Design and Analysis of Picking Cam for High Speed Shuttle Loom

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***Abstract*— Shuttle Loom is widely used in Indian Power loom industry due to its versatility to weave nearly all kind of fabrics. The only problem with this machine is that the speed is low compared to the other weaving machines. It can be enhanced by improving the speed of shuttle loom .This needs to increase the speed of picking mechanism which plays the main role in weaving of fabric. Detailed study was done on the working of the picking mechanism of present loom. Experiments have been carried out to study the motion of a shuttle in the present machine. Kinematic and Dynamic analysis are done for the present loom. Cam profile has been generated and Kinematic and Dynamic Analysis are carried out for the designed cam. A prototype is manufactured and from the practical experiment it has been observed that the accuracy, speed and force of picking mechanism have been increased.**

Keywords— Picking Mechanism, Kinematic and Dynamic Analysis, Cam design, Shuttle motion.

I. INTRODUCTION

Weaving is a method of fabric production in which two distinct sets of yarns or threads are interlaced at right angles to form a fabric or cloth. Loom is a machine used to weave fabric. There are mainly five operations of weaving namely (Shedding) opens the warp sheet into layers, (picking) causes the shuttle carrying weft to be propelled from one end of the loom to another,(Beat up) motion lays the previously laid weft to the fell of the cloth,(take up and let off)motions. Picking is the 2^{nd} primary weaving motion. The main function of the picking mechanism is to insert the weft through warp [1]. A pick cam toe is used to give motion to the picking mechanism in shuttle loom. In order to increase the length of the loom or to increase the speed of the loom, the speed of the shuttle has to be increased. Higher force, speed and accuracy can be attained by designing a cam follower system.

II. THEORETICAL AND EXPERIMENTAL ANALYSIS OF PRESENT LOOM

A. *Theoretical analysis:*

Theoretical analysis of the present mechanism at 98 r.p.m shows that the time for the travel of shuttle is 110 degrees of crank shaft rotation and thus the relevant time is 0.1867 second [2], the maximum velocity of shuttle comes up to be 9.48 m/s. Now as the shuttle is accelerated by the picking stick in 10 inches. The acceleration is 176.9 m/s^2. Considering the mass of the shuttle to be 0.450kg, the force required to accelerate the shuttle is 79.60N.

B. *Experimental analysis of motion of shuttle:*

A stroboscope was connected with programmable logic controller (PLC) guided by a modulator switch in experimental setup. A proximity sensor was mounted nearby crank shaft which senses the knob on crank shaft. A signal at 0 degree crank rotation was taken with the help of proximity sensor. This signal was input to the PLC to create a flash of a stroboscopic light. The flash can be delayed with the help of a modulator switch. This gives delayed flashes with respect to 0 degree position of crank. This facilitates to locate the shuttle while it is moving from one end to the other end. Same delayed flashes were used to locate the crank rotation.

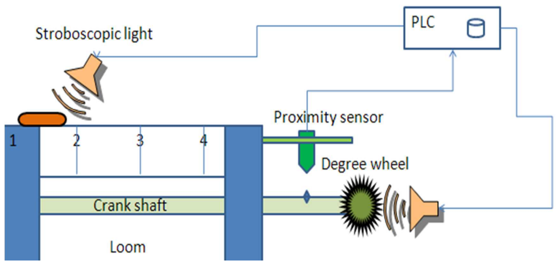

Figure 1. Experimental analysis of motion of shuttle.

For different crank rotation (like, 10^0 20^0 till 110^0) the location of shuttle was found and has been plotted as shown in the figure 2.1. For experimental setup shuttle was made to start at 0 degree crank rotation.

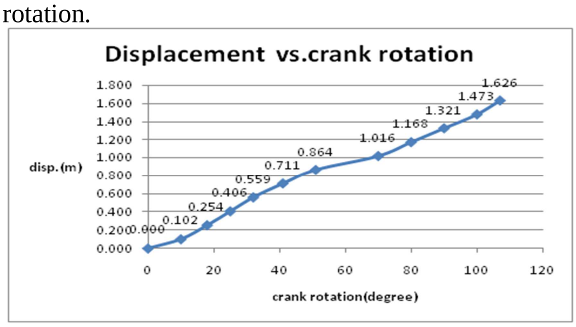

Figure 2.1.Displacement of shuttle

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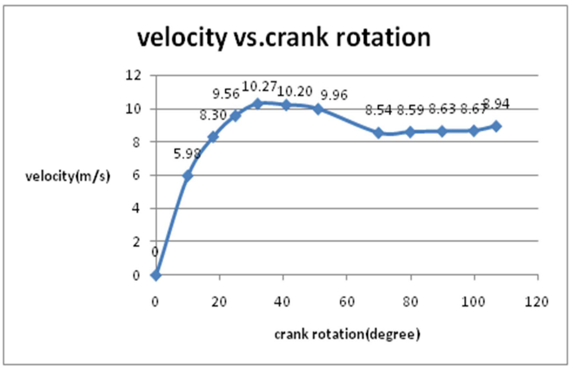

Figure 2.2.Velocity of Shuttle

As shown in the Figure 2.1 it can be seen that the displacement is uniform up to 30 degree of crank shaft rotation and there after the rate of change of displacement decreases and again follows a uniform path. Here the total displacement of the shuttle is 1.62 meter .Similarly in the Figure 2.2. It can be observed that the velocity increases constantly till 30 degree of crank shaft rotation and reaches up to a maximum of 10.27 m/s and then it follows a linear path i.e. the rate of change of velocity decreases uniformly. And so the average velocity comes out to be 8.14 m/s. The velocity of the shuttle remains nearly constant during its passage in shed, so the effect of friction can be neglected.

C. Theoretical analysis of picking cam profile:

The picking cam is mounted on the bottom shaft which is connected to a crank with gear ratio of 2:1, so the bottom shaft rotates once when the crank shaft rotates twice. The profile of cam was generated using solid edge and the profile was divided into equal number of parts of 5 degree. Following results were obtained as shown in the figures 3, 4, 5.

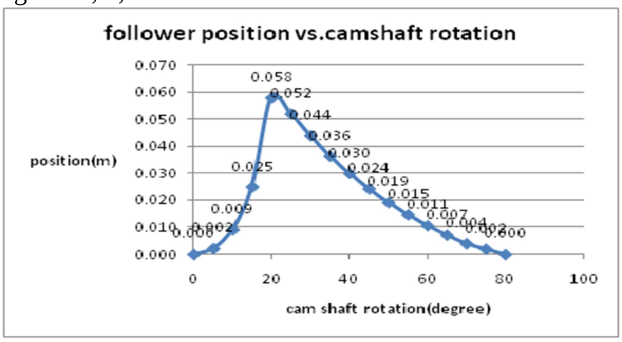

Figure 3.Displacement of Follower

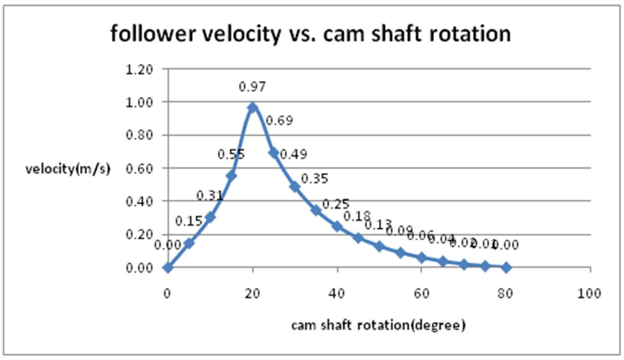

Figure 4.Velocity of follower

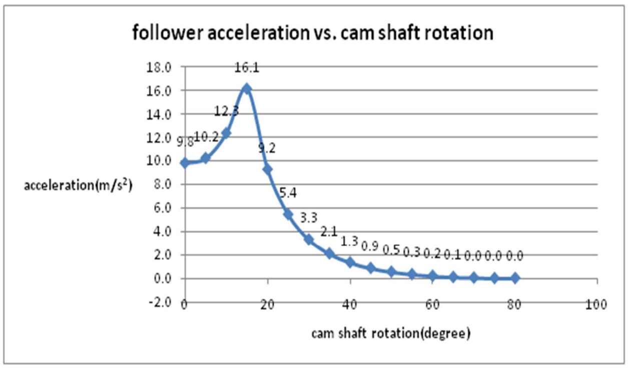

Figure 5.Acceleration of follower

It has been observed from the Figure 3. that the displacement of the follower increases constantly up to 20 degrees of cam shaft rotation and reaches up to a maximum of 0.058 m/s. Similarly from the Figure 4, it can be seen that the velocity of the follower also increases constantly up to 20 degrees of cam shaft rotation and reaches up to a maximum of 0.97 m/s and thereafter it starts decreasing till 0.Here the acceleration reaches a maximum of 16 m/s^2 at nearly 20 degrees of cam shaft rotation as shown in Figure 5.

D. Schematic presentation of picking mechanism:

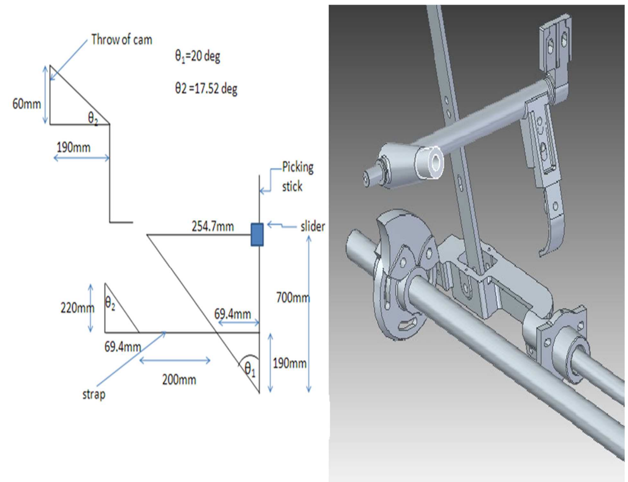

Figure 6. Schematic presentation and assembly of picking mechanism.

The picking mechanism is represented schematically in Figure 6.The movement of picker is 254 mm and the length of the picking stick is 700 mm, the angle of oscillation θ_1 is found to be 20 degree as shown in Figure 6.Now as the strap is connected at 190 mm upwards from the pivot of picking stick the required movement of strap is 69.4 mm. The strap being connected to the side shaft with flange length of 220 mm makes it to turn 17.52 degrees and the movement at the other end of the strap becomes 60 mm as the other flange is 190 mm long. This gives the displacement of the follower achieved through cam rotation. Now let ω_1 equals to angular velocity of stick, so we have

$$\omega_1 = \frac{\theta_1}{t} \qquad (2.1)$$

$$= 6.017 \text{ rad/sec.}$$

Similarly calculating the velocity and acceleration for whole mechanism, finally the velocity of follower comes out to be 0.986 m/s and acceleration of follower equals to $17 m/s^2$. From the above calculations it has been found that the velocity and acceleration found in section (I.B) are matching with the present calculations and the values occurs at 20 degrees of cam shaft rotation, when follower is fully raised. As the force required to throw the shuttle is 79.60N, the force at the cam is 339N.Now calculating the torque of motor with 0.5hp from the equation [3].

$$P = \frac{2\pi NT}{60000} \qquad (2.2)$$

$$T = \frac{PX60000}{2\pi N}$$

$$= 4.97 \text{ N-m}$$

And so the torque in the bottom shaft is 69.26 N-m and the force at the tip of the cam becomes 513 N which is higher than the obtained force. This difference occurs because spring force and friction has been neglected. In present mechanism this is overcome by adjusting the lug strap.

III. DESIGN AND SELECTION PROCEDURE FOR NEW CAM

The speed of the loom is to be increased up to 180 r.p.m and the length is to be increased by 1.8 meters. Various cam profiles have been generated using Matlab for co-relating the present cam profile. It has been found that the parabolic cam profile shown in figure 7 matches the requirements and here only h/2 of the profile is considered by keeping in mind maximum velocity. The displacement of the follower takes place in 14.5 degrees of crankshaft rotations.

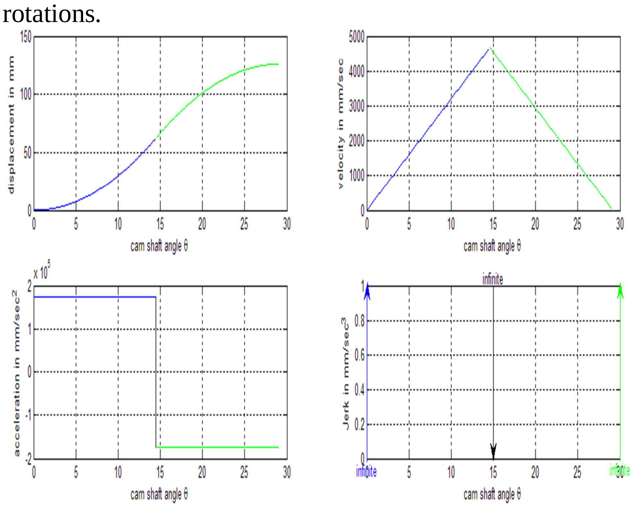

Figure 7.Parabolic cam

The problem with the parabolic cam is that the jerk is infinite at start, middle and the end. So in order to compensate with this, a circular arc is combined at the starting as shown in the figure 8. Such small modifications are used to retain simplicity and ease in manufacturing [4].The X and Y coordinates of whole profile are found by the following equation:

X-co-ordinate = (base circle radius + displacement)

* cos (θ * pi/180) (3.1)

Y-co-ordinate = (base circle radius + displacement)

* sin (θ * pi/180) (3.2)

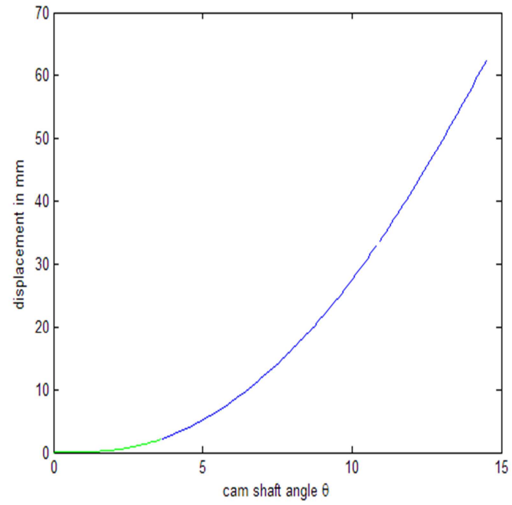

Figure 8. Modified parabolic cam profile

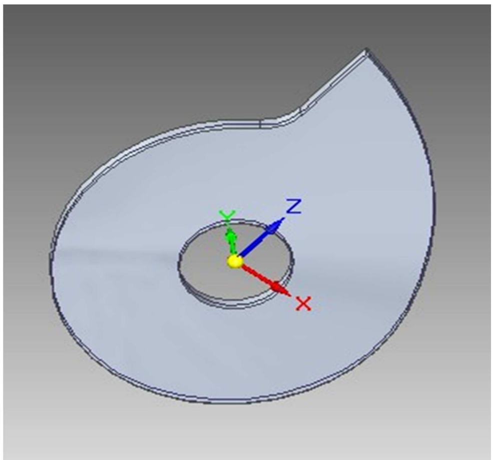

Figure 9. Model of designed cam

The profile thus generated in matlab is exported to solid edge and modeled as shown in figure 9.

A. Kinematic analysis of new cam follower :

From the design of parabolic cam done above, kinematic analysis of cam and follower has been carried here. Table.1. shows the displacement, velocity and acceleration of follower with respect to angle of cam rotation.

Table1. Kinematic analysis of new cam follower

Angle of rotation of cam in degree (θ)	Disp. of follower in mm(y)	Velocity of follower in mm/sec ($\dot{y}$)	Acceleration of follower in mm/sec^2 ($\ddot{y}$)
0	0.00	0.00	172.840
1	0.30	0.32	172.840
2	1.19	0.64	172.840
3	2.67	0.96	172.840
4	4.74	1.28	172.840
5	7.41	1.60	172.840
6	10.67	1.92	172.840
7	14.52	2.24	172.840
8	18.97	2.56	172.840
9	24.01	2.88	172.840
10	29.64	3.20	172.840
11	35.86	3.52	172.840
12	42.68	3.84	172.840
13	50.09	4.16	172.840
14	58.09	4.48	172.840
14.5	62.31	4.64	172.840

Here the velocity reaches a maximum of 4.64 m/s from 0 m/s with constant acceleration of 172.840 m/s and the follower gets lifted maximum of 62.3 mm.

B. Dynamic analysis of new cam follower:

From the displacement, velocity and acceleration found in the above section and from figure 10 [5], it can be found that

$$\dot{y} = \omega_2 \times O_2 P \quad (3.4)$$

$$F_{32} = P + (-m\dot{y}) \quad (3.5)$$

$$T_2 = O_2 P \times F_{32} \quad (3.6)$$

Here considering the mass of the follower =0.5 kg and constant acceleration 172.8 m/s^2**,** so the inertia force equals to 86.42 N. Also considering force due to load, p=68.67 N the resultant force equals to 155.1 N from equation(3.5).Torque applied by follower on cam is found by using equation(3.6) and is as shown in figure 11.

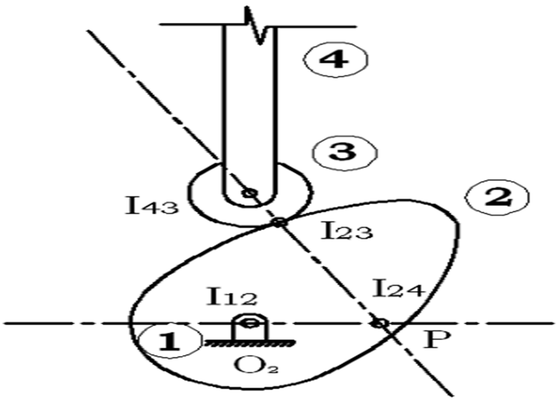

Figure 10.Instantaneous centers of cam

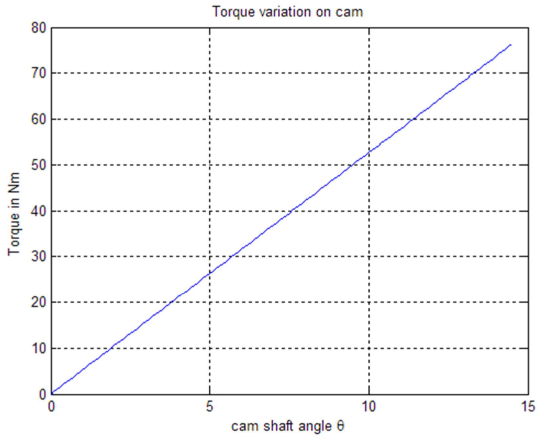

Figure 11.Torque variation on cam

IV. Simulation of Designed Mechanism

Assembly of the picking mechanism has been done in Pro-E and it has been found that the velocity of the picker reaches a maximum of 19.3 m/s as shown in figure 12 and the acceleration reaches up to 556 m/s^2 as shown in figure 13.

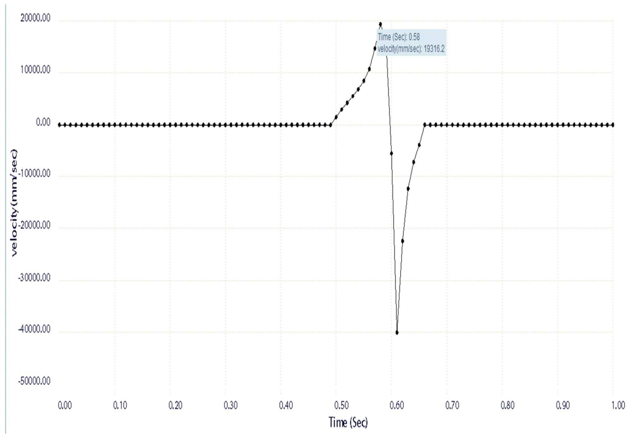

Figure 12.Velocity of picker

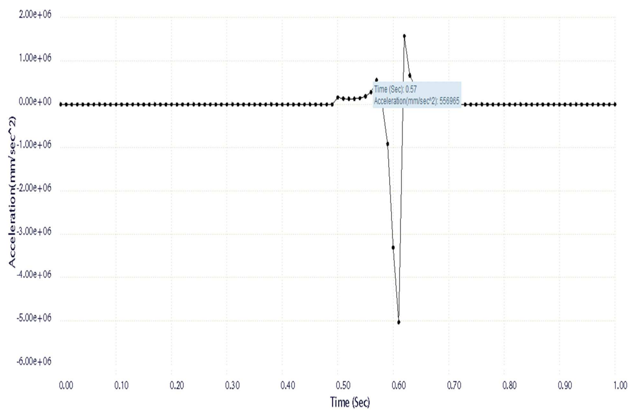

Figure 13.Acceleration of picker

The cam that has been modeled in solid edge is transferred to Ansys Workbench and load steps are applied for various intervals of time.

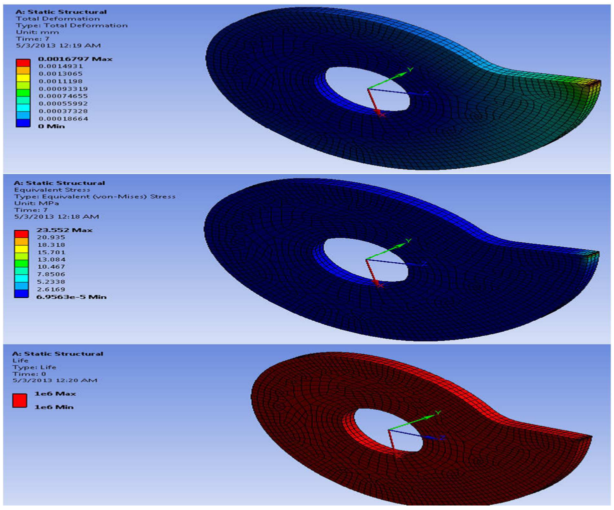

Figure 14.Analysis of cam using Ansys

From the figure 14.it can be seen that the deformation is nearly negligible and the von misses stress is equal to 23.55mpa at the tip of the cam also the cam design is safe for 10^6cycles.

V. MANUFACTURING AND TESTING OF PROTOTYPE

From the calculations done in previous sections, a cam was cut in a vertical CNC machine from a 10mm, SS304 plate as shown in figure 15 and then it was mounted on cam shaft and made to run at 180 R.P.M, as shown in the figure 16, it runs smoothly and the required velocity and force of the picker is achieved.

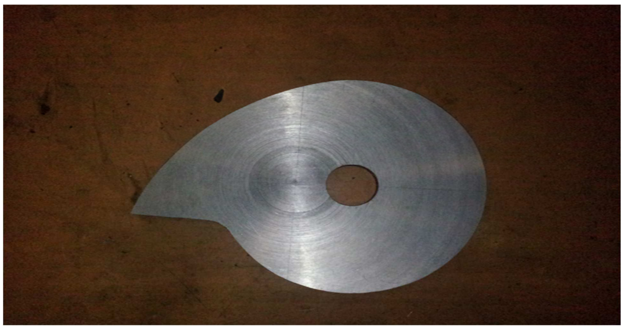

Figure 15.Prototype of cam

Figure 16.Speed testing of new mechanism

VI. CONCLUSION

The designing of pick cam toe by the use of modified parabolic profile will help the shuttle loom industry to get precise movement of the shuttle and will also help in increasing the speed of loom. Moreover it will help in eradicating the olden ways of trial and error method in designing of picking cam.

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