation could the loom be stopped safely in the required position.

The use of one driving point for the whole factory was an economic method of driving a large number of machines in spite of the fact that there were large transmission losses and an extensive loss in production if a breakdown occurred in the main power generator. The system was eventually and gradually replaced by several large electric motors, driving quite large groups of machines. At a later stage, it was preferred to drive much smaller groups of machines from one motor, until finally, in the early 1930s, the system of having one motor for each loom in the weaving shed gained acceptance. This was mainly because it minimized power losses, especially if one loom had to be stopped for a long period, and also because the individual motors required less space, which was usually within the confines of the machine. Shed conditions were thus improved by reducing the dirt and danger associated with the heavy rotating shafts and belts. Furthermore, light and visibility in the shed were considerably improved, but it should be realized that the fire hazard was increased.

Initially, the individual motors merely replaced the line shaft and retained the dangerous and unsatisfactory fast-and-loose-pulley drive, which required a flat belt. Although it was possible by deft use of the starting handle to inch the loom forward after stopping, there was an inherent element of danger in the practice, which was therefore discouraged. Any adjustment of the loom position had then to be made by the weaver manipulating one of the accessible wheels on the main shaft of the loom by hand. It did not take long for loom makers to show a preference for V-belts or direct-gear drives, which gave better transmission and made manipulation of the starting handle for inching a more practical proposition. This eased considerably the work necessary on the part of the weaver in the performance of this task. It was, however, still necessary for the weaver to reverse the loom by hand to its back-centre position, if it was not already in this position, in order to ensure that the highest possible cyclic loom speed was achieved at the commencement of weaving.

10.2 Clutch Systems

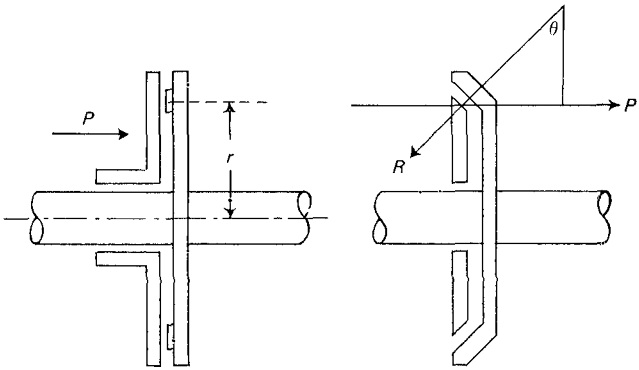
10.2.1 Direct and Indirect Drives

The use of direct-gear drive between the motor and the loom means that the motor has to be switched on and off whenever the loom is started or stopped. This system thus requires a motor having a high starting torque to ensure that the first pick can be successfully inserted. With a loom using this type of drive, there is a problem when the loom is stopped suddenly, in the event of a bang-off, for example. The momentum of the motor will cause it to continue running, and it is thus necessary to have a slipping clutch arrangement in the driving wheel on the main shaft of the loom. Such a clutch will slip and absorb the motor energy in the event of a sudden stoppage of the loom. The clutch should be adjustable in the event of wear occurring, because slipping must be avoided during normal starting, stopping, and running of the loom if the optimum loom speed is to be achieved. Furthermore, it is desirable that the loom should be brought to rest at the appropriate position for each type of loom stoppage (i.e., warp break or weft break) in order to minimize the work of the weaver.

Direct drives are not confined to gear transmission, and a similar arrangement incorporating a slipping clutch has been used extensively with V-belt transmission. The main advantage of direct drive is that there is a saving of power whenever the loom is stopped because the motor is switched off. A further advantage is that no sideways movement is required in the driving arrangement on the main shaft of the loom.

Sideways movement of either the driving wheel or the clutch (usually the latter), although minimal, is necessary if a friction-clutch arrangement is used between the motor and the loom. High initial torque is then quite unnecessary because the freely rotating clutch (or maybe driving wheel) is already rotating at the normal running speed. Provided that a highly efficient clutch system is used, the first pick will be projected at its full velocity if the loom is placed in the back-centre (180°) position before starting. The main types of driving clutch used in indirect driving systems are the plate clutch, the conical clutch, and the expanding clutch. The plate clutch has been used extensively because it is capable of spreading the area of wear over a larger area, but the conical clutch is capable of transmitting a greater torque (i.e., more power) for a given pressure applied to the frictional faces, as shown by Hanton[1].

In the simple plate-clutch system illustrated in Fig. 10.2*a*, the distance r can be assumed to be the average distance of the driving point from the centre of the shaft, and the friction force Fr will then be equal to μPr, where μ is the coefficient of friction and P the sideways thrust required to bring the clutch



(*a*) Plate clutch (*b*) Conical clutch

Fig. 10.2 Clutches

plate into effective contact with the driving wheel. In Fig. 10.2(*b*), however, the normal force must be related to the sideways thrust required and thus, since

$$F = \mu R$$

and

$$\sin\theta = \frac{P}{R},$$

then:

$$R = \frac{P}{\sin\theta},$$

and hence:

$$F = \frac{\mu P}{\sin\theta}.$$

Thus, if a frictional force of 400 N is required to drive a loom and μ is 0·50, then a sideways thrust of

$$P = \frac{F}{\mu} = \frac{400}{0{\cdot}5} = 800\ \text{N}$$

will be required with a plate clutch, but, with a conical clutch having an angle of 30°, the required thrust will be:

$$P = \frac{F \sin \theta}{\mu} = \frac{400 \times 0{\cdot}5}{0{\cdot}5} = 400 \text{ N},$$

and, if the cone angle is reduced to 10°, the required thrust will be:

$$P = \frac{F \sin \theta}{\mu} = \frac{400 \times 0{\cdot}1736}{0{\cdot}5} = 138{\cdot}88 \text{ N}.$$

Hence the conical clutch needs a lower sideways force, and, furthermore, the smaller the cone angle, then the lower will be the sideways thrust required to drive the loom. There will, however, be a greater amount of wear and an increased risk of jamming in the engaged position.

Sideways displacement involving forces of these magnitudes is generally undesirable in spite of the popularity of these systems because wear will occur in the main-shaft bearings unless adjustable-thrust bearings are used. Expanding clutches overcome this problem.

The clutch is usually mounted inside the driving wheel, and it is made to expand or contract by the starting handle of the loom, which gives a partial rotation to a rod. Each end of the clutch band carries an internally screwed sleeve, but the two sleeves are screwed in opposite directions. The ends of the rod are similarly screwed in opposite directions to pair with the sleeves. The linkage between the rod and the starting handle is so arranged that the clutch expands and the loom starts when the handle is moved to the 'on' position. Movement of the handle in the other direction will cause the clutch to contract and thus cease to drive the loom. In a more recent arrangement, the clutch band in this type of clutch is loaded with a series of springs, which force the band onto the driving wheel when the starting handle is moved to the 'on' position.

Both direct and indirect methods of drive require some type of flywheel on the main drive shaft of the loom to reduce the effect of cyclic variation in the speed of the loom on the frictional properties of the clutch. An additional wheel may be used, but the driving wheel or clutch wheel, whichever is fitted to the main shaft, is usually quite effective for this purpose. Such a wheel must not be too large or too heavy in order to prevent slow starting and overrunning on stopping.

10.2.2 Clutch Motors

More recently, motors have been introduced that incorporate the clutch. One such system (the Diehl) inverts the normal position of the stator and the rotor (see Fig. 10.3) so that the stator is fixed on the inside and the rotor rotates round it. Movement of the starting handle to the 'on' position will cause the clutch plate to be pushed into contact with the rotor because the centre shaft of the motor, to which the plate is connected, will receive a very slight sideways displacement. The loom is driven directly by gears. When the starting handle is returned to the 'off' position, pressure on the clutch plate is released, and the

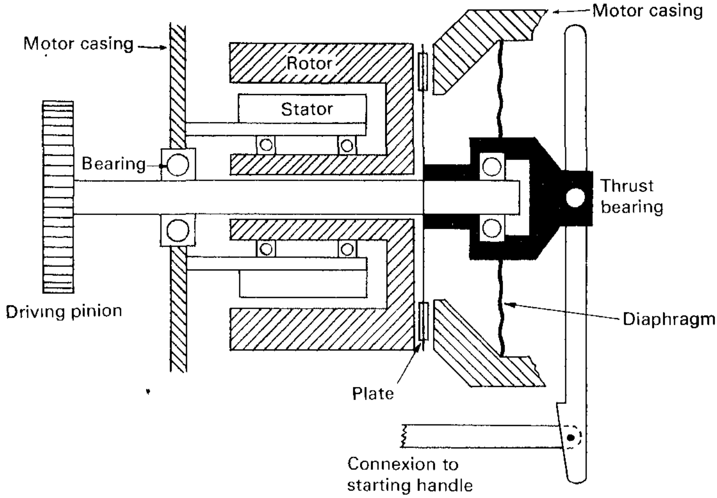


Fig. 10.3 The Diehl clutch motor

plate is moved off the rotor and onto the motor casing, which acts as a brake plate. This system has proved very effective in high-speed looms since it allows the necessary quick start, and cyclic variation in the angular velocity of the crankshaft is no problem. There is, however, a safety problem, because the loom rotor continues to revolve freely for up to 30 min after the isolator has been switched off. Under these circumstances, if an operative approaches a loom and notes that the isolator is 'off' and thus pulls the starting handle 'on' in order to perform some repair duties, the loom will turn over for two or three picks before gradually slowing to a stop. The revolving rotor can be heard if all the other machinery in the shed is stopped, but, since this is most unlikely, it is advisable that the shuttle should be removed from the loom and the loom then placed in the bang-off position (i.e., with the dagger in contact with the steel) when the starter has been switched off. The starting handle can then be pulled 'on' until all the energy in the motor has been dissipated.

10.3 Loom Control

10.3.1 Introduction

Whatever type of motor-and-clutch arrangement is used on a loom, it is essential that the weaver should be able to start, stop, and adjust the position of a stopped loom easily in the performance of his or her duties.

10.3.2 Starting Handles

Fig. 10.1 shows that, if a belt drive was used, a weaver standing in a central position in front of the loom would have had to push the starting handle to transfer the belt drive from the loose to the fast pulley. A pushing action was necessary because the fast pulley was always positioned nearer to the loom frame than the loose pulley. In addition to transferring the belt position, the starting handle was also required to release the brake in starting the loom and to reapply it in stopping the loom.

The advent of the individual motor operating in conjunction with a V-belt or gear drive created a slight modification to the function of the starting handle. In a direct-drive system, the handle is also required to switch the motor 'on' and 'off'. Alternatively, in an indirect-drive system, it is required to activate the clutch arrangement or give a sideways displacement to the driving wheel.

It has been found that a weaver makes less movement, uses less energy, and is quicker-acting if the starting handle is pulled to start the loom and pushed to stop it. The movement of the arm away from the body quickly to displace a spring-loaded starting handle is most useful when it is required to stop the loom in an emergency, and this system is now extensively preferred on modern looms.

Modifications to the starting-handle system include a handle at each side of the loom, a bar extending across the full width of the loom from the starting handle (particularly popular on very wide looms), or a connexion to the handle from a point at the back of the loom. In each of these cases, the loom can be stopped quickly if a fault or some other emergency arises, irrespective of the weaver's position around the loom. In some instances, the loom can be restarted without the weaver's being required to walk to the main starting handle. This is only advisable if the loom position is not critical on starting, and such a situation can only exist when the insertion device is positively controlled in the shed. A rapier loom is a typical example of this circumstance.

10.3.3 Loom Brakes

The principle of operating loom brakes is very similar in a wide range of looms. The movement of the starting handle from the 'on' to the 'off' position causes a brake band to close round the flat perimeter of a wheel (Fig. 10.4), which is mounted on the main shaft of the loom and is acting as a brake drum.

Such a system is likely to vary in efficiency, depending upon the angle of lap over which the brake band makes contact with the brake drum, the coefficient of friction, and the condition of the two surfaces. The presence of foreign matter, such as oil on the surface, is quite detrimental to efficient braking.

It is preferable that the braking system should allow the loom to be brought to rest at the desired position in order to allow a yarn break to be repaired without intermediate manual or mechanical adjustment of the loom position. Examples of this requirement are:

(*a*) with the healds level if the warp stop-motion stops the loom for an end-break;

(*b*) with the sley around the back-centre position on the pick after a weft break has been detected by the side-weft-fork motion; and

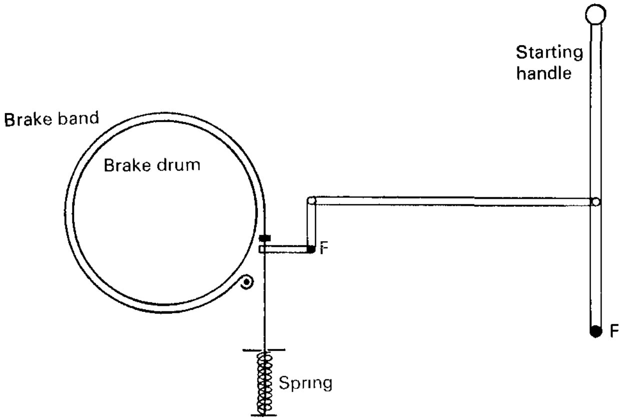


Fig. 10.4 A loom brake

(*c*) before beat-up on the actual pick of the weft break if the centre-weft-fork motion is responsible for stopping the loom in weaving a fabric in which pick-finding is almost certain to create a setting-on place.

The braking system in the last of these instances needs to be highly efficient, and, for this reason, brake drums on looms having centre weft forks usually have larger angles of lap, the brake band having a high coefficient of friction and covering almost the whole of the perimeter of the brake drum as shown in Fig. 10.4 in order to spread the wear over a greater area and dissipate the heat created.

10.3.4 Inching and Reversing

The assumption that the loom will always come to rest in the required position is quite unrealistic, and frequently it is necessary to adjust the position of the stopped loom manually by releasing the brake and turning the handwheel of the loom.

Alternatively, the starting handle may be drawn towards the 'on' position and released as soon as the sley has moved forward to the required position. A small movement of the sley (inching) is generally achieved with each pull of the starting handle, but it is a tricky manoeuvre, which lacks accuracy.

Mechanical reversing of the loom from the top- and front-centre positions becomes desirable after the redrawing of a broken end or the detection of a weft break. It saves time and effort on the part of the weaver. One such method was used on the Hattersley 382 loom, on which the depression of a foot pedal caused a small V-pulley to engage a segment on the driving wheel. The V-pulley running in reverse caused the loom also to run slowly in reverse until the pulley came to the end of the segment. Reverse running then ceased with the loom in its back-centre position. Saurer use a similar system, which allows the small reverse-running pulley to be engaged when the starting handle is pushed towards the back of the loom instead of being pulled towards the weaver for the 'on' position. Slow reverse-running continues until the starting handle is released into the 'off' position.

Another method of reversing the loom is to use the starting handle or some other pedal to make an electrical contact that will switch any two of the line connexions in a three-phase motor. This will cause the motor and thus the loom to run in reverse until the contact is released.

10.3.5 Push-button Control

It is generally desirable for the weaver to stop the loom with the shuttle in the left-hand shuttle-box of a single-shuttle loom and with the sley somewhere between the back- and top-centre positions. It is difficult for the weaver to see the shuttle on high-speed looms and so judge accurately when to knock off the starting handle. There can thus be no guarantee that the loom can safely be brought to rest in the most desirable position.

Push-button control has therefore gained wide acceptance with various objectives in mind. In the first place, it is the most reliable way of ensuring a quick and accurate start and a complete first pick. Secondly, it will always bring the loom to rest in the same desirable position, with the healdshafts approximately level. Thirdly, the loom can be made to run forward at normal or possibly slow speed to a predetermined position, and finally the loom can be made to run in reverse to the back-centre position so that restarting will take place from the most desirable position. Moreover, running in reverse can be continued for a longer period for the purpose of pick-finding. This latter circumstance requires that all the loom parts should run in reverse, and it is generally more acceptable on rapier looms with tappet shedding or specially designed dobbies capable of continuous running in reverse.

The Picanol MDC loom is an instance of a modern shuttle loom that gains the maximum benefit from the use of push-button control. Some adjustments to the timings are possible in the control panel, but, in the main, as a result of electrical contacts being made at specific times during the pick cycle, the loom will stop at the healds-level position, at back centre, at a predetermined position from the cloth fell, or just before front centre, depending on the reason for the stop or the button depressed. The healds-level position is preferred after pressing the forward-run button or after a warp break so that drawing-in can take place without further adjustment to the loom position. Pressing the stop, reverse, or one-pick button will bring the loom to rest at back centre so that it is ready for restarting. This position is also preferred after the detection of a weft break by the side-weft-fork motion so that the broken pick can easily be found. A centre weft fork limits the time of stopping to just before front centre, and the electronic shuttle-control sensor of the warp-protector motion (see Section 7.1.4) will cause the loom to come to rest a predetermined distance from the fell.

An electrically controlled clutch unit (Fig. 10.5) is necessary to operate this

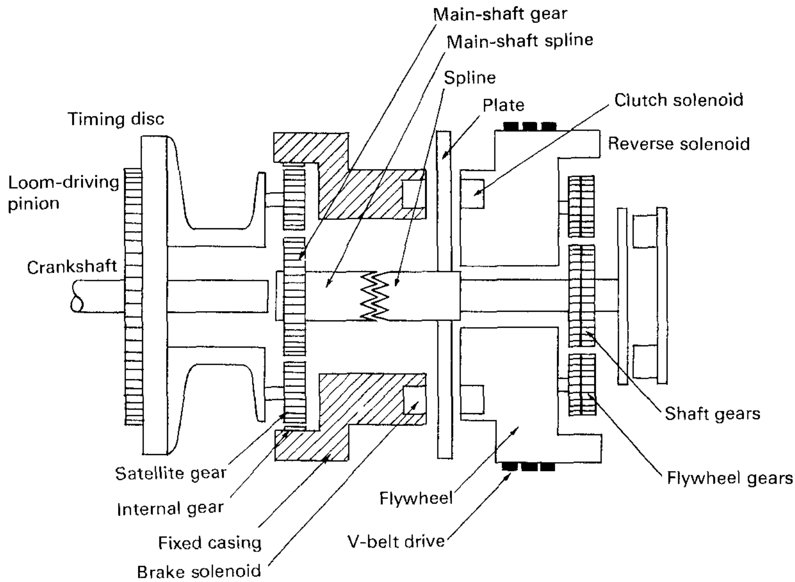


Fig. 10.5 An electromagnetic clutch drive

system. If, in any circumstance, the loom is required to run forward, the clutch solenoids will be energized so that the plate will be attracted onto the driven fly-wheel. This will cause the spline to drive the satellite gears from the main-shaft gear, and thus the loom-driving pinion will be rotated. There are three satellite gears, each one having 18 teeth and being driven from a gear having 60 teeth, so that there is a speed differential of 3·33:1, but, since the shaft itself makes one revolution to complete a drive cycle, then the fly-wheel is rotating 4·33 times as fast as the crankshaft of the loom. This is advantageous in reducing the possibility of variation in angular velocity of the crankshaft.

When the loom is to be stopped, the clutch solenoids are de-energized, and the brake solenoids become energized so that the plate is attracted onto the fixed motor casing. The timing is such that the loom is brought to rest at the appropriate position, depending on the reason for the stoppage.

A further set of solenoids is energized when it is required to run the loom in reverse. Instead of the fly-wheel gears driving a shaft gear with an additional tooth to give a forward drive, the shaft position is adjusted, and the drive occurs on a gear with one tooth less so that the loom will run in reverse.

This unit thus typifies the type of drive control that is made necessary by modern looms running at high speeds. It utilizes push-button control, which, through a control panel, indicates the cause of the stoppage on a light panel, it governs the control of the loom for all eventualities, and, at the same time, it assists in minimizing the problems of providing instant high velocity on the first pick and variation in angular velocity of the crankshaft during the pick cycle.

References

1. W. A. Hanton. 'Mechanics for Textile Students', the Textile Institute, Manchester, 1954.
2. W. A. Hanton. 'Mechanics of Textile Machinery', Longmans, London, 1924.
3. I. H. Thomas and J. J. Vincent. *J. Text. Inst.*, 1949, **40,** Tl.
4. I. H. Thomas and J. J. Vincent. *J. Text. Inst.*, 1949, **40,** T30.
5. I. H. Thomas and J. J. Vincent. *J. Text. Inst.*, 1949, **40,** T45.
6. I. H. Thomas. *Fibres*, 1949, 90.
7. M. G. Catlow and J. J. Vincent. *J. Text. Inst.*, 1951, **42,** T413.
8. P. R. Lord. *Text. Rec.*, 1967, **84,** Feb., 49.
9. M. G. Catlow. *J. Text. Inst.*, 1958, **49,** T424.
10. D. Morrison in 'Modern Mechanisms in Textile Machinery' (Proceedings of Symposium of the Institution of Mechanical Engineers, Nov., 1962) (edited by G. R. Wray and P. Grosberg), the Institution of Mechanical Engineers, London, 1963, p. 14.
11. A. Ormerod in 'Modern Developments in Weaving Machinery' (edited by V. Duxbury and G. R. Wray), Columbine Press, Manchester, 1962, p. 80.
12. P. R. Lord and M. H. Mohamed. *Text. Rec.*, 1964, **82,** Oct., 67.
13. R. M. Dawson. *Text. Mfr*, 1971, **97,** 99.
14. D. C. Snowden. *Text. Prog.* 1972, **4,** No. 2.
15. R. Marks, A. T. C. Robinson, H. Collier, and J. S. Dimelor. 'Fabric Technical-training Manual', Cotton and Allied Textiles Industry Training Board, Manchester.
16. R. M. Dawson. *J. Text. Inst.*, 1974, **65,** 639, 645, 648, 654.
17. M. W. H. Townsend. *Wool Sci. Rev.*, 1960, No. 18, 7.
18. F. T. Peirce. *J. Text. Inst.*, 1937, **28,** T45.
19. K. Greenwood and W. T. Cowhig. *J. Text. Inst.*, 1956, **47,** T241, T255; K. Greenwood and G. N. Vaughan. *ibid.*, T274.
20. K. Greenwood. *Silk & Rayon Rec.*, 1956, **30,** 1154.
21. K. Greenwood and G. N. Vaughan. *J. Text. Inst.*, 1957, **48,** T39.
22. K. Greenwood. *J. Text. Inst.*, 1959, **50,** P205.
23. R. Foster. 'Positive Let-off Motions', Wool Industries Research Association, Leeds, 1961.

Index

A

ATPR rapier loom 131, 134
Automatic looms, conventional 8
Acceleration of shuttle during picking 21–28
Air-jet looms 9
Alacrity of picking mechanism 44
All-metal reed 7

B

Back roller, forces acting 178–181
freely rotating 178, 179
friction 179–181
Backlashing, fault in fabric 201, 224, 227
Barber–Colman drawing-in machine 187
Bartlett let-off motion 170–172
Basic warp tension, definition 143
Beam-driving mechanism 167–170
Beat-up force, definition 143
Beating-up, definition 1
description 7
frequency 3
Bobbin change, single-shuttle automatic 220–225
Bobbin-change mechanism 220, 221
Bobbin loaders 225, 226
Bottom-closed shed 72
Bowl-and-shoe-underpick mechanisms 101, 102
Box motions 197–204
Bumping conditions 146, 147

C

Cam-and-cone picking mechanisms 97–101 97–101
Carpet-weaving, jacquards for use 81
Catch cord 117
Catch end 117
Centre-closed shed 73
Centre-weft-fork stop-motion 192–194
Checking, retardation of shuttle 28–31
Circular looms 8
properties 109–116
Circular weaving 134–138
Cleats, function 227
Cloth control 153–155
function 1
Cloth-fell position 142–146
Cloth formation 140–151
Clutch drive, electromagnetic 242, 243
Clutch motors 238, 239
Clutch systems 236–239
Coarse-pitch jacquard machines 74–77
Cone-overpick mechanism 97, 98
Cone-underpick mechanism 99–101
Continuous weft insertion 134–139
Controlled negative automatic let-off 164, 165
Crank 7
Crank circle, definition 11
Crankarm 7
Cranks and crankarms, length 17–19
Crompton & Knowles C5 loom 200
Crompton & Knowles C7 loom 184
Crompton & Knowles PAPA 4 × 3 box loom 230
Crompton & Knowles looms 11, 69, 167, 183

D

Dents, definition 7
Depth feelers 218
Dewas system of rapier-weaving 124–127, 208, 209
Diehl clutch motor 238, 239
Direct and indirect drives 236–238
Direct take-up 153
Disturbed weaving conditions 147, 148
Dobby shedding 47–49, 59–72
Dobcross woollen and worsted looms 19, 69
Dornier loom, weft-patterning system 208
Draper DSL loom 20
Draper looms 11, 55, 183, 193
Drives, direct and indirect 236–238
Drop wire 186–188
Dropper 186

E

Ejected bobbin control 224, 225
Elastic properties of picking mechanism 25–27
Electrical warp stop-motion 189, 190
Electromagnetic clutch drive 242, 243
Electromagnetic warp protection 184–186
Elitex air jet loom 130, 131, 133
English pitch jacquard machines 74–77

F

Fairbairn Lawson loom 135
Fast and loose pulley 235, 236
Fast reed warp protector 182, 183
Fayolle–Ancet loom 115, 135, 136
Feeler, depth 218
photo electric 219
side sweep mechanical 217
two prong electric 219
Feeler position 216
Feelers 215–220
classification 216
function 215
Fell, definition 1
Fine-pitch jacquard machines 77–80
Fischer ALV bobbin-loader unit 225, 226, 233
Float, fault in fabric 186
Floating back roller, forces acting 178–181
Floating swell 29, 30
Forces acting at floating back roller 178–181
Four box motions 200–202
Friction at back roller 179–181
Full-width temples 176, 177

G

Gabler system of rapier-weaving 124–127, 210
Geometry of warp shed 38–42
Griffe, function 74
Gripper Axminster loom, jacquard 81
Gripper-shuttle looms 8
Gusken loom 127

H

Hand-looms 8
Harness ties, jacquard machines 84
Hattersley 382 loom 161, 241
Heald shaft-reversing motions 52–55
Heald shafts, motion 12, 13
Healds, materials 4

I

IWER ONA loom 115, 139
Inching and reversing of loom 241, 242
Indirect take up 153
Inertia effect in checking 32, 33
Investa H and P air jet looms 115
Investa Kontis loom 139

J

Jacquard harness 81–83
Jacquard lift 81, 83
Jacquard machines, for carpet weaving 81
special types 74, 80, 81
Jacquard shedding 47–49, 74–87
card cutting instructions 85–87
casting out 85
coarse-pitch machines 74–77
fine pitch machines 77–80
Jet looms, properties 109–139
Jet weaving 130–134

K

Keighley dobby 59–68, 73
Kenyon undermotion 54, 64
Knowles positive dobby 69–73

L

Lacey top motion 53
Lashing in, fault in fabric 201, 224, 227
Leeming dobby 72
Length of cranks and crankarms 17–19
Let off motion 162–174
function 1, 140
Link pick 99, 100
London tie, jacquard harness 84
Loom, methods of driving 234–236
Loom brakes 240, 241
Loom control 239–243
Loom dimensions, typical 18
Loom drives 234–243
Loom speed and loom timing 33–36
Loom-timing, method of indicating 11, 12
Loom width 36, 37
Loom-winders 227–230
Looms, classification 8
Loose reed warp protector 183, 184
Lug strap, function 99

M

MAV rapier loom 125, 128, 210
Maxbo air jet loom 130, 131, 133
Mechanical feeler, side sweep 217
Mechanical warp stop motion 188, 189

Midget feeler 217
Motion of heald shafts 12, 13
Motion of shuttle 21–46
Motion of sley 13–21
Multifil 233
Multiphase looms 9, 114, 138, 139
Multiple gripper weaving machine, weft control 95, 96
Multishuttle weft replenishment 230–233
Murata–Maxbo air jet loom 115, 130, 131

N

Negative cams, shedding 50–52
Negative dobby shedding 59–68
Negative friction-type let-off motion 162–164
Negative shedding 49, 50
Negative take up 152, 153
Non automatic power looms 8
Northrop industrial blanket loom 19
Northrop looms 183, 188, 231
Northrop Sensamatic loom 184, 186
Norwich tie, jacquard harness 84

O

Open shed 73

P

Parallel pick 99, 100
Photo-electric feeler 219
Picanol MDC loom 115, 184, 242
Picanol cotton loom 18
Picanol looms 55, 183
Picanol President loom 107, 115
Picanol take-up motion 160
Pick, definition 1
Pick-at will 202–204
Pick spacing, causes of variation 148–151
  importance of uniformity 140
  method of changing 159–162
Picking, acceleration of shuttle 21–28
  definition 1
  description 5
  frequency 3
  power required 103–105
Picking mechanisms 88–105
  as simple elastic system 43–46
  conventional 97–102
  elastic properties 25–27
  functions 97
  types 88–96
Picking strap, function 99
Pin 186
Pirn, definition 6
Pitch baulk reed 7
Positive cams, shedding 55–58
Positive direct take up 159
Positive dobby shedding 68–72
Positive indirect take-up, method of driving 155–159
Positive let-off 165–167
Positive shedding, definition 49, 50
Positive take-up 153–159
Power required for picking 103–105
Primary motions of weaving, definition 1
  timing 37
Prince–Nissan water-jet loom 18, 115, 131–134
Prince–Nissan weft accumulator 112, 113
Prince–Nissan weft stop motion 194
Productivity of loom 9, 10
Push button control 242, 243

R

Rapier, jet, and continuous methods of weft insertion 109–139
Rapier looms 9
  properties 109–139
  weft patterning systems 208—210
Rapier weaving 123–130
Reed, all-metal 7
  description 7
  function 1
  pitch-baulk 7
Retardation of shuttle during checking 28–31
Ring temples 176
Ripple-shedding loom 9
Rocking shaft 7
Roller temples 176
Roper let-off motion 170
Ruti BANL loom 184
Ruti C loom 115, 184, 190
Ruti C 1001 loom 184
Ruti cam dobby 65, 66
Ruti cotton loom 19
Ruti looms 183
Ruti rapier loom 131, 133
Ruti TWR multiphase (ripple-shedding) loom 9, 114, 115, 138, 139
Ruti take-up motion 161, 162

S

SMIT loom 115
Safarti weft accumulator 112, 115
Saurer 100W loom 161
Saurer 300 Versaspeed loom 114, 115, 184
Saurer cotton loom 16–18, 55

Saurer let-off motion 172–174
Saurer looms 118
Savi weft accumulator 112, 113
Self-threading shuttle 225
Selvedges, conventional 117, 118
 dummy 121
 fringe 119–121
 leno 119
 requirements 116, 117
 tuck 118, 119
 twist 119–121
Setting-on places, fault in fabric 150
Semi-open shed 72, 73
Shed, advantages of different types 73, 74
 bottom-closed 72
 centre-closed 73
 definition 1
 different types 71–74
 open 73
 semi-open 72, 73
Shedding, definition 1
 definition of 'positive' and 'negative' 49, 50
 description 3–5
 frequency 3
Shedding mechanisms 47–87
Shirley loom, let-off motion 164
 picking mechanism 28
 take-up motion 158
Shuttle, acceleration during picking 21–28
 description 6
 rest position 31, 32
 retardation during checking 28–33
 self threading 225
 split 222
Shuttle-checking devices 105–108
 conventional shuttle looms 105–107
 multiple-gripper weaving machine 107, 108
Shuttle eye, description 6
Shuttle flight 33–37
Shuttle protection 221–223
Shuttleless looms 8, 9
Side lever mechanism 101, 102
Side-shaft mechanism 101, 102
Side sweep mechanical feeler 217
Side-weft-fork stop motion 191, 192
Single-gripper looms 8
Single shuttle automatic bobbin change 220–225
Size of weave repeat, limiting factors 58, 59
Skip draft 5
Sley, eccentricity 15, 16
 factors affecting motion 13–15
 function 3
 geometry 18
 motion 13–21
Sley-eccentricity ratio 15, 16, 18
Sleys that dwell 19–21
Split shuttle 222
Spring pick, principles 89–92
Spring tensioning 170
Spring undermotions 64
Starting handles 239, 240
Starting places, fault in fabric 150
Stäubli dobbies 72
Stitching, definition 4
Stop motions 182–194
Straight draft 5
Straight line nominal movement 44–46
Swell, functions 28
Swordpin 7
Sulzer weaving machine 8–10, 20, 21, 42, 57, 72, 88, 92–96, 105, 107, 108, 114, 115, 119, 124, 154, 160, 205–207
Sulzer weft accumulator 112
Sulzer weft-patterning systems 205–207
Svaty jet 131, 132

T

Take up motion 151—162
 function 1, 140
Tappet-and-cone picking mechanisms 97–101
Tappet shedding 47–59
 with negative cams 50–52
 with positive cams 55–58
Temple-cutting 224
Temples 174–177
 function 1
Te Strake rapier loom 131, 133
Tensioning by springs 170
Timing, method of indicating on loom 11, 12
 of primary motions 37
 of shuttle flight 33–37
Torsion picking, principles 92–95
Turbo-winder, function 138
Two-prong electric feeler 219

U

Unconventional looms, methods used for weft insertion 8
 weft-patterning 204–210
Unifil automatic loom-winder 213, 214, 227–230
Uniform acceleration of shuttle, factors producing 27, 28

V

Verdol jacquard machine 77–80
Versamat loom 124, 208
Vincenzi jacquard machine 77, 78

W

Warp and cloth control 140–177
Warp control, function 1
Warp line 40–42
Warp protection 182–186
electromagnetic 184–186
Warp protector, fast reed 182, 183
loose-reed 183, 184
Warp-protector motion 7
Warp shed, geometry 38–42
size 38–40
Warp stop-motion, electrical 189, 190
mechanical 188, 189
Warp stop motions 186–190
Warp tension, basic, definition 143
Water jet looms 9
Weave repeat, factors limiting 58, 59
Weaver, duties 211–214
Weaver's beam 1
Weaving, basic mechanisms 1–3
primary motions, definition 1
Weaving machinery, classification 8
Weaving resistance, definition 143
Weft accumulators 112
Weft control, in multiple-gripper machine 95, 96
Weft control timings on jet looms 132, 133
Weft-cutting, fault in plain cloths 142
Weft insertion, comparison of properties of different methods 109–116
continuous 134–139
in unconventional looms 8
rate 36, 37, 114–116
Weft-mixing motions 197–199
Weft-patterning 195–210
comparison of systems available 195–197
on unconventional looms 204–210
scope 116
Weft replenishment 211–233
Weft reserve 215, 216
Weft stop-motions 190–194
Weft-tension control 223
Weft velocity 109–114
Weft waste 121–123
Wilton loom, jacquard 81