13

AUTOMATION AND CONTROLS

Key words: *back rest, barré, battery, beam ruffle, bobbin loaders, box chain,* 2 x 1 *box motion, box motion, bunch, bunter, center filling fork, change gear, cloth roller, dagger, direct take-up, drop wire, droppers, feeler, filling transfer mechanism, fork, frog, give-way, hammer lever, knock-off device, knock-off motion, let-off motion, loose reed, magazine, midget feeler, multiplier chains, negative-feed-back, negative let-off, pawl and ratchet, pick-and-pick, pick at will, pinned, positive let-off, protector feeler, race-board, risers, semi-positive let-off, sheath, sinkers, side filling fork, smash, stop motions, stop rod finger, take-up roller, temples, tin fillets, transfer latch, unifil, warp stop motion, well, worm and wheel.*

In a loom there are many mechanisms to control the warp and fabric tensions, fabric width and color pattern in the filling direction. There are also some automatic protection devices such as warp and filling *stop motions.* One very important mechanism on the loom, which gives the loom the adjective "automatic", is the *filling transfer mechanism.*

Warp Let-off Motions

The function of a *let-off motion* is to apply tension on the warp yarns to help form a clear shed. The tension also has to be high enough to develop the forces required between the warp and filling to form the cloth. The tension ratio between the warp and filling has to be correct, otherwise the crimp levels are improperly balanced; this affects the appearance of the cloth. The let-off motion applies the tension by controlling the rate of flow of warp yarns.

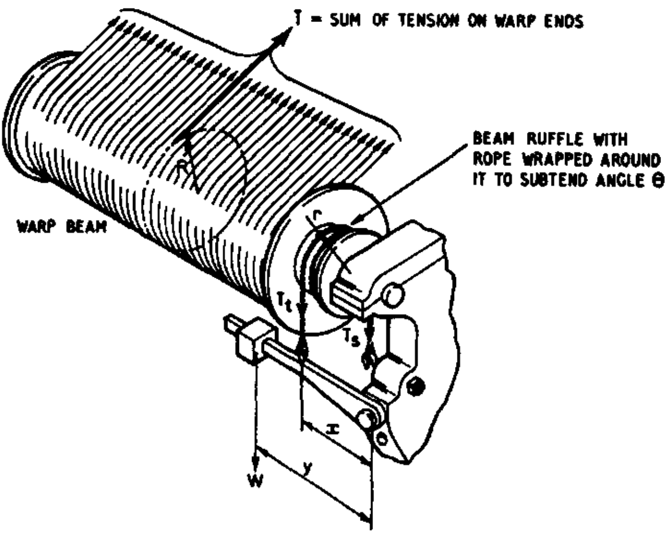

Fig. 13.1. Negative let-off ;not to scale.

There are three different types of let-off motions, namely:

(1) Negative.
(2) Semi-positive.
(3) Positive.

1. *Negative Let-off*

In this case the pull of the warp is purely against friction forces in the let-off motion. To demonstrate the principle of negative let-off, let the simple non-automatic mechanism shown in Fig. 13.1 be considered. The tension of the warp is regulated by the friction between chain or rope and the *beam ruffle*.

Taking moments about the center of the beam

$$T \times R = (T_t - T_s) \times r \qquad (13.1)$$

and by Amontons' law

$$\frac{T_t}{T_s} = e^{\mu\theta} \tag{13.2}$$

where μ = coefficient of friction between the chain or rope and the beam ruffle.

θ = the angle of lap

Taking moments about the hinge O of the lever:

$$T_t \times x = W \times y \tag{13.3}$$

$$\therefore \qquad T_t = W\frac{y}{x} \tag{13.4}$$

Substituting for T_t and T_s in eqn. (13.1)

$$\therefore \qquad T = \frac{r}{R} T_t \left(1 - \frac{T_s}{T_t}\right)$$

$$= \frac{r}{R} \times W\frac{y}{x}(1 - e^{-\mu\theta})$$

$$= K\frac{Wyr}{xR} \tag{13.5}$$

i.e.,

$$T \propto \frac{1}{R} \tag{13.6}$$

This means that the tension of the warp increases as the radius of the warp on the beam is reduced. This cannot be allowed in practice and the increase in tension must be balanced by moving the weight towards the fulcrum O, to reduce the distance y. Therefore, the condition needed to maintain a constant warp tension is that

$$\frac{y}{R} = \text{constant} \tag{13.7}$$

The movement of the weight can be either manual or automatic. In modern looms, only negative let-off motions of the automatic type are used. In this case, the weight is not moved along the lever, but the lever is fixed in a different way such that any change in tension causes a change in the moment applied.

2. *Semi-Positive Let-off*

In the case of negative let-off the warp is pulled off the beam and the tension is regulated by slippage in a braking system. In a positive let-off, the beam is driven through a positive mechanism where no slippage takes place. This latter type of mechanism is seldom used and in most cases the tension is controlled by a mechanism driving the warp beam, which allows a certain loss of motion (or slippage) whenever the tension increases. Basically these are crude *negative-feed-back* automatic-control systems which are related to controls such as are used in autoleveling during sliver production. These mechanisms are sometimes considered positive, but in reality they are semi-positive.

Figure 13.2 shows an example of a semi-positive let-off motion. The warp beam is driven through a *worm and wheel*

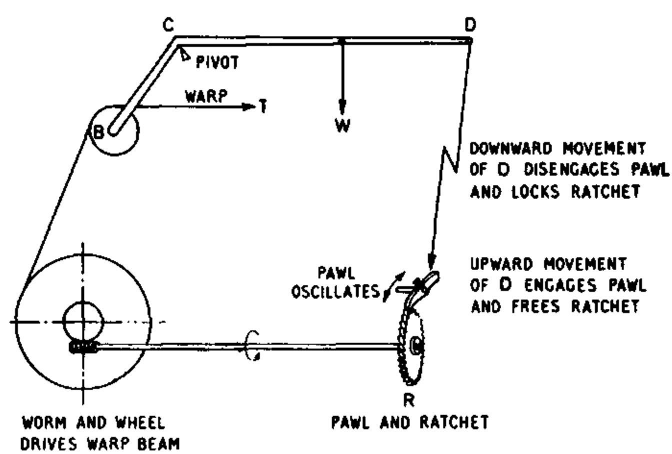

Fig. 13.2. Semi-positive let-off

which are turned by a *pawl and ratchet* (*R*). The warp passes over a moveable *back rest* (*B*). If the warp tension is reduced, the back rest is moved up under the influence of the weighted lever (*CD*). The motion is transmitted through levers to a *sheath* which locks the movement of the ratchet wheel and disengages the pawl until the tension increases again to the normal level under the influence of the cloth take-up. If the tension increases beyond normal, the back rest is lowered, which causes the sheath to be moved away from the ratchet wheel; the let-off is restarted and overfeeds slightly, causing the tension to drop. The tension remains nearly constant between limits, and does not depend on the warp diameter on the beam.

There are many semi-positive let-off mechanisms designed to control the warp tension and it is beyond the scope of this book to discuss all of them.

Fabric Take-up Motions

As the picks are inserted, the point of fabric formation has to be moved and, to maintain the same pick spacing, the rate of movement must be kept constant.

The fabric commonly follows one or other of two paths. These are used in the direct and indirect take-up systems, as shown in Fig. 13.3. In the indirect system, at (a) and (b) the fabric is passed over a *take-up roller* before being wound over the *cloth roller*. The cloth roller of (a) is driven through friction between it and the take-up roller. This is suitable for spun yarn fabrics, since the friction does not produce a critical defect in the fabric. In the case of the indirect motion at (b), the cloth roller is negatively driven and is kept away from the take-up roller; this makes this type of motion suitable for the more sensitive continuous filament yarn fabrics. One of the advantages of this system is the possibility of cutting the fabric and removing the cloth roller without stopping the loom.

The motion shown at (c) is known as "*direct take-up*"; the fabric is wound on the cloth roller directly with a press

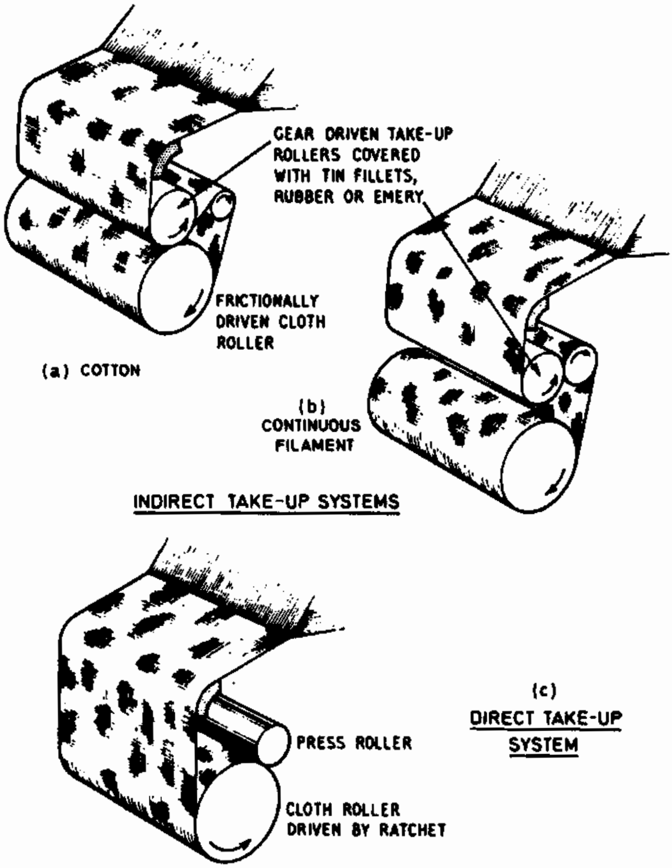

Fig. 13.3. Cloth take-up systems.

roller. The drive of the cloth roller is of the negative type so as to reduce the rotational speed as the diameter of the cloth on the roller increases. For this reason, the indirect take-up motions are normally considered positive whereas the direct motions are considered negative.

The take-up rollers are normally covered by *tin fillets* or rough rubber to increase the grip between the roller and the fabric. Also, the take-up roller is usually driven through a gear train which reduces a single movement of the pawl to the distance between two successive picks in the fabric. If the commercial range of pick densities is considered to

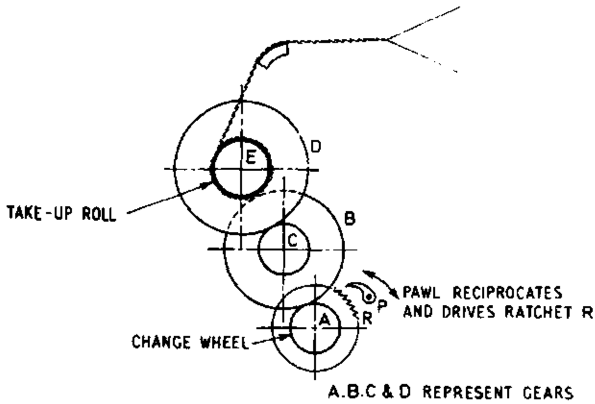

Fig. 13.4. Five wheel take-up system

be between 8 and 40 picks/cm (20 and 100 ppi), the distance moved by the cloth or take-up roller is between ¼ and 1¼ mm/pick (0.01 and 0.05 inch/pick). This movement is very small and a gear train of 5 or 7 wheels must be used to allow for the large reduction in movement and to provide for the changing of the pick density of the fabric. This is normally achieved by changing only one gear, called the *change gear*, in the train.

Figure 13.4 shows a 5-wheel take-up mechanism in which the ratchet wheel (*R*) is moved the distance of one tooth by the oscillating pawl (*P*) for every pick. The pawl gets its movement from the lay mechanism. If *R* is the number of teeth on the ratchet wheel, one revolution of the ratchet wheel corresponds to *R* picks inserted into the fabric.

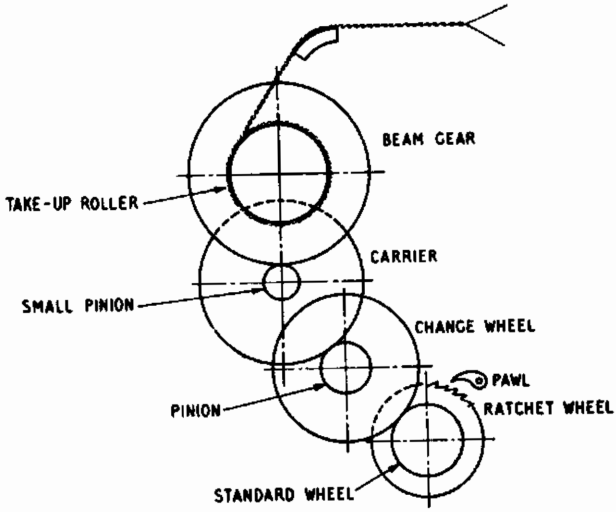

Fig. 13.5. Seven wheel take-up system

The length of cloth taken-up for every revolution of the ratchet wheel = $(A/B) \times (C/D) \times$ circumference of take-up roller (E).

Pick density = Picks inserted ÷ Length of Fabric Taken-up

$$= \frac{R}{(A/B) \times (C/D) \times \text{circumference of } E}$$

$$= R \times \frac{B}{A} \times \frac{D}{C} \times \frac{1}{\text{circumference of } E}$$

A is the number of teeth of the change wheel. The other gears usually are not changed. Therefore,

$$\text{Pick density} = \frac{\text{constant}}{A}$$

Due to fabric contraction after leaving the loom, the pick density in the fabric is normally increased by 1·5–2 per cent. This has to be taken into account when designing the gear train so that,

$$\text{Pick density} = \frac{\text{constant}\ (1 + K)}{A} \qquad (13.8)$$

where K = percentage contraction $\div$ 100.

Many looms use a 7-wheel gear train similar to that shown in Fig. 13.5, in which case the change wheel is a driven gear instead of a driving wheel (as in the 5-gear train). The pick density is then given by

$$\text{Pick density} = (\text{constant} \times \text{change wheel teeth})(1 + K) \qquad (13.9)$$

In this case, the gear train is usually designed so that the number of teeth on the change wheel is equal to the pick density. This means that the constant, after taking contraction into consideration, is equal to unity.

A very important point to be considered in the design of a take-up motion is the effect of imperfections in the cutting or mounting of the gears. Any eccentricity in a gear in the train produces cyclic variation in pick spacing which produces a defect in the fabric known as *barré*. If the spacing of the bars is either less than 2 mm or more than 25 cm, the effect is not readily seen in the fabric. Thus, in a well-designed take-up motion, the length of fabric woven during a full rotation of any gear in the train should be less than 2 mm (0.1 inch) or more than 25 cm (10 inches).

Figure 13.6 shows the Shirley take-up motion which was designed to satisfy this condition. The length of fabric woven for every rotation of the standard wheel A for a cloth of 25 picks/cm (60 ppi) is about 1.7 mm (0.07 inches). The length of fabric woven for one revolution of the carrier wheel B is 1.4 mm. Every rotation of the worm wheel E produces 25 cm of fabric. In this mechanism, also, the number of teeth on the change wheel is equal to the pick density. The minimum number of teeth on the change wheel to satisfy the previous condition concerning barre is 32. This good characteristic of the mechanism was made possible through the use of a worm and wheel with a speed reduction of 150:1.

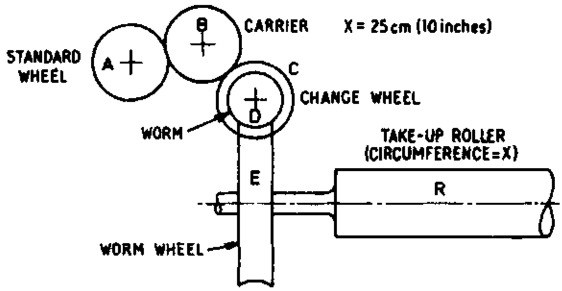

Fig. 13.6. The Shirley take-up.

Transverse Fabric Control

Due to the crimp in the filling yarns, a component of the tension exists in the filling direction (see Fig. 8.9). This force tends to bring the ends closer together, causing a contraction in fabric width. The contraction is exaggerated by the increase of warp tension or stiffness and also by increasing the twist in the filling yarn. When this contraction is allowed to be excessive, high frictional forces are created between the reed and warp yarns near the selvage. This, apart from affecting the fabric quality, also increases the warp breaks near the selvage and in turn reduces the weaving efficiency. To control the fabric width and maintain proper crimp levels for warp filling, the fabric has to be pulled at the selvages in the direction of the filling. This is done by using *temples* similar to those shown by Fig. 10.9. There are many types and designs of temples, but they all perform a similar function.

Warp Stop Motions

One of the important automatic protective devices on the loom is the *warp stop motion.* The main function of this motion is to stop the loom, in a very short period of time, when a warp yarn breaks. This helps to maintain the quality of the fabric, and to reduce the time to repair the breaks, thus improving weaving efficiency. Both considerations have a direct effect on the economics of the process.

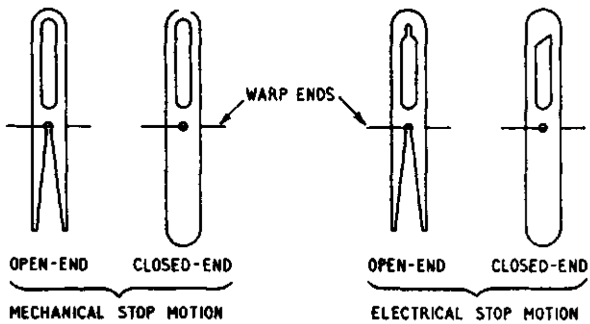

Fig. 13.7. Typical drop wires.

There are two types of warp stop motions, viz. mechanical and electrical. In both cases a *feeler* or *drop wire* is used for each warp end. The ends are drawn through the drop wires prior to weaving or they may be placed on the warp yarns on the loom. The drop wires (or *droppers* as they are sometimes called) have to be thin, because so many must be fitted across the width of the loom. Typical designs are shown in Fig. 13.7. The droppers used for both mechanical and electrical stop motions are basically similar, but may show slight differences. They are either open-ended or closed. The open-end dropper can be placed or "pinned" on the yarn without threading, but the closed-end droppers are better secured to the yarn.

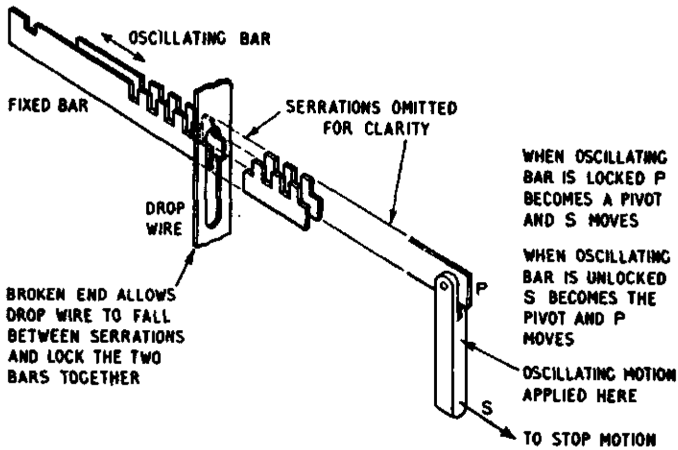

Fig. 13.8. A typical mechanical stop motion

The usual mechanical stop motions consist of two serrated bars, as shown in Fig. 13.8. One of the bars slides to and fro continuously with respect to the other. The droppers are held clear of the bars by the warp yarn. If a warp end breaks, the drop wire falls between the serrations and locks the two bars together. The movement of the drive is transmitted to a *knock-off motion*, i.e. it operates a device which stops the loom, usually by causing the starting handle to be moved to the "off" position. The principle of using a lightweight device to insert a link in a power mechanism to initiate an operation requiring considerable force is very common in weaving.

Electrical warp stop motions, which have been in use for many years, are to be found on most modern looms. Two bars are used as electrodes connected to a transformer and a magnetic *knock-off device*. When a yarn breaks, the wire drops and completes the electrical circuit. The current passing through the circuit operates the magnetic knock-off device which stops the loom.

The loom must stop very quickly before the lay moves forward to beat the filling, otherwise the weaver will have to reverse the loom for one complete revolution to remove the filling; this is necessary if the repaired warp end is to have the correct interlacing. The effectiveness of the braking system may be increased by disconnecting the clutch, so reducing the inertia of the system which is to be stopped.

Filling Stop Motions

Filling stop motions usually have a feeling device in the form of a *fork*. In some cases the fork is placed at the side of the reed and clear of the warp (*side filling fork*), and in other cases the fork is placed at the center of the warp (*center filling fork*). The side fork systems feel the filling every other pick, whereas center fork systems feel the filling every pick.

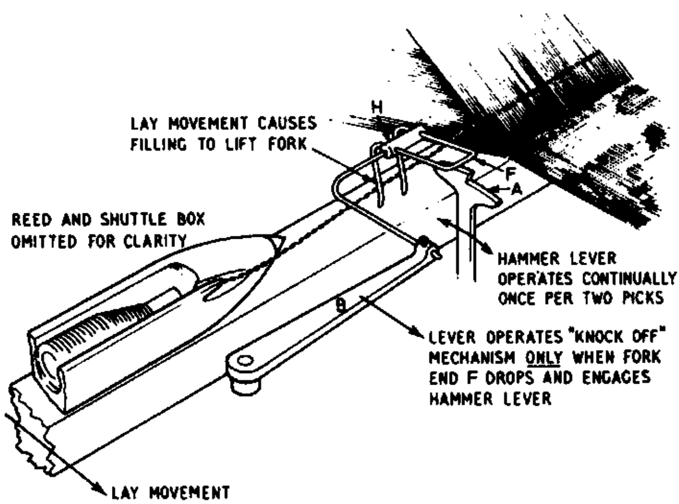

Fig. 13.9. Typical side filling fork mechanism

A typical filling fork mechanism is shown in Fig. 13.9. The forward movement of the reed during beating pushes the filling yarn forward. The yarn acts on the fork (*F*) which is hinged at the point (*H*) causing the bent end to be lifted, and this permits the loom to continue running. If the yarn is broken or the quill is exhausted, the fork stays with the bent end downward and this engages an oscillating *hammer lever* (*A*) which takes the fork lever (*B*) backwards. The fork lever then knocks the loom starting handle to the "off" position. The hammer lever gets its movement from a cam on the loom camshaft. This motion is robust and simple and its only disadvantage is its failure to feel every pick. One way of overcoming this is to use two side fork mechanisms, one at each side of the warp.

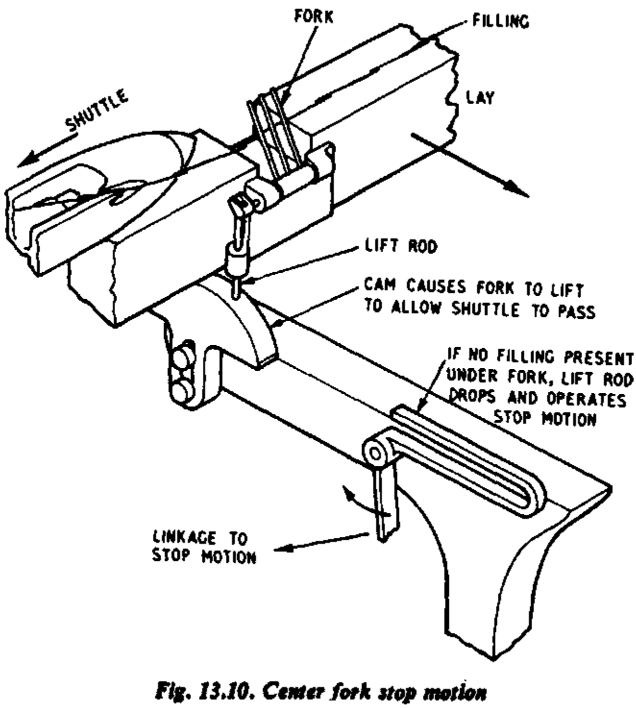

Fig. 13.10. Center fork stop motion

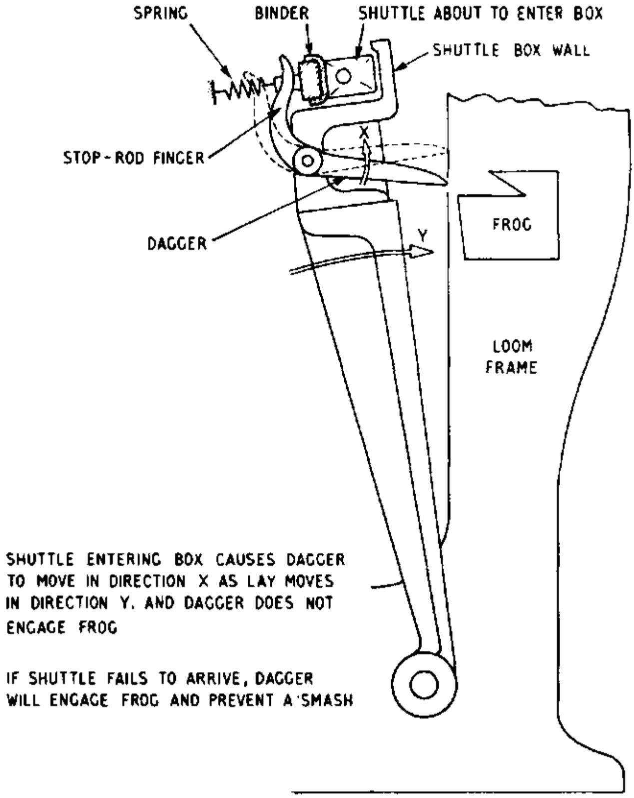

Fig. 13.11. Warp protector motion.

The center filling fork stop motion is more complicated and difficult to set. Several designs of center fork motions are available, but they are all basically the same. A *well* or channel is cut at the center of the *raceboard*. The fork is mounted on a bracket fixed to the front of the lay. Before the shuttle passes, the fork is raised clear of the shuttle. After the passage of the shuttle, the fork is lowered, and if the filling yarn is present the fork is prevented from dropping into the well and the loom is permitted to continue running. At á certain crank angle the fork must be withdrawn to allow the filling to be beaten into the fell of the cloth.

If no yarn is present, the fork drops quickly and operates the movement to stop the loom before the beating action is started. This simplifies repair and saves time. Figure 13.10 shows a typical system; in this case, if the filling yarn is not present, the dropping of the fork operates the knock-off device and the loom stops.

Warp Protector Motion

This motion is used on all conventional looms to stop the loom when the shuttle does not arrive at the shuttle box at the proper time. This action protects the warp for, if the loom continued to run, the reed would beat against the shuttle and cause damage to a large number of warp yarns as well as to the reed and shuttle (this is called a *smash*).

Two techniques are used. In the first, the reed can swing backwards when it meets a resistance greater than the normal reaction of beating-up (such looms are known as *loose reed looms*). In the second, a mechanism connected to the binders is used to knock off the loom when the shuttle fails to arrive at the proper time. The principle of operation of such a mechanism is shown in Fig. 13.11. When the shuttle arrives in the shuttle box, the binders move outwards by the impact of the shuttle. The outward movement of the binder produces a similar movement in the *stop rod finger* causing the *dagger* to be lifted to clear the *frog*, and the lay continues to move forward for beat-up. If the shuttle is late, the dagger comes in contact with the frog, which is moved; the movement is transmitted to the knock-off mechanism, which stops the loom. Contact with the frog results in a very severe deceleration to the system and the parts involved must be extremely rugged.

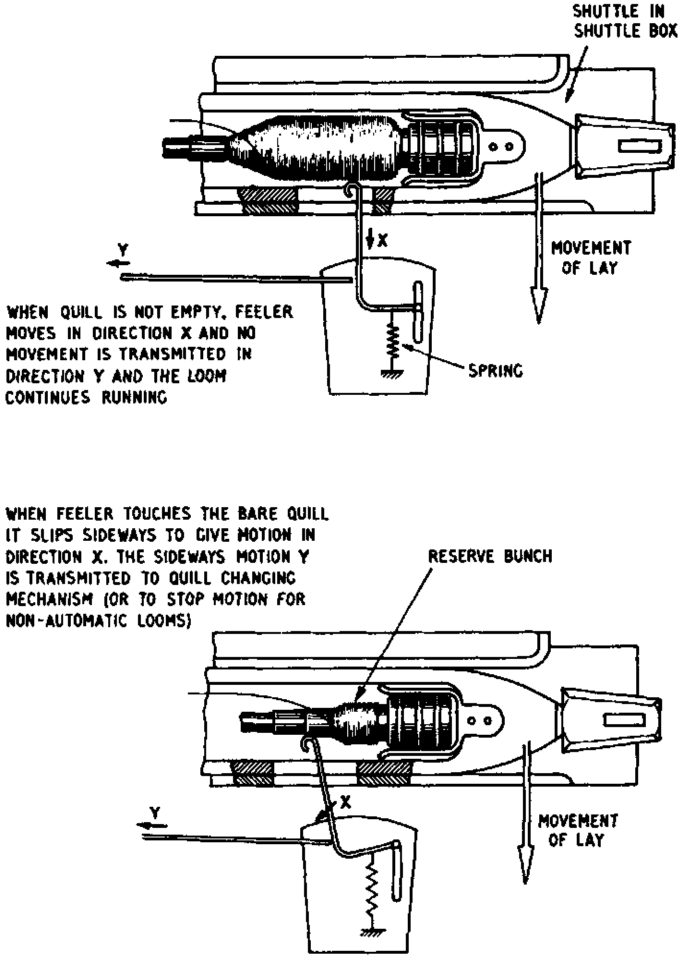

Fig. 13.12. Mechanical weft feeler

Filling Transfer Motions

Automatic filling supply is the principal feature of an automatic loom. Two main types of mechanism are used. In the first type, the whole shuttle is replaced. In the second type (which is now the most common) only the quill or bobbin is replaced; the exhausted quill is knocked out and replaced by a full one in the very short time that the shuttle is stationary in the shuttle box. Further discussion is limited to the *bobbin changer* type of transfer.

The mechanism is composed of three components; a feeling device, a transfer mechanism with associated linkage and a magazine or other quill storage.

The feeling device can be one of three types; mechanical, electrical or photo-electrical. The feeler detects the difference between the yarn surface and the base surface of the quill tube. It is usual to have a "*bunch*" of yarn on the quill not detected by the feeler and this allows sufficient yarn reserve to carry over between detection and transfer. Figure 13.12 shows the most commonly used mechanical feeler, the "*Midget*" *feeler*. When the filling yarn on the quill is down to the *reserve bunch*, the feeler blade comes in contact with the smooth surface of the quill and slides sideways rather than lengthways. This movement is transmitted to the transfer mechanism (this is fixed on the magazine side of the loom which is normally remote from the feeler).

In the case of electrical feeler, when the two probes merely touch the yarn, no transfer takes place; when the yarn runs down to the reserve bunch, the probes come in contact with the metal tube and complete an electric circuit. This passes current through an electromagnetic relay which initiates the quill transfer.

The photo-electrical feeler consists of a photocell and a lamp. The light beam incident on the quill does not pass through to the cell until the yarn on the quill is down to the reserve bunch. When this happens, the cell induces a voltage and current passes through an electric circuit which initiates the change.

In the case of the mechanical feeler, a linkage between the feeler and the transfer mechanism is needed to connect to the *magazine* or *battery*. This linkage usually consists of a shaft which rotates through a small angle to transmit the initiation of the change to the mechanism. Electrical and photo-electrical feelers do not need this type of linkage, since the initiation is transmitted by electric circuits.

The magazine can be of the rotary or the vertical stationary types. The first type is the most commonly used on single shuttle automatic looms, whereas the second type is more popular on multi-shuttle automatic looms. The magazine is normally mounted on one side of the loom in such a position as to be above the shuttle box when the lay is at front center.

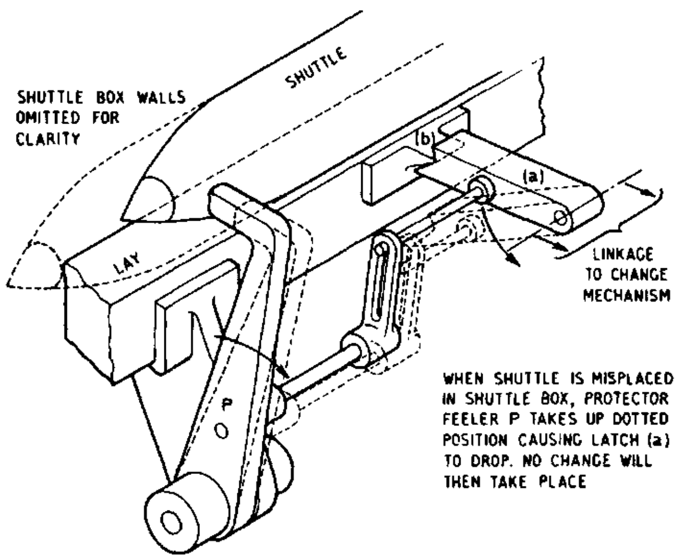

Fig. 13.13. Protector feeler for change mechanism

When the quill is exhausted, the feeler slips to one side and turns the shaft connecting the components of the mechanism. When the shuttle reaches the magazine side, a *protector feeler* moves forward to the mouth of the shuttle box. If the shuttle is not properly positioned, the protector is knocked away by the shuttle and the loom continues running without filling transfer. On the next pick the same procedure takes place, and if the shuttle is not properly positioned the same happens until the yarn is completely exhausted and the loom is stopped. If the shuttle is in the proper position, which is normally the case, the *transfer latch* (*a*) (Fig. 13.13) is then lifted and locked with the transfer hammer. As the lay moves forward to beat the filling, the *bunter* (*b*) engages the latch (*a*), turns the hammer about its fulcrum and presses a quill into the shuttle, pushing the empty quill out through the bottom of the shuttle. The backward movement of the lay returns all the parts to their normal position and this causes the magazine to rotate through an angle just enough to position a new quill under the hammer.

The movement of the shuttle from the magazine side partly threads the yarn, the threading being completed when the shuttle is picked from the other side. The end of the yarn from the old quill is cut by a cutter fixed at the temple and the end from the new quill is cut by the shuttle eye cutter.

In multi-shuttle looms, several shuttles are used, either for mixing of filling yarns to prevent barré or for the use of color. In the first case, where a 2 × 1 *box motion* is used to insert two picks from each shuttle; a rotary magazine is normally used. When color is introduced, a rotary multicolor magazine can be used, but it is preferable to use a stationary magazine. Every color of filling is stacked separately, as shown in Fig. 13.14. There are many designs for the filling transfer on multi-color magazines.

The main difference between multi- and single-color magazines is that a color selecting device, working in connection with the box motion, is used. Also provision is

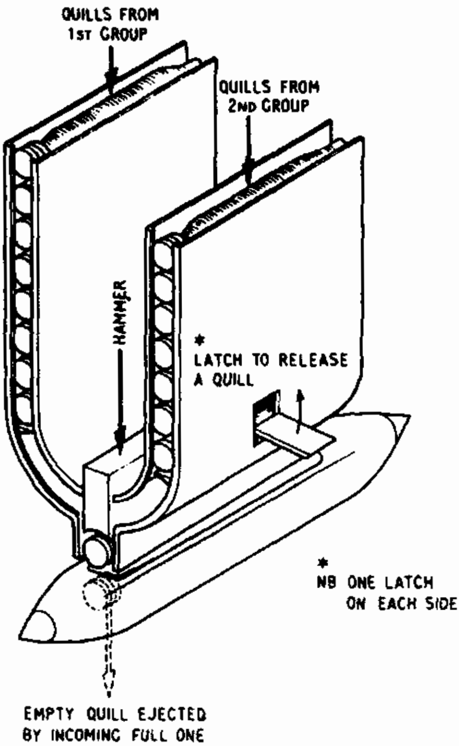

Fig. 13.14. U quill magazine

made to store the signal for transfer, if box changing takes place before the shuttle which has the empty quill is picked to the magazine side. In this case, the indication of the need for transfer is made by a feeler fixed at the magazine side.

In a weaving mill, the supply of quills to the magazine is normally done manually. This task involves the employment of labor to move the quills from winding rooms to the looms and keep supplying bobbins to magazines. This is a costly process, and it also makes the mixing of quills unavoidable. Two systems have been developed to minimize the labor cost and prevent mixing of quills. The *bobbin*

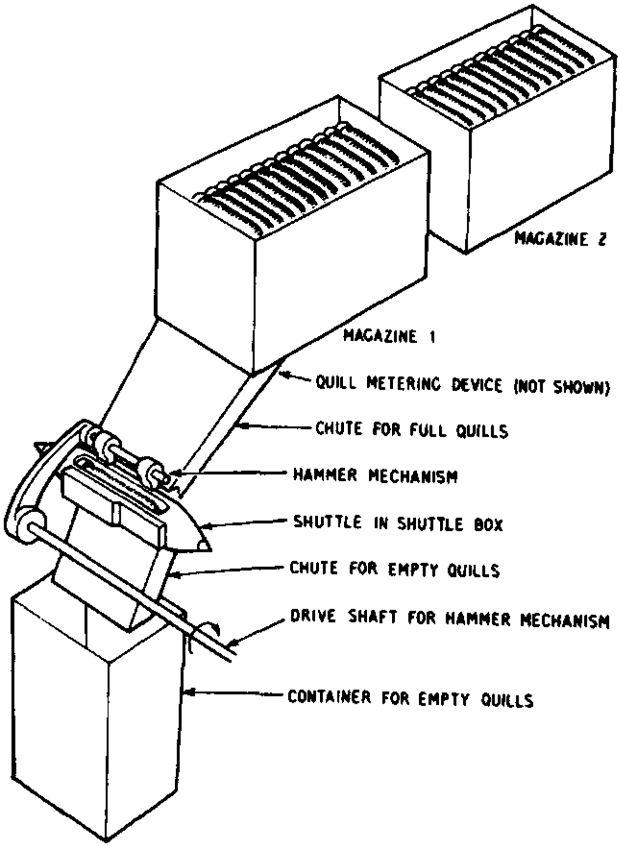

Fig. 13.15. Simplified bobbin loader system. (End finder, cutters and waste systems omitted for clarity)

loader uses containers to hold up to about 100 quills each and these are stacked during winding. Two containers are placed on the loom to act as magazines; one of these is active and the other is a reserve. When the first is exhausted, the second takes its place and a new full container is placed in position (see Fig. 13.15).

The second system which uses a winder at the loom, is known as the *Unifil*. In this case the filling yarn is supplied to the loom in the form of a large cone. The partly empty bobbins are stripped and returned to the winding position by a conveyor belt, as shown in Fig. 13.16. This system is advantageous in preventing barré caused by the mixing of quills, especially with filament yarns. Also, the system eliminates the need for quill winding machines and saves labor cost in transporting and supplying magazines. The use of Unifil also saves capital cost on quills since the number

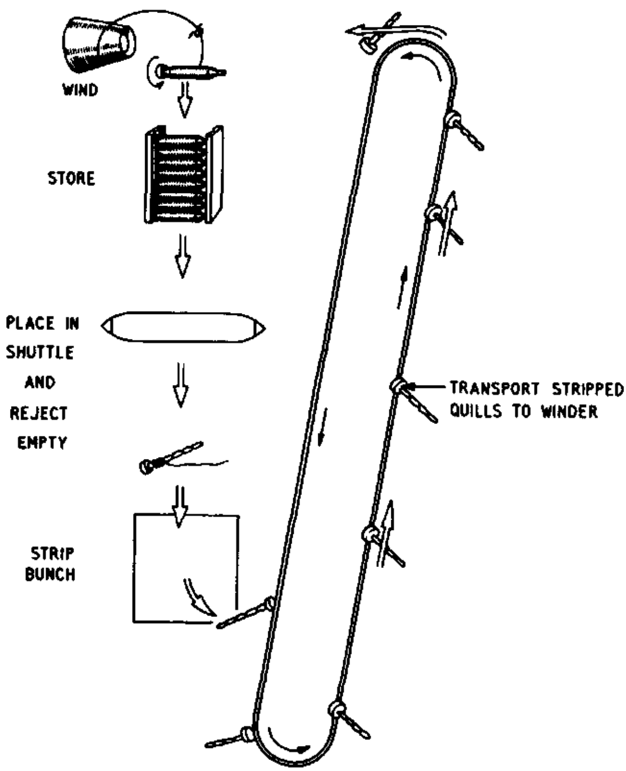

Fig. 13.16. Flow cycle for Unifil

of quills used is very much reduced. Unifil has a definite economic advantage in the case of coarse filling, because the rate of consuming quills is very high. On the other hand, it may not prove to be economical in the case of fine filling; the cost of the unit is considerable and it lies idle for a large proportion of the working time because the frequency of change of the quills is reduced. In many cases a compromise between cost and other advantages has to be made. The system is limited to single filling yarn color.

Multi-shuttle Looms

In the production of solid color fabrics, a single shuttle loom is normally used with one shuttle box on either side of the loom. With some fabrics, especially when filament yarn is used in the filling, two shuttles are used for the mixing of filling to prevent barré defects. In this case the loom must have at least two shuttle boxes on one side. If there is one box on the other side, the loom is usually denoted as a 2 × 1 loom. In this case every shuttle is used alternately for two picks.

If two or more colors are used in the filling, an equivalent number of shuttles must be used to give a series of looms with the appropriate number of shuttle boxes on one side. These are known as 2 × 1, 4 × 1, etc., looms. If multi-shuttle boxes are used on one side only, the maximum number of colors used is equal to the number of boxes on that side. In this case, the number of picks inserted from any color must be an even number, since the shuttle must be brought back to its box before any box changing can take place.

It is possible to have several shuttle boxes on each side of the loom and such looms are denoted as 4 × 4, 2 × 2, and 4 × 2 looms. They are sometimes used to permit the use of more colors and odd numbers of picks from each color; they are known as *pick-at-will* or *pick-and-pick* looms. The maximum number of colors used is equal to the number of boxes of the loom minus one.

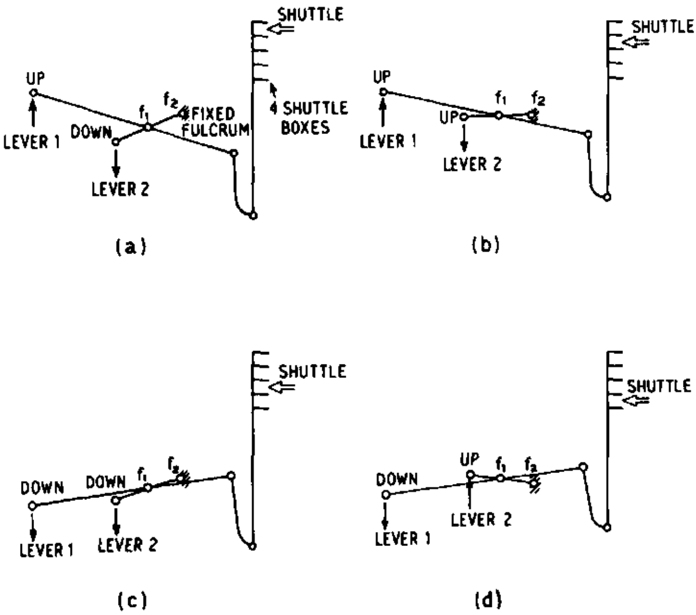

Fig. 13.17. Four-box motion in each of its positions

The most common mechanism used is the 4 × 1 box motion. This mechanism uses two levers to produce a compound movement at the box rod. This movement depends on which lever is moving and about which fulcrum the levers are moving. Lever 2 (Fig. 13.17) moves about the fixed fulcrum f_2 to move fulcrum f_1 on lever 1. The levers are usually moved by means of cams which are gear driven from the loom camshaft. When movement is required, clutches are engaged to cause the appropriate cam (or cams) to rotate sufficiently to cause the lever system to take up one of the configurations shown in Fig. 13.17 according to the demands of the pattern.

The control for selecting any particular box is initiated by a pattern chain which contains movable protuberances called *risers* and other links without protuberances called *sinkers.* A normal *box chain* has two rows, side by side, in which these risers can be fitted. There are four combinations in which the risers can be installed and these are translated into the appropriate lever settings by using a system of feelers and linkages. The combinations are:

	Row 1	*Row* 2
Combination 1	*R*	*S*
Combination 2	*R*	*R*
Combination 3	*S*	*R*
Combination 4	*S*	*S*

where R = riser and S = sinker.

To prevent these chains becoming unduly long where many picks are required in sequence from a single bobbin, it is usual to use *multiplier chains.* Consider first a single multiplier chain with one row of sinkers and risers similar to those used in a box chain. If there are x risers in succession on the multiplier chain, then the box chain is held stationary until these have passed (during which time $2x$ picks have been inserted). When the next sinker in the multiplier chain arrives, the box chain continues until it is again interrupted by a riser in the multiplier chain. It is usual to have two chains in the multiplier system which can be used in combination and this gives a range of multiplying possibilities. The two multiplier chains are both driven together (the pattern chains being stopped). One multiplier chain consists of all risers except one and the other chain consists of all sinkers except one. The pattern chains are only started when the odd riser in the one multiplier chain coincides with the odd sinker in the other. The multiplier chains are then stopped until a signal from the pattern chain causes the multiplier chains to restart and the pattern chains to stop.

If the multiplier chains each have different numbers of links, then they behave as a single long chain of $(N_1 \times N_2)$/H.C.F. as shown on page 262.

(N.B.: H.C.F. = highest common factor).

If chain no. 1 moves through z revolutions, and it contains N_1 links, then zN_1 pass during this time. If chain no. 2 contains N_2 links, then it moves zN_1/N_2 revolutions. Assuming that $N_1 = an_1$ and that $N_2 = an_2$, then

chain no. 2 moves through $\frac{z\,a\,n_1}{a\,n_2} = \frac{z\,n_1}{n_2}$ revolutions.

For proper registration, the odd riser has to coincide with the odd sinker, and starting from a proper registration, it is necessary for chain no. 2 to move a whole number of revolutions before registration can occur again. In other words, zn_1/n_2 must be a whole number. If n_1 and n_2 are prime numbers, then the requirement can only be met when $z = n_2$. In this case, $z\,N_1$ links pass but $z = n_2 = N_2/a$ and therefore a repeat occurs after every $(N_1 N_2/a)$ links pass. (Note: *a* is the H.C.F.). During the time these links pass, twice this number of picks are inserted. It is quite normal for the pattern lengths to be measured in picks and in this case the repeat is given by $[(M_1 M_2)/a]$ where M_1 and M_2 are the number of picks in each chain. The arrangement of colors in the shuttleboxes must satisfy two conditions; firstly, the color which appears most often should be placed in the top box, and secondly, skipping from box 1 to box 4 should be avoided because this puts more strain on the mechanism.

With all box motions, a safety device must be incorporated to prevent breakage of parts should the boxes jam. A *give-way* is normally built into the mechanism to allow relative movement between the parts in an emergency; this prevents further movement until the fault has been rectified.

14

POWER, ENERGY AND VIBRATION

Key words: *Alacrity, amplitude, back rest, beat effect, bottom shaft, buffer, capacitor, centroid, clutch, crank, damping, damping coefficient, damping pad, dynamic equivalence, dynamic magnifier, dwell, elasticity, electrical slip, equilibrium position, excited, fell, flywheel, forcing frequency, four bar chain, harmonic, induction motor, lay, lint, mass-elastic system, (mass) moment of inertia, natural frequency, offset, picker, power factor, radius of gyration. reed, resonance, rocking shaft, shuttlebox, simple harmonic motion, stiffness, structure-borne vibration, sword, synchronous speed, time constant, torque, torsion, vibration.*

The Loom as an Integrated Mechanism

The functions of a loom are interconnected and interrelated. Action at one place produces reactions elsewhere. These actions and reactions are correlated in the first part of this chapter.

Speed Variations

In a loom, it is necessary to use a great deal of reciprocating motion of heavy parts and these motions can involve considerable impulsive loading. Thus it is desirable to make the rotational parts heavy (to act as flywheels) and the reciprocating parts light. If the rotational parts are made too heavy, the loom will be slow to start and this is likely to cause cloth faults. These faults arise because the first few picks are not beaten and shedded in quite the same way as the rest. One solution to the problem is to insert a *clutch* between the loom and the drive. The motor may have a considerable *flywheel* attached to it because it does not matter much if the motor is slow in starting if it is not connected to the loom. Connecting the loom to a relatively massive motor/flywheel assembly causes the motor to slow down only slightly and the loom to come up to speed very rapidly. The situation is rather like the collision between two bodies as described in the section on checking in Chapter 11; instead of linear motion, the present case has rotary motion. The heavier the flywheel (strictly, the greater the *moment of inertia*) the more quickly will the loom be brought up to speed but also the greater will be the load on the clutch. The clutch limits the extent to which this can be carried, and the loom dictates how far it is desirable. A heavy duty loom requires more stabilization than a light silk loom.

Cyclic variations in loom speed are caused mostly by the reciprocating motions. Shedding has some effect and picking induces a sharp pulse once per pick, but the most important effect of all arises from the *lay* movement. The lay and all its

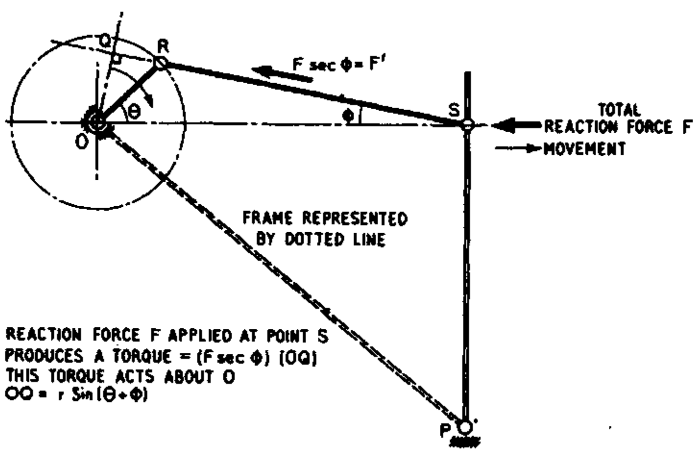

Fig. 14.1. Schematic diagram of crank/lay system

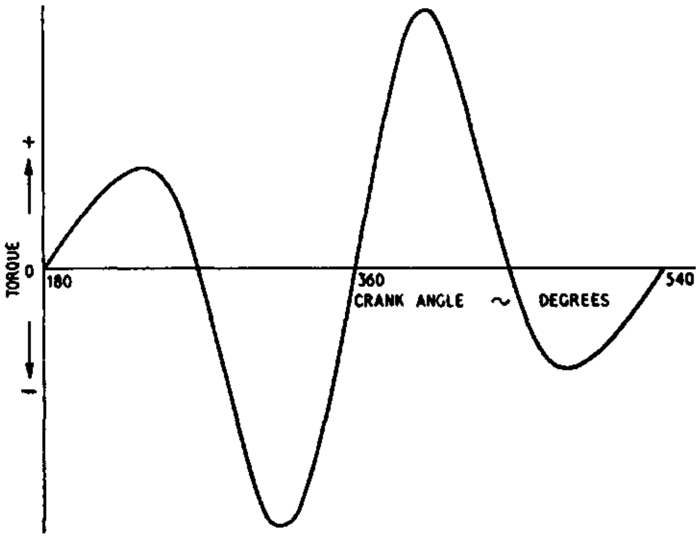

Fig. 14.2. Theoretical torque characteristic of lay drive mechanism. N.B. Bearing friction and windage ignored

associated parts may be regarded as a single large mass acting at the *reed*. This mass (M) is subject to the acceleration described by equation 11.6, i.e.,

$$f = \omega^2 r \left(\cos\theta + \frac{r}{l}\cos 2\theta\right) \qquad (14.1)$$

The force generated by this acceleration $(F) = M \times f$. As shown in Fig. 14.1, the torque needed at the crankshaft to produce this force $(T) = F^1 \times r \times \sin(\theta + \phi)$. The angle ϕ is small compared to θ; therefore, an approximate expression for the *torque* is as follows:

$$T = Fr \sin\theta$$

$$= Mfr \sin\theta$$

$$T = M\omega^2 r^2 \sin\theta \left(\cos\theta + \frac{r}{l}\cos 2\theta\right)$$

$$= \frac{M\omega^2 r^2}{2}\left(\sin 2\theta + \frac{r}{l}(\sin 3\theta + \sin\theta)\right) \qquad (14.2)$$

It will be noticed that the torque required is proportional to the equivalent mass of the lay and the square of the loom speed and *crank* radius. This means that the peak torque (which determines the motor size) depends on these parameters. A graph of eqn. 14.2 is given in Fig. 14.2. It will be seen that the torque required to drive the lay oscillates and is sometimes negative; this means that the lay might sometimes drive the motor by virtue of its own inertia. There is a strong double angle component (sin 2θ) which is referred to later.

When it is realized that the mass M may be as much as 200 kg (400 lb) and the maximum acceleration may be up to about 3 g, it will be seen that the force transmitted by

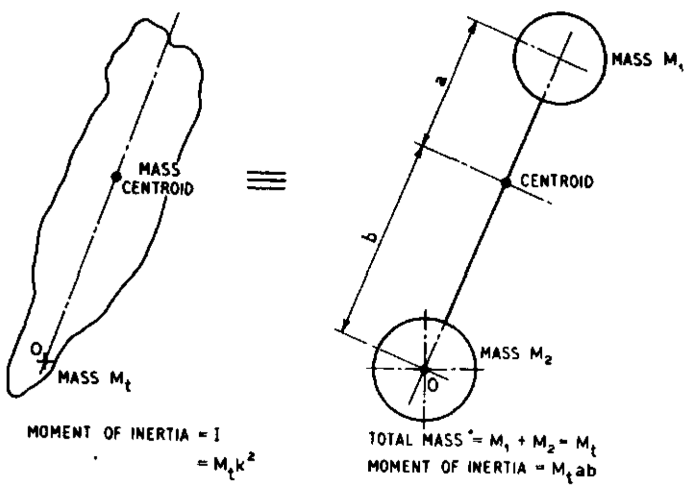

Fig. 14.3. Dynamic equivalents

the connecting rods could be up to 600 kg (i.e. roughly ½ ton). This emphasizes the importance of the lay mass.

In reality, the mass M is not a single mass concentrated at a point, but is distributed over the lay, *sword* and associated parts. The concept is, however, very useful (providing the nature of the simplification is fully realized). A complex component such as the lay assembly can be represented by the theoretical model shown in Fig. 14.3. To be *dynamically equivalent*, the following criteria have to be satisfied.

(a) The total mass of each must be the same.
(b) The first moment about a given external point must be the same for each.
(c) The second moment about a given external point must be the same for each.

In symbols, $M_1 + M_2 = M_t$, $M_1 a = M_2 b$,

and $$M_1 a^2 + M_2 b^2 = M_t k^2$$

(where k is known as the *radius of gyration* and I is known as the *moment of inertia).* The latter is a measure of the difficulty of imposing an angular acceleration to the assembly; it is roughly comparable to mass in the case of straight line movement. If the axis of swing is changed, so is the moment of inertia; therfore, the value of I only has meaning when the axis is defined; in this case, I is referred to its *centroid.*

The above conditions can only be met when $k^2 = ab$. For our purposes, we require only one mass in operation and this can be achieved by putting the other one at the pivot point. In this way the effective mass M_1 is situated at distance $(a + b)$ from the pivot and the centroid at distance b. Under these circumstances,

$$M_1 = \frac{b}{(a + b)} M_t \tag{14.3}$$

If the centroid of the assembly is near the lay, M_1 will be little different from M_t (which is the actual mass of the assembly).

On the other hand, if the mass is concentrated nearer the pivot, it has much less effect. Thus there is every incentive to reduce the mass of those parts which are remote from the pivot axis (*rocking shaft* axis). Consequently, care must be taken to keep the masses of the lay, *shuttleboxes* and other

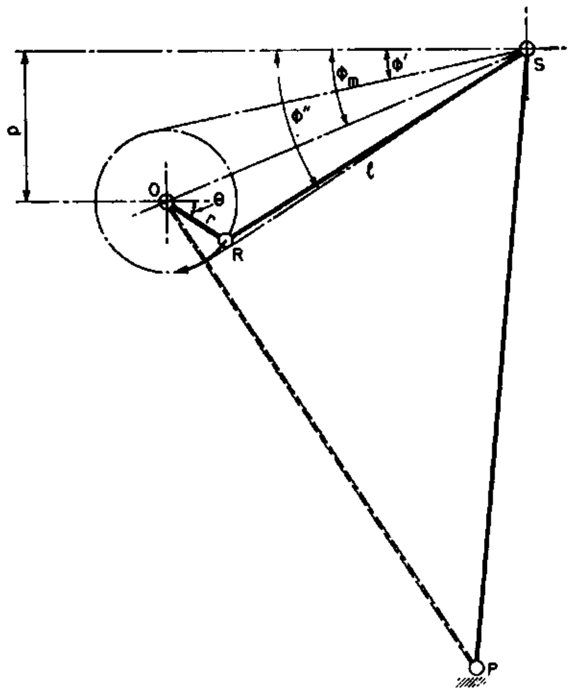

Fig. 14.4. Four bar chain with offset representing a lay mechanism

items at a similar radius, to a minimum. This is one reason why the lay is usually made of wood, which is light and can provide good stiffness without undue weight. It also provides a good running surface for the shuttle. It is apparent that the use of multi-shuttle boxes can be a disadvantage in this respect. With shuttleless looms, it is possible to make the whole mechanism lighter with the result that higher speeds can be used.

In the foregoing analysis, it was assumed for simplicity that the lay moves in a straight line. In actual fact the mechanism is really a *four bar chain*, as shown in Fig. 14.4, but because of the radius of the sword, straight line motion is a very good approximation. Furthermore, the line of movement of the lay does not have to pass through the center of the crank. An *offset* tends to distort the displacement curve and to produce a more pronounced *dwell* and, therefore, it is often used to enable the shuttle to traverse the loom without interference. It can be shown that the following equation holds approximately when p/l is small, i.e. when the offset is small.

$$\text{Acceleration} = d^2x/dt^2 = \omega^2 r\left(\cos\theta + \frac{r\cos 2\theta}{l\cos\phi}\right) \quad (14.4)$$

The symbols are defined in Fig. 14.4. Equation (14.4) may be compared to eqn. (14.1). It will be seen that the double angle component is still present. Since an increase in offset causes an increase in the mean value of ϕ, then it will further be seen that the offset tends to increase the double angle effect and thereby make the dwell more pronounced.

Power

An electric motor cannot deliver a varying torque without speed variation and a normal relationship is as indicated in Fig. 14.5. A motor is not 100 per cent efficient and some energy is dissipated; this energy appears as heat and is a function of the *electrical slip*. An induction motor, such as is used on a normal loom, works at a speed lower than the *synchronous speed* set by the a.c. electrical supply. The difference between the synchronous and actual speeds expressed as a proportion of the synchronous speed is called electrical slip. As the slip in such a motor is increased, the amount of heat generated in the motor increases; thus,

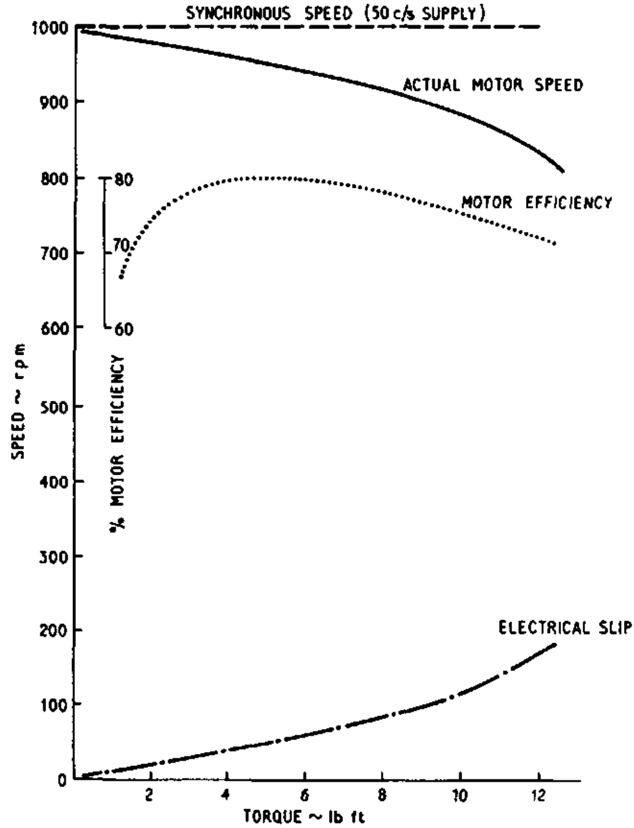

Fig. 14.5. Characteristic curves for a typical loom motor

running at high slip leads to overheating and an inefficiency unless the motor is specially designed. For electrical reasons, it is required that the motor should be kept below synchronous speed; therefore, where there are large swings in speed (as there are in a loom), the average slip has to be rather large. Hence it is necessary to use specially designed motors to drive a loom. Another reason for requiring special motors is that the *lint* from weaving is very easily ignited and the motors should be flameproof to prevent fires.

Since the swings in speed are a function of the inertias involved, a flywheel will lessen them; in extreme cases, however, it is desirable to use a special motor to meet the situation. This might have some advantage in that a smaller slip would be needed and the efficiency of the motor could be increased.

The loom motor works at a higher electrical slip than most other motors and this causes the *power factor* to be poor. The power factor is the ratio of power (in watts) to the mathematical product (volts × amps). It represents a sort of efficiency to the supplier of the electricity and, generally, a cost penalty is imposed for operating at a poor power factor. It is usually worthwhile to apply correction and this is most often done by installing electrical *capacitors* in parallel with sets of motors.

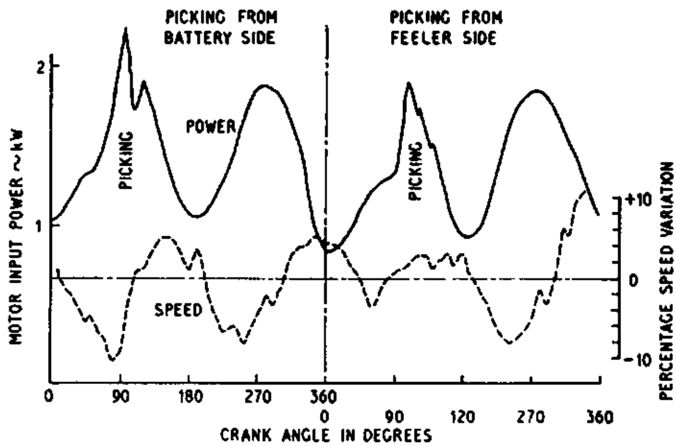

Fig. 14.6. The power and speed characteristics of a typical loom. Adapted from a Ph.D. thesis by M.H.M. Mohamed, University of Manchester, 1965

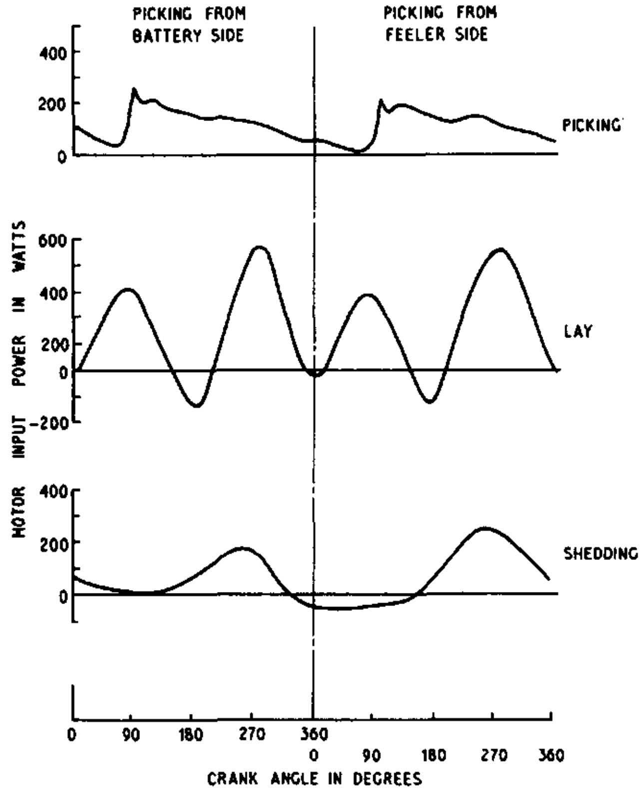

Fig. 14.7. Component power requirements. Adapted from a Ph.D. thesis by M.H.M. Mohamed, University of Manchester, 1965

The demands on the motor vary cyclically, as has been explained, and this results in a variation in power as indicated in Fig. 14.6. The motor, power lines, and switch gear must be able to cope with the peak currents rather than the average; also, a poor power factor yields greater current for a given power and thus all the electrical components have to be oversize by normal standards.

The power needs vary from instant to instant. Power and speed are related, as can be seen from Fig. 14.6. When the speed rises, the power absorbed declines; when the power demand rises, the speed drops. Referring to Fig. 14.2, it will be noticed that there is a strong similarity between the theoretical torque and the actual power demand; the double angle component is particularly noticeable, which indicates the importance of the lay mass.

In the past, attempts have been made to assess the power needed for each of the constituent mechanisms of the loom, but because of the interactions, it is not possible to determine these by progressive disconnection of various components without loss in accuracy. Unfortunately, it is difficult to measure the component behavior in any other way. The results by such disconnections and by difference between components have been used to obtain the curves given in Fig. 14.7. These should only be regarded as approximate; some other data are given in Table 14.1. If the component curves were added together, they would not produce the curve for the loom running with all components in use. This is because the behavior of both the motor and the loom depends upon the speed at the particular moment, as well as the accelerations developed by the other components. As can be seen from Fig. 14.6, picking has an effect on the battery side which is different from that on the other side. This is partly explained by the fact that the speed at the instant of picking is different in the two cases. The results referred to in this section apply to a Picanol President loom of 2.1 m (85 inch) width and are, therefore, particular in nature, but the pattern is somewhat similar to that found in other looms and may be taken as being reasonably typical of most.

Examining the component curves in more detail, it will be observed that the effect of beating and lay movement are the most important as far as power is concerned. The strong double angle effect arising from the sin 2θ term given in eqn. (14.2) is clearly evident. The deformation of the curve arising from beat-up can be detected, but to emphasize the point another set of curves (Fig. 14.8) is also given. It becomes increasingly difficult to beat up as the filling is forced into the fell of the cloth; also, a larger diameter filling (i.e. a coarser count) tends to make beating more difficult. This is also seen in Fig. 14.8; in one case, weaving was fairly normal and in the other a bumping condition existed.

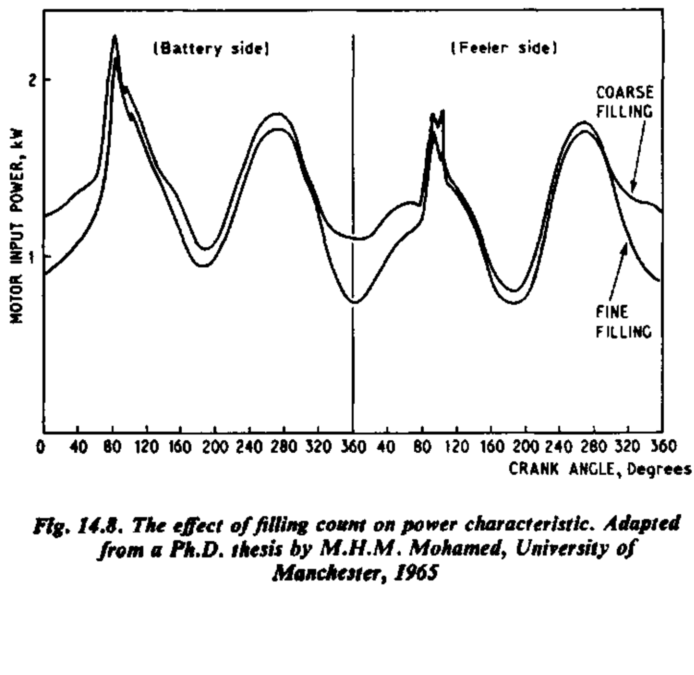

Fig. 14.8. The effect of filling count on power characteristic. Adapted from a Ph.D. thesis by M.H.M. Mohamed, University of Manchester, 1965

The power curve related to shedding is given in Fig. 14.7 and the peaky nature of the curve will be noted. This is because the cam profiles are shaped to give a rapid shed change once the shuttle has passed across the warp shed. The *amplitude* and shape of the power curve relating to this component are a function of the loom speed and the shape of the cam used for shedding.

Figure 14.7 shows the power required for picking. As explained earlier, picking needs only a short pulse of energy which might last only about 25 crankshaft degrees. In fact the sudden and rather large demand for energy causes the loom speed to drop and the character of the loom in combination with the motor determines how long it will take for the system to recover. In the case shown, the recovery takes almost 360°; if it had taken longer there would have been interference between one pick and the next (which could have led to instability). Any attempt to overspeed a loom can lead to this sort of difficulty, as can a large change in inertia.

The power required for picking is proportional to the cube of speed. If the shuttle velocity is taken to be proportional to the loom speed, then the kinetic energy required per pick is proportional to (loom speed)^2. The rate of using energy (i.e. the power) is proportional to (loom speed)^2 × (picks/minute) which in turn is proportional to (loom speed)^3. The bearing friction and windage losses increase as the square of the loom speed; thus the total power requirement of a loom is roughly proportional to (loom speed)^z, where z is between 2 and 3. The heavier the shuttle, and the more massive the loom, the nearer is the index z to 3·0. Conversely, by reducing the masses involved, it is possible to reduce the power required for a given loom speed and this can be translated in terms of cost.

Vibration (A Review of Theory)

Vibration occurs in a variety of ways (see Fig. 14.9). Longitudinal vibration is unimportant as far as this discussion is concerned and will be ignored.

Let the actual mass of the vibrating element be typified by an equivalent mass (M) situated at the points shown in the diagram. Let the stiffness of the system be defined as the force or torque needed to move that point by one unit. These parameters determine the *natural frequency* of the system and the system will tend to vibrate at that frequency if suitably excited.

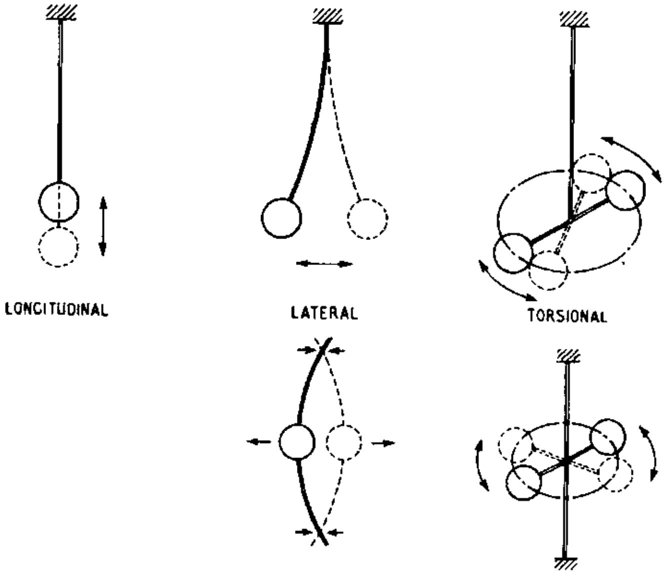

Fig. 14.9. Some forms of vibration

Consider the case of lateral deflection. Let

M = the equivalent mass (not weight),
s = *stiffness* of the vibrating member in the direction of movement,
= force required to produce unit displacement in the direction of movement,
n = natural frequency of vibration in cycle/second (Hertz),
x = displacement of the mass from its equilibrium position,
α = acceleration of the mass.

When the mass M is displaced from its *equilibrium position* by a distance x, the force needed to do this is $s \times x$ and the *elasticity* of the system tends to cause the mass to be returned to its original position. If the member is vibrating, this is still true and at the instant the mass is at distance x from the equilibrium point, there is a force equal to sx acting. This force can also be expressed in terms of the acceleration, i.e. force $= -M\alpha$. Hence

$$\frac{\text{acceleration}}{\text{displacement}} = -\frac{s}{M} = \text{constant}$$

This is *simple harmonic motion* and an exact mathematical solution to this arises when

$$\text{displacement} = x = A \sin \omega t \qquad (14.5)$$

where A = *amplitude*,

$\omega = 2\pi n$,

t = time in seconds.

$$\text{acceleration} = \frac{d^2x}{dt^2} = -A\omega^2 \sin \omega t = \alpha$$

therefore $$\frac{\alpha}{x} = -\omega^2$$

and $$\omega^2 = \frac{s}{M}$$

or $$n = \frac{1}{2\pi}\sqrt{\frac{s}{M}} \qquad (14.6)$$

In the *torsional* case, a very similar situation exists; if M is replaced by the *mass moment of inertia* (I) and the stiffness by the torsional stiffness (q):

$$\frac{\text{torsional acceleration}}{\text{torsional displacement}} = \frac{-q}{I} = \text{constant}$$

and $$n = \frac{1}{2\pi}\sqrt{\frac{q}{I}} \qquad (14.7)$$

In the case of eqn. (14.6), s and M must be expressed in consistent units. For instance, mass M can be expressed as

$$\frac{\text{weight}}{\text{gravitational acceleration}} = \frac{W}{g}$$

if W is in pounds and $g = 32$ ft/sec^2, then s must be expressed in lb/ft to give the natural frequency in Hertz (cycles/sec). In the metric system the mass is expressed in grams or millinewtons (mN), but the stiffness S must be expressed in gf/cm or mN/m (where gf and mN are measurements of force).

Similarly in eqn. (14.7) q and I must be in consistent units. (I is the mass moment of inertia and $I = Mk^2$) The mass moment of inertia (I) is

$$Mk^2 = \frac{W}{g} k^2$$

where k is the radius of gyration measured in meters or ft [Note: in the latter case I has the units lb ft sec^2]. In the *SI* system of measurements (metric), the torsional stiffness may be expressed in mN m/radian but in the Imperial system one would use lb ft/radian and in either case the natural frequency is expressed in Hertz (cycles/sec).

The value of I depends not only upon the mass, but also upon the position of the mass with respect to the axis of rotation. If the mass is concentrated in a rim, we have a very effective flywheel; the greater the radius of the rim, the more effective it is and, with a given torsional stiffness of spring, the lower will be its natural frequency.

In any machine, there are successive disturbances which occur at regular intervals. Each of these disturbances can set up one or more vibrations which continue after succeeding disturbances arrive and pass. The later disturbances might augment the previous ones or not if they do on a regular basis, the vibration might build up to dangerous proportions. In other words, when the *forcing* and natural frequencies coincide, the system *resonates* and the condition is known as *resonance*. The consequences of this are widespread and it is necessary to discuss it in more detail before discussing the part it plays in a loom.

A vibrating mass will not continue to vibrate for ever; the rapidity with which the amplitude declines depends on the nature of the material. Some materials absorb substantial amounts of irrecoverable energy when deformed (which appears as heat) and such materials do not vibrate readily. Other materials absorb little energy in this way and these materials resonate easily. Examples of the two classes are cast iron and spring steel, respectively. This is one good reason why a loom frame is made of cast iron, i.e. because the iron damps the vibrations.

Under conditions of *damping*, the equation of motion is modified to

$$x = Ae^{-t/\tau} \cos mt \tag{14.8}$$

where A = amplitude,

$e = 2{\cdot}718$,

t = time in seconds,

τ = time constant,

$= 2M/\delta$,

δ = damping coefficient,

= damping force/unit velocity,

$$m^2 = \omega^2 - \frac{1}{\tau^2}.$$

The *time constant* (τ) expresses the decay characteristic of the vibration in much the same way as the half life expresses the decay of a radioactive source. It is a function of the damping coefficient (δ). A further point to note is that the frequency of vibration (m) is a little less than the natural frequency (ω) described earlier; in other words, damping reduces the amplitude of vibration and slightly affects the frequency too. These facts are important in resonant systems because the amount by which the amplitude decays between one pulse and the next determines how much the resonance can build up. With little decay, it can build up to destructive proportions.

If a force of $F \cos \omega_f t$ is applied to the system, this will be balanced by the inertia, damping and elastic forces in the material. In symbols,

$$M \frac{d^2x}{dt^2} + \delta \frac{dx}{dt} + sx = F \cos \omega_f t$$

An approximate solution to this differential equation is

$$x = C \cos (\omega_f t - \beta) \qquad (14.9)$$

where β is a phase angle which need not concern us here: suffice it to say that since $\cos (\omega_f t - \beta)$ can never exceed $\pm 1{\cdot}0$, then the maximum value of x is C. It can be shown that

$$\left(\frac{C}{\Delta}\right)^2 = \frac{1}{(1 - (\omega_f/\omega)^2)^2 + ((\delta/M)(\omega_f/\omega))^2} \qquad (14.10)$$

where Δ = the deflection the part suffers due to its own weight under static conditions,
C/Δ = the *dynamic magnifier.*

When the forcing frequency (ω_f) is the same as the natural frequency (ω),

$$(C/\Delta)_r = M/\delta \qquad (14.11)$$

where the suffix r refers to resonance.

Thus if there were no damping, the dynamic magnifier would be infinite, which is clearly impossible. The physical reason for this is that all materials damp to some extent, no matter how small. However, the dynamic magnifier can be very large indeed and, when it is, the structure can vibrate very violently at resonance, even to the extent of self destruction.

In a very complex vibrational system such as a loom, each of the many components possesses a set of natural frequencies. Each of the motions produces a whole series of frequencies and their *harmonics*. Those natural frequencies which correspond with a forcing frequency or their harmonics will be accentuated; this is rather like a panel in a car which vibrates only at a given speed. Therefore, the structure has to be considered as well as the sources of vibration.

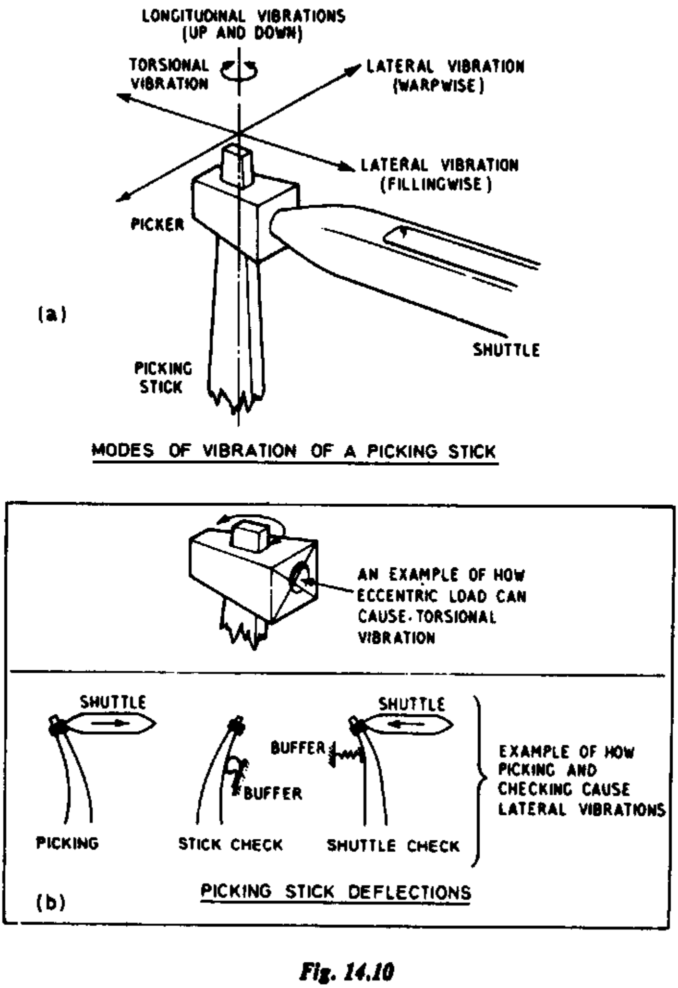

Fig. 14.10

The picking of the shuttle creates a very sharp pulse which occurs at regular intervals and these trigger a whole spectrum of vibrations throughout the loom. The picking action of a normal shuttle loom depends upon the *deflection* of the picking stick, and the *alacrity* of the system is very important. This is only another way of saying that the picking system is a vibratory system which can be explained by eqn. (14.9). As was stated in Chapter 11, the natural frequency of the system is extremely important to the proper working of the loom. Looking deeper into the subject as we are now able to do, it will become apparent that the damping character of the material of the system is also important. If the picking stick were to have a very high damping coefficient, it would not work properly; if it were to have too low a damping coefficient it might continue to vibrate during the next pick and cause difficulty. It is necessary to use some material such as wood (preferably laminated) which has good elastic properties and a suitable damping characteristic.

A picking stick can vibrate in a variety of ways (see Fig. 14.10). For example, if the picker is not set correctly, both picking and checking will induce torsional and lateral vibrations which might carry over from one pick to another. The buffer used to check the stick after picking can cause similar effects. An adverse attitude of the picker when it starts to accelerate the shuttle can cause the shuttle to be deflected from its proper path. This can create difficulties as the shuttle enters the shuttle box on the other side. An incorrect entry can impose very high stresses on the shuttle; it can cause vibrations of the quill within the shuttle which can lead to faulty unwinding at a later stage. The entry of the shuttle also affects the way it is checked and this in turn affects the following pick. The irregularity of picking caused by such disturbances tends to be cyclic over several picks and although the mechanisms are complex, it still remains a fact that this is another sort of instability related to resonance.

Not only is the picking stick excited, but so are other elements in the system; for example, the *bottom shaft* can be set into violent torsional and flexural vibrations (Fig. 14.11) each of which have their own set of natural frequencies. To simplify matters, consider only one of these, say the torsional case. The system (which is usually poorly damped) vibrates at its natural frequency because of the blow it received from a given pick, but the frequency of exciting disturbances is related to the loom speed (which varies somewhat from pick to pick) and not to the natural frequency. Consequently, there is a random *beat effect* where the excitation sometimes augments the vibration and sometimes opposes it. This sort of thing seriously affects the stability of the running loom; sometimes there are strong picks and sometimes there are weak ones. Usually the strong picks are late and the

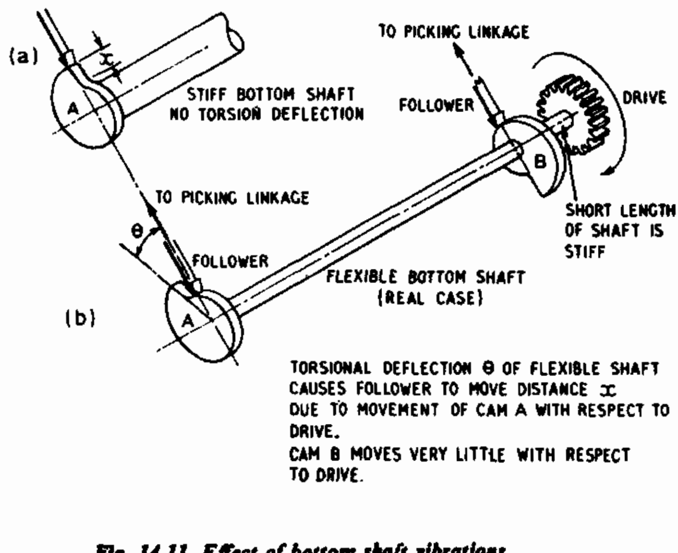

Fig. 14.11. Effect of bottom shaft vibrations

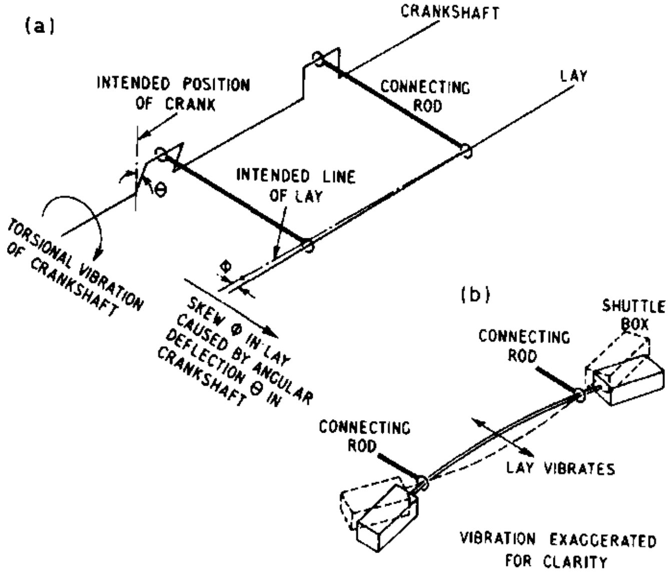

Fig. 14.12. Effects of vibration of crankshaft and lay

weak picks are early; this is because the vibration causes the cam to be displaced from its nominal position and it is this displacement that produces the extra energy (or the lack of it) to give the stronger (or weaker) pick. This leads to a most undesirable variation in speed and performance which tends to depress the acceptable running speed of the loom and thereby reduce its effectiveness.

When there is a strong pick, the motor slows down more than when there is a weak one; the loom also takes longer to recover from the strong pick. Thus these variations also affect the other motions; for example, a strong pick can be followed by a weak beat-up. If this is marked, the effect will show in the fabric and this is of some importance.

The lay beats up the filling and, in so doing, it suffers considerable force. To keep the fell of the cloth straight, the lay has to be rigid so that it does not deflect unduly. There is also another reason. The sort of shocks already discussed could produce a vibration in the lay, as shown in Fig. 14.12, and this would not be related to the loom motion. The vibrating lay could beat one part of a filling strongly and another not so strongly. At the next pick, the position could be reversed or there could be some other more complicated pattern of beating over the area of the cloth. Thus the lay should vibrate little, and one way to ensure this is to make it so stiff that its natural frequency is very high, in which case it is unlikely to vibrate strongly because all materials damp more readily at high frequencies.

In a multi-shuttle loom, the lay has heavy shuttle boxes at each end with considerable overhang in respect to the connecting rod pivots. This could lead to a heavy low-frequency vibration in the lay which would be very difficult to suppress. One solution to this is to use auxiliary cranks and connecting rods to support the extra masses during

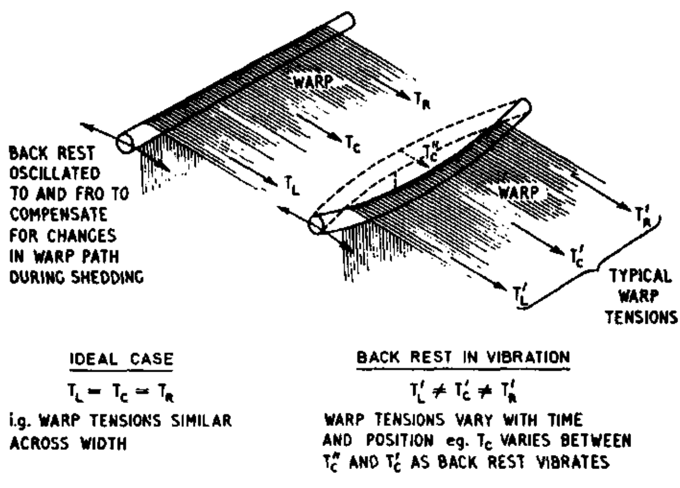

Fig. 14.13. Effect of back rest vibration

beating. If the crankshaft is set into vibration, by whatever means, this too can affect the beating. If one crank is in advance of the other (or others) the reed will be skewed (or bent) as indicated in Fig. 14.12 and this will give the beating irregularities previously described. Also, where a four crank loom is concerned, the vibration of the crankshaft can cause bearing trouble because of maldistribution of load and temporary bearing misalignment. A similar and more serious effect can be met with the bottom shaft; here, the bending and torsion is often quite violent and it has been known for shafts to be broken even though they may be solid steel bars of perhaps 5 cm (2 inch) diameter. Even if the difficulties mentioned do not result in immediate failure, there is always the possibility of an accelerated wear rate of the bearings.

The *back rest* in a loom is often oscillated deliberately to preserve, as nearly as possible, a constant warp tension. This movement not only adjusts the warp length to give the desired tension control but unfortunately it also introduces some disturbing forces. This and other excitations (such as from picking) cause the back rest to vibrate along its length, as shown in Fig. 14.13; this gives unevenness in warp tension across the width of the warp and the pattern is everchanging (because of the nature of the vibration). If large enough, this can produce patterning in the fabric, especially with fine synthetic materials.

A well-designed loom is made in such a way as to overcome most of these difficulties and they have been discussed in order that the design of the loom may be understood. A proper understanding of loom operation also helps in avoiding difficulties that may arise through unwise alterations or additions to the design of the loom.

The loom as a whole can vibrate on a springy floor, and it is often surprisingly difficult to find a suitably rigid floor; even "solid" earth is capable of acting as a spring in this respect. Thus we have a *mass-elastic system* in which the loom (or a set of looms) acts as the mass and the floor and the surrounding structure acts as the spring. (The surrounding structure is very important; *structure borne noise* and vibration from a loom can appear in parts of a building quite remote from the weave room.)

If a loom is mounted on pads to absorb the energy of vibration, it is necessary that there should be movement of the loom to permit the *damping pads* to work. Such movement means that, at best, only part of the vibration can be removed. Furthermore, the damping is a function of the velocity of movement; therefore, the higher frequency components will be damped out quite well but the lower frequency ones will not. The amount by which the loom can be allowed to move on its mountings must be limited, because an undue excursion of the frame whilst the shuttle is in flight could cause all sorts of trouble. Thus pads can do no more than give some relief from the problem; they cannot effect a cure. In general terms, they muffle the noise a little.

If a loom is mounted on flexible mounts, those vibrations arising within the loom which are considerably above the natural frequency of the mounts will be attenuated. This means that the level of vibration in the floor will be reduced for the given frequencies. If the forcing frequency is the same as that for the mounts, the assembly will rock violently because it will be at resonance. Taking into account the mass of the loom, it may be possible to obtain mounts whose natural frequency when installed is (say) 2 cycles/sec. The lowest frequency (in cycles/sec) of any great magnitude generated by a loom is that of the bottom shaft ($\frac{1}{2}$ × picks/min ÷ 60). This means that a normal shuttle loom has a spectrum of frequencies from about 1 cycle/sec upwards. Thus there is a very good chance of getting into disastrous resonance and, even if this is avoided, there will be little attenuation of the low frequency vibrations, However, with a shuttleless loom, the problem is not so severe. In those looms which have a picking mechanism which produces little or no external reaction, the half speed component is negligible and the major component is the double speed one from the lay motion. The frequency in cycles/sec, in this case, is 2 × (picks/min) ÷ 60. Bearing in mind that these looms run faster than conventional ones, the major component might well be at some 10 cycles/sec and such a component could be attenuated by flexible mounts. Even so, the looms would not be rigidly anchored and would tend to flop about; this could lead to operational difficulty. Another factor is the lack of the stiffness which a loom normally acquires by being secured (either by its own weight or by securing devices); this means that the frame of the loom can more easily vibrate. The gains obtained by using pads or flexible mounts or both are reduced by this fact. Extra stiffening without extra mass is needed in such cases.

The foregoing indicates the difficulties which are involved in reducing the vibration levels in a shuttle loom and it is quite clear that the shuttle has much to answer for in this respect. If noise becomes an over-riding factor, there will be a strong incentive to change over to shuttleless looms, which can meet legal specifications for the maximum permitted noise and vibration. This is apart from the other merits that these looms might have.