Analysis of Four-Bar Linkage Mechanism of a Power Tiller Operated 8-Row Rice Transplanter

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ABSTRACT

Four-bar linkage mechanism of a power tiller operated 8-row rice transplanter was analyzed. Position analysis, velocity analysis and acceleration analysis were done from both analytical as well as graphical methods. The acceleration and velocity of transplanting finger of a power tiller operated 8-row rice transplanter at different crank angles were evaluated. Different equations were derived to find the angle of coupler with the horizontal, angle of follower with the horizontal and velocity and acceleration of transplanting finger at different positions of the crank. The maximum acceleration and velocity were found to be 3866.5 cms^{-2} and 280.32 cms^{-1} respectively, at 125 rev/min crank speed. The angle of fixed link with the horizontal was 37.41^0. Total distance travelled by the finger in the vertical direction was found to be 20.05 cm. From the analysis, it has been seen that at this much acceleration (3866.5 cms^{-2}) and the angle of fixed link with the horizontal (37.41^0), the transplanting of rice seedlings was not proper.

Key words: Four-bar linkage, Kinematic analysis, Transplanting

INTRODUCTION

A four bar linkage is the simplest movable linkage which consists of 4 rigid bodies each attached to two others by single joint or pivot to form a closed loop. Four-bars are simple mechanisms and fall under the study of kinematics. Fixed link, crank (360^0 rotation), coupler and the follower are the four linkages in which crank is the input link. It is a very versatile linkage, which has been used extensively in a variety of machines, e.g. rice transplanter, Ackerman steering mechanism in automobiles, sewing machines, earth movers, packaging machines, automobile suspensions etc. Four-bar linkage mechanism is the heart of a rice transplanter. The quality and quantity of rice transplanting depends on the performance of four-bar linkage mechanism.

The four bar mechanism is the simplest closed kinematic chain of hinged links with a single degree of freedom (after fixing one link) and that more complex mechanisms can be built up by using one four bar mechanism to drive one or more others. Because of wide variety of motions which can be generated directly by four bar mechanisms. It is often found at the heart of machines and sub-systems such as punch presses, film transports, quick returns, analog computers, and function generators. The study of the four bar linkage is well justified not only because of its many direct applications but also because most of the basic problems encountered in more general linkages show up in a simpler and more understandable way in the four bar linkage.

Masurekar and Gupta (1988) studied that with increasing machine speed, rigid link assumption was no more valid; hence, links of the mechanism were subjected to vibration. Stability analysis of a vibrating system gives the speeds at which the system response was unbounded. It was necessary to avoid these speeds or to provide sufficient damping for better performance of the system. They have presented the theoretical work done for stability analysis, which was simple and could be extended to any mechanism in general. It gives the speed ranges, in which response was unbounded, and presents the study of the response at various speeds for undamped system.

Pennestri *et al*. (1990) presented a numerically efficient technique for the optimum balancing of linkages. Computationally efficient and fast methods of dynamic analysis were required for balancing linkages using general purpose optimization codes. In this approach, instead of solving directly the dynamic equations, a technique was introduced to solve the link dynamic equations in a 'shoe string' fashion. The method through the combined use of Newton's equations anti the principle of virtual work decouples the system of link equilibrium equations. They have also focuses on a new optimality criterion and on constraints on balancing parameters necessary to obtain solutions corresponding to physically feasible linkages.

Thomas (2002) designed a mechanism for analytical synthesis. A planar four-bar linkage with coupler extension was selected as the basic design. The path generated by the mechanism was plotted on a computer screen. By varying the dimensions of various links in the mechanism different paths of output motion of the coupler point were obtained. The potential link dimensions were identified based on the suitability of the path for picking, conveying and planting of seedlings as well as the return motion. A four-row self-propelled transplanter using the above mechanism and an optimized-planting finger was then developed and tested. The machine transplanting system was found to be technically viable.

Todorov (2002) described a new dimensional synthesis method. The position function of the four-bar mechanism was presented by the Freudenstein's equation and it was minimized by the Chebyshev's best approximation theory. The target function was used as an exactly satisfied equation and the Freudenstein's equation was considered as a Chebyshev's polynomial. In some cases the method provides possibilities to find simple solutions of the synthesis tasks. These solutions were known as the best Chebyshev's approach of the weighted target function. This approach was near the best one for the original target function.

Kuang and Tzong (2005) presented a computer-aided curvature analyses program of the planar four-bar linkage mechanism. The complete kinematics curvature analyses of the planar four-bar linkage mechanism depend on the variation of input link angle; angular velocity and angular acceleration were performed by this program. Since the kinematics analyses are routine works, the animation programming was needed. They have presented an animation program based on the curvature theory for the planar four-bar linkage mechanism.

Dado (2005) studied a synthesis and analysis procedure for the limit positions of compliant 4-bar mechanisms. The mechanism compliance was presented at the output link which was considered to be fixed to the ground and can experience large non-linear elastic deflection at its pinned end. Under this condition, the mechanism mobility and its limit positions were dependent on the output link compliance. In addition, an elastic potential energy was stored at each limit position with a magnitude depending on the mechanism geometry and output link compliance properties. The developed procedure provides means of determining the complete mechanism geometry for (1) specified limit positions, or (2) specified energy level at each limit position. On the analysis side, the procedure provides the limit positions and energy levels for specified mechanism geometry. The compliant output link was modeled using the variable parametric pseudo-rigid body model. In this model, the pseudo-rigid-body parameters vary for different loading conditions, thus, providing a more accurate model than that with fixed parameters. The procedure was presented in the form of charts with minimum computational effort by the user.

Bai and Angeles (2007) proposed a unified, robust algorithm for the input–output analysis of planar, spherical and spatial four-bar linkages. Robustness was needed to account for architecture and algebraic singularities that were likely to occur, for example, when conducting an iterative optimization of the linkage at hand. The unified feature of the algorithm was based on the algebra of dual numbers and the Principle of Transference, which allows the extension of the algorithm developed for spherical linkages to their spatial counterparts by a simple dualization of the real geometric relations derived for the former. Numerical examples were included to demonstrate the application of the algorithm.

Roy L. *et al* (2008) used a computer programme and genetic algorithm for the analysis and synthesis of a planar four-bar mechanism. Link dimensions were varied to obtain coupler path maps. Position, velocity, acceleration and the force at the coupler point at various input angles were obtained and the designed path was utilized in the working models. Plot of variation of the link angles, variation of velocity and acceleration of the coupler point with the crank angle were obtained. It was found that the ratio of velocity of crank and coupler point varies between 4.48 and 0.38, acceleration of the coupler point was maximum.

Alvi N.G. *et al* (2012) have analyzed a four bar linkage mechanism to determine values of velocity and acceleration at different positions of the crank. The analysis was done from both graphical as well as analytical methods. The analytical expressions were derived for displacement, velocity and acceleration in terms of general parameters. Also a computer program was developed to get the values of velocity and acceleration of a four bar mechanism at different positions of the crank.

MATERIALS AND METHODS

Four-bar linkage mechanism of a power tiller operated 8-row rice transplanter was analyzed. The measured parameters of four-bar linkage mechanism are given in Table 1. Position analysis; velocity analysis and acceleration analysis were done from both analytical method as well as graphical methods. The acceleration and velocity of planting finger at different crank angles were evaluated. The line drawing of planting mechanism and detailed drawings of four-bar linkage mechanism of power tiller operated 8-row rice transplanter are shown in Fig.1 and Fig. 2, respectively. Vector diagram of a four bar linkage mechanism is given in Fig 3**.**

Displacement analysis: The lengths of linkages of a power tiller operated 8-row rice transplanter

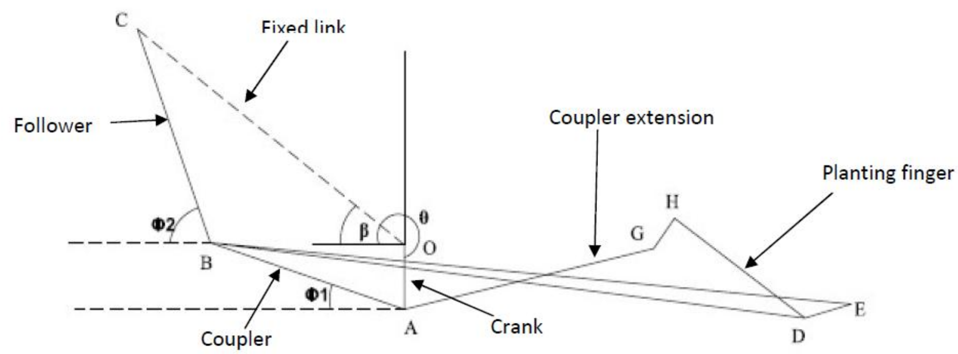

Fig. 1: Planting mechanism of power tiller operated 8-row rice transplanter

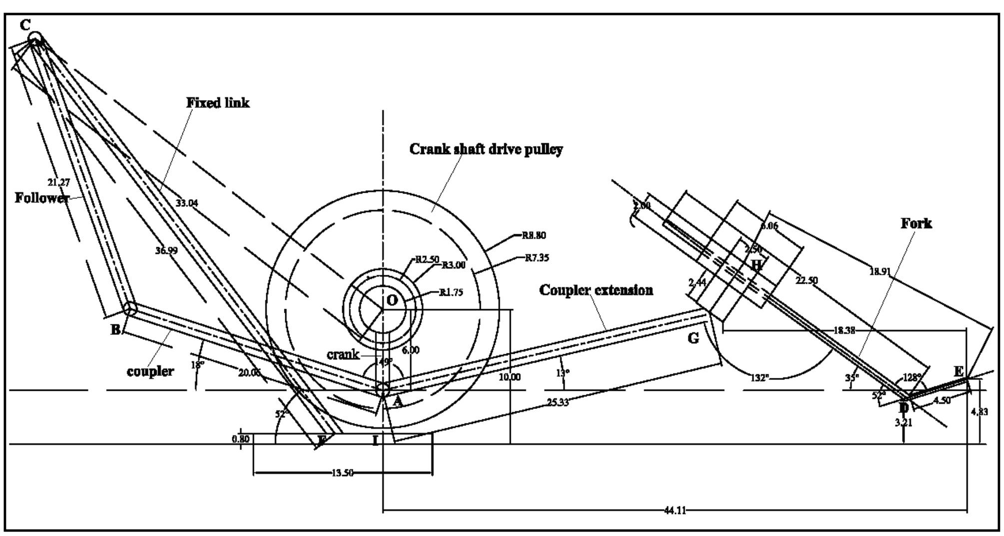

Fig. 2: Side view of planting unit of power tiller operated 8-row rice transplanter

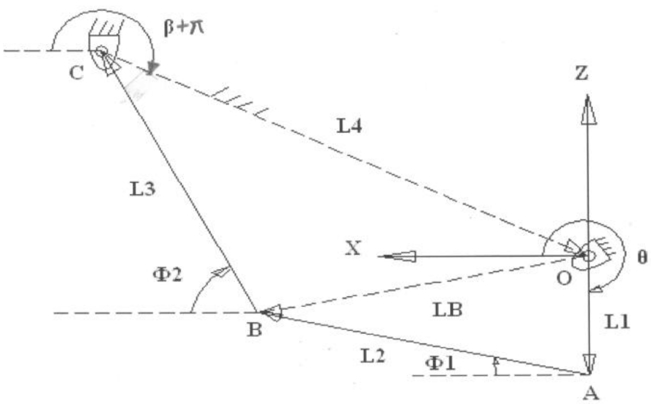

Fig. 3: Vector diagram of a four bar linkage mechanism

Table 1: Measured parameters of four bar linkage mechanism of a power tiller operated 8-row rice transplanter

Particular	Dimension
Length of crank (L_1)	6.0 cm
Length of coupler (L_2)	20.0 cm
Length of follower (L_3)	21.3 cm
Length of fixed link (L_4)	33.0 cm
Length of coupler extension (L_f)	25.3 cm
Angle of fixed link with the horizontal (β)	37.4^0

(Table 1) was analyzed mathematically. The angle of coupler with the horizontal (Φ_1) and the angle of follower with the horizontal (Φ_2) was calculated at different crank angles with the horizontal (θ) with the help of derived equations. Summing X and Z components of the link vectors (Fig 3):

$L_1\cos\theta + L_2\cos\Phi_1 + L_3\cos\Phi_2 - L_4\cos\beta = 0$ …(1)

$L_1\sin\theta + L_2\sin\Phi_1 + L_3\sin\Phi_2 - L_4\sin\beta = 0$ …(2)

From equation (1) & (2), we get

$K_1 (\sin\beta.\sin\Phi_1 + \cos\beta.\cos\Phi_1) + K_2 (\sin\theta.\sin\beta + \cos\theta.\cos\beta) + K_3 = \cos(\theta - \Phi_1)$ …(3)

Where,
$K_1 = (L_4 / L_1)$, $K_2 = (L_4 / L_2)$ & $K_3 = (L_3^2 - L_1^2 - L_2^2 - L_4^2) / 2L_1L_2$

From equation (3),

$K_1\sin\beta \sin\Phi_1 + K_1\cos\beta \cos\Phi_1 + K_2\sin\theta.\sin\beta + K_2 \cos\theta.\cos\beta + K_3 = \cos\theta.\cos\Phi_1 + \sin\theta.\sin\Phi_1$

$(K_1\cos\beta - \cos\theta).\cos\Phi_1 + (K_1\sin\beta - \sin\theta).\sin\Phi_1 + K_2 \cos(\theta - \beta) + K_3 = 0$

$A \cos\Phi_1 + B \sin\Phi_1 + C = 0$ …(4)

Where,
A = K1cosβ – cosθ, B = K1sinβ – sinθ & C = K2 cos (θ – β) + K3

Dividing equation (4) by $\cos\Phi_1$, we get

$A + B \tan\Phi_1 + C \sec\Phi_1 = 0$
$(A + B \tan\Phi_1)^2 = (- C \sec\Phi_1)^2$
$A^2 + B^2 \tan^2\Phi_1 + 2AB \tan\Phi_1 = C^2 \sec^2\Phi_1$
$A^2 + B^2 \tan^2\Phi_1 + 2AB \tan\Phi_1 = C^2 + C^2 \tan^2\Phi_1$
$(B^2 - C^2) \tan^2\Phi_1 + 2AB \tan\Phi_1 + A^2 - C^2 = 0$

$ax^2 + bx + c = 0$ …(5)

Where,
$a = B^2 - C^2$, $b = 2AB$, $c = A^2 - C^2$ & $x = \tan\Phi_1$

From equation (5),

$$\tan\Phi_1 = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

$$\Phi_1 = \tan^{-1}\left[\frac{-b \pm \sqrt{b^2 - 4ac}}{2a}\right]$$

Velocity and acceleration Analysis: Velocity and acceleration of the linkages have been analyzed and calculated with the help of respective derived equations. The sum of the vectors (Fig 3) is:

$L_1 + L_2 + L_3 + L_4 = 0$
$L_B = L_1 (e^{i\theta}) + L_2 (e^{i\Phi1}) = - L_3 (e^{i\Phi2}) - L_4 (e^{i\beta})$

Where,
eiθ = cosθ + isinθ and so are eiΦ1, eiΦ2 & eiβ

$V_B = L_1\omega_1 i (e^{i\theta}) + L_2\omega_2 i (e^{i\Phi1}) = - L_3\omega_3 i (e^{i\Phi})$

$A_B = [- L_1\omega_1^2 (e^{i\theta}) + L_1\alpha_1 i (e^{i\theta})] + [- L_2\omega_2^2 i^2 (e^{i\Phi1}) + L_2\alpha_2 i (e^{i\Phi1})]$

$= [L_3\omega_3^2 (e^{i\Phi2}) - L_3\alpha_3 i (e^{i\Phi2})]$

Where,
$\omega_1 = d\theta/dt$, $\omega_2 = d\Phi_1/dt$, $\omega_3 = d\Phi_2/dt$, $\omega_4 = d\beta/dt = 0$
$\alpha_1 = d\omega_1/dt$, $\alpha_2 = d\omega_2/dt$, $\alpha_3 = d\omega_3/dt$

The angular velocities ω_2 & ω_3 and angular accelerations α_2 & α_3 may be determined by equating the real and imaginary parts of the velocity and acceleration equations respectively.

$\omega_2 = [L_1\omega_1 \sin(\theta - \Phi_2)] / L_2 \sin(\Phi_2 - \Phi_1)$
$\omega_3 = -L_1\omega_1 \sin(\theta - \Phi_1) / L_3 \sin(\Phi_2 - \Phi_1)$

$\alpha_2 = [L_1\omega_1^2\cos(\theta - \Phi_2) + L_1\alpha_1\sin(\theta - \Phi_2) + L_2\omega_2^2\cos(\Phi_1 - \Phi_2) + L_3\omega_3^2] / L_2\sin(\Phi_2 - \Phi_1)$

$\alpha_3 = [L_1\omega_1^2\cos(\theta - \Phi_1) + L_1\alpha_1\sin(\theta - \Phi_1) + L_3\omega_3^2\cos(\Phi_2 - \Phi_1) + L_2\omega_2^2] / L_3\sin(\Phi_2 - \Phi_1)$

Complex Number method: The angular velocities and angular accelerations of different linkages of four bar linkage mechanism can also be determined with the help of this method. The position vector L_B for point B (Fig.3) of the linkages is the resultant in two independent vector equations: $L_B = L_1 + L_2$ and $L_B = L_4 + L_3$. These equations may be combined to determine the velocity V_B and acceleration A_B of point B.

$L_B = L_1(e^{i\theta}) + L_2(e^{i\Phi 1}) = L_4(e^{i\beta}) + L_3(e^{i\Phi 2})$

$V_B = L_1\omega_1(ie^{i\theta}) + L_2\omega_2(ie^{i\Phi 1}) = L_3\omega_3(ie^{i\Phi 2})$

$A_B = [L_1\omega_1^2(i^2e^{i\theta}) + L_1\alpha_1(ie^{i\theta})] + [L_2\omega_2^2(i^2e^{i\Phi 1}) + L_2\alpha_2(ie^{i\Phi 1})]$

$= L_3\omega_3^2(i^2e^{i\Phi 2}) + L_3\alpha_3(ie^{i\Phi 2})$

The angular velocities ω_2 & ω_3 and angular accelerations α_1, α_2 & α_3 may be determined by equating the real and imaginary parts of the velocity and acceleration equations respectively.

$\omega_2 = -[L_1\omega_1\sin(\theta - \Phi_2)] / [L_2\sin(\Phi_1 - \Phi_2)]$

$\omega_3 = [L_1\omega_1\sin(\theta - \Phi_1)] / [L_3\sin(\Phi_2 - \Phi_1)]$

$\alpha_1 = -[L_1\omega_1^2\cos(\theta - \Phi_1) + L_2\omega_2^2] / [L_1\sin(\theta - \Phi_1)]$

$\alpha_2 = [L_2\omega_2^2\cos(\theta - \Phi_1) + L_1\omega_1^2] / [L_2\sin(\theta - \Phi_1)]$

$\alpha_3 = \omega_3^2$

Calculation of crank speed: The crank speed and the speed of the ground wheel were calculated based on the measured parameters and assumed parameters.

Assume forward velocity of transplanter,

$V = 1.5$ km/h $= 0.4167$ m/s

$V = 3.14 \times D \times N_w / 60$

Where,

D = diameter of ground wheel of the power tiller = 55 cm (measured)

N_w = wheel speed in rev/min

Therefore,

$N_w = 60 \times 0.4167 / (3.14 \times 0.55) = 14.48$ rpm

Hill to hill spacing = 20 cm (assumed)

Therefore speed of crank,

$N_c = (3.14 \times 0.55 \times 14.48) / 0.2 = 125$ rpm

Angular velocity of the crank,

$\omega_1 = 2 \times 3.14 \times 125 / 60 = 13.08$ rad/s

RESULTS AND DISCUSSION

Four-bar linkage mechanism of a power tiller operated 8-row rice transplanter was analyzed. The measured parameters for the analysis of four-bar linkage mechanisms are shown in Table 1. The acceleration and velocity of planting finger of a power tiller operated 8-row rice transplanter at different crank angles and at crank speed 1 rad/s and 13.08 rad/sec are shown in Fig.4 & Fig.5 and Fig.6 & Fig7 respectively. The maximum acceleration and velocity in Z-direction at 1 rad/sec crank speed were 19.18 cm/s^2 and 17 cm/s respectively and at 13.08 rad/sec were 3572.54 cm/s^2 and 239.78 cm/s respectively. The maximum overall acceleration and velocity of transplanting finger were found to be 3866.5 cm/s^2 and 280.32 cm/s respectively at 125 rev/min crank speed. The angle of fixed link with

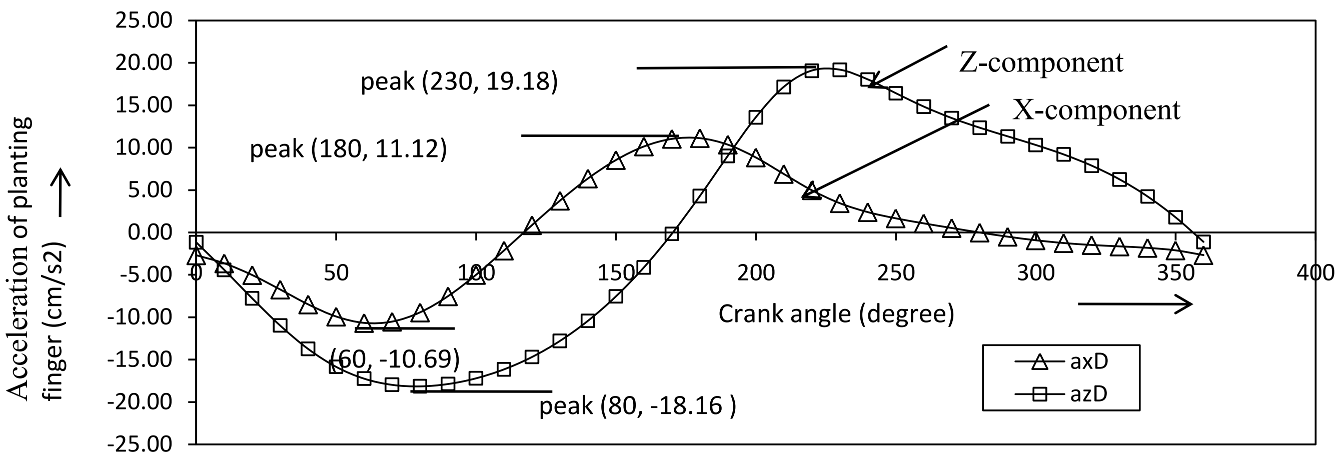

Fig. 4: Acceleration of planting finger at different crank angles (crank speed, ω1 = 1 rad/s)

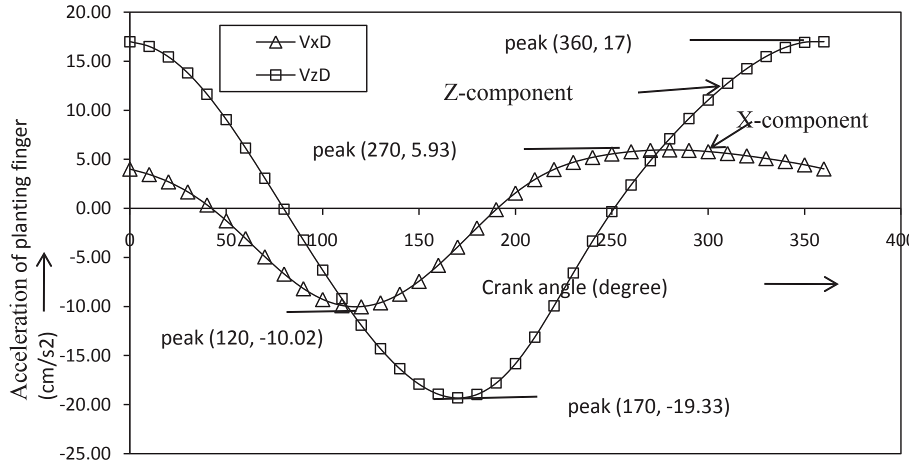

Fig. 5: Velocity of planting finger at different crank angles (crank speed ω1 = 1 rad/s)

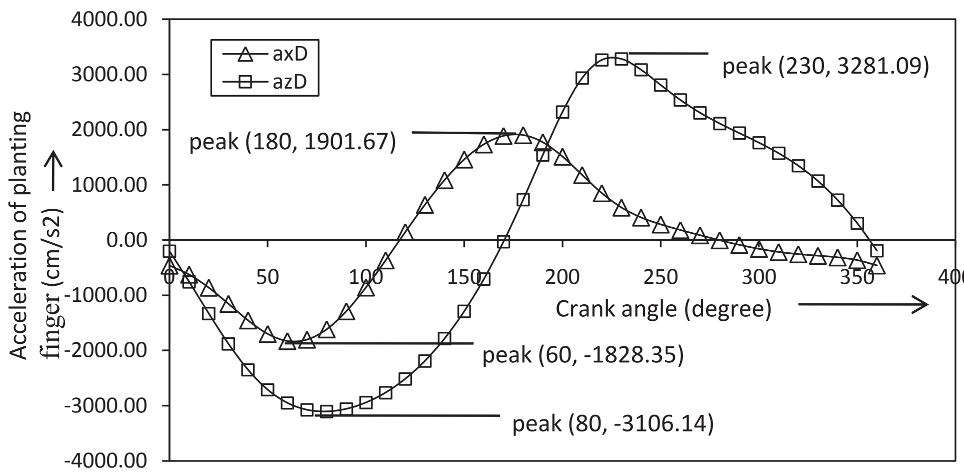

Fig. 6: Acceleration of planting finger at different angles (crank speed, ω1 = 13.08 rad/s)

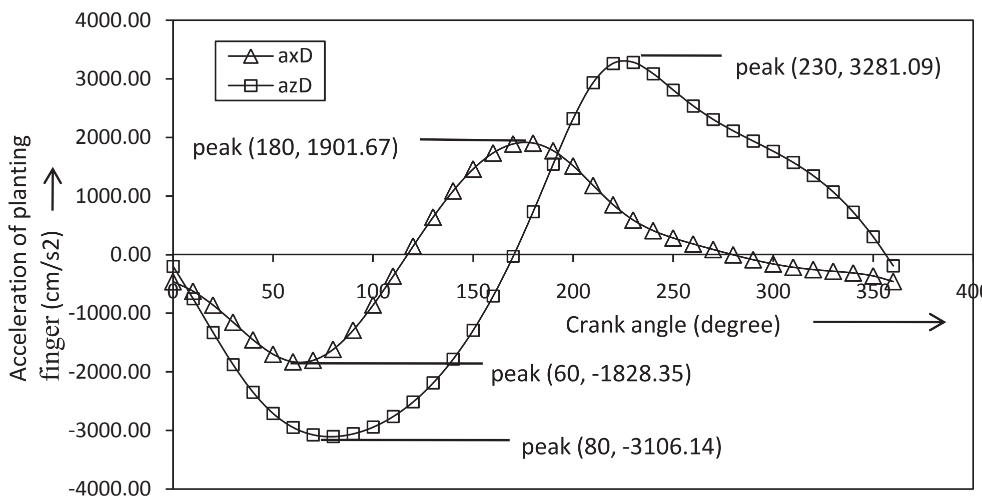

Fig. 7: Velocity of planting finger at different crank crank angles (crank speed, ω1 = 13.08 rad/s)

the horizontal was 37.41^0. Total distance travelled by the finger in the vertical direction was found to be 20.05 cm. The path traced by the transplanting finger (β = 37.4^0) is shown in Fig. 8.

The result shows that the maximum acceleration of transplanting finger in vertical direction is 3572.54 cm/s^2 and overall acceleration is 3866.5 cm/s^2. The values of accelerations are too high and due to this, the transplanting of rice seedlings was not proper. This may be due to improper lengths of four bar linkage mechanism and lower value of angle of fixed link with the horizontal. The lower value of angle of fixed link with the horizontal may be due to improper lengths of four bar linkage mechanism or the proportionate of the lengths of linkages was not proper.

Statistical Analysis

Descriptive analysis were performed for the variable under study like angle of crank with the horizontal, angle of coupler with the horizontal, angle of follower with the horizontal, velocity of transplanting finger and acceleration of transplanting finger using 37 observations. The descriptive statistics of the variables under study mentioned above is given with respective co-efficient of variation, standard deviation, range, median, lower quartile and upper quartile in Table 2. The minimum and the maximum values of acceleration of transplanting finger were 435.28 cm/s^2 and 3866.50 cm/s^2 at 350^0 and 70^0 crank angles respectively. Similarly the minimum and the maximum values of velocity of transplanting finger were 72.30 cm/s and 280.32 cm/s at 250^0 and 160^0 crank angles respectively. The average velocity and acceleration of transplanting finger were found to be 184.64 cm/s and 2398.91 cm/s^2 respectively. The angle of coupler with the horizontal (Φ_1) varies from -18.1^0 to +19.4^0 and the angle of follower with the horizontal (Φ_2) varies from +54.1^0 to +89.4^0.

The table shows that the median is more closer to mean than mode. The co-efficient of variation is higher for angle of coupler with the horizontal and lower for angle of follower with the horizontal. This shows that the variation in angle of coupler with the horizontal is higher and the consistency is lower as compared to angle of follower with the horizontal. Similarly the variation in acceleration of transplanting finger is more and the consistency is less as compared to the velocity of transplanting finger. Correlation analysis was also performed to examine any linear relationship among variable under study. The frequency distributions of angle of coupler with the horizontal, velocity of transplanting finger and acceleration of transplanting finger are shown in Fig. 9, Fig.10 and Fig. 11, respectively.

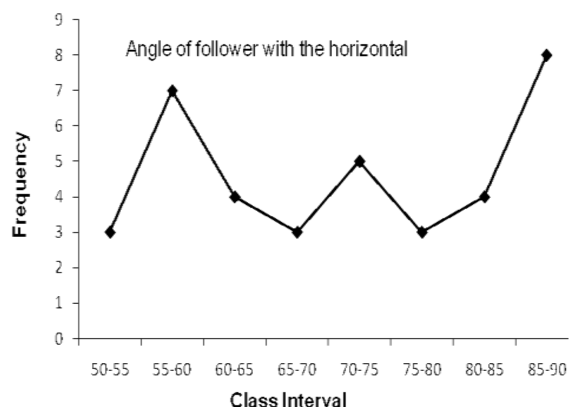

Fig. 9: Frequency distribution of angle of rotation at β coupler with the horizontal

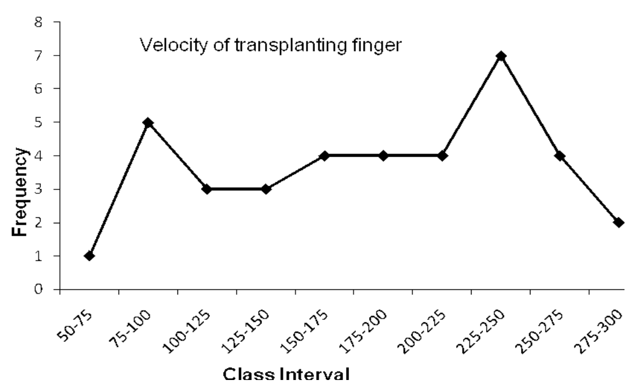

Fig. 10: Frequency distribution of velocity transplanting finger

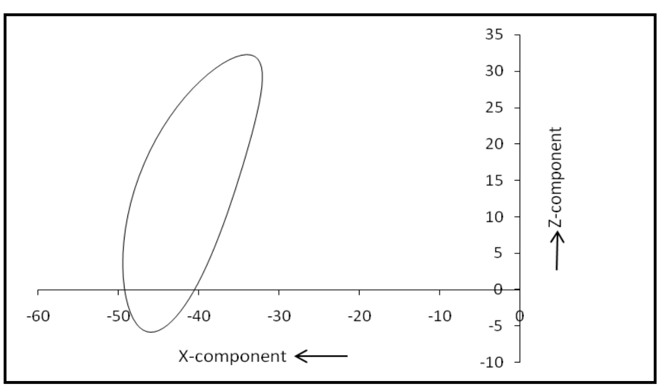

Fig. 8: Path of transplanting finger during crank 37.40

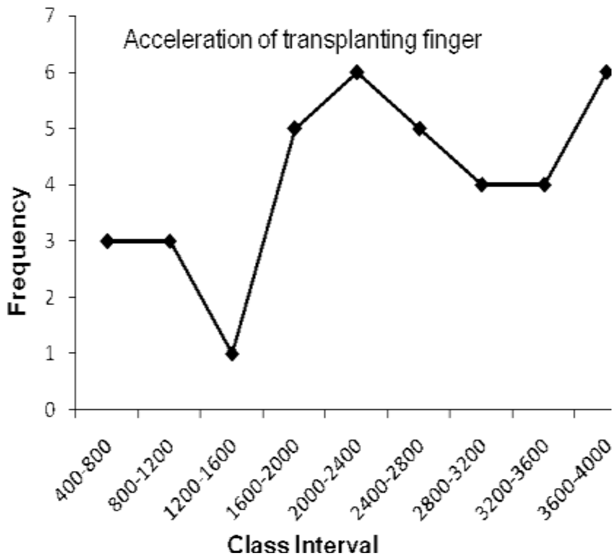

Fig. 11: Frequency distribution of acceleration of of transplanting finger

CONCLUSIONS

From the analysis, it has been seen that at this much acceleration (3866.5 cm/s^2) of transplanting finger and the angle of fixed link with the horizontal (37.41^0), the transplanting of rice seedlings was not proper. The acceleration of transplanting finger in Z-direction can be minimized by reducing the lengths of the four bar mechanism and increasing the angle of fixed link with the horizontal. Analysis can be made by varying the angle of fixed link with the horizontal at different link lengths of four bar linkage mechanism. The computer program can also be developed to find the maximum acceleration and velocity of transplanting finger at different positions of the crank.

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