11

SHEDDING AND BEATING

Key words: *baulk, beating, beat-up, bumping, cam, clear shed, closed shed, comber board, complete shed, connecting rod, crank, cylinder, dobby, dobby head, dobby loom, double-lift dobby, drawstrings, feelers, fell, float, grate, griffe, harness, heddle, hook, incomplete shed, jack, jammed fabric, knives, lay, lay sword, lift, lingoes, mails, needle, neck, open shed, pattern chain, peg, punched card, reed, semi-open shed, shed, shedding, shedding diagram, single-lift dobby, spindle, unclear shed, under-cam loom.*

The Interrelationship between Shedding and Beating

From a design point of view, *shedding* and *beating* are interrelated. Referring to Fig. 11.1, it is apparent that the shed opening $(H_1 + H_2)$, the distance B and the angles β_1 and β_2 are related as follows:

$$H_1 = B \tan \beta_1$$

$$H_2 = B \tan \beta_2$$

In a symmetrical system, $H_1 = H_2$, which implies that $\beta_1 = \beta_2$ and $h_1 = h_2$. For ease of reference, let these be referred to as H, β, and h, respectively; also for simplicity let only the simple symmetrical case be considered.

In general

$$h = \mathrm{b} \tan \beta \qquad (11.1)$$

It might be thought that the angle β should be as small as possible so as to reduce the *lift* of the *harnesses* (i.e. to reduce H), since this would reduce the strain on the warp.

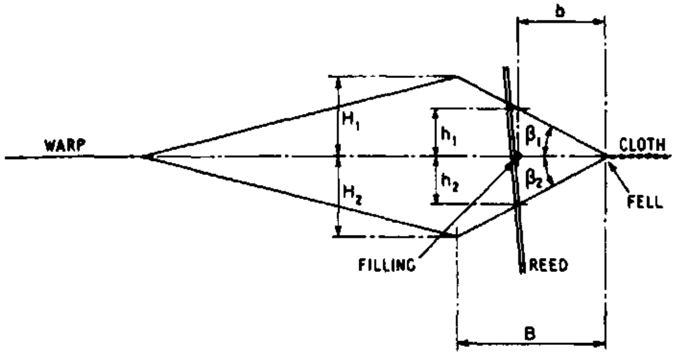


Fig. 11.1. Geometry of warp shed

A reduced lift would also make the mechanism easier to design and operate. However, if β is made too small, the warp sheets will not open properly and some ends are likely to catch up; such an obstructed opening is called an *unclear shed.* In such a case, there is a strong likelihood that the shuttle will rub the protruding warp end and cause it to break. Thus an unclear shed can lead to unacceptable increases in end breaks. Furthermore, the distance traveled by the *lay* in beating the filling into the *fell* of the cloth is a function of β. The lay and associated parts are heavy and appreciable amounts of energy are required to move them quickly over considerable distances; this affects the motor size and efficiency. Also the movement of such large masses gives rise to undesirable cyclic speed changes in the loom.

If the distance the lay moves is b_a, and b_s is a constant, then

$$b_a = b_s + b$$

but

$$b = h \operatorname{cotan} \beta$$

therefore

$$b_a = b_s + h \operatorname{cotan} \beta$$

$$= b_s + \frac{hB}{H} \tag{11.2}$$

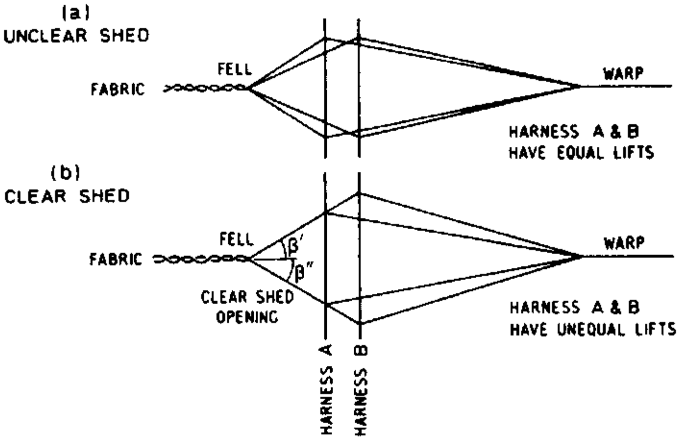


Fig. 11.2. Shed openings.

From eqn (11.2) it can be seen that if the harness lift H is made small, keeping the other parameters on the right-hand side constant, then the lay movement b_a must be increased in order to permit the shuttle to enter. The height of the front wall of the shuttle controls h and is itself controlled by the diameter of the quill which fits in the shuttle. This, in turn, involves the economics of weaving because a small quill has to be changed frequently and handling costs tend to go up as the size goes down. Also, every time a quill is changed there is a potential fault. Good machine design can minimize this potential but it cannot eliminate it completely. Hence for several reasons, the distance b must be limited and a balance has to be struck between the harness lift H and the lay movement b_a. It is desirable that h be as large as possible but that H and b should be as small as possible; clearly it is impossible to reconcile these requirements and a compromise has to be made. The art of achieving the best compromise is beyond the scope of this book.

Shed Forms

If the shed is unclear, as shown in Fig. 11.2(a), it is possible for the shuttle to cause many end breakages and it is usually desirable to keep the angle β the same for all ends in order to produce a *clear shed*. This requires an adjustment to the movements of the various *heddles* unless they can all exist in a plane at a fixed distance from the fell of the cloth. This is an impossible condition for a normal loom because the harnesses must be at different distances from the fell in order to permit their independent movement. To maintain a clear shed, the angle β has to be kept constant and to do this each harness must be given a slightly different lift from its neighbors; in fact, the lift must be proportionate to the distance of the harness from the fell. Thus the differences depend upon how closely the harnesses can be packed; if there are many harnesses, the differences can be large.

For certain looms operated by drawstrings (Fig. 2.7), positive movements which involve the direct drive of the elements occur only in one direction and deadweights return the shed to the lower position. In these cases the shed opening is rarely symmetrical and angles β_1 and β_2 are dissimilar; in fact β_2 can be very small in some cases. Such a system gives a so-called *incomplete shed*. The most common case of shedding, called a *complete shed*, is as shown in Fig. 2.8; the sheds are driven in both directions.

The timing of the opening of the fully open shed is important. Various systems have been designed to compensate for the change in warp length in shedding and so minimize the variations in warp tension (see Chapter 10). These systems operate on all warp ends, and if some ends are left in position whilst others are moved, the compensation will act on some more than others; this may cause some ends to go slack and some to be over-tensioned. The slackness might enable entanglements to form and the high tensions could lead to end breaks; the variations in tension

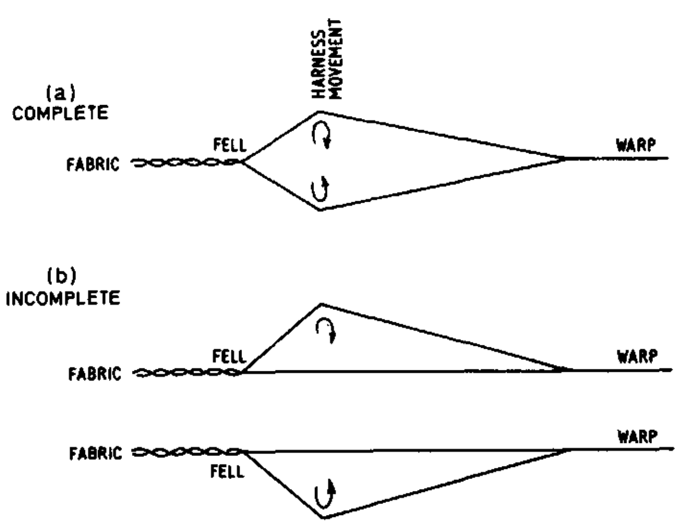


Fig. 11.3. Shed forms.

would also affect the fabric structure. In the simple case, where the shed is changed only as dictated by the fabric weave, the *shedding diagram* might be as shown in Fig. 10.1(a). Where compensation is used, the corresponding shed must be adjusted to avoid the difficulties cited; the diagram might be as in Fig. 10.1(b). The former is called an *open shed* and the latter a *closed shed*. It is also possible for the shed to be asymmetrical, in which case the shedding has to be adjusted to give minimum tensions and a *semi-open shed* is used, as shown in Fig. 10.1(c).

Shedding Motions

The lift of the heddles can be achieved by using *cam* or *dobby* operated harnesses or by *drawstrings* and each has its own limitations. Leaving aside all mechanical differences, it is useful to consider the operational limitations. Most cam systems tend to be inaccessible and it is sometimes inconvenient and time consuming to change the weave, especially with complex fabric designs. With such complex weaves (which need many harnesses), a change can involve replacement of some or all of the cams, which is a fairly lengthy process. Occasionally, alterations can be made by varying the cam-shaft speed in multiples so as to avoid cam changes. The most important advantages of the cam system are that the design permits a relatively high speed and the system is inexpensive. Some of the operational inflexibility of the cam loom can be circumvented by good organization if there is a sufficient number of looms at the disposal of the manager.

Moving to the other extreme, a Jacquard loom is very flexible as far as the weave is concerned because it is possible to control single ends if so desired. Fabric designs can be changed merely by inserting a series of punched cards into the *pattern chain*. However, the production rate is very low and consequently these looms are used only for complex weaves. The Jacquard heads are expensive and therefore there is considerable incentive to find cheaper ways of producing patterned fabrics, especially if the required pattern is not too complex.

Intermediate between the Jacquard and the cam looms is the *dobby loom*; this loom is one on which small geometric and regular figures can be woven. Originally a dobby boy sat on top of the loom and drew up the warp as required to form the pattern. This function has now been automated and a device which replaces the dobby boy has become known as a *dobby head*. The origin of the word dobby is obscure.

The inaccessibility of the multiple *under-cam* system limits the design possibilities. One solution is to bring the cams to the side of the loom where they are more accessible. This permits an increase in the number of harnesses from about eight to about sixteen, but the system tends to be cumbersome. It is an obvious advantage to miniaturize the system so that the drive to the harnesses is triggered by an actuating device rather than being driven directly. This, in fact, is what a dobby head does; instead of a cam and lever system, a peg or some other discontinuity in a pattern chain triggers the appropriate shed changes. The pattern chain determines the

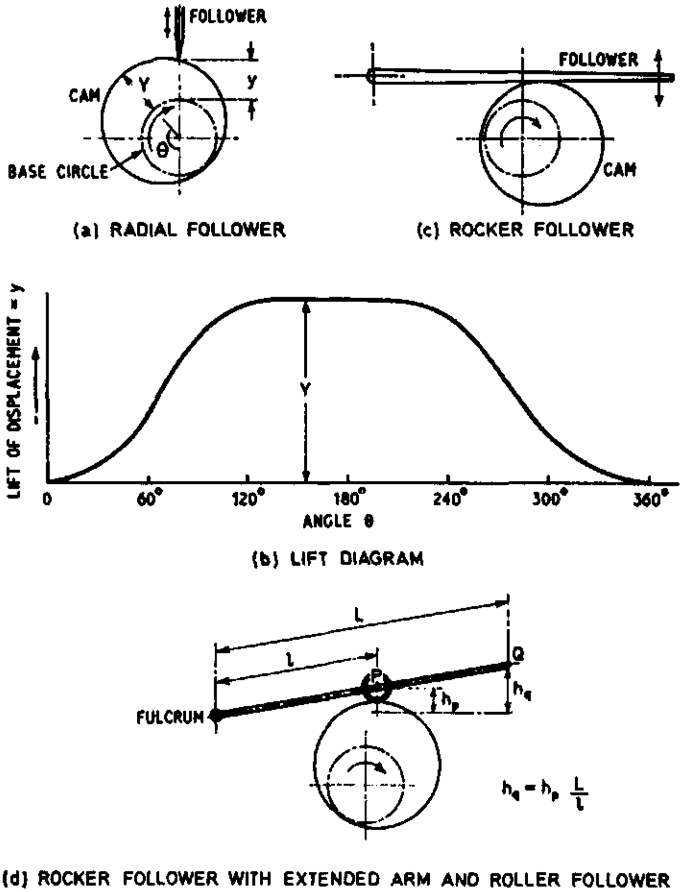


Fig. 11.4. Simple cam systems

weave. The variety of patterns and the ease of change is increased considerably. Against this must be set a slight decrease in the maximum permissible operating speed; this arises because of the excessive stresses set up in the dobby head when it is operated over the maximum safe speed.

To understand some of the restrictions, it is necessary to consider some elements of cam design. First, an explanation may be given in terms of a cam with a radial knife edged follower, even though this is a gross oversimplification. Referring to Fig. 11.4(a), the rotation of the cam causes the follower to lift along a radius and its vertical position is determined by the angle θ; the lift diagram is shown in Fig. 11.4(b) where any ordinate refers to the corresponding radial distance between the base circle and the outside operating surface of the cam. The relationship between these two parameters is linear, but if the follower does not move along a radius or if a knife edged follower is not used, this linear relationship no longer holds. Since practical cam systems do have nonlinear relationships, the actual cam shape is dissimilar to the lift diagram and this implies that cams from different systems are not interchangeable. In practice, interchanges are possible in many cases but care has to be taken in this respect, especially where high performance systems are involved.

If the case of the simple system, it is possible to have a rocker follower (Fig. 11.4(c)) which will give a lift diagram similar to the one previously mentioned, providing the distance l is adequate. If the arm is extended as in Fig. 11.4(d), the point Q will move a greater distance than the point P but the shapes of the respective displacement diagrams will be similar. The lift at Q

$$= (\text{the lift at } P) \times \frac{L}{l} \tag{11.3}$$

With a given mechanism, it is possible to alter the position of Q with respect to P and thereby change the lift at Q. If the harnesses are connected at Q, then the lift can be adjusted as required by altering either L or l and the varying lifts that are needed to give a clear shed can be obtained by this means without having to have a wide range of cams.

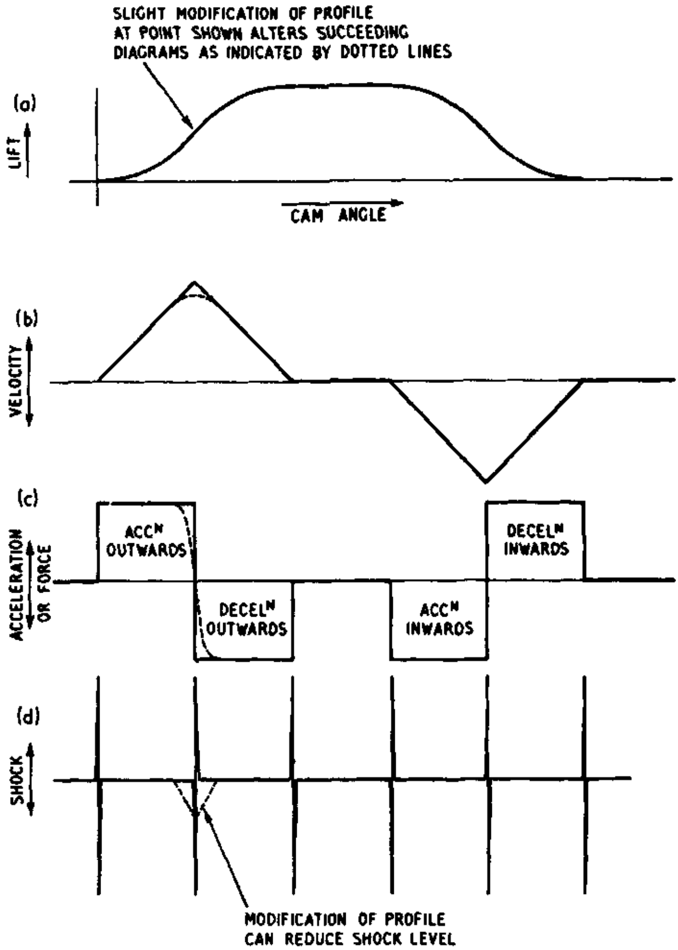


Fig. 11.5. Lift diagram with corresponding velocity, acceleration and shock diagrams.

A rocker follower which operates about a fixed fulcrum would cause the point Q to move in an undesirable arcuate path but if the fulcrum is allowed (or caused) to move to compensate, a desirable straight lift can be obtained. A similar problem exists in picking and since it is so important in that context, the matter will be dealt with under that heading.

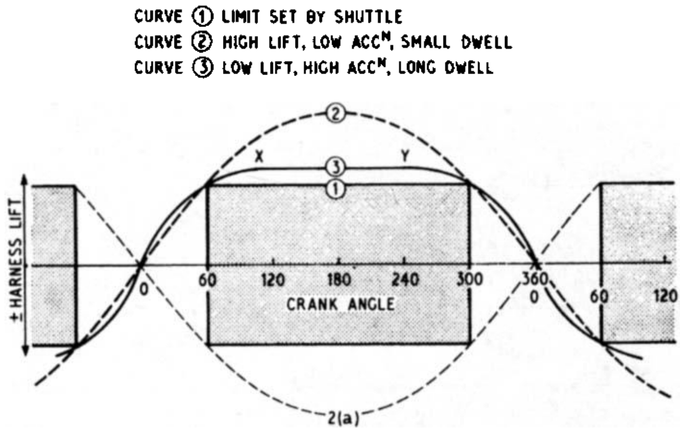


Fig. 11.6 (a) Shedding diagrams and shuttle interference. N.B. Sheds are not always symmetrical

A lift diagram such as that shown in Fig. 11.5(a) implies that the corresponding velocity diagram will have steep portions, which in turn implies an acceleration as shown in Fig. 11.5(c). A high acceleration acting on a mass will produce a large force, and a sudden change in acceleration produces a shock which will produce noise and vibration. Thus the cams must be designed to minimize these forces as well as the shocks, and the faster the loom runs the more important this becomes. (In mathematical terms, if the lift is h, the harness velocity is $\partial h/\partial t$, the acceleration is $\partial^2 h/\partial t^2$ and the shock level is proportional to $\partial^3 h/\partial t^3$.) Suppose the dimensions of the shuttle and the timing dictate that the harness lifts should lie outside the shaded area shown

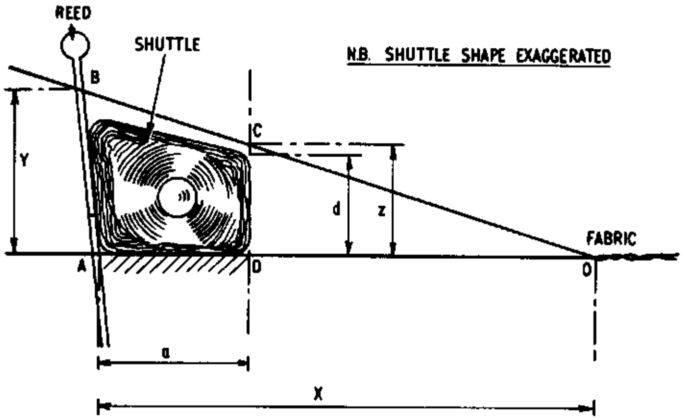


Fig. 11.6 (b). Shuttle interference at C.

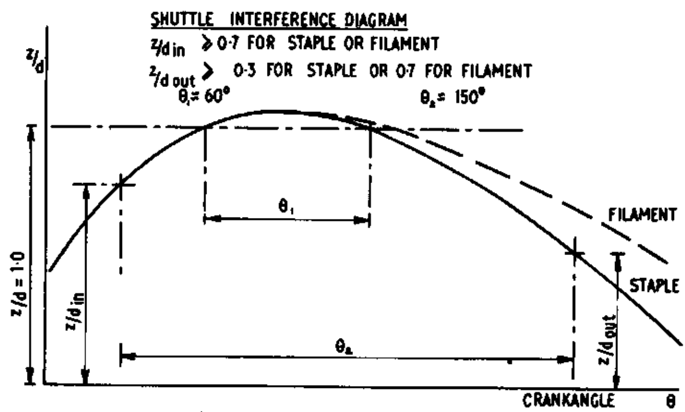


Fig. 11.6 (c).

in Fig. 11.6; in such a case there are several alternative solutions. The acceleration can be minimized by using a curve such as 2, but this would be at the expense of an increased harness lift (which would strain the warp ends and lead to increased breakage rate). It is also possible to maintain the dwell between Y and X as shown in curve 3, but this would be at the expense of generating extra forces in the mechanism (which might limit speed). Thus a compromise is usually needed, but the band of acceptable solutions will narrow as the picking rate increases and, of course, high picking rates are preferred. The solution also depends upon several other factors; for instance, with a closed shed there are some rather rapid changes which generate high forces and it is not usually possible to operate a closed shed as fast as an open one.

The warp shed opening (Z) may be calculated from the similar triangles OAB and OCD shown in Fig. 11.6(b).

$$z = Y(1 - a/x)$$

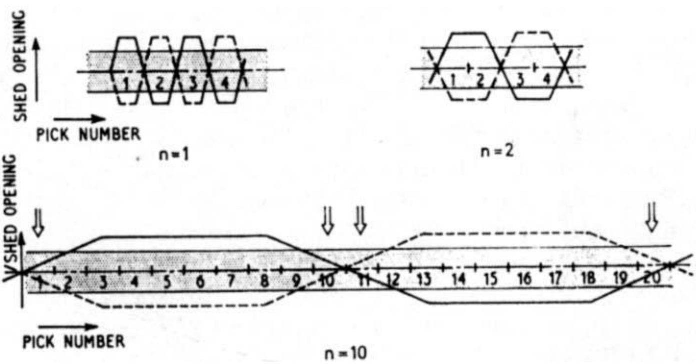
The warp shed opening needed to clear the shuttle at point C as it traverses the loom depends upon the lay position and the dimensions of the shuttle. Should the warp shed fail to clear the shuttle, there will be a degree of warp abrasion especially at the selvages. However, it is common practice to accept a degree of shuttle interference for the sake of increased productivity as will be explained later.

For example, consider a case where a loom runs at 200 picks/min, the shuttle velocity is 20 m/sec (66 ft/sec) and the crankangle over which the shed is open sufficiently for the shuttle to pass without shuttle interference is 80°. The loom speed is (200 x 360)/60 = 1200 degrees/sec. Hence the warp shed is open for 80/1200 = 0.06666 sec and, in this time, the shuttle travels 20 x 0.06666 = 1.3333 meters. The width of fabric that could be woven without shuttle interference is 1.333 $-\ell_{sh}$ (where ℓ_{sh} is the effective length of the shuttle). Thus if ℓ_{sh} is 0.3 meters, the maximum fabric width would be 1.0333 meter if shuttle interference were completely eliminated. Any attempt to weave wider fabric would lead to a tendency for warp abrasion and this would get worse as the width increased. Similarly with speed, if the speed were increased to (say) 250 picks/min, the loom speed would be 1500 degrees/sec, the shed opening time 0.053 sec and the shuttle would travel 1.0666 meter in this time; thus the maximum width would decrease from 1.0333 to 0.7666 meter for the particular timing and shuttle speed. Thus it can be seen that any increase in speed or width (both of which tend to increase productivity) can only be achieved at the expense of shuttle interference. All the factors are closely related and are NOT independent variables.

By suitably shaping the shuttle, the effects can be minimised. It is usual to make the front wall height (d) less than the back wall height (the wall in contact with the reed is larger), and the shuttle edges are rounded. Also the shuttle ends are pointed, which reduces the effective shuttle length (ℓ_{sh}) and by these means it is possible to carry the largest possible quill (shown dotted in Fig. 11.6 (b)) for the minimum amount of shuttle interference.

Even when all these precautions are taken, the avoidance of all shuttle interference would unnecessarily restrict productivity. Because of the smooth pointed ends to the shuttle, it is quite possible to allow the shuttle to enter a warp shed which is too small but opening. It is also possible to allow the warp shed to close on the departing shuttle. The extent of these interferences depends mainly upon the warp yarns. A typical warp shed opening diagram is given in Fig. 11.6(c) and it will be observed that the crankangle interval ($\Delta\theta$) is much enlarged by permitting a controlled amount of shuttle interference. For typical staple yarn $(Z/d)_{in} > 0.7$ and $(Z/d)_{out} > 0.3$, whereas with most filament yarns $(Z/d)_{in}$ and $(Z/d)_{out}$ are both greater than 0.7. Limited adjustments to $\Delta\theta$ can also be obtained by varying the loom timing.

In practice, knife edged followers are not used. A knife edge would wear too rapidly and it is usual to use a roller



NB. Shaded areas indicate minimum shed opening to permit passage of shuttle. Interference indicated by arrows in the case of n=10

Fig. 11.7. Shedding diagrams from one given cam.

follower as shown in Fig. 11.4(d). The roller is limited in its ability to follow the cam surface when there are very rapid changes in profile; for instance, a roller cannot produce a vee-shaped lift diagram because the roller surface cannot make contact with the apex of the vee. The larger the roller or the sharper the vee, the less will be the conformity. The diameter of the roller thus determines, in part, the profile of the cam and this is another reason why care should be taken in interchanging cams between different sorts of looms.

A gear train is normally used to connect the crankshaft and the camshaft to maintain proper synchronization. In one shedding cycle, the harness remains up for one dwell period and down for the other. With a plain weave, two picks are inserted in this time and the camshaft has to rotate at half the crankshaft speed. If 4 : 1 gearing were used, two picks would be inserted with the harness in the up position and two in the down position. If $2n$: 1 gearing were used there would be n picks inserted between changes. Such a procedure is used to make twills and other structures which require *floats*, but there are limitations. Consider an extreme case; let $n = 10$ and let the cam profile shown earlier be used. As shown in Fig. 11.7, the changeover of the shed would still be in progress at the time that picks number 1, 10, 11, 20, etc., should be inserted. The shed would be insufficiently open to allow the shuttle to pass and it would be impossible to weave unless the cam profile were changed to give a very rapid shed change. Such a profile could only be run at a diminished speed.

With a given fabric structure, the pattern repeats even though it might only be one up, one down. Whatever the pattern, the gear ratio should be equal to the number of picks per complete repeat. For example, with a plain weave the repeat is over 2 picks and the gear ratio is 2 : 1. For a 2 × 2 basket weave, the repeat is over 4 picks and the gearing ratio is 4 : 1.

To simplify matters, normal fabrics woven on a cam loom are usually restricted so that all cams are similar but merely displaced in terms of their relative angular position. When the fabric structure is changed it is sometimes necessary to

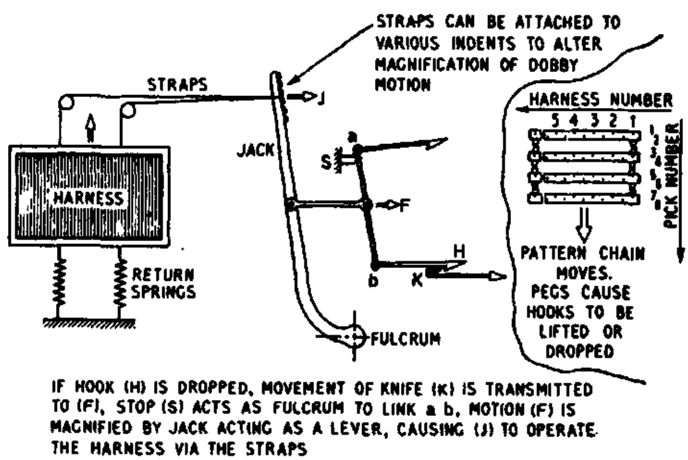


Fig. 11.8. Basic dobby motion. Not to scale. Certain parts omitted for clarity

change cams, gearing or both; therefore, accessibility to these parts is important.

Dobby Shedding

The cams used in a cam loom fulfill two functions, namely:

(1) To position the harnesses as required from time to time
(2) To transmit the power needed to cause the desired movement.

In a dobby mechanism, these functions are separated, with the result that the actuating mechanism can be very much lighter in construction and much more compact and accessible. These improvements permit the production of more complex weaves under normal mill conditions without the sacrifice in productivity that follows the use of a Jacquard loom. The Jacquard loom is used primarily for weaves which are too complex for the dobby loom.

Basically, the dobby mechanism consists of two sections; one section is concerned with power transmission, the other with the connection and disconnection of the harnesses to and from the power source at the appropriate time. The latter device may be regarded as a sort of mechanical equivalent to an electrical switch. On the power side of the switch, there is a permanently connected set of moving *knives* which reciprocate along fixed paths. As shown in Fig. 11.8, there are hooks (*H*) which hinge on a *baulk* (*ab*); when these hooks are lowered, they engage the knives and move with them. In one mode, the stop (*s*) acts as a fulcrum and thus the movement of the knives causes the central link to move, which in turn causes the *jack* to move. The motion is magnified by the lever action of the jack and is transmitted to the harnesses. The connection and disconnection of the hooks is caused by *needles* or *feelers* which contact the pegs in the pattern chain.

Other modes are possible. Figure 11.9 shows four modes,

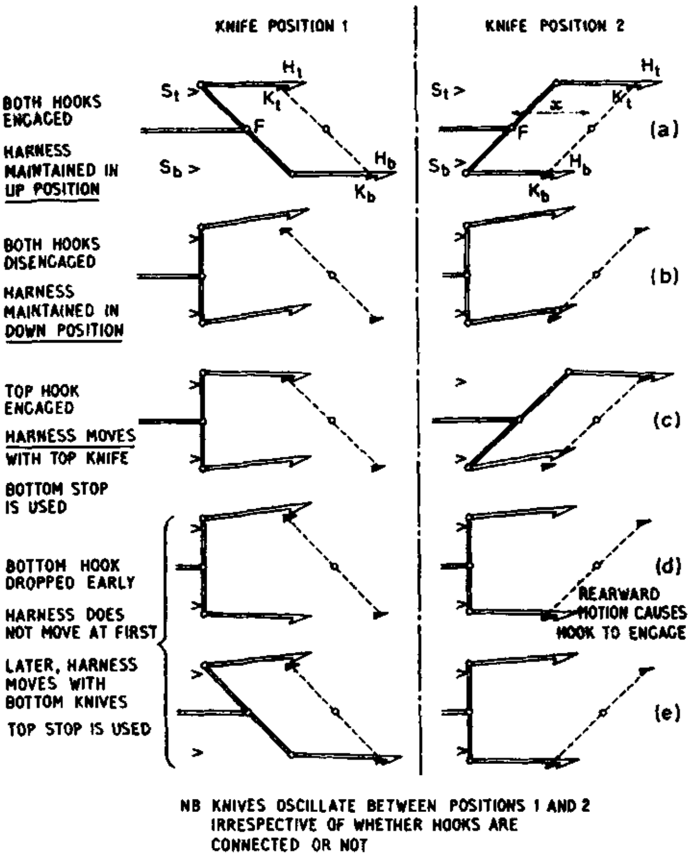


Fig. 11.9. Different modes of action of a dobby.

the knives being shown in each of the extreme positions. When both hooks are engaged, the baulk merely seesaws about point F and the harnesses are maintained in the up position. When both hooks are disconnected, the baulk is held against both stops and the harnesses are maintained in the down position. When one hook is disengaged, the harness will move up and down in sympathy with the other knife. It is the practice to lift the rearward knife because at that time the appropriate portion of the baulk is held by the stop and the hook can be raised without damage. A hook can be dropped at any time, but there may be a time lag before it latches into proper sequence as shown in sections (d) and (e) in Fig. 11.9.

By altering the distance x (Fig. 11.9(a)), it is possible to operate with semi-open sheds. Therefore it is possible to obtain any desired shedding diagram by having two sets of pegs, hooks, and knives per harness; this is known as a *double lift dobby*. It is also possible to use a single set, but *single lift dobbies* are now comparatively rare.

Single Lift Jacquard

Pattern cards are presented to a four-sided "*cylinder*" in such a way that every card fits one side (Fig. 11.10). Every card on the chain represents one pick in the weave. Thus the cylinder speed of rotation is one-quarter that of the loom crankshaft. The cylinder moves away from the needles, rotates one-quarter of a revolution to present a new card and then moves again towards the needles. A hole on the card indicates that the corresponding hook has to be lifted. The movement of the needle through the hole, under spring pressure, causes the hook to be ready to engage with the knife. The knife therefore lifts the hook during its upward movement, which in turn lifts the cord attached to the hook. The ends passing through the heddles (or mails) attached to the hook will then be lifted. When no hole faces the needle, it will be pushed by the card against the spring and the hook will be kept away from the knife.

In order to prevent the rotation of the hook, the lower end is bent and passes through a narrow slit in the *grate*. When the hook is not engaged with the knife the bent end rests on a *spindle* which controls the lower position of the warp yarns.

The capacity of a Jacquard head is indicated by the number of hooks (each hook has one needle). The single-lift Jacquard discussed here is the simplest and gives the necessary basic

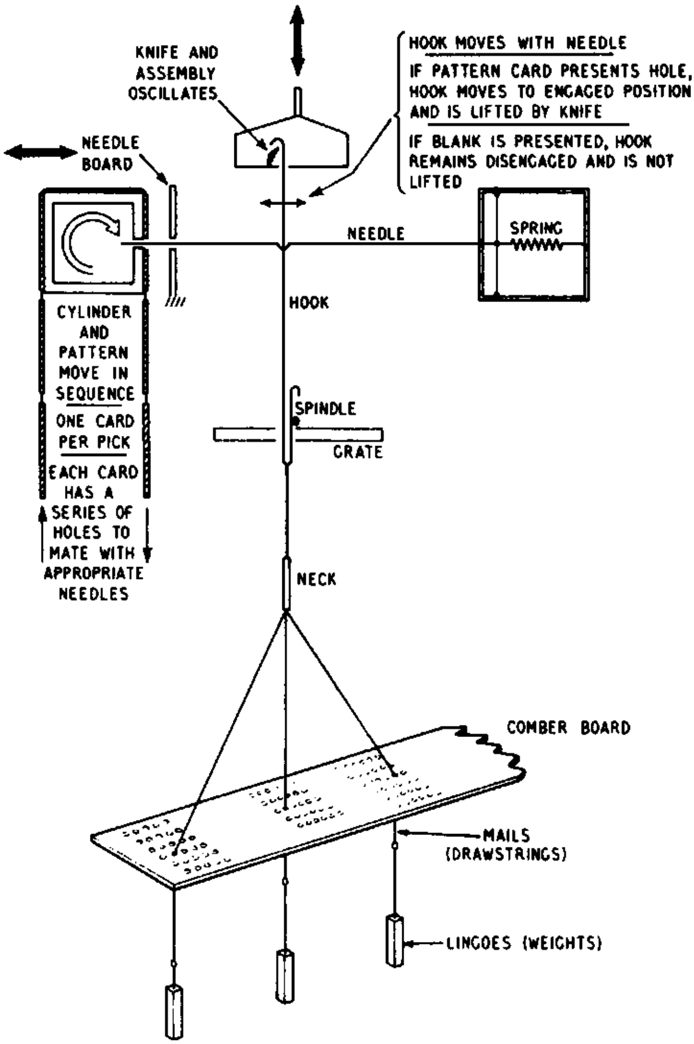


Fig. 11.10. Single lift Jacquard

principle of Jacquard operation, but there are also double lift Jacquards of a variety of designs which are more complex; these are beyond the scope of this book. Suffice it to say that double lift machines usually have two cylinders (one on each side of the needles), two card chains, and no spring box.

Selvage Motion

In the case of Jacquard and dobby looms there is no problem in weaving the selvage, which in all cases has a design different from the weave in the body of the fabric. With cam looms, however, the situation is different, since the number of harnesses is limited. In this case, it is necessary to use a separate shedding motion for the selvage yarns. In its simplest form, the selvage motion uses two pairs of small frames with heddles, one on either side of the loom. These frames get their motion from eccentrics or cams fixed on the loom camshaft, or on a separate shaft. The most common weaves used for the selvage are plain, 2 × 2 rib or 2 × 2 basket weaves.

Beating Motions

The Motion of the Lay

A lay mechanism is shown in Fig. 10.4. It will be seen that the mechanism is in reality a four bar chain in which the loom frame forms the link AD. The link AB is the *crank*, the link BC is the *connecting rod* and the link CD is the *lay sword.* As the crank AB rotates, it causes the point C to oscillate and if the reed is attached to the link CD (or an extension of it), it, too, will oscillate as required. If the crankshaft is geared to the camshaft (and auxiliary shaft where appropriate) in the proper manner, then the essential synchronism is achieved.

It is highly desirable that the lay should stay in the back position as long as possible so as to give the shuttle more time to pass. This requires that the lay should move in a fairly complex harmonic motion. To understand this it is necessary to analyze the mechanism. To simplify the motion, it is possible to assume that the point C moves along a straight line rather than along an arc (as the radius CD is always large compared to AB, this is a reasonable assumption). For the purposes of explanation, let it be further assumed that the line of action of C passes through A and that the loom speed remains constant. Let

r = length of crank
l = length of connecting rod
h = instantaneous perpendicular distance of B from line AC
θ = crank angle relative to inner dead center (in radians)
ϕ = inclination of connecting rod to AC (in radians)
x = displacement of lay from the beginning of its stroke
v = velocity of lay
= dx/dt
f = acceleration of lay
= d^2x/dt^2
ω = angular velocity of crank in radians/second

Displacement
$$x = l + r - l\cos\phi - r\cos\theta$$
$$= r(1 - \cos\theta) + l(1 - \cos\phi) \tag{11.4}$$

but
$$h = r\sin\theta$$
$$= l\sin\phi$$

therefore
$$\sin\phi = \frac{r}{l}\sin\theta$$

By Pythagoras
$$\cos\phi = \sqrt{1 - \sin^2\phi}$$
$$= \frac{r}{l}\sqrt{\left(\frac{l}{r}\right)^2 - \sin^2\theta}$$

Substitute in eqn. (11.4)

$$\frac{x}{r} = 1 + \frac{l}{r} - \cos\theta - \sqrt{\left(\frac{l}{r}\right)^2 - \sin^2\theta}$$

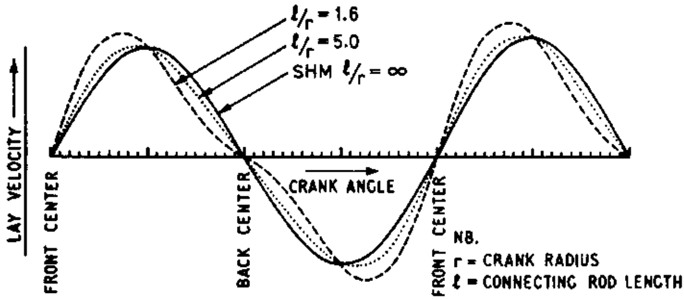


Fig. 11.11. Effect of short connecting rod

but if $$D^2 = \left(\frac{l}{r}\right)^2 - \sin^2\theta$$

$$x = r\left(1 + \frac{l}{r} - \cos\theta - D\right)$$

Differentiating $$v = \frac{\mathrm{d}x}{\mathrm{d}t}$$

$$= \frac{\mathrm{d}x}{\mathrm{d}\theta}\cdot\frac{\mathrm{d}\theta}{\mathrm{d}t}$$

$$= \omega\cdot\frac{\mathrm{d}x}{\mathrm{d}\theta}$$

$$= \omega r\left(\sin\theta + \frac{\sin 2\theta}{2D}\right)$$

$\mathrm{Sin}^2\,\theta$ is small compared to l^2/r and it is a close approximation to write $D = l/r$, whence

$$v \simeq \omega r\left(\sin\theta + \frac{r}{2l}\sin 2\theta\right) \qquad (11.5)$$

TABLE 11.1

Effect Of Varying the Connecting Rod Length

l/r	*Connecting Rod*	*Type of Movement*	*Type of Fabric*
Greater than 6	Long	Very smooth with low acceleration forces	Fine cotton Silk Continuous filament
Between 6 and 3	Medium	Smooth	Medium density cottons
Less than 3	Short	Jerky with high acceleration forces	Heavy cottons Woolen

Figure 11.11 shows a series of curves for various values of l/r. It will be seen that as the connecting rod is made shorter there is an increased deviation from the pure sine wave (the sine wave represents simple harmonic motion or SHM). This is another way of saying that the lay lingers in the back position because of the shortness of the connecting rod, and it is this which is used to advantage in the loom.

There is a limit to the amount of deviation which can be achieved because as $l \to r$ the mechanism becomes unworkable; when $l = r$, the crank and the connecting rod could spin round a common center and the lay would not move at all. In fact, when the connecting rod is made too short, the system will jam and it is normal for $l > 2r$.

The value of l/r used in a loom depends upon whether a smooth action is required or whether an impulsive jerky type of beat-up is needed; this in turn depends upon the fabric to be woven and the width of the loom. A fine delicate fabric should not be roughly handled, whereas a coarse staple yarn may require a sharp beat to be effective. With a wide loom, the shuttle transit time is long; this leaves less time for the other functions and therefore the beat-up has to be short if the loom speed is to be kept up. For example, l/r for a 45 in. loom might be 4, whereas for an 85 in. loom it might be as low as 2. The range of values for various conditions is shown in Table 11.1.

Generally, the long connecting rod enables a light loom construction to be used whereas the short connecting rod demands a heavy construction which adds to the weight of the reciprocating masses and further increases the forces. To understand how the high accelerations arise, it is necessary to differentiate eqn. (11.5).

The lay acceleration

$$f = \mathrm{d}v/\mathrm{d}t$$

$$= \omega \,.\, \mathrm{d}v/\mathrm{d}\theta$$

$$= \omega^2 r \left(\cos\theta + \frac{r}{l}\cos 2\theta\right) \qquad (11.6)$$

It will be noted that pure SHM would give an acceleration of $\omega^2 r \cos\theta$ and the difference caused by the short connecting rod is $\frac{\omega^2 r^2}{l}\cos 2\theta$. An infinitely long connecting rod would make the difference zero and the movement would be SHM; however, if $l/r = 2$, the peak accelerations will be more than 50 per cent higher than with SHM. (When $\theta = 0$, 180°, 360°, etc., both $\cos\theta$ and $\cos 2\theta = 1.0$; thus the portion of eqn. (11.6) in brackets becomes $1 + \frac{1}{2}$, i.e. only slightly less than the maxima which occur at angles slightly different from those quoted. With SHM the maximum value is 1·0.) Since force = mass × acceleration, and since the lay is relatively massive, these differences are important because the forces can become very large.

The foregoing has been developed on the basis of several simplifying assumptions, but even in the more complex cases, there are similar results; however, any further such mathematical development is beyond the scope of this book.

The Beating Action

The motion of the *reed* positions the filling, but in so doing it has to push against frictional forces imposed by the

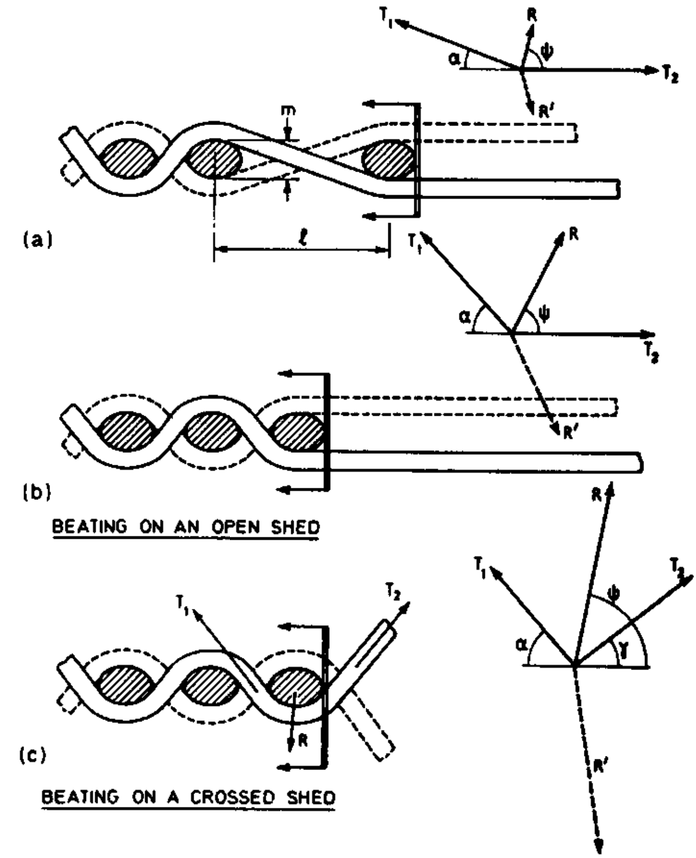


Fig. 11.12. The forces in beating

warp. The magnitude of these forces depends on the coefficient of friction between the warp and filling, as well as the reaction between them (this reaction arises from the interlacing).

Consider a filling yarn about to be moved towards the fell of the cloth as illustrated in Fig. 11.12. The yarn tensions on a single end are as shown on the right-hand side of the diagram. In addition, there are other reactions from the opposite interlacings as indicated by R'. As the filling is moved towards the fell, the angle α steepens, the magnitude of R and R' increases and ψ moves away from 90°. When moved far enough into the fell, the angle ψ becomes so acute that the filling would be squeezed out if it were not restrained. There is a critical value for ψ which is determined by the coefficient of friction. Thus the minimum pick spacing which can be obtained by beating on an open shed is determined by the coefficient of friction between filling and warp.

When beating on a crossed shed, γ is more nearly equal to α and there is a smaller tendency for the filling to be squeezed out. Hence closer pick spacings can be obtained.

Let the general case illustrated in Fig. 11.12(c) be considered. The force needed to move the filling over a single warp end consists of two parts; the first is a component of the reaction R resolved in the direction of beating, and the second is a frictional component which also has to be resolved in the same direction.

The force needed to cause the filling to slip $(F) = \mu R$ and this acts perpendicularly to R; thus the resolved component is $F' = \mu R \sin \psi$.

Resolving the direct forces vertically,

$$R \sin \psi = T_2 \sin \gamma + T_1 \sin \alpha \tag{11.7}$$

therefore $$F' = \mu T_2 \sin \gamma + \mu T_2 \sin \alpha$$

Resolving the direct force horizontally (in the direction of beat)

$$T_2 \cos \gamma - T_1 \cos \alpha = R \cos \psi \tag{11.8}$$

Beating force/warp end

$$F'' = R \cos \psi + \mu R \sin \psi \tag{11.9}$$

$$F'' = T_2 \cos \gamma - T_1 \cos \alpha + \mu R \sin \psi$$

Substitute for $R \sin \psi$ and we get

$$F'' = T_2(\cos\gamma + \mu\sin\gamma) - T_1(\cos\alpha - \mu\sin\alpha)$$

but from Amontons' law of wrap friction

$$T_1 = T_2 e^{\beta}$$

$$\text{where } \beta = -\mu\,[\pi - (\alpha + \gamma)]$$

therefore

$$F'' = T_2\{(\cos\gamma + \mu\sin\gamma) - Ke^{-\mu(\alpha+\gamma)}(\cos\alpha - \mu\sin\alpha)\} \tag{11.10}$$

When the angle γ is small, as it often is, then eqn. (11.10) can be simplified as follows

$$F'' = T_2\{1 - Ke^{-\mu\alpha}(\cos\alpha - \mu\sin\alpha)\} \tag{11.11}$$

If further approximation is tolerable then this can be reduced to $F'' = 2\mu(\sin\alpha)T_2$ or even to $F'' = 2\mu\alpha T_2$.

Bearing in mind that the angle α is determined by the crimp levels, the yarn dimensions and the pick spacing, it will be realized that the beat-up forces are affected by them too. Also, the tension T_2 is affected by the beat-up and, as demonstrated above, the beat-up force is also a function of T_2; this is another reason why it becomes progressively more difficult to beat up as the filling is pushed into the fell of the cloth. Indeed, too heavy a beating action to achieve a close pick spacing will produce a condition known as *bumping*; here the whole beat-up force is taken by the warp and the fabric ahead is temporarily relieved of tension. Bumping usually indicates a *jammed* fabric; that is, one which cannot have a closer pick spacing due to its construction.

Referring again to Fig. 11.12(a), it will be seen that the angle α is fixed by the dimensions m and l. At the beginning of beat-up, l is long and α is small but as the process proceeds, l gets shorter with the result that α becomes more acute. Hence, as beat-up proceeds, the beating forces increase until the lay is reversed or the bumping condition is reached (see

Fig. 10.8). Because of this, it is important to set the loom properly to prevent undue stress; high stresses lead to increased ends down and consequent loss of production. Also it is possible to damage the machinery, particularly if it is designed only for light fabrics.

12

SHUTTLE PICKING AND CHECKING

Key words: *Alacrity, binder, buffer, checking, coefficient of restitution, controlled binder, kinetic energy, lug strap, natural frequency, nominal movement, overpick loom, picker, picking, picking stick, picking cam, pitch, roll, shuttle box, shuttle box length, simple harmonic motion (SHM), stick checking, underpick motion, yaw.*

The Energy of Picking

To accelerate a shuttle to the necessary speed in the very short time available requires that considerable amounts of energy should flow in that short time. This means that there has to be a source of energy as near to the shuttle as possible, with as little intervening mass as possible. Such masses impede the flow of energy and make it more difficult to accelerate the shuttle. In the conventional loom, a transient source of energy is the picking stick itself; this is normally made of wood, which is a very resilient material capable of storing remarkable amounts of strain energy. The primary source of energy is the electric motor, but the energy is transmitted through a cam/linkage system which has considerable inertia. Without the resilient stick to store energy at appropriate times (i.e. with a perfectly stiff mechanism), such a cam/linkage system would be unsatisfactory. If sufficient energy were transmitted to give adequate acceleration, the forces generated in some parts would be above the safe working limits and failures would occur.

A conventional system works, in the first part of the operation, by causing the drive end of the mechanism to start with little or no motion by the shuttle and picker. This is achieved by the action of the *binder* which temporarily restrains the shuttle. During this first phase, various parts of the mechanism are strained (especially the *picking stick*) and strain energy is built up. During the subsequent phase, the direct motion of the drive end continues but the shuttle now moves. The direct motion and that due to the sudden unbending of the picking stick are superimposed to give a very rapid acceleration. There is a sort of whiplash effect, which is extraordinarily effective in producing the acceleration required without producing unacceptably high stresses in the mechanism.

The Motion of the Shuttle during Acceleration

The force applied to the shuttle is proportional to its mass and the acceleration involved (i.e. force = mass × acceleration). For a given shuttle, it is necessary to restrict the acceleration if the limiting force is not to be exceeded. The limiting force is determined by the strength and durability of both the shuttle and the propulsion mechanism. In this respect it is the peak acceleration that is important (rather than the average value), and this peak should be kept as low as possible. On the other hand, to get a high loom speed it is necessary to have a high average acceleration. These requirements clash, but the conflict can be lessened by making the ratio of peak to average acceleration as low as possible. This implies that the shuttle displacement curve should be parabolic, as shown in Fig. 12.1. This can be explained in mathematical terms as follows.

Let

$$x = at^2 + bt + c \tag{12.1}$$

where a, b, and c are constants and the eqn. (12.1) describes a parabola

t = time ($t \propto \theta$ approx.)

θ = angular position of crankshaft

x = position of shuttle from some reference

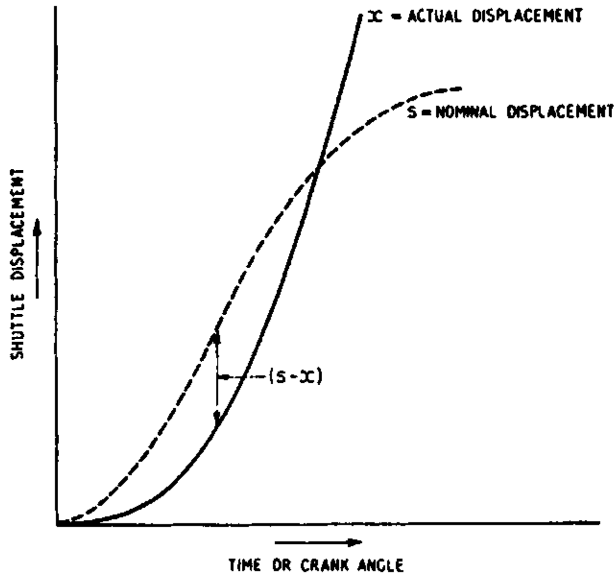


Fig. 12.1. Shuttle displacement

Differentiating

$$\text{shuttle velocity} = \dot{x} = 2at + b$$

$$\text{shuttle acceleration} = \ddot{x} = 2a = \text{constant}$$

where $\dot{x} = \mathrm{d}x/\mathrm{d}t$

$\ddot{x} = \mathrm{d}^2x/\mathrm{d}t^2$

Under these circumstances, the ratio of peak to average is unity, which is the lowest possible value; this means that for a given loom speed, the shuttle acceleration (and the force applied) will be the lowest possible.

The force pulse applied to the shuttle in this case will be rectangular, similar to that shown in Fig. 11.5; the steps in force levels are likely to create unwanted shocks which lead to the production of noise and vibration. To reduce shocks, it is necessary to round off the flanks and this implies that the displacement curve has to depart a little from the simple parabola.

The accelerating distance x is of some importance. From space considerations, it ought to be as short as possible, but from force considerations it ought to be as long as possible. Bearing in mind that there are geometric limitations in any picking mechanism and that the maximum forces acting on the shuttle arises elsewhere, it is usual to make x about 20 cm (8 inches). This determines the *shuttle box length* as well as the difference between the loom and fabric widths.

The motion of the picker is never geometrically similar to the shape of the *picking cam*; however, to overcome this in the present discussion, let the cam shape be defined by the motion of the picker, it being recognized that the actual cam shape has to be modified to take into account the geometry of the linkage.

There is another and very important factor to be considered. The picking stick and other parts deflect under the loads encountered under running conditions; it is necessary, therefore, to make another modification to the cam profile if the actual shuttle motion is to approximate to the parabolic ideal. If the loom is turned over very slowly with no shuttle, there will be little or no load on the system; it will behave as if the components were perfectly stiff. Let the movement obtained like this be referred to as the *nominal movement*. The actual movement, obtained under normal running conditions, differs greatly from this because of the deflections of the various components in the system (see Fig. 12.1).

Assuming the system to be perfectly elastic:

$$\text{force} \propto \text{acceleration}$$

but also

$$\text{force} \propto \text{deflexion}$$

therefore

$$\text{acceleration} \propto \text{deflexion}$$

or in symbols,

$$\ddot{x} = n^2 (s - x) \qquad (12.2)$$

where n = a constant which is known as the *alacrity*

$\ddot{x}$ = shuttle acceleration

$(s - x)$ = elastic picker displacement

s = nominal displacement of picker

x = actual displacement of picker

Equation (12.2) is mathematically identical to that for simple harmonic motion. This implies that the picking stick with its various associated masses is in fact a vibratory system which has its own particular *natural frequency*. This vibratory motion, which is superimposed upon the applied motion derived from the cam, is fairly heavily damped so that it scarcely carries over from one pick to another; nevertheless, it is very important in determining the behavior of the picking system as a whole.

In a normal loom, it is necessary to apply an average force F_s to the accelerating shuttle for a time t_s at the beginning of each pick. Since it is required that the strain energy be released in the time t_s, it is most desirable that the natural frequency should be such that at least a quarter cycle of vibration should be completed in the time t_s. Since the alacrity is proportional to the natural frequency (which in turn is a function of the mass and elasticity of the system), the foregoing really fixes the flexibility of the picking system for a given shuttle weight. In a typical case, the force F_s might be of the order of 50kg (110 lb), the time over which this force has to be applied (i.e. t_s) might be some 0.02 sec and the relative deflection of the picker might be over 5 cm (2 inches); the natural frequency might be about 20 c.p.s., which is equivalent to an alacrity of about 120/sec.

The energy stored in deflecting the stick by the stated amount might represent as much as a quarter of that needed to propel the shuttle at the required speed. The strain energy released at the appropriate time gives that extra impulse that makes the system so effective. If the stiffness of the system is too low (i.e. the alacrity is too low), the propulsion phase of the pick will be completed before all the energy has been released and this will reduce the effectiveness of the system. If the system is too stiff, the energy is released too quickly and the energy is largely dissipated in noise and useless vibration. Thus it is important that the picking mechanism be properly designed to meet the requirements of the particular loom working with its particular shuttle at its particular speed. Of course there is some band of tolerance and it is, for example, possible to speed up a loom to some degree without redesigning the picking system; to get the best possible advantage, however, it is worth making sure that the design is reasonably adequate in this respect.

Another factor must be considered. The rest position of the shuttle determines the amount of energy that is built up and this in turn determines the shuttle velocity. In normal weaving, this rest position varies from pick to pick and therefore the shuttle velocity also varies. The arriving shuttle tends to bounce away from the picker and to leave a gap just before picking starts. The size of the gap is dependent on the arrival velocity of the shuttle; thus there is an unstable situation which is characteristic of most looms. The instability is made worse by certain other factors (such as the torsional oscillation of the bottom shaft) but these other factors are rather difficult to control without redesigning the loom; consequently, they will not be further discussed here. The gap, however, can be controlled; for example, a good controlled buffer or other device can return the stick and the shuttle to their proper starting positions and thus eliminate some of the instability. A smoother running loom tends to be more efficient in all senses and it is usually worthwhile to eliminate the causes of irregularity.

It was suggested earlier that a parabolic motion was usual and that this would produce a rectangular force pulse on the shuttle. Equation (12.2) suggests that the value $(s - x)$ is proportional to the force applied to the shuttle;

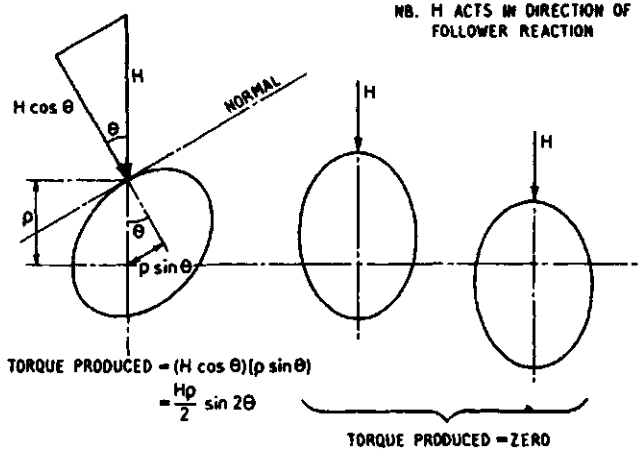


Fig. 12.2. Varying mechanical advantage of a cam

however, Fig. 12.1 shows this to be untrue. One reason for this is that the alacrity is not a constant; it varies from time to time within the cycle. The alacrity is dependent on the mass-elastic behavior of the whole system and this means that the driving shafts and linkages are all involved to some extent. The cam and follower in this system act as a link which transmits some of these effects to a varying extent dependent on the angular position of the cam. Referring to Fig. 12.2, it will be seen that when the follower is climbing the flank of the cam, forces may be transmitted either way through the system. When the follower is at either extreme; no such transmission is possible; rotation of the cam causes no movement of the follower and a force exerted on the follower merely tends to dent the surface of the cam. In fact, the extent of the force transmission depends on the local angle between the cam surface and the follower and thus it varies from time to time. Hence the effect of the bottom shaft, and other systems connected to it, is not constant.

Where the drive to the cam is stiff, this is not of great significance, but where the drive is flexible, the effect can be quite noticeable.

In the normal loom, the driving motor is located to one side, the motion being carried to the other side by shafts. In picking, it is necessary to have a cam system on either side of the loom; one is driven by a short stiff shaft and the other by a long flexible extension of that shaft. This affects the behavior quite markedly and it is usual to fit unlike cams, i.e. the cam at the end of the flexible shaft is modified to take the difference into account. This is a further reason for care being taken with cams. Even with the modified cam profiles, picking is frequently uneven and must be adjusted to give equal "strengths" of picking by altering the position of the *lug strap*.

Parallel Motions

If a picking stick were to swing about a simple fulcrum at one end, the other end would move in an arc; however, the shuttle moves in a straight line. The older *overpick looms* (which have mechanisms such as that shown in Fig. 12.3) solve the problem by constraining the picker by guides and allowing the motion of the strap to compensate. Some looms with an *underpick* motion also have guided pickers, and in this case the picker can slide on the picking stick to achieve the necessary compensation. A different approach is to use a curved foot (Fig. 12.4) or an equivalent linkage to cause the stick to lift as it rocks, and to do so in such a way that the free end of the stick moves in a straight line. In this way it is possible for the picker to be attached firmly to the free end of the stick. It is essential, however, that the stick should be properly restrained so that it cannot lift except as desired. One consequence of an unsuitable lift is that the shuttle may be projected in a wrong direction and fly out of the loom; this is highly dangerous. An unsuitable lift might also damage adjacent mechanisms such as the quill changing device.

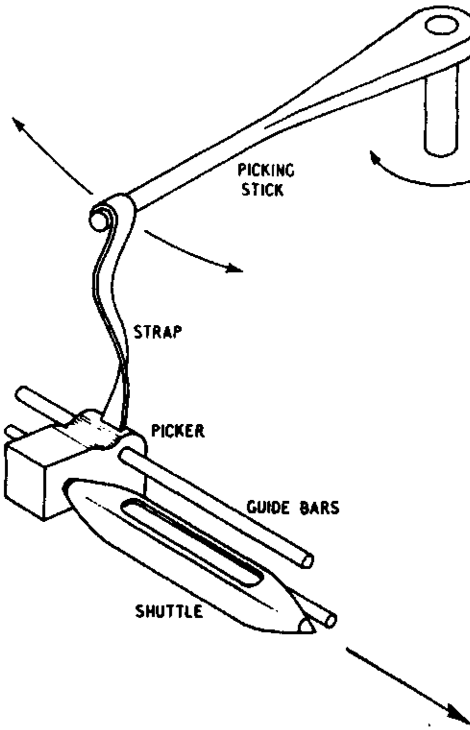


Fig. 12.3. Overpick loom picking mechanism

Stick Checking

At the time that the shuttle leaves contact with the picker, the picking stick has completed its task but it still possesses appreciable *kinetic energy*. This energy has to be dissipated before it can be returned to its starting position in readiness for the next pick from that side. The simplest way of doing this is to place a resilient buffer in the path of some portion of the stick, but this produces a heavy concentrated load at the impact point. Furthermore, the load caused by the decelerating stick is distributed on either side of the impact point, with the result that there is severe bending as the stick tries to wrap itself round the *buffer*. When it is realized that the kinetic energy possessed by the stick just before checking is about the same as that possessed by the shuttle, and that it is decelerated in a fraction of the time that was

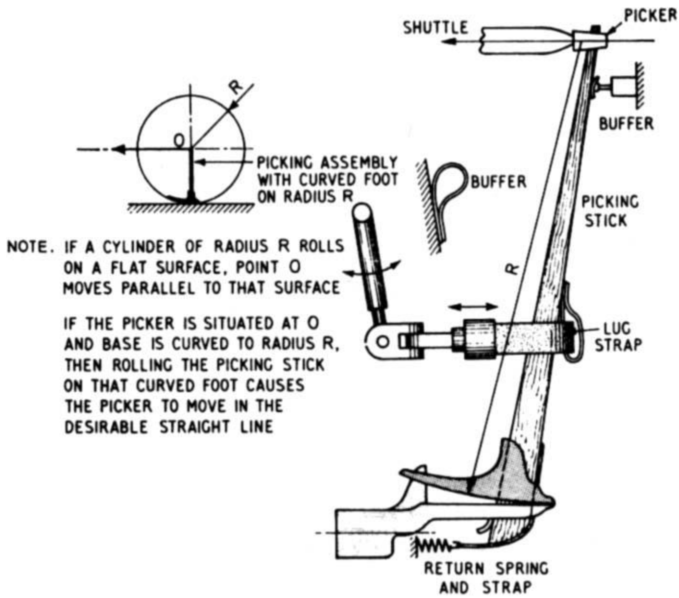


Fig. 12.4. A normal underpick motion with a curved foot

spent in accelerating the system, it will become evident that the forces involved must be very large. The highest stress levels in the picking stick are encountered at this time, stick failures occur most frequently at this time and stick damage builds up due to the impacts met during this phase of the stick motion. The stick vibrates within its own length and a number of unintentional movements are generated; the stick is usually thrown backwards and forwards within the constraints of the linkage and it may also lift. These movements can cause subsidiary damage; more important, they cause considerable noise and vibration through the whole loom.

Shuttle Checking

After the shuttle has traversed the warp shed and has left its trail of filling behind, it too has to be decelerated very rapidly. It is not possible to use a check as severe as that used for the stick, as this would cause the yarn to move on the quill (which would make it impossible to unwind properly and would lead to loom stops). The system almost universally used is shown in Fig, 12.5. The *binders* (swells) rub on

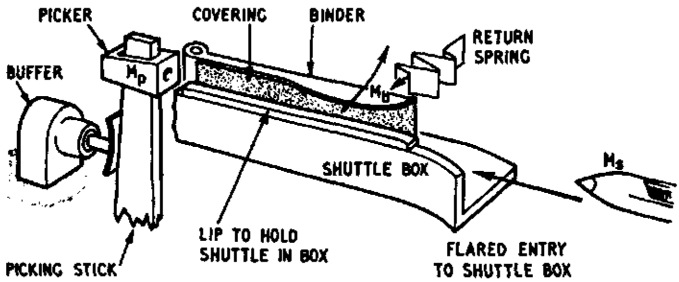


Fig. 12.5

the incoming shuttle and the frictional force slows down the shuttle but does not stop it. The final braking action is obtained as the shuttle collides with the picker, which at that time is backed with a suitable buffer. The binders are not very effective as a brake but they do serve another purpose (see page 220); they rarely remove more than about 20 per cent of the kinetic energy from the incoming shuttle.

At the risk of considerable oversimplification, conside the following theoretical model. To avoid the difficulties of asymmetrical loading of the shuttle, let the binder be split into two equal parts and let one part act on one side of the

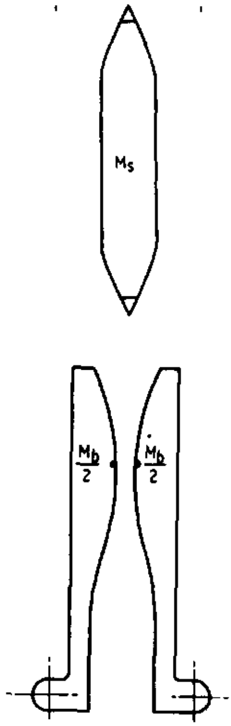


Fig. 12.6.
Theoretical model

shuttle and the other on the opposite side as shown in Fig. 12.6. Furthermore, let both portions of the binder act in the same way so that at any given time they both have the same velocity. Let subscripts s and b refer to shuttle and binder, respectively.

In general,

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

Retarding force acting on the shuttle

$$= F_s = M_s \times \mathrm{d}V_s/\mathrm{d}t$$

Also $$F_s = \mu(F_b' + F_b'')$$

but $$F_b' = F_b'' = F_b$$

and $$F_b = \tfrac{1}{2}M_b \times (\mathrm{d}V_b/\mathrm{d}t)$$

Therefore $$M_s \times \mathrm{d}V_s = \mu M_b \times \mathrm{d}V_b \tag{12.3}$$

This equation has the same form as that which describes the collision of two bodies traveling along the same line of action, except the two masses are M_s and μM_b instead of their real values. Using the classical solution to the simple collision problem and defining the *coefficient of restitution* (e) as the ratio $(V_s' - V_b')/V_s$, we may formulate the change in shuttle velocity as follows.

(N.B. the superscripts refer to the stated quantities just after impact and V_s is defined below.)

$$\frac{\mathrm{d}V_s}{V_s} = \frac{(1+e)\mu M_b}{(M_s + \mu M_b)} \tag{12.4}$$

where e = the coefficient of restitution
μ = the coefficient of friction
M_s = the mass of the shuttle
M_b = the mass of the binder
$\mathrm{d}V_s$ = the change in shuttle velocity
V_s = the velocity of approach of the shuttle

Apart from the somewhat sweeping assumptions made, the coefficient of restitution is not a constant and if there is vibration present, it can vary with conditions (between 0·2 and 0·6). For the present purpose, this is of no great importance since the equation is much more sensitive to changes in μ and M_b. There is only a limited range for the coefficient of friction; therefore, the binder mass is seen to be the most important factor.

Experimental work has demonstrated that changing the nature of the binder covering has little effect. Traditionally, the covering was leather and this provided durability and a reasonably high coefficient of friction against the wood of the shuttle. Attempts to replace leather by materials of different coefficients of friction sometimes yielded strange results. For instance, substituting a steel strip for the leather produced a more effective check even though the coefficient of friction was reduced; this was because the mass of the binder was increased and this more than compensated for the change in frictional character of the surface. Resilient backings altered the time of contact between the binder and shuttle and thereby decreased the average pressure, but this had little or no effect on the amount of energy extracted from the shuttle.

In the normal system, the remaining kinetic energy in the shuttle has to be dissipated by striking the picker attached to the picking stick at the arrival side of the loom. Apart from the energy extracted by the buffer, the impact with the picker and its associated masses causes energy to be dissipated. In fact, this is the classical impact case and eqn. (12.4) can be rewritten as

$$\frac{\mathrm{d}V_s}{V_s} = \frac{(1 + e)M_p}{(M_s + M_p)} \qquad (12.5)$$

where M_p is the mass of the picker and associated masses. Other symbols are as for eqn. (12.4).

Since M_s and M_p are roughly of the same magnitude, the change in shuttle velocity is very large; it is possible for the change to be as high as 70 per cent of the arrival velocity. This may be compared to a normal change of about 20 per cent caused by the binders. Some of the energy of the approaching shuttle (but not all of it) is converted into kinetic energy in the picker and its associated masses. This energy is dissipated by the buffer and the stick is brought to rest only to suffer another blow from the shuttle. However, this blow is a light one and has little significance. If the loom is set correctly with a good buffer, the separate actions just described may merge into one continuous action. A typical checking characteristic is given in Fig. 12.7 and this also shows the acceleration for comparison. It also shows the loss in velocity during transit across the warp shed. The mass M_p is significant and an increase in mass could give a more rapid checking action but any change in M_p will alter the alacrity; thus there is little chance of securing a gain in this

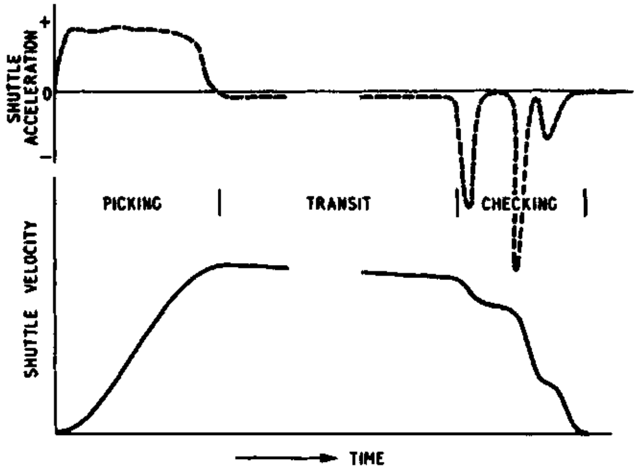


Fig. 12.7. Variation of shuttle velocity during one pick

direction. Furthermore, the checking distance is important. Most of the shuttle deceleration is confined to the last inch or so of its travel. The shuttle was accelerated over a distance of several times this, hence the forces involved in shuttle checking are considerably larger than in picking. The acceleration (or deceleration) of the shuttle has to be limited to prevent damage to the yarn on the quill. As loom speeds rise, this problem becomes more acute.

The shuttle is a projectile whose path is not subject to exact restraint. The path is not linear, because of the movement of the lay. To this must be added the effects of contact with the reed, lay and warp. It is normal for the shuttle to *roll*, *pitch* and *yaw* (see Fig. 12.8) as it passes across the warp shed. Consequently, the shuttle rarely enters a shuttle box cleanly; often it enters obliquely with the result that appreciable transverse forces are generated. Measurements have shown that these can be as much as ten times the worst force acting along the path of the shuttle. These forces can be very damaging to the yarn on the quill and sometimes even to the shuttle. Careful setting of the picking mechanism can minimize this difficulty. In particular, the path of the picker can be very important and any tendency for it to rise during picking will cause the shuttle to deviate from its intended path. Hence, the parallel motions described earlier are seen to be very important.

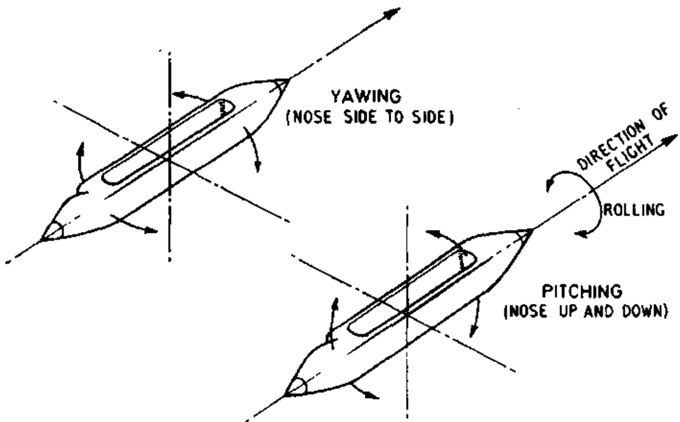


Fig. 12.8. Shuttle movements during flight

An interesting concept in checking is to use multiple binders so that the shuttle is retarded in steps. This distributes the load and instead of a single blow to reduce the shuttle speed before it collides with the picker, it is possible to have several lesser blows. The shocks arising from these lesser blows are more tolerable than those from the single one. An established variant of this is to use a *controlled binder* system comprising two units. One of these acts independently like the normal binder and the other has a variable spring pressure which can be applied or released at various times in the weaving cycle. Although its main purpose is to control the shuttle during picking, the auxiliary binder also helps in checking.