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# Recent innovations in loom shedding mechanisms

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The critical analysis of the recent research developments in the loom shedding mechanisms has been reported. The use of angle shedding disks without dynamic warp loading in multi-phase weaving machines enables to weave higher pick densities and difficult varieties of fabrics. Other researches have been directed towards more effective heald movements and better design of shedding cams. The paper also highlights the various developments related to these areas and provides scope for further research in the shedding mechanisms.

**Keywords**: Cycloidal, Loom, Microprocessor, Pressure angle, Shedding mechanism

## 1 Introduction

During the recent years, considerable research has been done on loom shedding mechanisms. Shedding systems have been suitably modified so that it has now become possible to weave fabrics with varied profiles to suit specific technical applications[1-6]. Different areas of research have been explored. In one of the areas, shedding has been done without dynamic warp loading. This has opened the possibilities of weaving the new textile structures which are difficult to weave on conventional machines. Cams have been designed that deviate from the simple harmonic motion and the results on weaving performance are found to be comparable to the conventional cams. Newer methods of regulating heald frame motions have been developed. In another interesting development, MEMS has been used to investigate the warp breakage rate and this is found to be useful for a wide range of weaves[7-9]. Kinematic study has been done on rotary dobby and it has been found that eccentric mechanism has a significant effect on heald frame motion[10-12]. Dobby with microprocessor has been developed. This paper critically analyses the different research attempts made in this area and highlights their merits and limitations over existing shedding systems.

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## 2 Requirement of New Systems

### 2.1 Drawbacks with Conventional Shedding Mechanism

In conventional shedding mechanism, the warp yarns are deformed compulsively. As the force causing the deformation tends to act perpendicular to the direction of the warp yarns, it raises the warp tension. Also the warp yarns are made to pass over parts of the loom which make the warp to move transversely, creating friction and shear on the moving warp yarns. Moreover, the shedding has a particular duration, which determines and restricts the operation of other important loom mechanisms. All these movements result in a further increase in the dynamic loading including the warp. Other disadvantages are destruction of warp and jamming of the threads due to the presence of thick places and other irregularities. This makes it difficult to weave fancy yarns. It is thus evident that the conventional loom construction restricts the choice of the fabric manufacture and hence there was need to develop new systems.

### 2.2 Development of New Systems – Shedding without Dynamic Loading

#### *2.2.1 Underlying Concept*

The drawbacks highlighted in the previous section have provided the basis for research to overcome the shortcomings associated with the conventional shedding mechanism. Thus, the longitudinal deformation of warp threads is to be avoided, threads are not displaced, and ultimately disturbances in the weaving process are eliminated even during the event of foreseen or unforeseen occurrence of periodic variations in the yarn. The use of multi-angled discs (Fig.1) appears to be the appropriate solution in order to overcome the aforementioned problems[13]. During the shed formation, the warp threads lie on the surface of the multi-angled discs. The main objective has been directed towards attainment of a new loom construction that could utilize the concept with the major merit of forming a warp shed without dynamic warp loading. This has enabled weaving of newer types of fabrics, which has been considered otherwise impossible. Thus, new variety of fabrics could be woven from raw materials that are of a very poor quality and difficult to spin in to yarn. It therefore results in the improvement in the economy of raw material processing and also to weave fabrics for specific applications such as heat and acoustic insulation, and in the case of reinforcing composites.

#### *2.2.2 Trial by Use of Shedding Discs*

Quadrangular discs have been used in shedding in the case of multiple shed weaving machines. Also 8 segment discs have been used for increasing the weft density. The two types of discs are in pairs and positioned at 45° to each other. The weaving drum consisting of the discs is shown in Fig. 2. Though a number of methods are available for feeding the warp sheet, one method has been explained. Air has been used as the medium of weft insertion and cam control. Use of electronic control enables the weaving drum to rotate without interruption at a constant preset velocity. This helps to carryout trials in weaving different types of fabrics.

### 2.3 Merits of New System

Fabrics have been woven from very low tenacity warp yarns and loose thread structure with strongly developed thread surface, which could practically have been impossible with conventional looms. The only practical difficulty encountered has been in feeding the warp yarns in the form of roving. This necessitated the use of a band transporter. It has been possible to weave entirely new varieties of fabrics very effectively by adoption of this technology. Thus, the fabrics could be woven economically with low density and these could be used for thermal and acoustic insulations.

As the mobility of warp threads in the structure is unrestricted, it has been possible to weave new types of woven fabrics having warp yarns of any dimensional structure. An example is the weaving of terry fabric with loop piles. Such a technology enables to form woven fabrics that could have varied applications such as geotextiles, insulation and reinforcements. Adjusting the shed depth is more difficult. Also accessibility is less. The system, however, has its own limitations. The shed change is not effective enough.

An additional advantage with this technology has been in the manufacture of woven grids with the use of very simple arrangements and with the possibility to produce specific products. The design and construction of new weaving machine can be carried out without much problem and any amount of shedding disc assemblies consisting of 2 or 3 discs can be multiplied to form an appropriate shedding. Due to the continuous rotational movement, the dynamics of the rotational weaving machine is advantageous for the weaving process as well as for conditions during usage. There is scope to obtain differentiated woven fabric forms and structures with the use of different raw materials.

## 3 Analytical Approach in Shedding Cam Design

Shedding cams are designed for weaving simple structures having repeat of up to 8 picks. A graphical

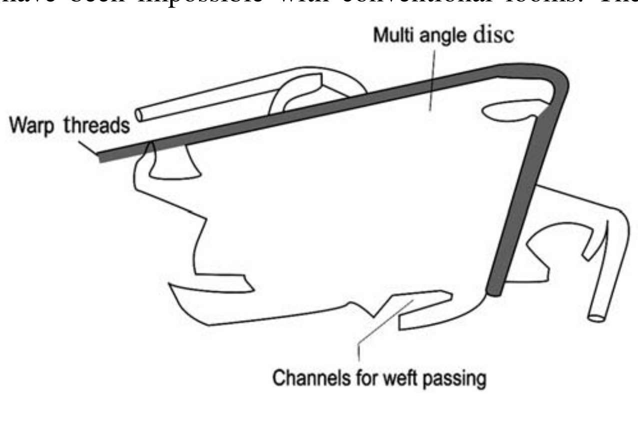


Fig.1—Multi-angled disc[13]

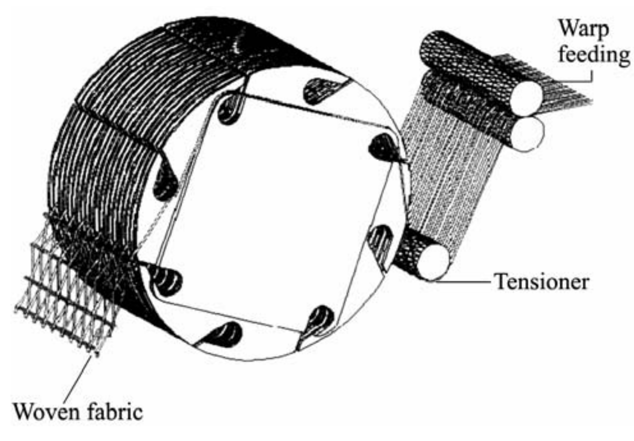


Fig.2—Weaving drum[13]

method has been evolved for designing shedding cams considering the simplest example of a plain weave[13]. Subsequently, a method has been developed for calculating the pressure angle of the shedding cam and the same has been applied for the case of a plain weave shedding cam with roller reversing mechanism located under the loom[14]. Investigations have also been done on the influence of follower motion curves and treadle pivot location on the pressure angle and lateral force affecting treadle pivot. Also, a geometrical method has been used for deciding the factors which restrict the number of picks in the weave repeat for shedding cam mechanisms[15]. Another approach provides a formulation method for shedding cam follower displacement diagrams and graphical construction of shedding cams under the hypothesis that the follower moves on a straight line instead of an arc[16,17]. It is intriguing to note that none of these researches provide a method for the determination of cam shedding mechanism dimensions for proper running and optimum force and motion transmission. Further increase in machine speeds demands a higher level of accuracy in the design and manufacture of shedding cams. Accordingly, an analytical method has been developed for the design of shedding cam mechanism based on the design requirements of pressure angle and minimum cam radius of curvature[18].

**3.1 Stages in Cam Designing**

A mathematical approach has been used in designing the cam shedding mechanism[19] using the stages as discussed hereunder.

***3.1.1 Displacement Diagram and Motion Curves of Cam Follower***

A cycle of the heald frame movement is related to the number of picks in the repeat of the weave, during which cam shaft makes one complete revolution. Hence, the shedding cam has to be divided into as many sections as the number of picks in a weave repeat. Each section corresponds to a heald frame movement for one pick, so as to synchronise its movement with picking and beat up mechanisms. This also involves reduction in the speed of main shaft of the loom correspondingly. Now-a-days, shedding cam motions are based upon the open shed principle. The following equations have been used to obtain the follower displacement diagrams:

$$\theta_1 = \frac{360s}{n} \qquad \ldots (1)$$

$$\theta_2 = \frac{360[p-s]}{n} \qquad \ldots (2)$$

where $\theta_1$ is the cam shaft angle of rotation during shed change; $\theta_2$, the cam shaft angle rotation during dwell; *s*, the portion of one loom revolution used for shed change; *p*, the number of picks at which warp is over or under the weft; and *n*, the number of picks in the weave repeat.

Motion curves relating to simple harmonic, cycloidal, modified sine and modified trapezoidal have been mathematically expressed and their influence on the shedding cam design studied[20].

***3.1.2 Determination of Dimensions of Cam***

The dimensions of the cam are restricted by the pressure angle and the minimum cam radius of curvature. The pressure angle is the angle between the direction of follower motion and the normal to the pitch curve at the contact point between cam and follower roller. It is a critical parameter in cam design and its maximum value is limited. The mathematical expression of the pressure angle can be derived by many methods[21]. Another crucial factor to be considered in the cam design is the minimum cam radius of curvature. Its value should be higher than the minimum value so that the follower roller follows the cam contour safely and also stresses at the contact area between cam and follower roller are not excessive. Hence, the minimum radius of curvature should be above zero to ensure smooth movement between cam and follower roller at the contact point. The cam radius of curvature can be obtained using the following relationship:

Cam radius of curvature = Pitch curve radius of curvature – Radius of follower

***3.1.3 Determination of Cam Profile Coordinates***

An analytical method has been used for the determination of cam profile by use of envelope theory[20,21]. The first and second matched shedding cam profile coordinates can be accurately calculated with respect to the angle of the cam shaft, by determination of the follower displacement diagram and the cam mechanism dimensions. This holds valid for positive cam shedding motions. In the case of negative cam shedding motions, either the first or the second shedding cam profile coordinates can be determined, since there is one cam contour.

**3.2 Analysis of Design**

In order to design the cam, the follower displacement diagram has to be determined with regard to the angle of cam shaft rotation. The follower displacement diagram is derived based on the weave pattern. The cam shaft angles for the upward or downward duration of the follower, as shown in Table 1, are based on the pick number in the repeat of the weave[19].

Simple harmonic motion gives the best cycloidal motion curve, the lowest minimum cam radius of curvature and highest pressure angle radius. The results have revealed that the increase in maximum velocity causes increase in maximum pressure angle. Increase in the maximum acceleration reduces the minimum cam radius of curvature. Hence, the choice of a motion curve to the lowest possible maximum acceleration and maximum velocity enables to design shedding cams with the higher minimum cam radius of curvature and lower maximum pressure angle. Motion curves with the lower maximum velocity and lower maximum acceleration tend to give a lower maximum pressure angle and a higher minimum cam radius of curvature respectively and vice versa. Such motion curve characteristics are also necessary for lower maximum shaft velocity and acceleration. Symmetrical motion curves are generally used in shedding cam design, whereas asymmetrical motion curves are necessary only in special cases.

The modified trapezoidal motion curve has been used in carrying out design trials with different cam dimensions. The studies have revealed that there is a significant increase in maximum pressure angle and decrease in minimum cam radius of curvature with the increase in the number of picks in a weave repeat. This is due to the decrease in the angle of cam shaft rotation relating to rise and return periods of follower motion which makes the cam profile steeper. The calculation has been based for a maximum 8 picks in the repeat with a dwell of 1/3. The maximum pressure angle and minimum radius of curvature are much improved without dwell. Thus, for a given size of shedding cam the maximum number of picks in the weave repeat is limited. Manufacturers of shedding cams normally restrict the maximum number of picks to 6 (refs 22,23).

The influence of cam dimensions on the maximum pressure angle and minimum cam radius of curvature has been investigated through various design trials, using the modified trapezoidal motion curve. The maximum pressure angle is found to reduce during the rise period and increase during the return period. With the increasing values of cam base circle radius, the minimum cam radius of curvature increases in both the periods. An identical effect is also observed by changing the radius of the roller. The maximum pressure angle reduces during the rise period and increases during the return period, with the increase in roller radius. Also the increase in roller radius reduces the minimum cam radius of curvature during the rise as well as return periods. This tends to improve the maximum pressure angle but affects the minimum cam radius of curvature. It has been observed that the same maximum pressure angle is obtained as the sum of cam base circle radius and constant roller radius. Hence, shedding cams can be designed for a particular maximum pressure angle by maintaining roller radius as large as possible, based on the consideration that the minimum cam radius of curvature is maintained just above zero so as to prevent under cutting and also reduce stresses due to contact.

Comparison of the maximum value of the pressure angle during rise and return periods shows that with the increase in cam dimensions, there is reduction in maximum value of pressure angle during the rise period and increase during the return period. With the selection of appropriate cam dimensions, it is possible to get a greater minimum radius of curvature and a lesser maximum pressure angle.

Further design trials have been conducted with the use of modified sine and modified trapezoidal motion curves with up to 6-8 picks in the weave repeat, based on different dwell periods. This would give a maximum cam diameter of 22-26 cm, which is found to be industrially suitable. A weave repeat of 6 picks is found to give a very convenient maximum pressure angle and also a minimum cam radius of curvature can be obtained using this size of shedding cams. The maximum pressure angle and the minimum cam radius of curvature are safely satisfied with the lower

Table 1— Angle of cam shaft rotation for different pick numbers in weave repeat[19]

| No. of picks | Pressure (α), deg 1 revolution | Pressure (α), deg 2/3 revolution | Dwell, deg 2/3 revolution |
|---|---|---|---|
| 2 | 180 | 120 | 60 |
| 3 | 120 | 80 | 40 |
| 4 | 90 | 60 | 30 |
| 5 | 72 | 48 | 24 |
| 6 | 60 | 40 | 20 |
| 7 | 51.4 | 31.2 | 17.2 |
| 8 | 45 | 30 | 15 |

cam base circle radius. Increase in the center of follower in rotation results in an increase in other cam mechanism dimensions that sufficiently meet the requirements of the maximum pressure angle and the minimum cam radius of curvature, thus larger size shedding cams are required.

## 4 Innovative Method of Heald Frame Motion Regulation

### 4.1 Review of Existing System

It is a fundamentally known fact that the heald frames in a loom have to move both up and down vertically so as to move the warp threads accordingly and cause interlacement to the weft yarns[24]. Hence, the movement of the heald frame has considerable effect on the shedding motion and warp tension. The modern weaving trend is aimed at moving the heald frames at a higher speed with variability and high density[25]. At higher loom speeds the acceleration of the heald frames increases drastically and hence it becomes necessary to design the heald frame so as to regulate its motion in a more reasonable manner. The two commonly used methods of heald frame motion regulations in modern looms are the simple harmonic and 7 order polynomial motions[26,27]. However, a simple harmonic type of motion regulation is unsuitable for high speed looms due to its poor dynamical performance (such as acceleration). The 7 order polynomial motion regulation is superior in this aspect and is thus widely used in high speed looms.

### 4.2 The New Concept

An innovative method of heald frame motion regulation has been developed and is based on the 8 order polynomial motion regulation[28]. It has been found to be far superior to both the simple harmonic as well as 7 order polynomial motion regulations. The following two criteria need to be fulfilled by the regulating motion of the heald frame so as to reduce the negative effects of the heald frames motion on the warp yarns and also to enable efficient weft insertion:

(i) The warp tension is minimum during the closure of the warp shed and is maximum when the shed is fully open. The velocity of the heald frame should accordingly change in a gradual manner. In other words, this should be the highest at the closer of the shed and minimum when the shed is fully opened. This can avoid the warp breakage caused by the sudden increase in tension or acute friction between the warp and the heald.

(ii) When the heald frame begins to change from motion to dwell or vice versa the acceleration has to be reduced to the minimum. During the rest of the movement, the change in acceleration should be mild so as to reduce the warp tension due to vibration of heald frame and also to adjust the shedding mechanism to accommodate the high speed of looms.

The displacement equations for 8 order polynomial motion regulation, 7 order polynominal motion regulation and simple harmonic motion regulation of heald frame are given below.

- 8 *Order Polynominal Motion Regualtion*

$$s = 8l[(35-4c)(\varphi/\alpha)^4 - (252-32c)(\varphi/\alpha)^5 + (728-96c)(\varphi/\alpha)^6 - (960-128c)(\varphi/\alpha)^7 + (480-64c)(\varphi/\alpha)^8] \quad \text{when } 0 \le \varphi \le \alpha/2$$

$$s = l - 8l[(31-4c) - (368-48c)(\varphi/\alpha) + (1890-248c)(\varphi/\alpha)^2 - (5460-720c)(\varphi/\alpha)^3 + (9695-1284c)(\varphi/\alpha)^4 - (10836-1440c)(\varphi/\alpha)^5 + (7448-992c)(\varphi/\alpha)^6 - (2880-384c)(\varphi/\alpha)^7 + (480-64c)(\varphi/\alpha)^8] \quad \text{when } \alpha/2 \le \varphi \le \alpha$$

- 7 *Order Polynomial Motion Regulation*

$$s = l[35(\varphi/\alpha)^4 - 84(\varphi/\alpha)^5 + 70(\varphi/\alpha)^6 - 20(\varphi/\alpha)^7]$$

- *Simple Harmonic Motion Regulation*

$$s = l/2[1-\cos(\pi\varphi/\alpha)]$$

where $\varphi$ is the angle of drive shaft of the loom; c, the pending coefficient; s, the displacement of heald frame; l, the scope of heald frame; and $\alpha$, the pressure angle.

The characteristics of the above 3 kinds of motion regulations (8 order polynominal motion regualtion, 7 order polynomial motion regulation and simple harmonic motion regulation) of heald frame, namely the velocity, acceleration and jerk are given in Table 2. The motion regulation curves for the three types of motions are given in Fig. 3.

It could be inferred that for a given condition of heald frame and the angle of the drive shaft**,** the curves 1 and 2 show higher velocity as compared to curve 3, during the commencement of shed opening. The velocity reduces when the healds begin to close. Hence, it is preferable to make the shed clear and allow the weft to pass through the warp shed effectively. The peak value of the curve 1 is slightly less than that of 2. However, at the point where the shed is fully open or begins to close the curve 1 makes the heald frames move slower. This could reduce the warp breakage rate and also increase the dwell of the healds, thereby enabling the weft to be inserted through the warp shed effectively.

Table 2— Details of motions for different types of cams[28]

| Type of motion regulation | Velocity | | Acceleration | | Jerk | |
|---|---|---|---|---|---|---|
| | Peak cm/s | Position deg | Peak $cm/s^2$ | Position deg | Peak $cm/s^3$ | Position deg |
| 8 order polynomial | 1.6667 | 120 | 0.0239 | 60/180 | $6.5351\times10^{-4}$ | 18.7663/101.2337/ 138.7663/221.2337 |
| 7 order polynomial | 1.8229 | 120 | 0.0261 | 66.3344/172.6656 | $7.5955\times10^{-4}$ | 120 |
| Simple harmonic | 1.3090 | 120 | 0.0171 | 0/240 | $2.2429\times10^{-4}$ | 120 |

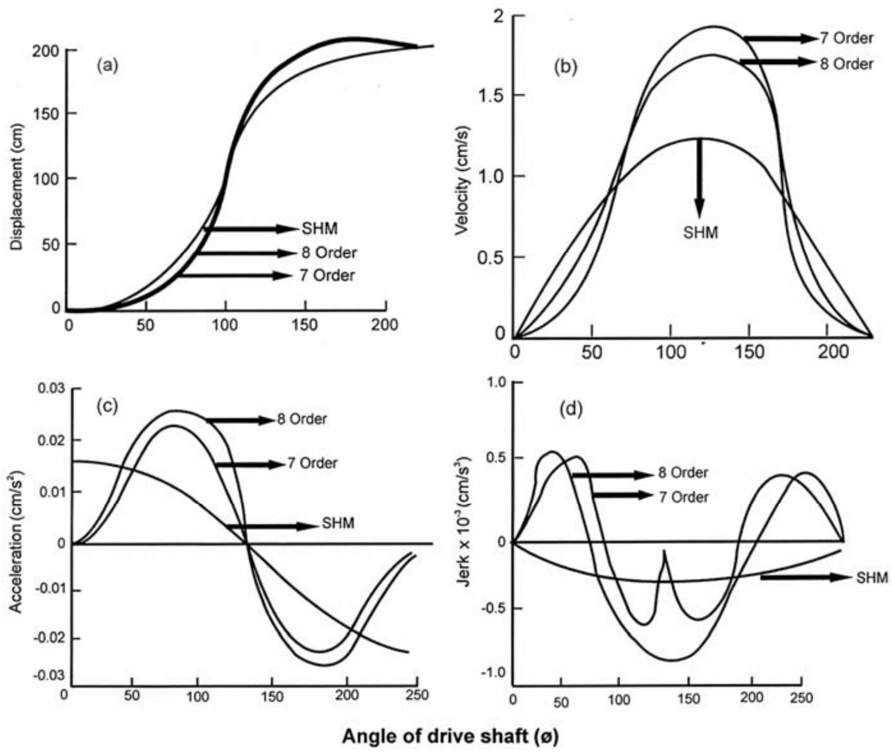


Fig.3—Curve for 3 kinds of motion[28]

In the case of curve 3 the peak value of acceleration occurs at either extremes of the heald frame's motion. The acceleration of the heald frame is much greater when it changes from static state to motion state or vice versa, which creates greater inertia force resulting in a little vibration. Hence, for this reason the simple harmonic motion is unsuitable for higher speeds. On the other hand the values of acceleration for curves 1 and 2 are zero at either extremes of the heald frame motion, which avoids jerks. For this reason both the 8 and 7 order polynomial motions are suited for high speed looms. Moreover in the case of curve 1, the acceleration changes lesser than that of curve 2 and also has a lesser peak value. Hence, the curve 1 (8 order polynomial) is most suited for high speed looms. Also, it gives the best dynamic performance at higher speeds.

## 5 Dobby Shedding with Microprocessor

In the case of positive dobby shedding, both upward as well as downward movements of the heald frames are directly controlled. Hence, it is suitable for weaving heavier varieties of fabrics at higher loom speeds. However, recent models of positive dobbies such as Staubli 2232 and Staubli 2600 are far advanced in the design and also prove to be costly. Hence, a need has been felt to develop a new type of positive dobby that is simple in design and also economical. Accordingly the research has been directed on model Staubli 2521, which is a negative dobby that can run at comparatively higher speeds. This model is found to be ideal for conversion into a positive dobby so as to fulfill the aforementioned need.

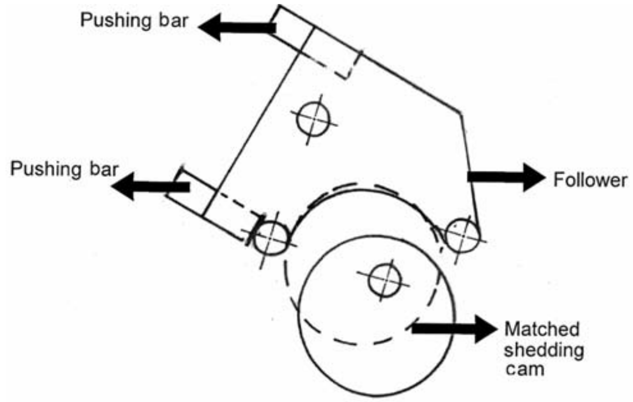


Fig.4—Driving mechanism of Staubli 2521 (ref. 29)

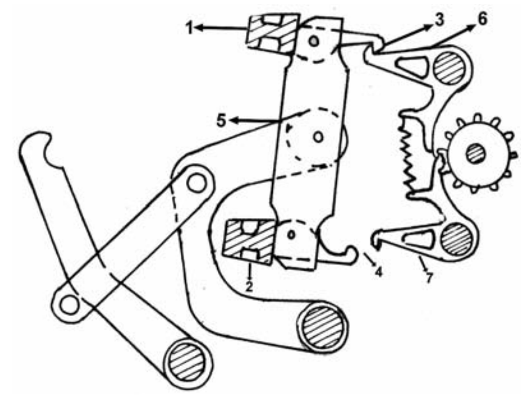


Fig.5—Control mechanism of Staubli 2521 (ref. 29) [1,2 – pushing bars; 3,4–hook; 5–baulk; and 6,7– control hook]

Table 3— Position of heald frame at different action modes of solenoids[29]

| Action mode | First solenoid | Second solenoid | First dual hook fixed to first hook knife | Dual hook fixed to second hook knife | Moving direction of first bar | Moving direction of second bar | Heald frame |
|---|---|---|---|---|---|---|---|
| 1 | Energized | De-energized | Engaged | Disengaged | Left | Right | Lifted |
| 2 | Energized | De-energized | Engaged | Disengaged | Right | Left | Lowered |
| 3 | De-energized | Energized | Disengaged | Engaged | Right | Left | Lifted |
| 4 | De-energized | Energized | Disengaged | Engaged | Left | Right | Lowered |
| 5,6 | Energized | Energized | Engaged | Engaged | Left<br>Right | Right<br>Left | Remain in top position |
| 7,8 | De-energized | De-energized | Disengaged | Disengaged | Left<br>Right | Right<br>Left | Remain in bottom position |

**5.1 Principle of Existing Dobby**

The driving mechanism of Staubli 2521 is shown in the Fig. 4. Matched shedding cams cause movement of the pushing bars fixed on the pivot, through respective followers. The pushing bars are fulcrumed and cause movement of the heald frame shedding control mechanism (Fig. 5). The upper hook of the dobby engages with the upper control hook. When the upper pushing bar moves towards the left and the other moves towards the right, the lower bar pushes the baulk lever to swing to the right, which, in turn, pulls the jack lever rightwards causing lifting of the respective heald frame. In case the heald frame is to be lowered for the successive pick, the lower control hook does not engage with its respective hook of the baulk lever and when the upper pusher moves rightward and lower pusher bar moves leftward the spring reverses the motion forces when the baulk lever to swing to the left and lower the heald shaft. The other motion possibilities can be analyzed or can be referred to the foregoing introduction of the novel type of positive dobby.

**5.2 Principle of Microprocessor Dobby**

The developed dobby comprises three components, namely driving mechanism, selection mechanism and heald frame shedding control mechanism[29]. The driving mechanism is similar to that of the existing dobby (Staubli 2521) with respect to the matched shedding cams. However, the two pushing bars are converted into pushing and pulling bars. In the existing dobby the pushing bars push the end of the baulk lever, whereas in the new type of dobby the bars not only push the baulk lever but also pull it by the left notch. The speed of the cam is half of the loom speed.

The selection mechanism comes into action during the dwell period of the matched shedding cams. A cam pushes down the selection bar through another bar. If the heald frame is to be lifted for the successive picks, a solenoid is energized, attracting the selection bar. Lowering of the heald frame for the next pick is done by de-energising the solenoid. Hence, the solenoid is unable to attract the selection bar. The solenoids need only small circuits to attract the selection bars. Moreover, the small circuit shortens the time of solenoid de-energising. Such arrangements enable the entire selection mechanism to quickly respond to the selection instruction from the microprocessor. The selection mechanism is designed in such a way that it avoids any mishap to make the dual hook dis-engage with its fixed hook knife during shedding, thereby rendering the mechanism to be more reliable. The control mechanism of the microprocessor dobby has eight modes of action as given in the Table 3**.**

The action of the selection mechanism for a pick commences at zero degree of the timing diagram. A double voltage DC power supply is used to accelerate the response of the solenoid. At the initial stage of energizing the solenoid, a high voltage is exerted on the solenoid and when the selection bar is attracted, low voltage is exerted to lower the circuit to eliminate the solenoid heat generation and also to lessen the duration to de-energise the solenoid.

#### 5.3 Motion Simulation

Fortron programming language has been used to analyse the motion of individual parts of the machine. AutoCAD has been applied so as to simulate the motion of the mechanisms based on the data obtained from kinematic analysis. The use of AutoCAD has enabled animation of the entire process of shedding, in which the mechanisms move as expected. The motion simulation becomes the primary aspect of the dobby design. A simplified prototype model has been developed after the simulation, and it has been proved to be effective. The newly developed dobby is comparable well with the Staubli 2521 and Staubli 2232 versions. The Staubli 2521 version of the microprocessor controlled dobby has the following advantages:

(i) Change from a single to a double tip hook.
(ii) The pushing action of the pusher bar has been converted into a pushing and pulling action.
(iii) The multiple control hooks have been replaced by a fixed hook knife.
(iv) New selection mechanism has been designed that enables quick response of the dobby, and also makes it more reliable and controllable due to the microprocessor.

With regard to Staubli 2232 version, the microprocessor dobby offers the following advantages:

- The main construction of the dobby is made simpler.
- Arrangements have been made to accelerate the response of the selection mechanism.
- Dobby can run at a higher speed.
- Owing to simplicity of construction and ease of manufacture, it is very cheap.

#### 5.4 Newer Versions of Electronic Dobbies

There are basically four models of electronically controlled dobbies from Staubli. Types 2561/2571 are negative dobbies suitable for use on jet weaving machines, particularly water-jets. Type 2580 is for low position mounting to air-jet weaving machines. It provides greater stability and exceptionally compact design. Side plates and direct connection to the weaving machine by means of a support base plate prevent inherent vibration. Type 2581 has a compact monoblock construction permitting optimal high position mounting to all air and water-jet machines. Type 2660/2670 are rotary dobbies for universal application on high speed rapier, projectile and air-jet weaving machines. This type ensures absolutely play-free, precise and positive action of the dobbies, even under maximum loading and satisfies all imposed demands. Control for all four types is electronic, with control unit either separate or integrated in the weaving machine. Repeat length for all is up to 6400 picks, or dependent on the control unit.

## 6 Use of Cycloidal Cam

#### 6.1 Review of Earlier Work

In the case of cam shedding the relation between the displacement and the time of the follower determines the profile of the cam. Attempts have been made to minimize the lateral force to the follower for which it causes side thrust and wear of the shedding mechanism[30]. It has also been proved that the lateral force on the follower in the case of cycloidal cam is initially less than that for simple harmonic cam during the change periods of the healds. The dynamic force exerted by the cycloidal cam to the follower is in the form of a sinusoidal curve[31]. The movement of the follower has been compared for both cycloidal and simple harmonic cams through one unit of distance in 1 radian of cam rotation, and it is found that the initial displacement is slower in the former case. Hence, the initial heald velocity is low in cycloidal cams[32]. With the increase in velocity the gradual displacement also increases. However, the follower ultimately again attains the end point slowly in the case of cycloidal cam. This kind of heald motion has a definite influence on the variation in warp tension and the abrasion of warp with the picking elements.

#### 6.2 Recent Work

In the recent attempt[33], all the other cam parameters such as displacement and time of cam follower have been kept constant in comparison with the existing simple harmonic cams of a particular type of loom. A computer programme coupled with graphic interface has been used to draw the profile of the cycloidal

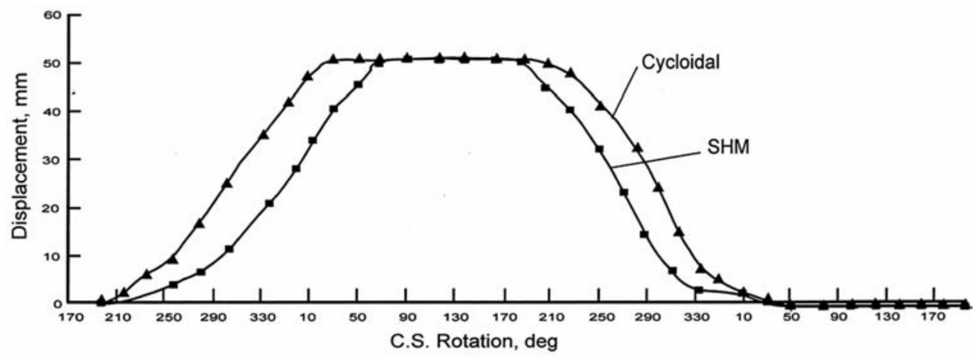


Fig.6—Cam follower displacement[33]

cams and the cams are thereby designed. The cam has been manufactured and the displacement of the cam followers compared and investigated along with the warp breakage rate and basic tension and its variations. The actual heald displacement behaviour of simple harmonic as well as cycloidal cams has been obtained by measuring actual displacement of cam follower for both the pairs of cams. The powerloom with negative tappet attachment has been selected. The variation in warp tension for both sets of cam has been measured by using rothschild tension meter at the position between back rest and lease rods. The warp breakage rate has been observed for both the cycloidal as well as the simple harmonic cams (Table 4). It is evident that the breakage rate is comparatively lesser by about 10% in the case of cycloidal cam. This could be attributed to the fact that a smoother cam follower motion at the areas of commencement and ending helps to distribute the stress better in the warp yarn. Thus, the tension on the warp yarn is somewhat smoother for cycloidal cam in comparison to the simple harmonic motion.

The nature of displacement of both the types of cams is shown in Fig. 6. The cam follower in the case of cycloidal cam commences in a slower and smoother manner as compared to its simple harmonic counterpart. It then moves faster than the simple harmonic cam follower and at the final position of the displacement the cycloidal cam follower stops comparatively more gradually.

The warp tension variation during the loom running without the weft insertion has been compared for both types of cams, as shown in Fig.7. The change in the warp tension during the heald change period is smoother during the initial and final part of the change period. Similar trend is also observed when the warp tension is measured dynamically with weft insertion during loom running. The optimum values of warp tension at the beat up point and greater level of tension at bottom position of heald as compared to that at top position agree with the earlier research findings[34].

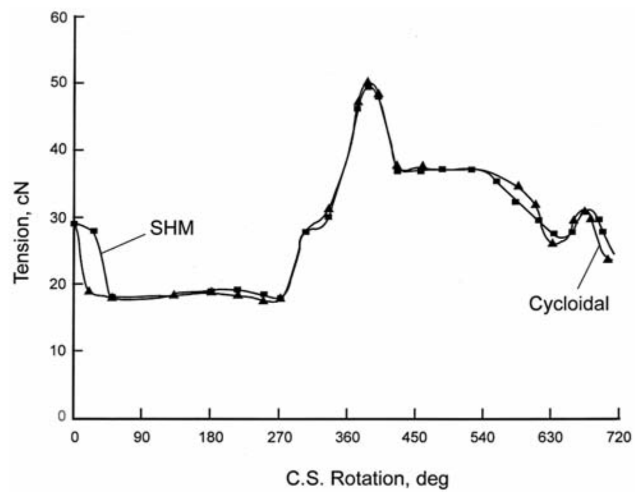


Fig.7—Variation in warp tension (without shuttle insertion)[33]

Table 4— Comparison of warp breakages in cycloidal and simple harmonic motion cams[33]

| Type of cam | Duration[a] min | No. of filling picks[a] | Mean warp breakage rate/1000 warp ends/10000 weft picks |
|---|---|---|---|
| Simple harmonic | 66 | 7940 | 2.11 |
| Cycloidal | 61 | 7860 | 1.94 |

[a]Mean of 5 observations.

## 7 New Shedding Concepts in 3D Weaving Method

The recent concept of 3D weaving[1] utilizes the dual directional shedding operation, which forms the core of the 3D weaving method. The principle of this shedding method is best explained by considering the simplest example of a plain weave. In order to move the warp yarns arranged in a grid-like manner so as to form multi-layer sheds along column- and row-wise directions, it is necessary to separate them from each other a little distance apart in the shedding area. It is to be noted that the sheds that are formed column-wise and row-wise are lifted alternately one after the other. This is due to the fact that the filling yarns in the vertical and horizontal directions need to be inserted in the corresponding warp sheds respectively. Such a method has been able to produce fabrics for a wide range of technical applications.

## 8 Conclusions

Considerable research has been done to improve the motion of the healds and thereby to reduce the wear and tear of the parts. Shedding without dynamic warp loading by the use of multi-angle shedding discs causes increase in the weft density and also provides possibilities to weave new variety of fabric structures that are considered to be difficult to manufacture through the conventional method. Also, there is a scope to use unspinnable yarns, roving and selvedges. This method, however, is applicable to only multi-phase weaving machines. Development of a new microprocessor controlled negative dobby has also enabled the loom to run at a higher speed than the existing positive dobbies. This new dobby method is proved to be economical and easy to manufacture owing to simplicity of design. Design of shedding cam by analytical method has enabled to use symmetrical as well as asymmetrical motion curves in the shedding cam design. An 8 order polynomial motion regulation of heald frame has been developed which improves the dynamic performance of the heald frame and is suitable for running at high speeds with lesser jerk. However, the jerk is not totally avoided. Trials on cycloidal cam have proved that the warp breakage rate is reduced as compared to that of simple harmonic cam. However, further trials are necessary to justify this fact.

Though the recent researches pave the way towards better shedding performance in shuttle looms in some cases, more research will be required so as to meet requirements of the higher speeds of unconventional weaving machines. Also, a method is to be devised to eliminate the jerkiness in heald movements.

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