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Analytical design method for cam shedding motions in weaving

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An analytical design method for cam shedding motions has been developed using mathematical method to derive the follower displacement diagrams for a given weave and the mathematical equations for pressure angle, cam radius of curvature and matched (or conjugate) cam profile coordinates. The effect of maximum pick number in the weave repeat, shedding cam mechanism dimensions, swing angle of the follower and four different motion curves on the maximum pressure angle and minimum cam radius of curvature has also been studied. The cam shedding mechanism dimensions are determined according to the critical values of maximum pressure angle and the minimum cam radius of curvature. It is observed that the cam shedding mechanism dimensions must be determined with maximum pick number in the weave repeat by considering the effects of motion curves and follower swing angle. Finally, four picks plain weave and twill 3/1 +1/1 shedding cam plots are presented.

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1 Introduction

Shedding is one of the main operations of the weaving process which separates warp yarns into two layers to obtain an openness called shed for weft insertion. There are different types of shedding motions varying in patterning capability. Cam shedding motions are used to weave fabrics from simple weaves up to six or eight picks weave repeat at high speeds exceeding 1000 rpm with air-jet and water-jet weaving machines. They are generally mounted to weaving machines from outside at bottom level and therefore also called outside cam shedding motions. Fig. 1 shows a cam shedding motion which consists of shedding cams (C and D), oscillating follower (L) with two rollers and motion transmission mechanism to a shaft. There are as many shedding cams and motion transmission mechanisms as the number of shafts. The cam shedding motion shown in Fig. 1 has matched or conjugate cams. Each cam has its own roller and the rollers are mounted on a common follower with a certain angle. When the cam rotates in clockwise direction continuously, the follower swings in clockwise and anti-clockwise directions and dwells at the end positions when necessary, depending on the weave. The clockwise rotation of the follower causes the link H to rotate in the clockwise direction and similarly anti-clockwise rotation of the follower rotates the link H in anti-clockwise direction. The motion of the link H is transmitted to a heald frame from the link J. Shafts move up when the followers rotate in clockwise direction and lifted down with the anti-clockwise rotation of the followers. The dwell of the followers at the end positions corresponds to the dwell of the shafts at upper and lower shed positions. Points A and

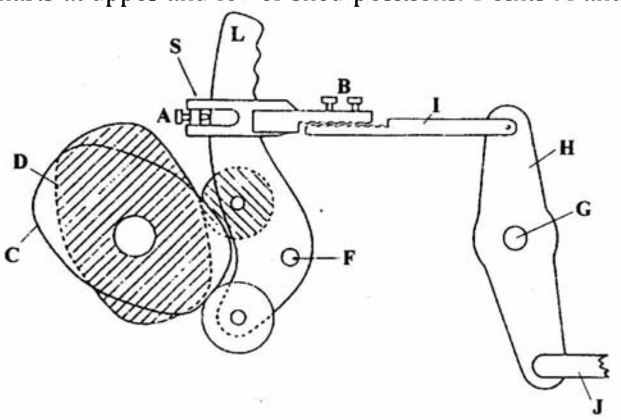

Fig. 1—A positive cam shedding motion[1] [A – Shed lift adjustment point; B – Shed bottom position adjustment point; C and D – Conjugate cams; F – Follower center of rotation; G – Center of rotation of the link H; H – Oscillating link; I – Connection link; J – Motion transmission link; L – Follower ; and S – Connection point of the link I to the follower]

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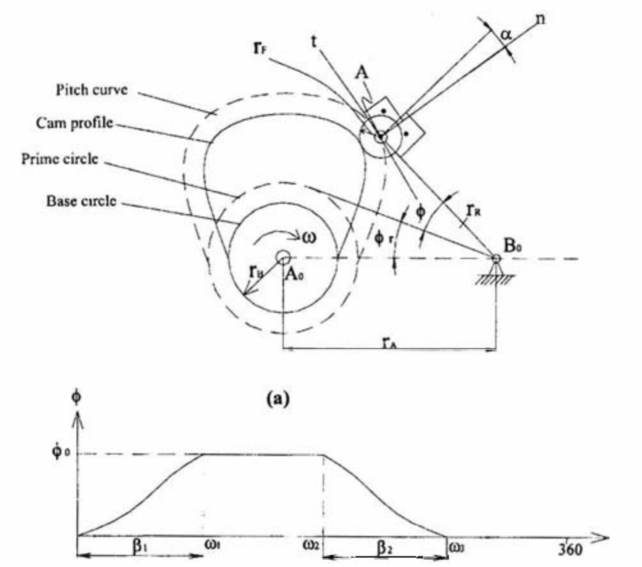
B show adjusting points for the amount of shed lift and position of the shed respectively. A new set of cams is placed on the cam shaft when weave changes. As the distance between the cam and the follower centers of rotation, the distance between the follower and the roller centers of rotation and the radius of the rollers remain the same for a cam shedding motion, all the shedding cams, irrespective of the weave to be designed, have to have the same minimum and maximum radii.

Swales[1] introduced a graphical method for constructing shedding cams with the plain weave example. Gu[2] developed a method for calculating shedding cam pressure angle and applied it to a plain weave cam shedding mechanism with roller reversing motion placed under the loom. The effects of follower motion curves and treadle pivot location on pressure angle and lateral force affecting treadle pivot have also been studied[2]. Marks and Robinson[3] expressed their views on the factors that limit the number of picks in the weave repeat for shedding cam motions using a geometrical approach. Talavasek and Svaty[4] and Alpay[5] gave a formulation method for shedding cam follower displacement diagrams and graphical construction of shedding cams with the assumption that the follower moves on a straight line rather than on an arc.

None of these publications presents a method for the determination of cam shedding mechanism dimensions for proper running and optimum force and motion transmission. Further increase in machine speeds impose even more accuracy on the design and manufacturing of shedding cams. This paper reports an analytical method for designing a cam shedding motion according to the design criteria of pressure angle and minimum cam radius of curvature.

Fig. 2 shows a cam nomenclature and a follower displacement diagram for swinging roller follower type of cam. Some cam mechanism parameters are defined below:

r_A — the distance between cam center of rotation (A_0) and follower center of rotation (B_0); r_R — the follower arm length (i.e. the distance between the roller center A and the follower center of rotation B_0); r_B — the cam base circle radius; r_F — the roller radius; α — the pressure angle (the angle between the normal to the pitch curve and the instantenous direction of the follower motion); ϕ — the follower angular displacement from its zero position in relation to the angle of cam shaft rotation (φ). The zero position of the follower is taken as the position at which the follower roller is in touch with cam base circle; and ϕ_0 — the maximum angular displacement of the follower from the zero position.



(b)

Fig. 2—(a) Cam nomenclature and (b) A displacement diagram for oscillating roller follower type of cam [A_0 – Cam center of rotation; B_0 – Follower center of rotation; A – Follower roller center; r_F – Follower roller radius; r_B – Cam base circle radius; r_R – Follower length; r_A – Distance between A_0 and B_0; φ – Angle of cam shaft rotation; ϕ – Follower angular displacement; ϕ_r – Angular position of follower when follower roller touches cam base circle; α – Pressure angle; t – Tangential direction; n – Normal direction; ϕ_0 – Follower angular stroke; β_1 – Cam shaft rotation of rise period; β_2 – Cam shaft rotation of return period; φ_1 – Cam shaft angle at the end of rise period; φ_2 – Cam shaft angle at the start of return period; and φ_3 – Cam shaft angle at the end of return period]

A follower displacement diagram expresses the motion of follower with respect to the angle of cam shaft rotation for one revolution of cam shaft. The follower displacement diagram can be described by the following three types of motion:

Rise — Motion of the follower away from the cam center. Fig. 2 shows that it corresponds to the clockwise rotation of the follower.

Dwell — It means that the follower is at rest.

Return — Motion of the follower toward the cam center. Fig. 2 shows that it corresponds to the anti-clockwise rotation of the follower.

β angle in motion curves (given in section 2.1.1) refers to the cam shaft rotation during which rise or return motion of the follower takes place.

2 Materials and Methods

2.1 Design Method for Cam Shedding Mechanisms

Necessary steps to design the cam shedding mechanisms mathematically are explained as under:

2.1.1 Follower Displacement Diagrams and Motion Curves

The number of picks in a weave repeat corresponds to one period of a heald frame (shaft) movement and cam shaft completes one revolution during this period. Therefore, a shedding cam must be divided into as many sections as the number of picks in a weave repeat and each section has to produce a heald frame movement for one pick (one loom revolution) to syncronise the movement of heald frames with other loom mechanisms such as weft insertion and sley mechanisms. Also, the speed of the main shaft of a loom has to be reduced by the number of picks in the weave repeat and transmitted to the shedding cam shaft to achieve the syncronisation.

The following relationship can be established between the angle of rotation of the loom main shaft and the shedding cam shaft:

$$\varphi=\frac{\theta}{K} \qquad \ldots (1)$$

where θ is the angle of loom main shaft rotation; and K, the number of picks in the weave repeat.

Open shed principle is used in shedding cam motions today. Sometimes, a portion of a loom revolution is used for shed change (i.e. shaft movement from the upper to lower or lower to upper positions) and in some applications, one loom revolution is fully utilised for shed change.

Shed change and shaft dwell periods are expressed in terms of the angle of cam shaft rotation as follows:

$$\varphi_c=\frac{360}{K}\eta \qquad \ldots. (2)$$

$$\varphi_d=\frac{360}{K}(k-\eta) \qquad \ldots. (3)$$

where φ_c is the cam shaft angle of rotation during which shed changes (the rise or return motion period); η, the portion of one loom revolution used for shed change; φ_d, the angle of rotation of cam shaft during which shafts dwell; and k, the number of picks at which the warp is over or under the weft.

The follower displacement diagrams can be obtained for any weave using Eqs (2) and (3).

Four different motion curves (simple harmonic, cycloidal, modified sine, and modified trapezoidal) expressed mathematically as under were used for this study and their effects in shedding cam design investigated. A large number of motion curves including these have already been discussed in the literature[6].

Simple Harmonic Motion Curve

$$\phi=\frac{\phi_0}{2}\left[1-\cos\left(\frac{\pi\varphi}{\beta}\right)\right] \qquad \ldots (4)$$

Cycloidal Motion Curve

$$\phi=\phi_0\left[\frac{\varphi}{\beta}-\frac{1}{2\pi}\sin\left(\frac{2\pi\varphi}{\beta}\right)\right] \qquad \ldots (5)$$

Modified Sine Motion Curve

$$\left.\begin{array}{ll}
\phi=\phi_0\left\{0.44\frac{\varphi}{\beta}-0.035\sin\left(4\pi\frac{\varphi}{\beta}\right)\right\} & for \quad 0\langle\frac{\varphi}{\beta}\langle\frac{1}{8} \\
\phi=\phi_0\left\{0.28+0.44\frac{\varphi}{\beta}-0.315\cos\left(\frac{4\pi\varphi}{3\beta}-\frac{\pi}{6}\right)\right\} & for \quad \frac{1}{8}\langle\frac{\varphi}{\beta}\langle\frac{7}{8} \\
\phi=\phi_0\left\{0.56+0.44\frac{\varphi}{\beta}-0.035\sin\left(4\pi\frac{\varphi}{\beta}\right)\right\} & for \quad \frac{7}{8}\langle\frac{\varphi}{\beta}\langle 1
\end{array}\right] \qquad \ldots (6)$$

Modified Trapezoidal Motion Curve

$$\left.\begin{array}{ll}
\phi=0.09724612\phi_0\left\{4\frac{\varphi}{\beta}-\frac{1}{\pi}\sin\left(4\pi\frac{\varphi}{\beta}\right)\right\} & for \quad 0\langle\frac{\varphi}{\beta}\langle\frac{1}{8} \\
\phi=\phi_0\left\{2.444062\left(\frac{\varphi}{\beta}\right)^2-0.22203\left(\frac{\varphi}{\beta}\right)+0.00723\right\} & for \quad \frac{1}{8}\langle\frac{\varphi}{\beta}\langle\frac{3}{8} \\
\phi=\phi_0\left\{1.6110154\left(\frac{\varphi}{\beta}\right)-0.0309544\sin\left(4\pi\frac{\varphi}{\beta}-\pi\right)-0.3055077\right\} & for \quad \frac{3}{8}\langle\frac{\varphi}{\beta}\langle\frac{4}{8} \\
\phi=\phi_0\left\{1.6110154\left(\frac{\varphi}{\beta}\right)-0.0309544\sin\left(4\pi\frac{\varphi}{\beta}-2\pi\right)-0.3055077\right\} & for \quad \frac{4}{8}\langle\frac{\varphi}{\beta}\langle\frac{5}{8} \\
\phi=\phi_0\left\{-2.444062\left(\frac{\varphi}{\beta}\right)^2+4.666092\left(\frac{\varphi}{\beta}\right)-1.2292648\right\} & for \quad \frac{5}{8}\langle\frac{\varphi}{\beta}\langle\frac{7}{8} \\
\phi=\phi_0\left\{0.6110154+0.3889845\left(\frac{\varphi}{\beta}\right)+0.0309544\sin\left(4\pi\frac{\varphi}{\beta}-3\pi\right)\right\} & for \quad \frac{7}{8}\langle\frac{\varphi}{\beta}\langle 1
\end{array}\right] \qquad \ldots (7)$$

2.1.2 Determination of Shedding Cam Mechanism Dimensions

There are following two criteria that limit the cam mechanism dimensions:

2.1.2.1 Pressure Angle

As illustrated in Fig. 2a , the pressure angle (α) is defined as the angle between the direction of follower motion and the normal to the pitch curve at the contact point between cam and follower roller. The force transmitted from the cam to the follower acts in the normal direction and can be resolved into two components (Fig. 3). $F.\sin\alpha$ is the force component applied to the bearing of follower which causes vibration and wear. $F.\cos\alpha$ is the driving force for the follower. As α increases, the force component applied to the follower bearing increases and the driving force component decreases. Therefore, the pressure angle is a critical parameter in cam design and its maximum value must be limited. In this research, a maximum value of 40° is allowed for the pressure angle as this value is recommended for swinging roller follower type of cams.

There are different methods for deriving the pressure angle expression. The following expression is used in this study[7]:

$$\alpha=\tan^{-1}\left\{\frac{1}{\tan(\phi+\phi_r)}-\frac{r_R}{r_A}.\frac{1-d\phi/d\varphi}{\sin(\phi+\phi_r)}\right\} \quad \ldots (8)$$

ϕ_r represents the position of follower when the follower roller is in touch with cam base circle and can be expressed as follows using the law of cosines, keeping the other parameters as defined in section 1:

$$\phi_r=\cos^{-1}\left(\frac{r_A^2+r_R^2-(r_B+r_F)^2}{2r_Ar_R}\right) \quad \ldots (9)$$

2.1.2.2 Minimum Cam Radius of Curvature

Other critical parameter in cam design is the minimum cam radius of curvature. The minimum cam radius of curvature must be larger than a minimum value to ensure that the follower roller follows the cam counter safely and that the stresses at the contact area between the cam and the follower roller are not excessive. If the follower roller cannot follow the cam counter at some points, a condition known as "undercutting" results[6]. Therefore, the minimum cam radius of curvature must be safely greater than zero to prevent undercutting conditions as well as to reduce contact stresses between cam and follower roller at the contact area (Fig. 4). Calculation of the contact stresses is not studied here as it requires to conduct dynamic force analysis of the mechanism for the determination of force acting between cam and follower roller. However, the Hertz equation can be used to calculate contact stresses[6,7] .

The following expression is used to calculate the pitch curve radius of curvature[8] (ρ):

$$\rho=\frac{\left\{\begin{matrix}r_A^2+r_R^2(1-d\phi/d\varphi)^2-\\2r_Ar_R(1-d\phi/d\varphi)\cos(\phi_r+\phi)\end{matrix}\right\}^{3/2}}{\begin{matrix}r_A^2+r_R^2(1-d\phi/d\varphi)^3-r_Ar_R(1-d\phi/d\varphi)\\(2-d\phi/d\varphi)\cos(\phi_r+\phi)+d^2\phi/d\varphi^2\sin(\phi_r+\phi)\end{matrix}} \quad \ldots (10)$$

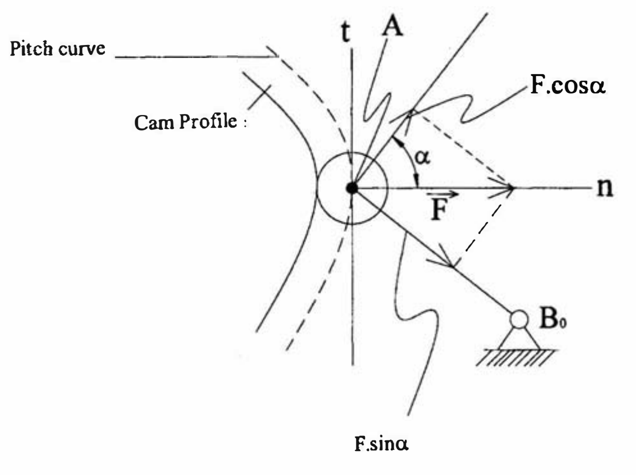


Fig. 3—Pressure angle and force components affecting to the follower [A – Follower roller center, B_0 – Follower center of rotation, α – Pressure angle, t – Tangential direction, n – Normal direction, and F – Force acting from cam to follower]

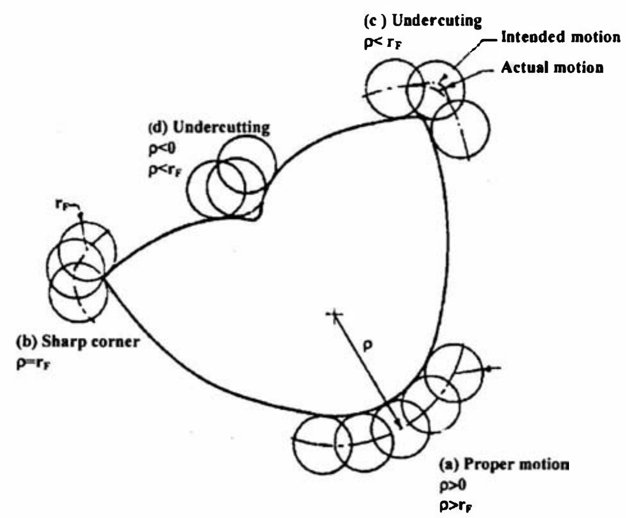


Fig. 4—Cam radius of curvature and undercutting conditions[6]

Cam radius of curvature (ρ_c) can be obtained using the following relationship:

$$\rho_c = \rho - r_F$$

2.1.3 Analytical Method for Determining Cam Profile Coordinates

Analytical method for cam profile determination makes use of the envelope theory[6,7]. $x1$ and $y1$ can be obtained using the following equations derived based on envelope theory according to Fig. 5:

$$x1 = x1_F \pm \frac{r_F}{\sqrt{1+\left(\dfrac{dx1_F/d\varphi}{dy1_F/d\varphi}\right)^2}} \quad \ldots (11)$$

$$y1 = y1_F - \frac{dx1_F/d\varphi}{dy1_F/d\varphi}(x1 - x1_F) \quad \ldots (12)$$

where

$$x1_F = r_A \cos\varphi - r_R \cos(\varphi - \phi_r - \phi)$$
$$y1_F = r_A \sin\varphi + r_R \sin(\varphi - \phi_r - \phi)$$

Similarly, the following expressions were obtained for the second cam profile coordinates ($x2$, $y2$):

$$x2 = x2_F \pm \frac{r_F}{\sqrt{1+\left(\dfrac{dx2_F/d\varphi}{dy2_F/d\varphi}\right)^2}} \quad \ldots (13)$$

$$y2 = y2_F - \frac{dx2_F/d\varphi}{dy2_F/d\varphi}(x2 - x2_F) \quad \ldots (14)$$

where

$$x2_F = r_A \cos\varphi - r_R \cos(\varphi - \phi + \phi_r + \phi_o)$$
$$y2_F = r_A \sin\varphi - r_R \sin(\varphi - \phi + \phi_r + \phi_o)$$

The plus or minus sign in Eqs (11) and (13) establishes the outer or inner envelope respectively.

After determining the follower displacement diagram and the cam mechanism dimensions, matched shedding cam profile coordinates ($x1$, $y1$) and ($x2$, $y2$) can be calculated accurately with respect to the angle of cam shaft by very small steps using Eqs (11)-(14). In the case of negative cam shedding motions, it will be sufficient to determine either ($x1$, $y1$) or ($x2$, $y2$) as there is one cam counter.

3 Results and Discussion

3.1 Follower Displacement Diagrams and Kinematic Characteristics of Motion Curves

The first step in cam design is to determine the follower displacement diagram with respect to the angle of cam shaft rotation. In a cam shedding mechanism, the follower displacement diagram is derived according to the weave. Table 1 shows cam shaft angles for rise or return period of the follower, depending on the pick number in the weave repeat. η =1 means that one loom revolution is fully used for

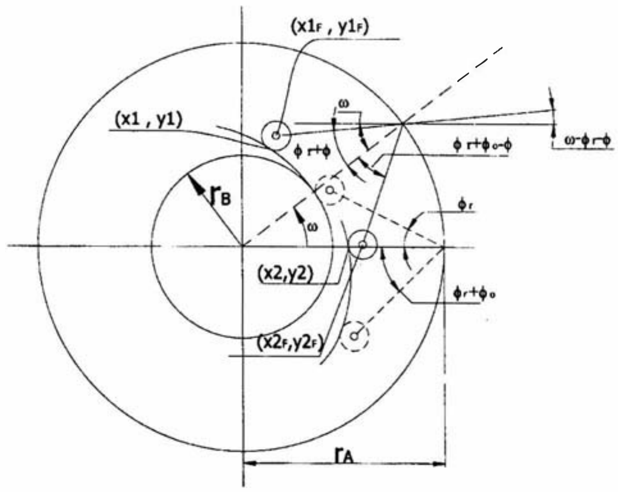


Fig. 5—Matched (conjugate) cam profile coordinates with oscillating roller type of follower [r_B – Cam base circle radius, r_A – Distance between follower and cam center of rotations, φ – Angle of cam shaft rotation, ϕ – Follower angular displacement, ϕ_r – Angular position of follower when follower roller touches cam base circle, ϕ_0 – Follower angular stroke, x1 and y1 – First cam profile coordinates, x2 and y2 – Second cam profile coordinates, $x1_F$ and $y1_F$ – First follower center coordinates, and $x2_F$ and $y2_F$ – Second follower center coordinates]

Table 1—β-angles for different pick numoers in the weave repeat

No. of picks	2	3	4	5	6	7	8
β (η =1), deg	180	120	90	72	60	51.4	45
Dwell (η =1), deg	0	0	0	0	0	0	0
β (η =2/3), deg	120	80	60	48	40	34.2	30
Dwell (η =2/3), deg	60	40	30	24	20	17.2	15

shed change or shaft (i.e. heald frame) movement and no dwell period exists within one loom revolution. In the case of η =2/3, 240 degree of a loom revolution is used for shaft movement and 120 degree for shaft dwell. The reflection of these loom main shaft angles to cam shaft angle is given in Table 1 for weaves up to 8 picks in the weave repeat. If a shaft is required to remain in upper or lower shed position for certain number of picks, the loom revolutions after the first one are completely reserved for shaft dwell in upper or lower shed position. The shaft dwell period after a rise or return motion can be calculated from Eq (3). Figs 6 and 7 are the follower displacement diagrams for four picks plain weave and twill 3/1+1/1 respectively with η =2/3. It is assumed in these diagrams that the heald frame is at the bottom position initially.

Kinematic characteristics of motion curves used for rise and return periods of the follower motion are important not only for controlling shaft movement kinematic characteristics (i.e. shaft displacement, velocity, acceleration and jerk) but also for determining cam mechanism dimensions. Normalised maximum velocity, acceleration and jerk values of four motion curves are given in Table 2 for comparison purpose. Normalised values are obtained by taking β and ϕ_0=1 radian in motion curve equations. The simple harmonic motion has the lowest maximum velocity and acceleration values but at the beginning and at the end of the motion, the jerk goes to infinite (i.e. there is a discontinuity in acceleration curve). This makes the motion curve unsuitable for high speeds as there is an upper or lower dwell in the displacement diagrams of the

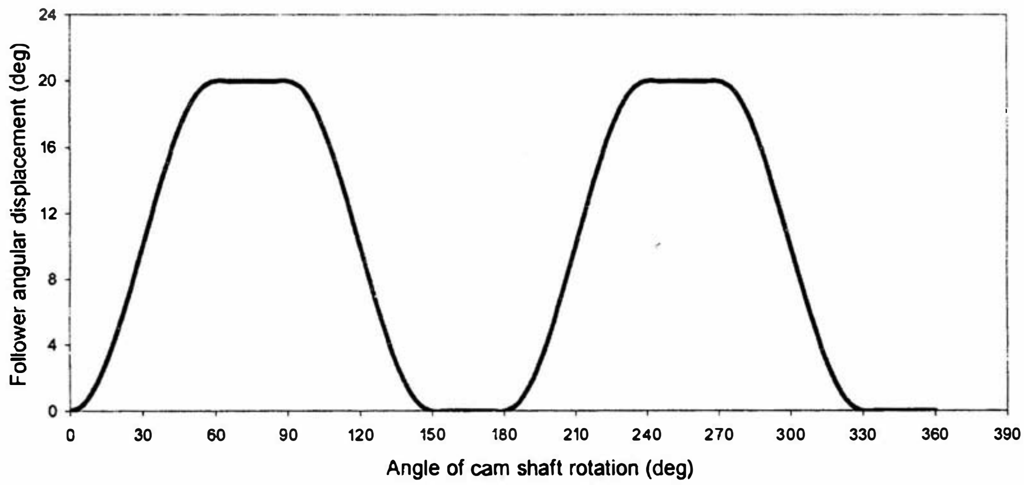


Fig. 6—Follower displacement diagram for four picks plain weave

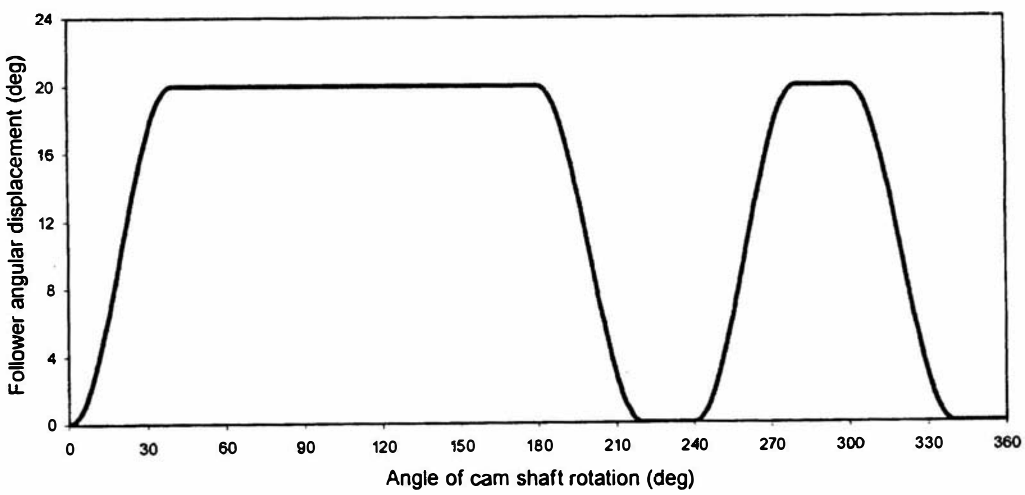


Fig. 7—Follower displacement diagram for twill 3/1+1/1 weave

weaves, except the plain weave cams designed with $\eta = 1$. Although the cycloidal motion curve has the highest maximum velocity and maximum acceleration values among the four motion curves, it can be preferred for high speed applications because there is no discontinuity in its acceleration curve. Modified sine curve has lower maximum velocity and acceleration values than cycloidal motion curve. There is no discontunity in the acceleration curve and the jerk is finite. This makes the motion curve attractive for high speed applications. The modified trapezoidal curve has the lowest maximum acceleration and finite jerk values. But, the maximum velocity of trapezoidal curve is the same as that of cycloidal motion curve. It is also suitable for high speed applications.

3.2 Effect of Motion Curves on Pressure Angle and Cam Radius of Curvature

Table 3 shows the effect of motion curves on the minimum cam radius of curvature and maximum pressure angle for β=30° and β=40°. The maximum pressure angle and the minimum cam radius of curvature are given for both the rise and return periods of the follower motion. Simple harmonic motion produces the best cycloidal motion curve, the worst minimum cam radius of curvature and maximum pressure angle values. The results show that the higher the maximum velocity the higher is the maximum pressure angle and the higher the maximum acceleration the lower is the minimum cam radius of curvature. Therefore, choosing a motion curve with as low a maximum acceleration and maximum velocity as possible helps in designing shedding cams with a higher minimum cam radius curvature and a lower maximum pressure angle.

Table 2—Motion curve kinematic characteristics

Motion curve	Simple harmonic	Cycloidal	Modified sine	Modified trapezoidal
Normalised max. velocity	1.57	2.00	1.76	2.00
Normalised max. acceleration	4.93	6.28	5.53	4.89
Normalised max. jerk	Infinite	40.0	69.0	61.0

3.3 Effect of Follower Angular Stroke on Maximum Pressure Angle and Minimum Cam Radius of Curvature

The maximum angular displacement of the follower was changed between 15 and 25 degrees at three steps and the maximum pressure angle and the minimum cam radius of curvature were calculated. The modified sine and trapezoidal motion curves were used in the calculations with β=40°. The results and mechanism dimensions are given in Table 4. As the follower angular stroke (ϕ_0) increases, the maximum pressure angle increases and the minimum cam radius of curvature decreases in both rise and return periods. This is an expected result because it increases the distance between the base circle and the maximum cam radius and therefore the rise and return periods of cam profile become steeper. The follower angular stroke must be taken into account in determining linkage ratio between followers and shafts. Reducing the follower angular stroke makes the design criteria less critical and the smaller size shedding cams can be obtained.

3.4 Effect of Number of Picks in Weave Repeat on Maximum Pressure Angle and Minimum Cam Radius of Curvature

The number of picks in the weave repeat is the most important factor in the shedding cam design

Table 3—Effect of motion curves on minimum cam radius of curvature and maximum pressure angle
[r_A =160 mm, r_R =80 mm, r_B =95 mm, r_F =30 mm and ϕ_0 = 20°]

Parameter	Simple harmonic		Cycloidal		Modified sine		Modified trapezoidal	
	Rise	Return	Rise	Return	Rise	Return	Rise	Return
				β =30°				
$(\rho - r_F)_{min}$, mm	5.4	6.1	-0.7	4.8	1.6	5.2	4.7	8.4
α_{max}, deg	32.8	-31.7	38.6	-37.9	35.5	-34.6	38.2	-37.8
				β =40°				
$(\rho - r_F)_{min}$, mm	23.5	24.2	14.1	19.9	18.3	22.1	22.0	26.1
α_{max}, deg	26.6	-25.4	31.4	-30.6	28.8	-27.7	30.9	-30.4

Table 4—Effect of follower angular stroke (ϕ_0) on maximum pressure angle and minimum cam radius of curvature [r_A=160 mm, r_R=80 mm, r_B=90 mm and r_F=30 mm]

Parameter	ϕ_0=15°		ϕ_0=20°		ϕ_0=25°	
	Rise	Return	Rise	Return	Rise	Return
			Modified sine			
$(\rho - r_F)_{min}$, mm	22.3	22.6	16.2	18.5	11.9	16.7
α_{max}, deg	28.9	-17.4	32.7	-25.6	36.0	-32.6
			Modified trapezoidal			
$(\rho - r_F)_{min}$, mm	26.3	24.8	20.1	22.4	15.4	20.4
α_{max}, deg	30.6	-19.9	34.6	-28.5	38.0	-35.7

Table 5—Effect of number of picks in the weave repeat on minimum cam radius of curvature and maximum pressure angle [r_A =160 mm, r_R =80 mm, r_B =95 mm, r_F =30 mm and ϕ_0 =20°]

Parameter	3[a]		4[a]		5[a]		6[a]		8[a]	
	Rise	Return	Rise	Return	Rise	Return	Rise	Return	Rise	Return
					η =1					
$(\rho - r_F)_{min}$, mm	88.7	88.9	74.8	75.3	61.6	68.6	49.9	55.6	30.0	34.4
α_{max}, deg	13.9	-13.1	16.5	-15.8	19.4	-18.8	22.4	-21.9	28.1	-27.7
					η =2/3					
$(\rho - r_F)_{min}$, mm	68.0	68.7	49.9	55.6	34.5	39.1	22.0	26.1	4.7	8.4
α_{max}, deg	17.9	-17.3	22.4	-21.9	26.7	-26.3	30.9	-30.4	38.2	-37.8

[a]Number of picks in the weave repeat.

because it determines β angle (the angle of cam shaft rotation which corresponds to the rise and return periods of the follower motion). β angles for η =1 and η =2/3 are given in Table 1 up to eight picks. After many design trials with different mechanism dimensions, the results presented in Table 5 were found using modified trapezoidal motion curve. It is clearly observed that as the number of picks in the weave repeat increases the maximum pressure angle increases and the minimum cam radius of curvature decreases very significantly. This is because of the fact that the decreasing β angle makes the cam profile steeper. Based on this result, it can be concluded that the shedding cam mechanism dimensions must be determined for the maximum number of picks in the weave repeat. The cam shedding mechanism dimensions given in Table 5 were calculated for maximum eight picks with η =2/3. Using these dimensions, a much better maximum pressure angle and the minimum cam radius of curvature values are obtained with η =1. The results in Table 5 explain the earlier reported fact that the maximum number of picks in the weave repeat is limited by the shedding cam size or for a given shedding cam size the maximum number of picks in the weave repeat is limited. But this comment is made with regard to the pressure angle. However, the results in Tables 5 - 10 show that the minimum cam radius of curvature is a critical parameter as much as the pressure angle in the shedding cam design. This observation is not mentioned in the literature. Limiting the maximum number of picks in the weave repeat to 6 allows to design shedding cams with a lower cam base circle radius compared to that given in Table 5. The cam shedding motion manufacturers generally limit the maximum pick number to 6 (refs 9 & 10).

3.5 Effect of Cam Mechanism Dimensions on Maximum Pressure Angle and Minimum Cam Radius of Curvature

Different design trials were carried out to study the effect of cam mechanism dimensions on the maximum pressure angle and the minimum cam

Table 6—Effect of cam base circle radius on minimum cam radius of curvature and maximum pressure angle
[r_A =160 mm, r_R =80 mm, r_F =30 mm and ϕ_0=20°]

Parameter	70[a]		80[a]		90[a]		100[a]	
	Rise	Return	Rise	Return	Rise	Return	Rise	Return
$(\rho - r_F)_{min}$, mm	11.3	3.3	15.6	15.0	20.1	22.4	23.5	30.0
α_{max} , deg	50.7	-20.6	42.3	-24.7	34.6	-28.5	27.2	-32.2

[a] Cam base circle radius in mm.

Table 7—Effect of roller radius (r_F) on minimum cam radius of curvature and maximum pressure angle
[r_A =160 mm, r_R =80 mm, r_B =95 mm and ϕ_0=20°]

Parameter	15[a]		25[a]		35[a]		45[a]	
	Rise	Return	Rise	Return	Rise	Return	Rise	Return
$(\rho - r_F)_{min}$, mm	30.6	30.0	25.1	27.4	18.5	25.0	10.9	23.1
α_{max} , deg	42.3	-24.7	34.6	-28.6	27.2	-32.2	20.0	-35.9

[a] Roller radius in mm.

Table 8—Effect of follower length (r_R) on minimum cam radius of curvature and maximum pressure angle
[r_A =160 mm, r_B =95 mm, r_F =30 mm and ϕ_0=20°]

Parameter	60[a]		80[a]		90[a]		100[a]	
	Rise	Return	Rise	Return	Rise	Return	Rise	Return
$(\rho - r_F)_{min}$, mm	30.0	24.3	22.0	26.1	18.2	25.5	14.8	25.2
α_{max} , deg	36.6	-12.9	30.9	-30.4	29.8	-36.4	29.5	-41.4

[a] Follower length in mm.

Table 9—Maximum pressure angle and minimum cam radius of curvature
[r_A =160 mm, r_R =90 mm, r_B =85 mm, r_F =30 mm and ϕ_0=20°]

Parameter	Modified sine				Modified trapezoidal			
	40[a]		45[a]		40[a]		45[a]	
	Rise	Return	Rise	Return	Rise	Return	Rise	Return
$(\rho - r_F)_{min}$, mm	11.5	14.9	18.8	22.0	15.0	18.4	22.4	26.0
α_{max} , deg	34.0	-30.8	31.4	-28.0	36.3	-34.0	33.4	-30.9

[a] β value in deg.

Table 10—Maximum pressure angle and minimum cam radius of curvature
[r_A =180 mm, r_R =110 mm, r_B =90 mm, r_F =35 mm and ϕ_0=20°]

Parameter	Modified sine				Modified trapezoidal			
	40[a]		45[a]		40[a]		45[a]	
	Rise	Return	Rise	Return	Rise	Return	Rise	Return
$(\rho - r_F)_{min}$, mm	8.0	12.7	15.8	20.5	11.7	16.5	19.6	24.8
α_{max} , deg	35.2	-34.7	32.3	-31.8	37.8	-38.0	34.7	-34.9

[a] β value in deg.

radius of curvature (Tables 6 -8). All the values were calculated using the modified trapezoidal motion curve with β =40°. Table 6 show that the maximum pressure angle decreases in the rise period and increases in the return period with the increasing values of cam base circle radius and the minimum cam radius of curvature increases in both the periods with the increasing values of the cam base circle radius. A similar effect is observed when the roller radius is changed. Increasing the roller radius causes the maximum pressure angle in the rise period to decrease and in the return period to increase. The minimum cam radius of curvature decreases in both rise and return periods with an increase in roller radius (Table 7). This has positive effect on the maximum pressure angle but negative effect on minimum cam radius of curvature. On comparing the results reported in Tables 6 and 7, it is observed that the same maximum pressure angle is obtained as the sum of cam base circle radius and roller radius remains constant. For a given maximum pressure angle, smaller size shedding cams can be designed by keeping the roller radius as large as possible on condition that the minimum cam radius of curvature remains safely greater than zero to prevent undercutting and to reduce contact stresses. The effect of follower length on the maximum pressure angle and the minimum cam radius of curvature is given in Table 8. Increasing the follower length decreases the minimum value of cam radius of curvature in the rise period and has no significant effect in the return period. The maximum value of pressure angle decreases in the rise period and increases in the return period with increasing follower length.

Comparison of the maximum value of pressure angle in the rise and return periods reveals that as the mechanism dimensions increase, the maximum value of the pressure angle decreases in the rise period while it increases in the return period. The minimum cam radius of curvature is generally different in the rise and return periods. Putting the data of Tables 6 - 8 together, it can be concluded that when the maximum values of pressure angle and the minimum values of cam radius of curvature in the rise and return periods are made closer to each other by choosing mechanism dimensions properly, a lower maximum pressure angle and a higher minimum cam radius of curvature can be obtained.

Further design trials were carried out with β=40° and β=45° using modified sine and modified trapezoidal motion curves. These β values correspond to η =7/8 and η =1 for maximum eight picks in the weave repeat and to η =2/3 and η =3/4 for maximum six picks in the weave repeat. The cam mechanism dimensions, the maximum pressure angle and the minimum cam radius of curvature are given in Table 9. In this case, the maximum pressure angle and the minimum cam radius of curvature are safely satisfied with a lower cam base circle radius.

Finally, the distance between the cam shaft center and the center of follower rotation (r_A) was increased to 180 mm and other cam mechanism dimensions (r_B ,

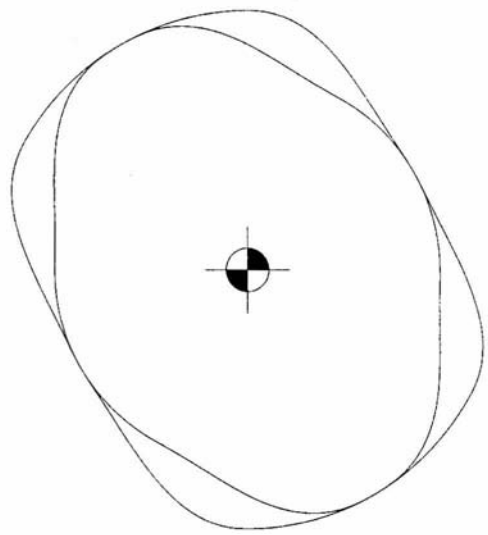


Fig. 8—Four picks plain weave matched cam plot

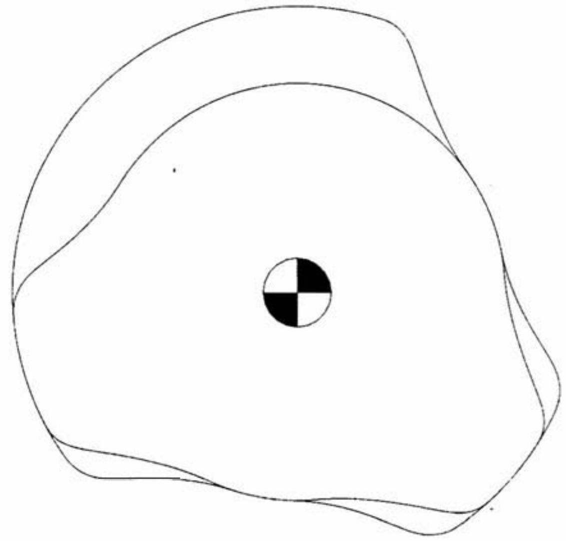


Fig. 9—Twill 3/1+1/1 matched cam plot

r_R, r_F) were determined so as to satisfy the maximum pressure angle and the minimum cam radius of curvature criteria. The results are presented in Table 10. Increasing r_A causes an increase in other cam mechanism dimensions to satisfy the maximum pressure angle and the minimum cam radius of curvature. As a result, larger size shedding cams are needed.

3.6 Shedding Cam Design Examples

Four picks plain weave and twill 3/1+1/1 shedding cam plots are given in Figs 8 and 9 with the scale of 1:2. The follower displacement diagrams for these shedding cams are shown in Figs 6 and 7. The cam shedding mechanism dimensions given in Table 9 are used in the four picks plain weave cam design and in the twill 3/1+1/1 cam design, the cam mechanism dimensions given in Table 4 are used.

4 Conclusions

4.1 With the maximum pick number in the weave repeat, the maximum pressure angle and the minimum cam radius of curvature become critical. Therefore, shedding cam mechanism dimensions must be determined for the weave with maximum pick number in the weave repeat.

4.2 Motion curves affect the maximum pressure angle and the minimum cam radius of curvature. Motion curves with a lower maximum velocity and a lower maximum acceleration produce a lower maximum pressure angle and a higher minimum cam radius of curvature respectively and vice versa. These motion curve characteristics are also required for lower maximum shaft velocity and acceleration.

4.3 There is a difference in the maximum pressure angle and the minimum cam radius of curvature values in the rise and return periods of the follower motion, depending on the cam mechanism dimensions. In general, the increase in maximum pressure angle in the rise period causes a decrease in the return period and vice verse. These values can be made closer to each other if the cam mechanism dimensions are properly chosen.

4.4 The cam shedding mechanism designed with maximum pick number of 6 or 8 can be obtained with a maximum cam diameter of 22-26 cm. This complies with the maximum shedding cam size used on industrial weaving machines. Especially with maximum six picks in the weave repeat, a very safe maximum pressure angle and a minimum cam radius of curvature can be obtained using this size of shedding cams.

4.5 Symmetrical motion curves are used in the shedding cam design. Asymmetrical motion curves and shaft dwell at only bottom position are required in some practices. As long as the shedding cam follower displacement diagram is formed to meet these motion curve requirements, the method presented here can be used to design this kind of shedding cams as well.

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