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Dynamic Analysis of Beat-up Mechanism for High Speed Shuttle Loom

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***Abstract*— To increase the productivity of clothes without selvage, shuttle loom is necessary, which produce clothes at lower cost. The only drawback of shuttle loom is its low speed, current shuttle looms are running at 120 ppm(pick per minute), and due to this its productivity is less. Therefore, in this paper kinematic and dynamic analysis has been done for present and proposed mechanism, to design high speed Beat-up mechanism, which is 3rd primary operation of shuttle loom. Basically beat-up mechanism is the reciprocating motion of the reed which is used to push every weft thread to the fabric fell.**

***Keywords*—Beat-up mechanism, shuttle loom, sley mechanism, weft insertion period.**

I. Introduction

Beat-up mechanism is also known as sley mechanism. Beat-up mechanism is very much similar with the crank-rocker mechanism. But here one wooden sley with reed is attached on the top of the rocker, which is used to push that weft into to the warp. In this paper, kinematic and dynamic analysis has been done for present beat-up mechanism at a speed of 120 rpm to 180 rpm respectively. Then wooden sley is replaced by aluminum channel, and for that new proposed mechanism, kinematic and dynamic analysis has been done.

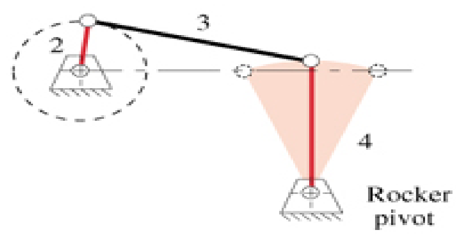

Fig. 1 Schematic view of Beat-up Mechanism

II. Synthesis of Beat-up Mechanism

The motion of the sley (rocker) during the beat-up operation is called Oscillation angle. It is also known as sley swing angle. It is generally between 10° to 30°. Besides the sley angle, the mechanism must satisfy some mechanical criteria such as optimum transmission angle and optimum crank-coupler ratio [1].

A. Optimum transmission angle (μ)

It is defined as the angle between the output link (rocker arm) and the coupler, shown in fig.2. For a crank-rocker mechanism, the angle will be constantly changing during the motion cycle of the mechanism, so the optimum value of the transmission angle is 90^0 [4].

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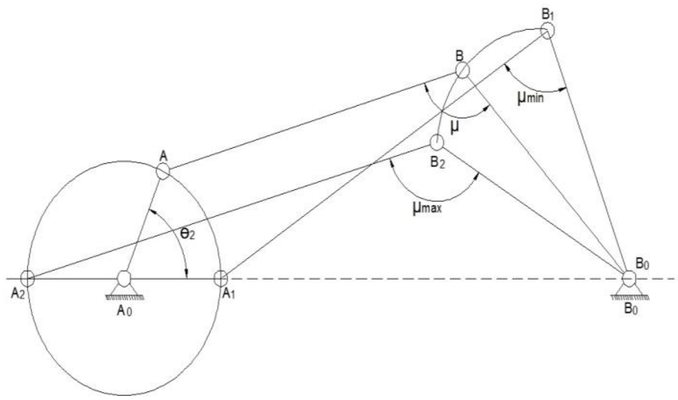

Fig.2. Transmission Angle of Crank-rocker Mechanism

B. Optimum crank coupler ratio

The ratio r/l, where r is the radius of the crank circle and l is the length of the crank arm, is called the sley eccentricity ratio (e). If the sley eccentricity is increased then the sley remains for longer time near its most backward position, and more time is available for the passage of the shuttle. But if the sley eccentricity increases many mechanical problems occur in loom. So, for this reason, mostly loom makers tend to avoid eccentricity ratio greater than about 0.3(particularly in shuttle loom) [3][4].

III .Design Calculation for the Link Lengths of Beat-up Mechanism

In this method, the link lengths (i.e. r_2, r_3, r_4) of a crank rocker mechanism are determine for a given sley angle (Ψ) and the transmission angle (μ). Here r_1 is an independent design parameter. And r_2, r_3, r_4 are three parameters calculated from the below equations [1].

$$\frac{r_2}{r_1} = \frac{r_4}{r_1}\sin\frac{\psi}{2} \qquad \ldots(1)$$

$$\frac{r_3}{r_1} = \frac{\sin\frac{\psi}{2}}{\cos\mu_{min}} \qquad \ldots(2)$$

$$\frac{r_4}{r_1} = \frac{\sqrt{1-\left(\frac{r_3^2}{r_1^2}\right)}}{\sqrt{1-\sin\frac{\psi}{2}\sin\frac{\psi}{2}}} \qquad \ldots(3)$$

TABLE I
Based on Minimum Transmission Angle (μ)(Please Refer Last Page)

IV. Kinematic Analysis for the Crank-Rocker Mechanism

As shown in Table no.I, mostly selection is done by the sley eccentricity ratio, because for the smooth weaving operation, eccentricity ratio 0.3 should be taken by loom designers. After the dimensions of the beat-up mechanism are calculated, the position, velocity and acceleration are determined by using loop closer equation [5][6].Here current loom dimensions are selected for kinematic analysis, which are mentioned in Table no.II.

TABLE III
Kinematic Analysis of Current Beat-up Mechanisms

μ=94°	**current loom**
$r_1 = 0.778m$	Ψ=13.34°
r_2	0.065 m
r_3	0.445 m
r_4	0.578 m
e $=\frac{r_2}{r_3}$	0.15

A. *Displacement, Angular Velocity and Angular Acceleration of Sley Mechanisms*

In this analysis, we have assumed that the weft insertion is completed during the period when the sley is in the "back zone" between its original (most backward) position and one-half of its maximum displacement. The corresponding period is referred to as the insertion period [2]. As shown in fig.3.(a), the weft insertion period of current loom is 91°+(360°- 277°)=174°.

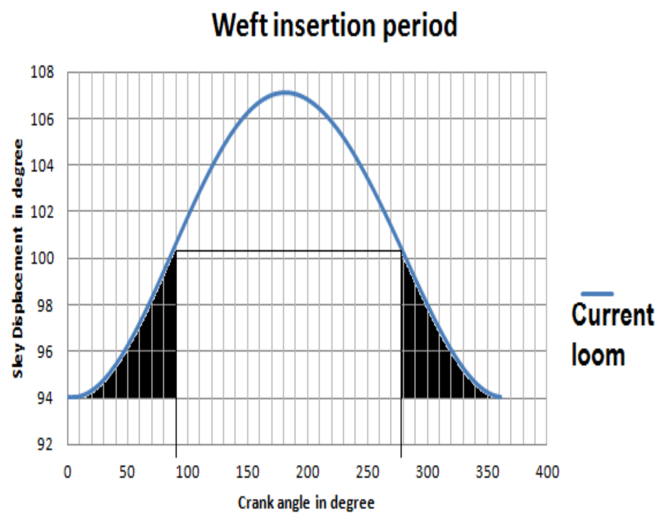

Fig.3 (a) Sley Displacement

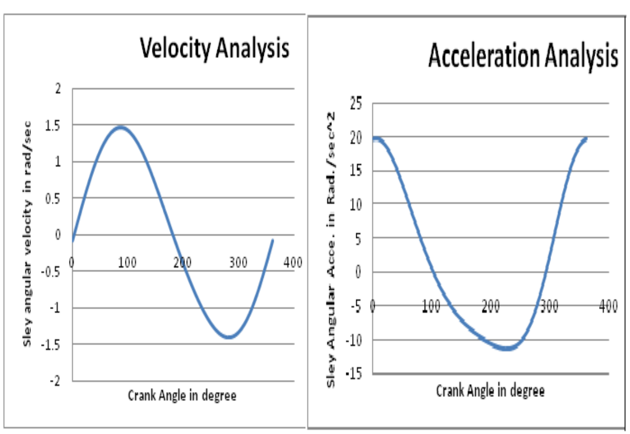

Fig.3 (b) Sley Angular Velocity and Angular Acceleration

V. Dynamic Force Analysis of Beat-up Mechanism

In order to design links and joints one must determine the worst loading conditions of each link and joint. In order to select the driving motor characteristics, input torque for the whole cycle is required. In such cases analytical methods suitable for numerical computation is utilized. After kinematic analysis of the beat-up mechanism, now using the below equations,forces of each link has to be calculated [7].

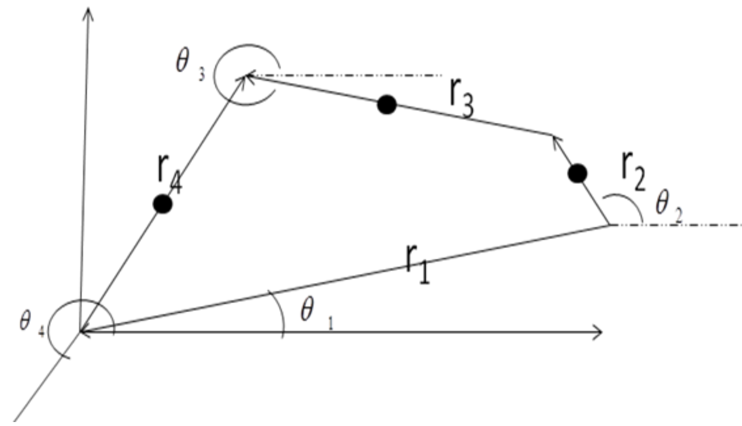

Fig.4 Acceleration of the centers of gravity

A. *Free Body Diagram of Beat-up Mechanism*

The system is in dynamic equilibrium under the action of these forces. We would like to determine the input Torque and the joint forces. The free body diagrams of each moving link can be drawn and the equilibrium equations can be written

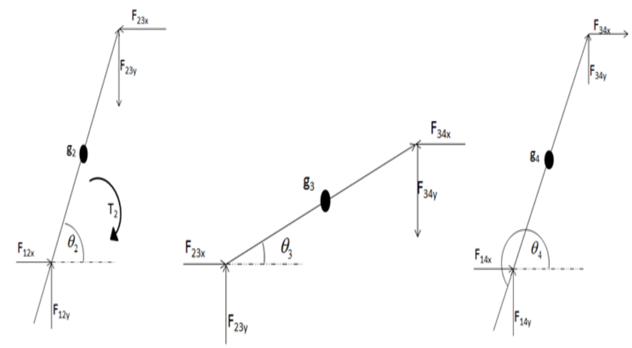

Fig.5 Free body diagrams of Beat-up Mechanism

1 For link number 2:

$$-F_{23x} + F_{12x} - m_2 a_{G2x} = 0 \quad ...(1)$$

$$-F_{23x} + F_{12x} - m_2 a_{G2x} = 0 \quad ...(2)$$

$$F_{23x}\, r_2 \sin(\pi - \theta_2) + F_{23y} r_2 \sin\left(-\frac{\pi}{2} - \theta_2\right) + T_2 - I_2 \alpha_2 - m_2 a_{G2x} g_2 \sin(-\theta_2) - m_2 a_{G2y} g_2 \sin\left(\frac{\pi}{2} - \theta_2\right) = 0 \quad ...(3)$$

2 For link number 3:

$$F_{23x} - F_{34x} - m_3 a_{G3x} = 0 \qquad ...(4)$$

$$F_{23y} - F_{34y} - m_3 a_{G3y} = 0 \qquad ...(5)$$

$$F_{34x}\, r_3 \sin(\pi - \theta_3) + F_{34y} r_3 \sin\left(-\frac{\pi}{2} - \theta_3\right) - I_3\alpha_3 - m_3 a_{G3x} g_3 \sin(-\theta_3) - m_3 a_{G3y} g_3 \sin\left(\frac{\pi}{2} - \theta_3\right) = 0 \qquad ...(6)$$

3 For link number 4:

$$F_{34x} + F_{14x} - m_4 a_{G4x} = 0 \qquad ...(7)$$

$$F_{34y} + F_{14y} - m_4 a_{G4y} = 0 \qquad ...(8)$$

$$F_{34y}\, r_4 sin\left(\frac{\pi}{2} - \theta_2\right) + F_{34x} r_4 \sin(-\theta_3) - I_4\alpha_4 - m_4 a_{G4x} g_4 \sin(-\theta_4) - m_4 a_{G4y} g_4 \sin\left(\frac{\pi}{2} - \theta_4\right) = 0 \qquad ...(9)$$

Hence, we obtain nine linear equations and nine unknowns (F_{l4x}, F_{l4y}, F_{34x}, F_{34y}, F_{23x}, F_{23y}, F_{l2x}, F_{l2y} and T_{12}). If a computer subroutine for the matrix solution is available, these equations can be solved directly for the unknowns [8]. However, it is much simpler to solve equations (6) and (9) simultaneously for F_{34y} and F_{34x} and then solve for each unknown from the remaining equations.

$$A = I_4\alpha_4 + m_4\, g_4[a_{G4y} \cos(\theta_4) - a_{G4x}\sin(\theta_4)]$$

$$B = I_3\alpha_3 + m_3\, g_3[a_{G3y} \cos(\theta_3) - a_{G3x}\sin(\theta_3)]$$

$$F_{34x} = [Aa_3\cos\theta_3 + Ba_4\cos\theta_4] / [a_3a_4\sin(\theta_3 - \theta_4)]$$

$$F_{34y} = [Aa_3\sin\theta_3 + Ba_4\sin\theta_4] / [a_3a_4\sin(\theta_3 - \theta_4)]$$

$$F_{14x} = -F_{34x} + m_4a_{G4x}$$

$$F_{14y} = -F_{34y} + m_4a_{G4y}$$

$$F_{23x} = F_{34x} + m_3a_{G3x}$$

$$F_{23y} = F_{34y} + m_3a_{G3y}$$

$$F_{12x} = F_{23x} + m_2a_{G2x}$$

$$F_{12y} = F_{23y} + m_2a_{G2y}$$

$$T_2 = F_{23y}a_2\cos(\theta_2) - F_{23x}a_2\sin(\theta_2) + I_2\alpha_2 + m_2 g_2[a_{G2y}\sin(\theta_2) - a_{G2x}\sin(\theta_2)]$$

With the help of above equations, the crank torque and the beat-up force of current mechanism can be calculate.

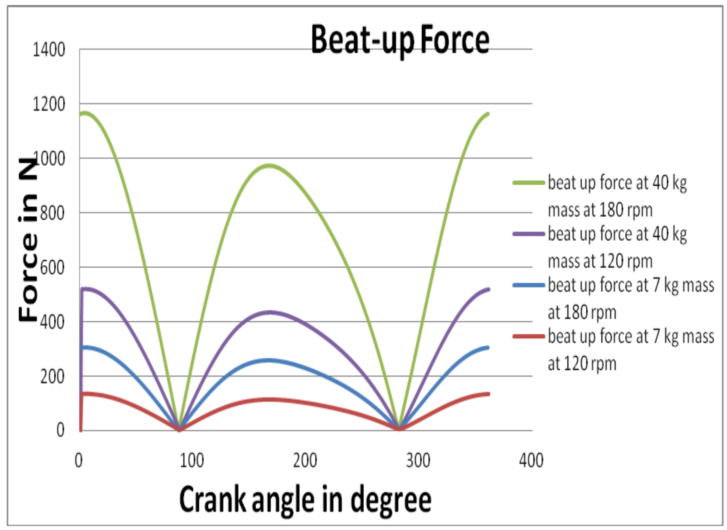

Fig.6 Comparison of Beat-up Forces at different speed &weight

VI. Results and Discussions

Fig.7 (a) Present Loom with wooden Sley

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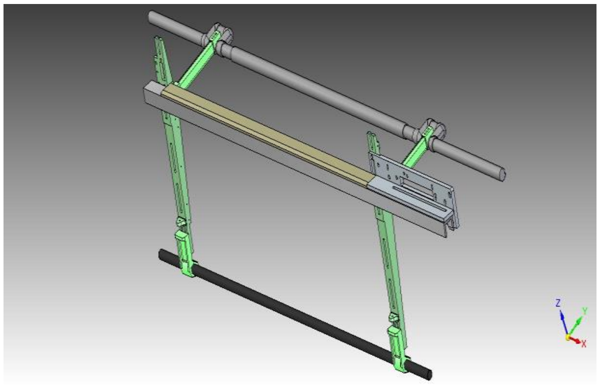

Fig.7 (b) Proposed Loom with Aluminum-channel Sley

TABLE IIIII
COMPARISON BETWEEN PRESENT AND PROPOSED LOOM)(PLEASE REFER LAST PAGE)

VII. CONCLUSIONS

In order to understand the working of beat-up mechanism and motion of sley, kinematic and dynamic analysis of current beat up mechanism has been carried out at a speed of 120rpm, also it has been done for 180 rpm respectively. It was found that due to high speed of 180 rpm various force in the beat up mechanism got increased, which could result in mechanical breakage. To reduce the forces the wooden-sley weighing 80 kg is replaced by the aluminum channel weighing 14 kg; here the proposed mechanism then analyzed for dynamic loading conditions, due to this the forces are reduced. So, the proposed mechanism can run easily at 180rpm without changing the driving motor characteristics. Here in proposed beat-up mechanism, the dimensions of linkages are to be taken as it is.

References

[1] R. Eren and A. Aydemir. "An Approach to Kinematic Design of Four-bar Sley Drive Mechanisms in Weaving", The Journal of The Text.Inst., Turkey, 2004. P 193-205.

[2] Youjiang Wang and Hui Sun," computer Aided Analysis of Loom Beating-up Mechanism", The Journal of The Text.Inst., 1998. P 631-634.

[3] H.I.Celik & M.Topalbekiroglu, "Kinematic analysis and synthesis of the beat up mechanism for Handmade Carpet loom", The Journal of The Text.Inst., Turkey, 2010.p.882-889.

[4] H.I.Celik &M.Topalbekiroglu."Kinematic analysis of the beat up mechanism for handmade carpet looms", Indian Journal of Fibre & Textile Research,(Vol.34),Turkey.2009.pp. 129-136.

[5] K.J.Waldron & G.L.Kinzel."Kinematic,Dynamics and Design of machinery",2nd edition,Library of Congress Cataloging-in-Publication Data,2004.Chap.5

[6] Jack T. Kimbrell,'Kinematic Analysis and Synthesis', Singapore, 1991 Chap.3,4.p.69-71,96-99.

[7] John J. Uicker, JR, Gordon R. Pennock, Joseph E. Shigley, ' Theory of Machines and Mechanisms' ,3rd edition, Oxford University Press,2011.p.466

[8] R. L. Norton, 'Kinematics and Dynamics of Machinery', 1st edition, Worcester, Massachusetts, U.S.A.2010.p.555.

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TABLE IV
BASED ON MINIMUM TRANSMISSION ANGLE (μ)

Links	μ=50°			μ=60°			μ=74°		
	Ψ=10°	Ψ=20°	Ψ=30°	Ψ=10°	Ψ=20°	Ψ=30°	Ψ=10°	Ψ=20°	Ψ=30°
r_2	67.37	132.02	190.76	66.99	128.60	178.40	65.0	106.66	73.132
r_3	105.42	209.98	312.91	135.44	269.81	402.14	245.30	488.73	728.51
r_4	773.80	760.65	737.38	769.05	740.96	689.46	741.12	614.62	282.72
$e=\frac{r_2}{r_3}$	0.63	0.63	0.60	0.49	0.47	0.44	0.26	0.22	0.10

TABLE VII
COMPARISON BETWEEN PRESENT AND PROPOSED LOOM

Sr.	Present Loom (fig.7.a)	Proposed Loom (fig.7.b)
1.	Beat-up Force is 520 N when wooden-sley weight is 40kg at 120 rpm	Beat-up Force is 135 N when aluminum-sley weight is 7kg at 120 rpm
2.	Beat-up Force is 1160 N when wooden-sley weight is 40kg at 180 rpm	Beat-up Force is 300 N when aluminum-sley weight is 7kg at 180 rpm
3.	Crank shaft torque is 40 N-m when wooden-sley weight is 40kg at 120 rpm	Crank shaft torque is 16 N-m when aluminum-sley weight is 7kg at 120 rpm
4.	Crank shaft torque is 92 N-m when wooden-sley weight is 40kg at 180 rpm	Crank shaft torque is 35 N-m when aluminum-sley weight is 7kg at 180 rpm
5.	Wooden sley weight = 70kg	Aluminum Channel weight = 11kg
6.	Shuttle box weight = 5*2=10kg	Shuttle box weight = 1*2 =2kg
7.	Total weight of sley =80kg	Total weight of sley = 14kg (1kg for wooden plate)