An Analytical Method for Kinematic Analysis of The Planting Mechanism of A Rice Transplanter

移秧機 植付機構의 機構學的 分析을 위한 解析的方法

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要 約

一般的으로 機構의 分析法은 圖解的 方法으로 大別할수 있다. 圖解的 方法이 簡便하지만 그 正確性이 不足하고 解析的 方法은 複雜한 計算過程을 要求한다. 最近 많은 콤퓨터 施設은 解析的 方法의 活用을 可能케 하였으나 簡單한 機構의 分析은 또한 經濟的인 面에서 콤퓨터의 廣範圍한 使用을 어렵게 하고있다.

本 硏究는 小型 計算機를 利用하여 크랭크로커 機構를 分析할수 있는 解析的 方法을 위한 方程式을 誘導하고 이 方法을 動力 移秧機의 植付機構의 分析에 適用하였다. 機構 表示法으로 크랭크—로커 機構를 심볼 方程式으로 나타내고, 機構上의 各 링크에 固定된 座標系를 3×3 行列式을 利用하여 座標系를 轉移 시키는 方法으로 方程式들을 誘導하였다. 크랭크—로커 機構의 링크상의 어떤 한 點의 位置벡타를 行列 方程式으로 表示하고 이 行列 方程式을 一次, 二次 微分하여 그 點에 대한 速度와 加速度 方程式은 誘導하였다.

1. Itroduction

Kinematic analysis of a mechanism is used to investigate the motion of machine parts without regard to the forces which produce that motion. It includes relative displacement, velocity and acceleration of a particular point or a particular link on the mechanism. A point on the coupler link of a four-bar mechanism, where on crank rotates through a complete revolution while the second crank only oscillates through a certain angle, produces a specific coupler curve. The use of the coupler curve has numerous applications in machine design. The variety of the shapes obtainable using the coupler curve is infinite. A number of coupler curves, each with its own kinematic characteristics can be obtained by varying the linkages or the position of the coupler point on the mechanism. The coupler curve necessary for the mechanism to satisfactorily perform a specific constrained motion has to be obtained, and the accurate displacements, velocities and accelerations of the linkages as well as the coupler point must be analyzed for further improvement of the mcehanism.

Two methods are used to analyze mechanisms: graphical and analytical. In recent years, there has been extensive development in both methods. In general, the analytical method is considered more tedious and time consuming than the graphical method due to its complicated solving procedures. However,

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a number of computer programs and calculating machines now available have enabled the analytical methods to be more widely used for fast and accurate computations. Further, the analytical method can be of value from the standpoint of systematic design.

Most planting devices of the power rice transplanting machines can be classified to crank and rocker mechanism of the four-bar linkages. A planting fork which is part of the connecting rod (coupler link) of the mechanism separates seedlings from a seedling board and places them in the soil. The coupler curve traced by the planting fork may have an influence on the stability of the planted seedlings. Mechanical damage to seedlings and high torque requirements for the planting devices may result from high velocities and accelerations of the mechanisms. The kinematic analysis of the planting mechanism is essential not only to an understanding of planting phenomena but also to further improvement of the mechanism.

In this study, an analytical method for determining the kinematic quantities of crank and rocker mechanism is developed and applied to analysis of the planting mechanisms.

The equations for linear and angular velocity and acceleration of a point on the mechanism were derived algebraically using 3×3 matrices which define the linear transformations of the Cartesian coordinate systems in two dimensions. Substitution of numerical data into the equations gives the kinematic quantities of interest. Computations to solve the equations can easily be carried out with a small desk calculator rather than computer programming. In addition to the above components of the mechanism the external torque necessary to drive the mechanism was obtained experimentally.

2. Review of literature

A generalized analytical method for determination of positions, velocities, and accelerations in mechanisms was presented by Francis H. Raven (13).

In his method of analysis, a particular point or a particular link on a mechanism are expressed as position equations which are mathematically independent. The position equation is a vector equation which expresses the position of the point or link as a function of the geometry of the mechanism. Successive differentiation of the indipendent position equations with respect to time will yield equations for obtaining the velocity and acceleration of the point or the angular velocity and acceleration. The desired kinematic quantities may be determined by simultaneously solving these independent equations.

Raven later developed a similar but more direct method which employs loop equations (17). A loop equation is also a mathematically independent position equation which expresses a closed path in the mechanism as complex numbers in exponential form. The independent loop equation may be separated into real and imaginary components to obtain algebraic equations from which unknown position terms may be determined. The first and second time derivatives of these equations give the equations for the angular velocity and acceleration of a linkage in the mechanism.

Use of the complex plane for kinematic analysis was initiated by Bayer, Bloch, and Rosenauer, particularly in the field of kinematic synthesis.

G.H. Martinand M.F. Spotts developed analytical method for determining velocities and accelerations in mechanisms by means of geometry of the complex plane (10). This method of analysis is based upon the equations for relative motion of machine parts. Velocity and acceleration equations are expressed in vector form as though one were going to make a graphical analysis. These vectors are then written as complex numbers, and a mathematical solution is derived so that ultimately the velocities and accelerations of each of the links in the mechanism can be related to those of the driving member.

An approach to the problem of rationalizing kinematics into a science by means of a symbolic language was proposed by Reuleaux in 1875(2). He intended that a symbolic language would permit a complete description of the kinematic properties of a mechanism and that this would be additionally useful not only for the analysis of existing mechanisms but also for the synthesis of new mechanisms.

J. Denavit and R.S. Hartenberg presented a kinematic notation which permits the complete description of the kinematic properties of the low-pair mechanism by means of equations (2). This symbolic notation furnishes a powerful and reliable analytical procedure since the equations involved are based on matrix algebra. They later extended the algebraic method using 4×4 materices to the analysis of velocities and acclerations in one-degree-of-freedom, single loop, spatial linkages consisting of revolute and prismatic pairs, either singly or in combination. The velocities and accelerations are then obtained by differentiating the matrix equation or position equation. The matrix operations involved in this method of analysis lead directly to machine computation.

Although there has been a considerable improvement in some parts of transplanter such as adjusting device, feeding mechanism and wheel and float control system. Few systematic studies on the planting mechanism have been done. K.H. Ryu and K. Namikawa analyzed the planting mechanisms of various transplanters using computer programs (11, 14). Namikawa investigated the effect of changing the length of each link in the mechanism (11). His results showed that the effect to the coupler curve was great, but that to the velocity was relatively small.

In a limited number of cases, it is possible from the geometry of the mechanism to obtain equations which express the position vector of any point on a mechanism as a function of a single independent variable. This variable usually defines the position of the driving member. When such equations can be differentiated twice with respect to time, the complete kinematic behavior of any point on a machanism can be described.

3. Development of equations

A. Transformation of Cartesian coodinates in two dimensions

Geometric transformations are part of the mathematical notion of a function (16). A particularly simple group of geometric transformation which is useful in kinematic analysis is the affine transformation in which points located on a straight line in one space are transformed into the corresponding points on a straight line in a second space. Transformation of the Cartesian coordinates of a general point in twodimesional space is defined analytically when X_2, Y_2 are linear function of X_1, Y_1.

In two Cartesian coordinate systems $X_1Y_1O_1$ and $X_2Y_2O_2$ as shown in Fig. 1, a point M given by its coordinates (X_2,Y_2) relative to the coordinate system $X_2Y_2O_2$ is transformed into the coordinates (X_1,Y_1) relative to the coordinate system $X_1Y_1O_1$.

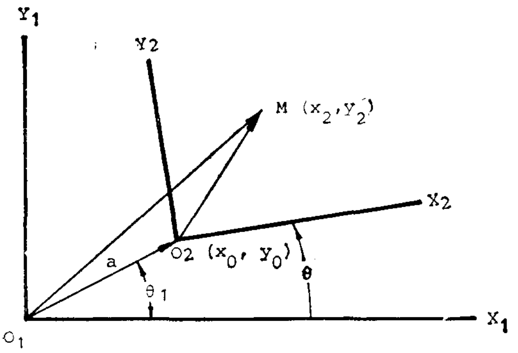


Fig. 1. Transformation of Cartesian coordinates in two dimensions.

The coordinates (X_1, Y_1) of the point M relative to $X_1Y_1O_1$ can be specified completely by the coordinates (X_0, Y_0) of the origin O_2 of $X_2Y_2O_2$ relative to $X_1Y_1O_1$, and and the inclination θ of O_2X_2 relative to O_1X_1. Since three displacement vectors $\vec{O_1O_2}$, $\vec{O_2M}$ and $\vec{O_1M}$ construct a closed vector polygon, the vector equation can be written as:

$$\vec{O_1M}=\vec{O_1O_2}+\vec{O_2M}$$

which represents the addition of vector $\vec{O_2M}$ to vector $\vec{O_1O_2}$ to produce the resultant vector $\vec{O_1M}$. Each term of the vector equation can be resolved into the X and Y component vectors.

Expressing these component vectors in terms of the coordinates of point M yields the following set of equations:

$$X_1=X_0+X_2\cos\theta-Y_2\sin\theta$$
$$Y_1=Y_0+X_2\sin\theta+Y_2\cos\theta.$$

The coordinates (X_0, Y_0) can be specified by the magnitude a and direction θ_1 of vector $\vec{O_1O_2}$. Thus, $X_0=a\cos\theta_1$, $Y_0=a\sin\theta_1$. The above set of equations may be expressed in the homogeneous matrix equation form as:

$$\begin{vmatrix} 1 \\ X_1 \\ Y_1 \end{vmatrix} = \begin{vmatrix} 1 & 0 & 0 \\ a\cos\theta_1 & \cos\theta & -\sin\theta \\ a\sin\theta_1 & \sin\theta & \cos\theta \end{vmatrix} \begin{vmatrix} 1 \\ X_2 \\ Y_2 \end{vmatrix}. \quad [1]$$

The coefficient determinant of matrix equation [1]

$$\begin{vmatrix} 1 & 0 & 0 \\ a\cos\theta_1 & \cos\theta & -\sin\theta \\ a\sin\theta_1 & \sin\theta & \cos\theta \end{vmatrix}$$

is called a displacement matrix. If θ and θ_1 are coincident, the coefficient determinant of equation [1] can be written in general form as:

$$A_i=\begin{vmatrix} 1 & 0 & 0 \\ a_i\cos\theta_i & \cos\theta_i & -\sin\theta_i \\ a_i\sin\theta_i & \sin\theta_i & \cos\theta_i \end{vmatrix}$$

which defines the Cartesian coordinate transformation from the coordinate system $X_i+1Y_i+1O_i+1$ into the system $X_iY_iO_i$.

Rigid body displacement without distortion can be considered as a special case of an affine transformation. The algebraic method using 3×3 displacement matrices will be extended to the derivation of equations for the displacements, velocities and accelerations of the crank and rocker mechanisms.

B. Description of crank and rocker mechanism

Symbolic notation devised by Revleaux in 1875 provided several concepts that appear to be fundamental for a description of the kinematic properties of a mechanism.

Description of a mechanism will involve a description of the relative positions of the successive pair axis of the mechanism which may be done with use of the unique common perpendicular between successive pair axis.

To describe the crank and rocker mechanism, the following set of conventions is established and illustrated in Fig. 2:

R_i=Number of particular pair. Input pair is taken as 1 and remaining pairs are numbered consecutively around closed loop.

X_i=Axes formed by common perpendicular directed from Z_{i-1} to Z_i.

Y_i=Axes implicitly defined to form right-handed Cartesian coordinate systems

$X_iY_iZ_i$.

Z_i = Characteristic axis of motion about R_i pair perpendicular to the plane of motion.

a_i = Length of common perpendicular from Z_{i-1} to Z_i, always positive.

θ_i = Angle from X_{i-1} to X_i measured in counter-clockwise direction about positive Z_{i-1}.

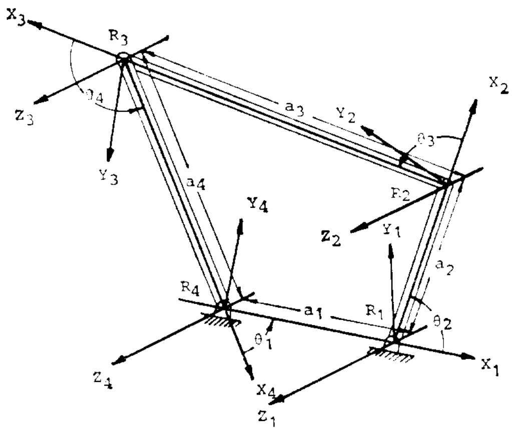


Fig. 2. Crank and rocker mechanism showing pairs and and axes.

Crank and rocker mechanism is a plane mechanism and composed of four machine parts which are connected by revolute pairs R_1, R_2, R_3 and R_4. For the sake of convenience, the machine parts are identified as 1, 2, 3 and 4 and designed in Fig. 2 as link 1, link 2, link 3, and link 4. As the driving link 2 roates about the fixed axis Z_1, the driven link 4 oscillates about the fixed axis Z_4. The position of each link, therefore, may be specified by the length of common perpendicular between Z_i axes and the angle between X_i axes of the coordinate systems fixed in each link. Once the geometries of an individual link position are provided, the description of a complete mechanism can be easily obtained by successively specifying the relative positions of the successive linkage.

Because crank and rocker mechanism consists of only revolute pairs, two parameters may be necessary and sufficient to completely describe the mechanism although four parameters are necessary to describe the low-pair mechanisms (2).

The common perpendicular between Z_i axes is equal to the constant link length between them. The angle between X_i axes is the pair variable of the revolute which varies with the angular displacement of driving link of the mechanism. The positive sense of X_i axis is defined to be from Z_{i-1} to Z_i, and the positive rotation of the coordinate system would be in the counter clockwise direction about positive Z_i axis. These parameters are illustrated in Fig. 3.

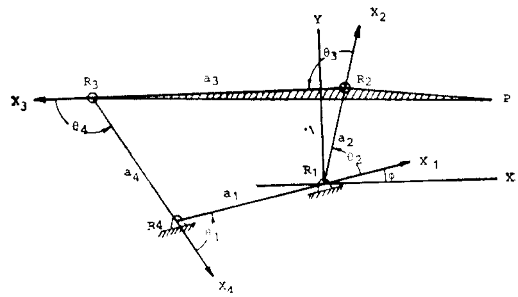


Fig. 3. Schematic of crank and rocker mechanism showing pair variable θ_i.

C. Symbolic equation

Once two parameters a_i and θ_i are determined for each pair of linkages, the mechanism can be symbolically represented by a block diagram which assembles all parameters into blocks, with one block to each linkage as shown in Fig. 4.

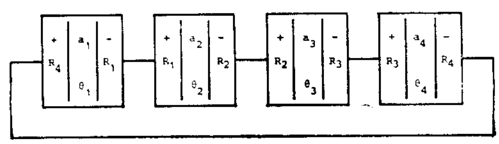


Fig. 4. Block diagram representing crank and rocker mechanism.

In the block diagram, each box represents an individual linkage of the mechanism; in each block are indicated its pair elements and parameters determing the position of the linkage and relation of the linkage to the preceding one. Since the chain of blocks so obtained forms a closed loop, the description of the block diagram may start from any pair element and proceed in either direction through the links and then return to the initial position. The relation of early linkage to the preceding one may be represented by the two parameters a_i and θ_i which are determined by the relative position between successive coordinate system fixed in those linkages. Such a relation is a linear transformation between two coordinate systems involved. A linear transformation of the coordinate system can be represented by a displacement matrix whose elements are functions of a_i and θ_i. If the linkages of the mechanism form a closed loop the product of all displacement matrices taken in succession around the closed chain must equal the unit matrix (3). Therefore, the block diagram can be represented by a symbolic equation as follows:

$$R_1(\theta_1)\begin{vmatrix} a_1 \\ \\ \theta_1 \end{vmatrix} R_2(\theta_2)\begin{vmatrix} a_2 \\ \\ \theta_2 \end{vmatrix} R_3(\theta_3)\begin{vmatrix} a_3 \\ \\ \theta_3 \end{vmatrix} R_3(\theta_4)\begin{vmatrix} a_4 \\ \\ \theta_4 \end{vmatrix} = \mathrm{I} \quad [2]$$

which would be more concise but equally definitive description of the mechansism.

This symbolic equation shows the pair variables of the mechanism and leads to a convenient matrix equation. Since four coordinate systems can be established in the crank and rocker mechanism four linear transformations taken in succession would give a return to the original coordinate system. The product of the corresponding four displacement matrices is equal to the unit matrix. The matrix loop equation so obtained would be

$$A_1A_2A_3A_4 = I$$

where A_i is the displacement matrix which represents the linear transformation of Cartersian coordinates from the the $X_iY_iZ_i$ coordinate system into the $X_{i-1}\ Y_{i-1}\ Z_{i-1}$ coordinate system and I is the unit matrix. In Fig. 3, if link 1 is a fixed link used as a reference coordinate system, the matrix loop equation obtained by taking the transformations from the $X_1Y_1Z_1$ coordinate system into its initial position through the counter clockwise direction will be

$$A_2A_3A_4A_1 = I \qquad [3]$$

Equation [3] gives the solution to the displacement relation between adjacent links of the mechanism. With the aid of Fig. 3, displacement matrices A_i are established as follows:

$$A_1 = \begin{vmatrix} 1 & 0 & 0 \\ a_1\cos\theta_1 & \mathrm{cas}\,\theta_1 & -\sin\theta_1 \\ a_1\sin\theta_1 & \sin\theta_1 & \cos\theta_1 \end{vmatrix}$$

$$A_2 = \begin{vmatrix} 1 & 0 & 0 \\ a_2\cos\theta_2 & \cos\theta_2 & -\sin\theta_2 \\ \mathrm{a}_2\sin\theta_2 & \sin\theta_2 & \cos\theta_2 \end{vmatrix}$$

$$A_3 = \begin{vmatrix} 1 & 0 & 0 \\ a_3\cos\theta_3 & \cos\theta_3 & -\sin\theta_3 \\ a_3\sin\theta_3 & \sin\theta_3 & \cos\theta_3 \end{vmatrix}$$

$$A_4 = \begin{vmatrix} 1 & 0 & 0 \\ a_4\cos\theta_4 & \cos\theta_4 & -\sin\theta \\ a_4\sin\theta_4 & \sin\theta_4 & \cos\theta_4 \end{vmatrix}$$

Among the elements of the displacement matrices, θ_2 is the known value as the the angular displacement of link 2. θ_1, θ_3 and θ_4 are the relative angular displacements between adjacent links, which are determined by θ_2 a_i is the constant link length of the mechanism.

D. Velocity equtation

The velocities of the pair variables can be obtained by differentiating the displacement matrix loop equation with respect to time.

To differentiate the matrix equation, it may be convenient to introduce the derivative operator matrix Q. The derivative operator matrix Q performs the indicated differentiation through the following definition:

$$\frac{dA_i}{dt}=QA_i$$

Under this definition, Q matrix is found to be

$$Q=\begin{vmatrix} 0 & 0 & 0 \\ 0 & 0 & -1 \\ 0 & 1 & 0 \end{vmatrix}$$

At any given angular displacement θ_2, the matrix loop equation $A_2A_3A_4A_1=\mathrm{I}$ can be differentiate by means of the operator Q to give

$$\frac{d}{dt}(A_2A_3A_4A_1)=0$$

in other expression

$$QA_2A_3A_4A_1\dot{\theta}_2+A_2QA_2A_4A_1\dot{\theta}_3+A_2A_3QA_4A_1\dot{\theta}_4+A_2A_3A_4Q_1A_1\dot{\theta}_1=0. \qquad [4]$$

Let Bi be represented by the definitions

$$B_1=A_2A_3A_4QA_1$$
$$B_2=QA_2A_3A_4A_1$$
$$B_3=A_2QA_3A_4A_1$$
$$B_4=A_2A_3QA_4A_1$$

Using these definitions and rearranging terms, equation [4] is reduced to

$$B_3\dot{\theta}_3+B_4\dot{\theta}_4+B_1\dot{\theta}_1=Q\dot{\theta}_2 \qquad [5]$$

where matrix B_2 is expressed as matrix Q since the product of matrices $A_2\cdot A_3\cdot A_4\cdot A_1$ is equal to the unit matrix.

Matrix equation [5] implies nine linear equations. However, the equalities of the three elements below the major diagonal of each matrix term are sufficient to satisfy the entire matrix equation. The three linear equations so obtained appear in a matrix form as follows:

$$\begin{vmatrix} B_{321} & B_{421} & B_{121} \\ B_{331} & B_{431} & B_{131} \\ B_{332} & B_{432} & B_{132} \end{vmatrix} \begin{vmatrix} \dot{\theta}_3 \\ \dot{\theta}_4 \\ \dot{\theta}_1 \end{vmatrix} = \begin{vmatrix} 0 \\ 0 \\ -1 \end{vmatrix} \dot{\theta}_2 \qquad [5-1]$$

The elements of the 3×3 matrices of equation [5—1] may be determined by substituting matrices A_i and Q into the matrix product B_i and equating the corresponding elements. The multiplications of matrices A_i and Q gives B_i as follows:

$$B_1=A_2A_3A_4QA_1=\begin{vmatrix} 0 & 0 & 0 \\ 0 & 0 & -1 \\ a_1 & 1 & 0 \end{vmatrix}$$

$$B_3=A_2QA_3A_4A_1=\begin{vmatrix} 0 & 0 & 0 \\ -a_3\sin(\theta_2+\theta_3)-a_4\sin(\theta_2+\theta_3+\theta_4) & 0 & -1 \\ a_3\cos(\theta_2+\theta_3)+a_4\cos(\theta_2+\theta_3+\theta_4)+a_1 & 1 & 0 \end{vmatrix}$$

$$B_4=A_2A_3QA_4A_1=\begin{vmatrix} 0 & 0 & 0 \\ -a_4\sin(\theta_2+\theta_3+\theta_4) & 0 & -1 \\ a_4\cos(\theta_2+\theta_3+\theta_4)+a_1 & 1 & 0 \end{vmatrix}$$

From the geometry of the crank and rocker mechanism shown in Fig. 3, it is noted that $\sin(\theta_2+\theta_3+\theta_4+\theta_1)=0$ and $\cos(\theta_2+\theta_3+\theta_4+\theta_1)=1$. Substituting the corresponding elements into equation [5—1] gives

$$\begin{vmatrix} -a_3\sin(\theta_2+\theta_3)-a_4\sin(\theta_2+\theta_3+\theta_4) & -a_4\sin(\theta_2+\theta_3+\theta_4) & 0 \\ a_3\cos(\theta_2+\theta_3)+a_4\cos(\theta_2+\theta_3+\theta_4)+a_1 & a_4\cos(\theta_2+\theta_3+_4)+a_1 & \theta_1 \\ 1 & 1 & 1 \end{vmatrix} \begin{vmatrix} \dot{\theta}_3 \\ \dot{\theta}_4 \\ \dot{\theta}_1 \end{vmatrix} = \begin{vmatrix} 0 \\ 0 \\ -1 \end{vmatrix} \dot{\theta}_2 \qquad [5-2]$$

Transforming matrix equation [5—2] into the set of simultaneous equations and solving with use of the Cramer's rule yield

$$\dot{\theta}_3=\frac{a_1\sin(\theta_2+\theta_3+\theta_4)}{a_3\sin\theta_4}\dot{\theta}_2$$

$$\dot{\theta}_4=-\frac{a_1\sin(\theta_2+\theta_3)}{a_4\sin\theta_4}\dot{\theta}_2-\dot{\theta}_3$$

$$\dot{\theta}_1 = -(\dot{\theta}_2 + \dot{\theta}_3 + \dot{\theta}_4)$$

where $\dot{\theta}_3$, $\dot{\theta}_4$ and $\dot{\theta}_1$ are the angular velocities of link 3 with respect to link 2, link 4 with respect to link 3 and link 1 with respect to link 4 respectively. $\dot{\theta}_2$ is the absolute angular velocity of link. 2.

Since $\dot{\theta}_3$ is the angular velocity of link 3 with respect to link 2, the anuglar velocity of link 3 with respect to the fixed link 1 can be obtained by solving the relative velocity equaty equationsas follows:

$$\omega_3 = \dot{\theta}_2 + \dot{\theta}_3 \qquad [6-1]$$

which states that the absolute angular velocity of link 3 is equal to the absolute angular velocity of link 2 plus the angular velocity of link 3 with respect to link 2. Similarly, the absolute angular velocity of link 4 is

$$\omega_4 = \dot{\theta}_2 + \dot{\theta}_3 + \dot{\theta}_4 \qquad [6-2]$$

Substituting $\dot{\theta}_2$, $\dot{\theta}_3$ and $\dot{\theta}_4$ into equations [6—1] and [6—2] gives

$$\omega_3 = 1 + \frac{a_1 \sin(\theta_2+\theta_3+\theta_4)}{a_3 \sin \theta_4}\omega_2 \qquad [6-3]$$

$$\omega_4 = 1 - \frac{a_1 \sin(\theta_2+\theta_3)}{a_4 \sin \theta_4}\omega_2 \cdot \qquad [6-4]$$

In these equations, ω_2, ω_3 and ω_4 are the absolute angular velocities of link 2, link 3, and link 4. Equations [6—3] and [6—4] give the absolute angular velocities of link 3 and link 4 in terms of the absolute angular velocity of link 2.

E. Acceleration equation

The time derivatives of the velocity equations [6—3] and [6—3] lead to the acceleration equations for link 3 and link 4. Differentiating equations [6—4] and [6—4] with respect to time and rearranging terms yield

$$\alpha_3 = \frac{a_1\omega_2\ \{\cos(\theta_2+\theta_3+\theta_4)\sin\theta_4\cdot\omega_4 - \sin(\theta_2+\theta_3+\theta_4)\cos\theta_4.\ (\omega_4-\omega_3)\}}{a_3 \sin^2\theta_4} + \frac{\omega_3}{\omega_2}\alpha_2 \qquad [7-1]$$

$$\alpha_4 = \frac{-a_1\omega_2\ \{\cos(\theta_2+\theta_3)\sin\theta_4.\ \omega_3 - \sin(\theta_2+\theta_3)\cos\theta_4\cdot(\omega_4-\omega_3)\}}{a_4 \sin^2\theta_4} + \frac{\omega_4}{\omega_2}\alpha_2 \qquad [7-2]$$

where α_2, α_3 and α_4 are the angular accelerations of link 2, link 3 and link 4. Equations [7—1] and [7—2] give the angular accelerations of link 3 and link 4 of the crank and rocker mechanism.

F. Displacemect of a coupler point

For the sake of definiteness, the right handed Cartesian coordinate system $X_iY_iZ_i$ is assumed to be fixed in each linkage of the crank and rocker mechanism as shown in Fig. 3. Any point on link i can easily be described by its coordinates $(X_iY_iZ_i)$ with respect to its coordinate system. Its position with respect to the reference coordinate of the mechanism can also be obtained by successively taking the linear transformation of the coordinate system from $X_iY_iZ_i$ into the reference coordinate system. If X_i represent the position vector of a point on link i with respect to the coordinate system $X_iY_iZ_i$, the position vector $\overline{X_i}$ of that point with respect to the reference coordinate can be determined by solving the relation between these two vectors as follows:

$$\overline{X}_i = T_i\, X_i \qquad [8-1]$$

where T_i is the matrix product of the displacement matrices which specify the linear transformations of the coordinate systems between the two position vectors.

$$T_i = A_1 \cdot A_2 \cdot A_3 \cdots\cdots A_{i-1}$$

The instantaneous position of any point on the moving link may be found from this matrix product.

In the crank and rocker mechanism shown in Fig. 3, the coupler point p can be specified by its coordinates (X_3Y_3) with respect to the $X_3Y_3Z_3$ coordinate system. The position vector of this point whith respect to $X_3Y_3Z_3$ may be defined by the column matrix as follows:

$$X_3=\begin{vmatrix} 1 \\ X_3 \\ Y_3 \end{vmatrix}.$$

In the plane mechanisms, the coordinates of Z_i are always zero or a constant. In this study it is assumed that the coordinates of Z_i are unity. From equation [8—1], the position vector of the coupler point with respect to the $X_1Y_1Z_1$ coordinate system can be expressed as:

$$\overline{X}_3=A_2A_3\begin{vmatrix} 1 \\ X_3 \\ Y_3 \end{vmatrix}. \qquad [8\text{—}2]$$

More generally, if the teference coordinate of the mechanism rotated about the origin of the $X_1Y_1Z_1$ coordinate system through an angle ϕ measured in counter-clockwise direction from the horizontal to the new position, the position vector of the coupler point with respect to the horizontal may be obtained by multiplying equation [8—2] by the rotation displacement matrix D as:

$$D=\begin{vmatrix} 1 & 0 & 0 \\ 0 & \cos\phi & -\sin\phi \\ 0 & \sin\phi & \cos\phi \end{vmatrix}$$

Multiplying equation [8—2] by the matrix D gives

$$\overline{X}_3=\begin{vmatrix} 1 \\ \overline{X}_3 \\ \overline{Y}_3 \end{vmatrix}=DA_2A_3\begin{vmatrix} 1 \\ X_3 \\ Y_3 \end{vmatrix} \quad [8\text{—}3]$$

If XYZ is the reference coordinate system established at the origin of $X_1Y_1Z_1$ with horizontal as X axis and vertical as Y axis, equation [8—3] gives the position vector of the coupler point with respect to the XYZ coordinate system. Substuting the matrices D, A_2 and A_3 into equation [8—3] and equating the corresponding elements, we get

$$\overline{X}_3=a_2\cos(\theta_2+\phi)+a_3\cos(\theta_2+\theta_3+\phi)$$
$$+U\cos(\theta_2+\theta_3+\phi+\psi)$$
$$\overline{Y}_3=a_2\sin(\theta_2+\phi)+a_3\sin(\theta_2+\theta_3+\phi)$$
$$+U\sin(\theta_2+\theta_3+\phi+\psi) \qquad [8\text{—}4]$$

where $U=\sqrt{X_3^2+Y_3^2}$

$\psi=\tan^{-1}\dfrac{Y_3}{X_3}$: degrees measured in counter clockwise direction

$\overline{X}_3$, $\overline{Y}_3$: coordinates of coupler point with respect to the XYZ coordinate system.

From equation [8—4], the coordinates of any point on the coupler link with respect to XYZ coordinate system can be determined.

G. Velocity and acceleration of a coupler peint

In general, epuations for velocity and acceleration can be obtained by successive differentiation of the equation for displacement with respect to time. Differentiating equation [8—4] yields

$$v_x=-a_2\omega_2\sin(\theta_2+\phi)-a_3\omega_3\sin(\theta_2+\theta_3+\phi)-U\omega_3\sin(\theta_2+\theta_3+\phi+\phi)$$
$$v_y=a_2\omega_2\cos(\theta_2+\phi)+a_3\omega_3\cos(\theta_2+\theta_3+\phi)+U\omega_3\cos(\theta_2+\theta_3+\phi+\psi). \qquad [9]$$

Equation [9] gives the equations for the velocity components of the coupler point in X and Y directions with respect to the XYZ coordinate system. The second time derivative of equation [8—4] is

$$A_x=-a\{\alpha_2\sin(\theta_2+\phi)+\omega^2{}_2\cos(\theta_2+\phi)\}-a_3\{\alpha_3\sin(\theta_2+\theta_3+\phi)+\omega^2{}_3\cos(\theta_2+\theta_3+\phi)\}-U\{\alpha_3\sin(\theta_2+\theta_3+\phi+\psi)+\omega^2{}_3\cos(\theta_2+\theta_3+\phi+\psi)\} \qquad [10]$$
$$A_y=a_2\{\alpha_3\cos(\theta_2+\phi)-\omega^2\sin(\theta_2+\phi)\}+a\{\alpha_3\cos(\theta_2+\theta_3+\phi)-\omega^2{}_3\sin(\theta_2+\theta_3+\phi)\}+U\{\alpha_3\cos(\theta_2+\theta_3+\phi+\psi)-\omega^2{}_3\sin(\theta_2+\theta_3+\phi+\psi)\}.$$

The X and Y components of the acceleration of the coupler point can be obtained using equation [10].

The epuations for determining the kinematic quantities are developed for the crank and rocker mechanism. These equations are used in the kinematic analysis of the planting mechanisms of the rice transplanters.

Angles θ_1, θ_2, θ_3, and θ_4 defined in Fig. 3 may be measured with a protractor. Measurements with a protractor alone will yield retults which are sufficiently accurate for the computation of the equations derived in the study.

4. Material and experiment

A planting mechanism of the power rice transplanter is shown in Fig. 5. The planting mechanism is a case of the four-bar-linkages where one crank is capable of rota-tion through a complete revolution while the second crank can only oscillate. The apex point on the planting fork which is part of link 3 in Fig. 5 may be called a coupler point in kinematics terms and produces a specific curve varying depending upon the lengths and arrangement of the linkages of the mechanism. The planting fork separates seedlings from a seedling board and plants them in the soil trailing along the specified path during each planting stroke.

Most planting devices employed in the power transplanters dan be classified to crank and rocker mechanism of the four-bar linkages similar to those mentioned. Analysis of the planting motion and related aspects of the planting devices, may be desirable not only to understand the motion involved but also to further improve the planting mechanisms.

The equations derived in the previous chapter are used to analyze the kinematics of the planting mechanism. For this study,

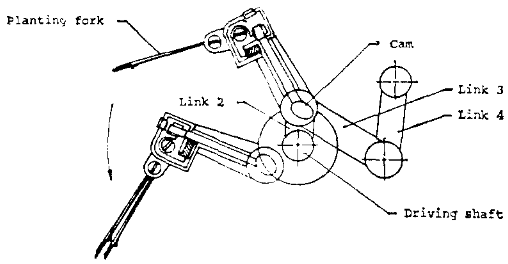


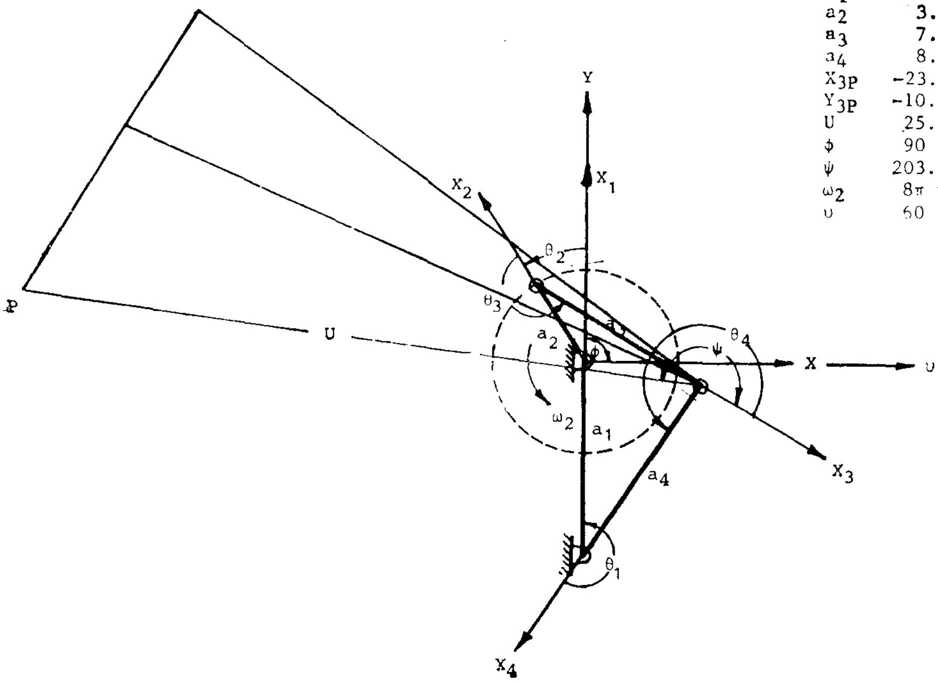
Fig. 5. Planting mechanism of rice transplanter.

three existing planting mechanisms were used: mechanism A and mechanism B from the Japanese power transplanters and mechanism C from an IRRI transplanter. Schematic diagrams of the mechanisms are shown in Fig. 6 through Fig. 8.

A. Analysis mechanisms

Mechanism A shown in Fig. 6 is employed in an engine powered two row transplanter made in Japan. The characteristic of this mechanism is that the driving link rotates backward in relation to the machine wheels. Mechanism B shown in Fig. 7 has been introduced by another Japanese two row power transplanter. The driving link rotates in the same direction as the machine wheels. Mechanism C shown in Fig. 8 is designed at the Agricultural Engineering Department of the International Rice Research Institute. The mechanism is to be employed in the planting system of the rice transplanter being developed at the Institute.

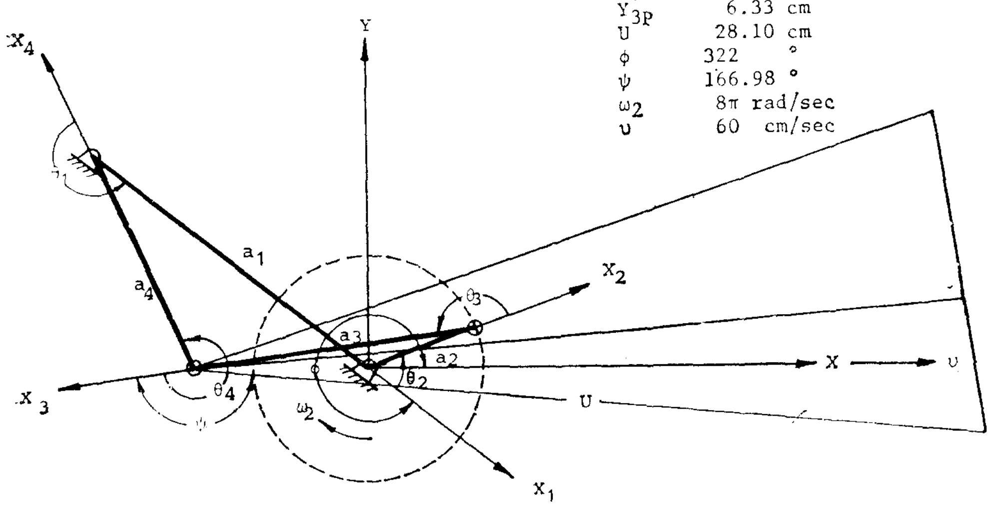
As presented in the schematic diagrams, the right handed Cartesian coordinate systems are established to be fixed in each linkage of the mechanisms. The horizontal and vertical lines passing through the center of the rotation of link 2 are designated as X and Y axes of the reference coordinate system in the mechanism. The other conventions rega-



Planting mechanism A

a_1	7.5	cm
a_2	3.5	cm
a_3	7.5	cm
a_4	8.0	cm
X_{3P}	−23.78	cm
Y_{3P}	−10.17	cm
U	25.86	cm
φ	90	°
ψ	203.15	°
ω_2	8π	rad/sec
υ	60	cm/sec

Fig. 6. Schematic diagram of planting mechanism A.



Planting mechanism B

a_1	12.5	cm
a_2	4	cm
a_3	10	cm
a_4	8.5	cm
X_{3P}	−27.38	cm
Y_{3P}	6.33	cm
U	28.10	cm
φ	322	°
ψ	166.98	°
ω_2	8π	rad/sec
υ	60	cm/sec

Fig. 7. Schematic diagram of planting mechanism B.

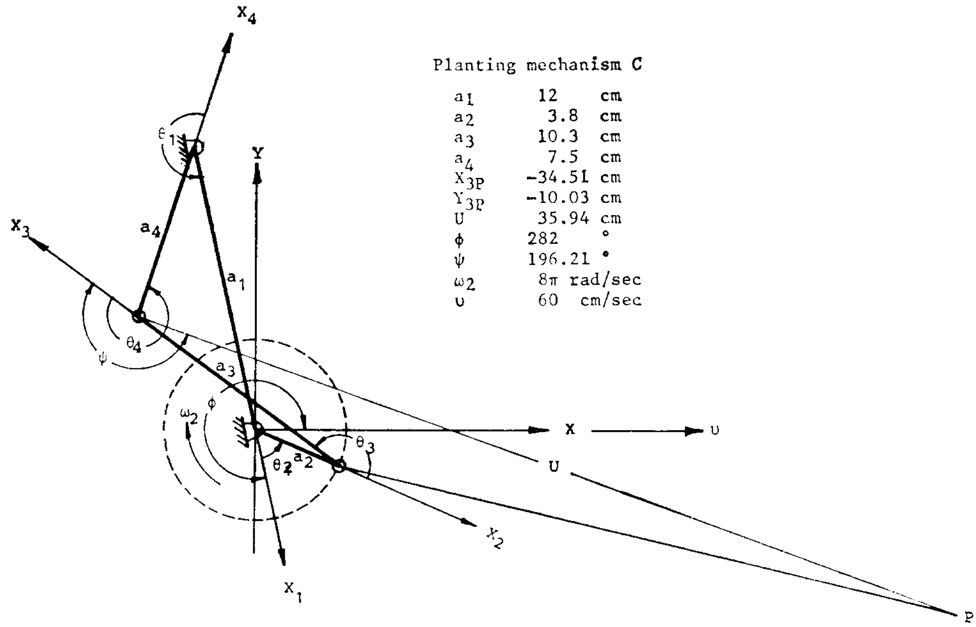


Fig. 8. Schematic diagram of planting mechanism C.

Table 1. Mechanism parmeters used for analysis of mechanism.

Notations	Description	Mechanism A	Mechanism B	Mechanism C
a_1	Length of link 1 (cm)	7.5	12.5	12
a_2	Length of link 2[a] (cm)	3.5	4	3.8
a_3	Length of link 3 (cm)	7.5	10	10.3
a_4	Length of link 4 (cm)	8.0	8.5	7.5
X_{3p}	X coordinate of coupler point with respect to $X_3Y_3Z_3$ (cm)	—23.78	—27.38	—34.51
Y_{3p}	Y coordinate of coupler point with respect to $X_3Y_3Z_3$ (cm)	—10.17	6.33	—10.03
U	Magnitude of position vector of coupler point (cm)	25.86	28.10	35.94
ϕ	Angle of orientation of $X_1Y_1Z_1$ (degree)	90	322	282
ψ	Direction of position vector of coupler point with respect to $X_3Y_3Z_3$ (degree)	203.15	166.98	196.21
ω_2	Angular velocity of link 2 (rad/sec)	8π	8π	8π
ν	Machine forward speed (cm/sec)	60	60	60

[a]Driving link

rding the signs of the axes and rotations are similar to those established in the previous chapter.

The dimensions of the links and angles were measured using a scale and protractor. Table 1 shows the dimensions of the links and angles which were actually measured from the three mechanisms.

Computations were performed at intervals of 10 degrees of the crank angle using a portable programmable calculator. The forward velocity of the mechanism relative to the ground was assumed to be 60 cm per second and link 2 turned with a constant angular velocity of 240 revolutions per minute. The positive sense of X axis of the reference coordinate was assumed to be the same direction as the machine wheeled.

B. Torque measurement

The external torque necessary to drive the planting mechanism was measured experimentally using a Shinkoh TM 5B torque meter. The torque meter had a measuring capacity of 5 kg-m and mounted on the driving shaft of the mechanism. The angular velocity of the driving shaft was changed using a variable speed motor and indicated on the tachometer connected to a transducer installed in the driving shaft. The torque obtained at three levels of the angular velocities were automatically recorded on Beckman RS dynograph.

To determine the torque when the planting fork entered the soil, the planting device and torque meter were installed in a small carriage and pulled through the soil bin by a motor with a constant forward velocity of 60 cm per second. The soil bin, 367cm long and 32cm wide, was uniformly supplied with water 2 hours before the measurement began. Zero cm of water depth was maintained during the test. The soil conditions of the soil bin were kept the same as those of the field by providing the same falling cone depth that had been obtained at the field to be transplanted. A cone with 127.5 gram weight, 3.6 cm diameter and a 45 degree apex angle was used. The reading of the falling cone depth from 50 cm of falling height was 15.73 cm both in the field and in the soil bin.

5. Analysis and results

A. Displacement

As the angular displacement θ_2 of the driving link increased by 10 degree intervals, the corresponding relative angular displacements between link 2 and link 2 and link 3 and link 3 and link 3 and link 4 were measured with a protractor The relative angular displacements were defined as pair variables θ_3 and θ_4 in chapter 3.

The coordinates of point P on the planting fork were calculated using equation [8—4] through a complete revelution of the driving link. Connecting these coordinates successively produces a path traced by point P relative to the XYZ coordinate system used as a reference coordinate of the mechanism. When the machine moves with a constant forward velocity v, the position of point P relative to the ground can be specified by the equations as follows:

$$X=a_2\cos(\theta_2+\phi)+a_3\cos(\theta_2+\theta_3+\phi)+U\cos(\theta_2+\theta_3+\phi+\varphi)+v\frac{\theta_2}{\omega_2}$$
$$Y=a_2\sin(\theta_2+\phi)+a_3\sin(\theta_2+\theta_3+\phi)+U\sin(\theta_2+\varphi_3+\phi+\psi) \quad [11]$$

Fquation [11] expresses the coordinates of point P relative to the ground as a function of angular displacement θ_2. The paths of the coupler point P relative to the XYZ coordinate system and relative to the ground are shown in Fig. 9 through Fig. 11 respectively. Because the planting devices used for the analysis equipped with a control system of planting depth it was assumed that planting depth would be four centimeters. The coupler curve relative to the ground obtained from mechanism A was 7.4 cm wide and 25 cm high. Those of mechanisms B and C were 9.8 cm wide and 22.3 cm high and 12.7 cm wide and 25 cm high, respectively. The planting fork of mechanism A retracted directly from the planting position after releasing seedlings. In mechanisms B and C, the planting fork made a small loop at

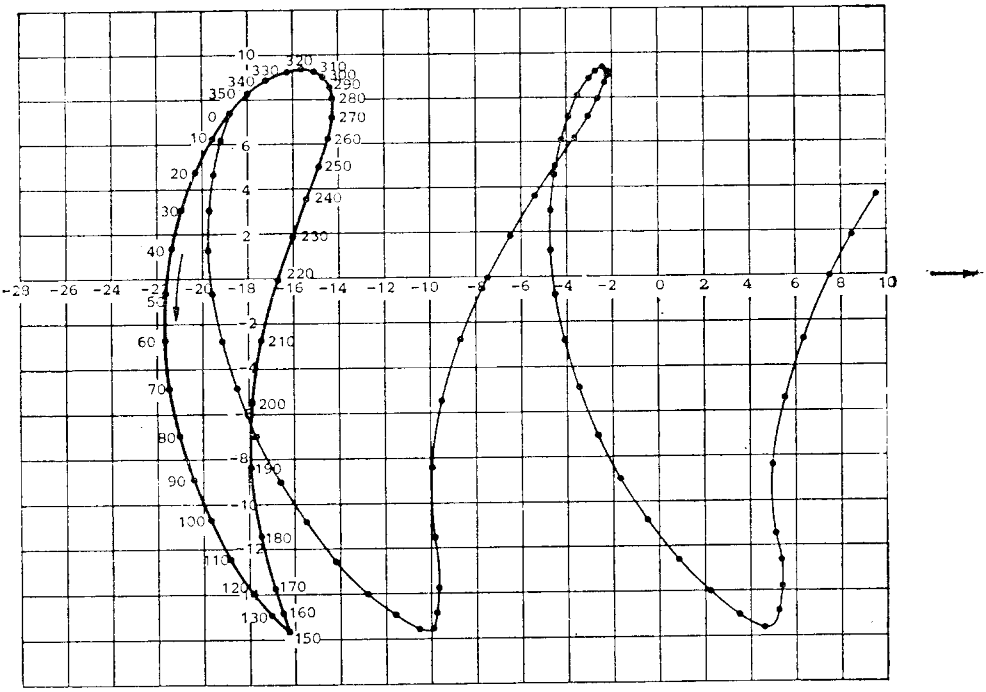


Fig. 9. Coupler curve of mechanism A relative to machine frame and ground.

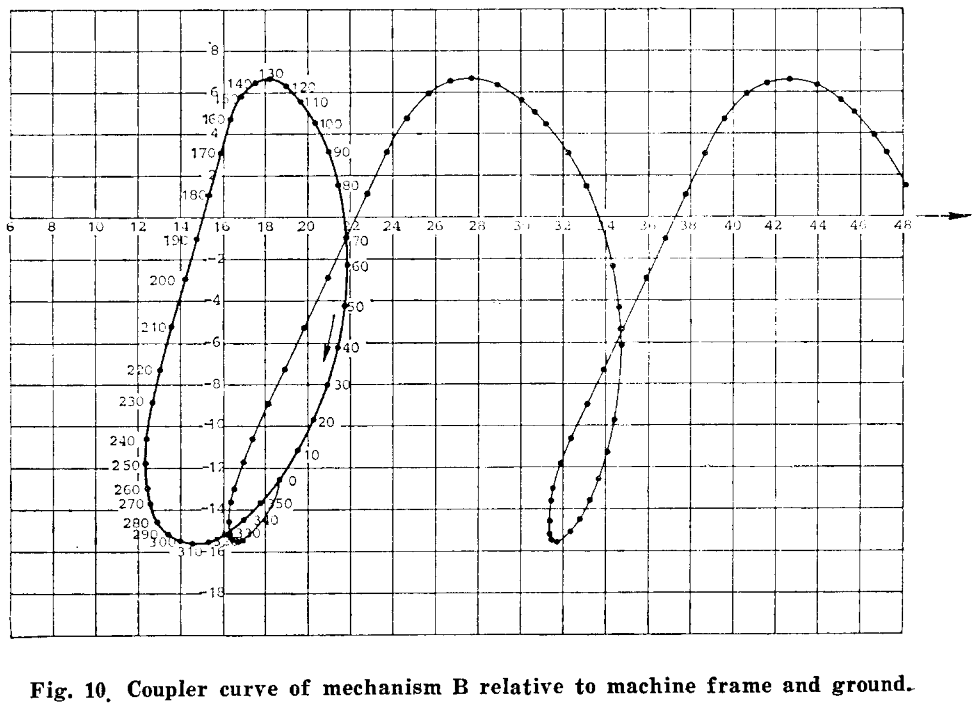


Fig. 10. Coupler curve of mechanism B relative to machine frame and ground.

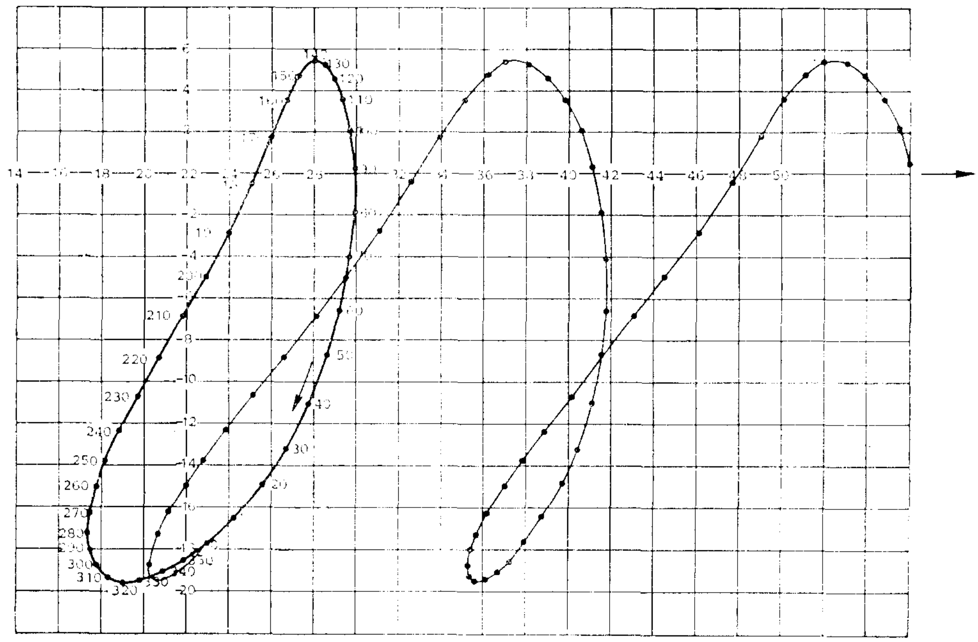


Fig. 11. Coupler curve of mechanism C relative to machine frame and ground.

the moment of seedling release. The loop makes the seedlings more straight after planted though it may cause more soil resisting force and damage to seedlings by the fork. The lengths of the path traced by the fork through the soil relative to the ground were 10.6 cm in mechanism A, 9.2 cm in mechanism B and 9.8 cm in mechanism C. During this phase, the driving link of mechanism A rotated 25 degrees, 113 degrees for mechanism B and 112.5 degrees for mechanism C. The planting fork of mechansm A moved a longer distance in a shorter period. The frictional force between the planting fork and the soil may be proportional to the amount of displacement which the planting fork traveled through the soil. From this point of view, mechanism B has a coupler curve producing less frictional force. The shape characteristics of the coupler curve generated by the fork during each planting stroke influence the straightness of the planted seedlings and seedling damage by the fork. It may be desirable for the approach angle of the fork to the seedling to be paralled with each other to reduce the seedling damage.

B. Velocity and acceleration

The angular velocity ratios of the driven link to the driving link ω_4/ω_2 and the coupler link to the driving link ω_3/ω_2 were obtained by dividing both sides of equations [6—3] and [6—4] by the angular velocity of the driving link. The equations thus obtained are:

$$\omega_3/\omega_2=1+\frac{a_1\sin(\theta_2+\theta_3+\theta_4)}{a_3\sin\theta_4}$$

$$\omega_4/\omega_2=1-\frac{a_1\sin(\theta_2+\theta_3)}{a_4\sin\theta_4}\quad .$$

The maximum of velocity ratio ω_3/ω_2 oc-

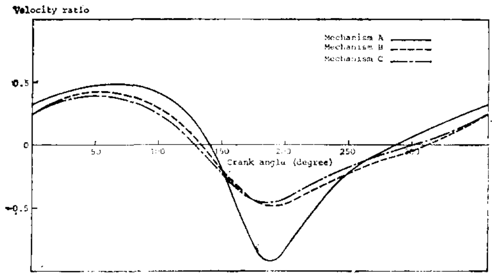


Fig. 12. Angular velocity ratio, ω_3/ω_2.

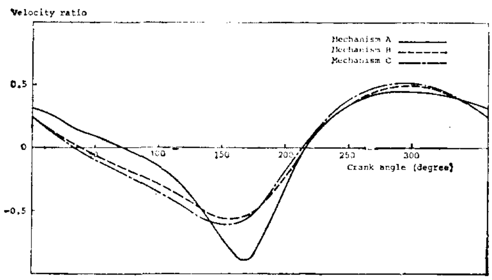


Fig. 13. Angular velocity ratio, ω_4/ω_2.

curred when the driving link had an angular displacement of 190 degrees in mechanism A, 195 degrees in mechanism B and 180 degrees in mechanism C. The velocity ratio at this phase was −0. 92, −0. 49 and −0. 47 respectively. The maximum value of ω_4/ω_2 occurred at 180 degrees in mechanism A, 155 degrees in mechanism B and 155 degrees in mechanism C and was −0. 89, −0. 57, and −0. 62 respectively. The negative values indicate that the angular velocities of link 3 or link 4 are opposite in sense to that of link 2.

Fig. 12 and Fig. 13 present velocity ratios ω_3/ω_2 and ω_4/ω_2 for one cycle motion of the driving link. The velocity components of point P in the x and y directions with respect to the reference coordinate were obtained using equation [9]. The resultant velocity can be then determined by substituting the velocity components into the equation as follows:

$$v=\sqrt{v^2_x+v^2_y}$$

and its directon is given by

$$\theta=\tan-1\frac{v_y}{v_x}.$$

Velocity of the planting fork, when entering the soil was 2. 7 m per second in mechanism A, 2. 33 m per second in mechanism B and 2. 95 m per second in mechanism C. As the fork reached the lowest position of the coupler curve, the velocity decreased to 0. 28 m per seconcd, 0. 97 m per second and 1. 09 m per second respectively. The fork left the soil surface with a velocity of 3. 94 m per seond in mechanism A, 1. 8 m per second in mechanism B and 1. 36 m per second in mechanism C.

Acceleration at the entry positon was 49. 6 m/sec^2 in mechanism A, 49. 7 m/sec^2 in mechanism B and 61. 8 m/sec^2 in mechanism C. Accelerations at the soil exit position were 146. 1 m/sec^2, 50m /sec^2 and 50 m/sec^2 in mechanisms A, B and C respectively. The velocity diagrams of the planting fork and its X and Y components in each mechanism were plotted with the velocity as the ordinate and the crank rotation angle as the obscissa. These are shown in Fig. 14 through Fig. 16. The acceleration diagrams are shown in Fig. 17 through Fig. 19. The velocity and acceleration diagrams provide the complete pictures of the velocity and acceleration characteristics of the planting mechanism. High acceleration of the planting fork while in the soil may result in floating hills because the resulting inertia forces may disturb the ability of soil to retain the planted seedlings. High flucturations in acceleration may also cause a severe vibration during high speed operation. Table 2 presents the kinematic quantities of the mechanism.

Table 2. Kinematic characteristics of mechanism.

	Mechanism A	Mechanism B	Mechanism C
Planting depth (cm)	4	4	4
Coupler curve			
Height (cm)	25	22.3	25
Width (cm)	7.4	9.8	12.7
Angular displacement of crank in the soil (degree)	75	119	112.5
Length of path traced in the soil (cm)	10.6	9.2	9.8
Horizontal distance from entry to exit (cm)	5	2	2.7
Entry position			
Crank angle (degree)	104	7.5	16.5
Velocity X component (m/sec)	1.32	−1.2	1.8
Y component (m/sec)	−2.42	−2.0	−2.34
Resultant (m/sec)	2.7	2.33	2.95
Entry position			
Acceleration X component (m/sec^2)	16	−13	−10
Y component (m/sec^2)	47	48	61
Resultant (m/sec^2)	49.6	49.7	61.8
Exit position			
Crank angle (degree)	179	248.5	264
Velocity X component (m/sec)	−0.85	0.1	−0.32
Y component (m/sec)	3.85	1.8	1.32
Resultant (m/sec)	3.94	1.8	1.36
Acceleration X component (m/sec^2)	51	30	40
Y component (m/sec^2)	137	41	30
Resultant (m/sec^2)	146.1	50	50
Lowest position			
Crank angle (degree)	150	310	320
Velocity X component (m/sec)	0.12	0.97	1
Y component (m/sec)	0.27	0.05	0.1
Resultant (m/sec)	0.28	0.97	1.09
Acceleration X component (m/sec^2)	−89	15.5	33.7
Y component (m/sec^2)	142	45.7	50.8
Resultant (m/sec^2)	167.8	48.2	61
Maximum velocity			
Crank angle (degree)	190	195	190
Magnitude (m/sec)	4.43	3.24	3.68
Direction (degree)	93.58	75.44	63.73
Minimum velocity			
Crank angle (degree)	150	300	135
Magnitude (m/sec)	0.298	0.914	0.858
Direction (degree)	66.36	160.13	−30.56
Maximum acceleration			
Crank angle (degree)	170	150	140
Magnitude (degree)	189.91	106.67	133.43
Direction (degree)	100.07	−82.24	−92.17
Minimum acceleration			
Crank angle (degree)	85	190	190
Magnitude (m/sec^2)	44.85	8.85	21.03
Direction (degree)	22.09	−119.4	116.56

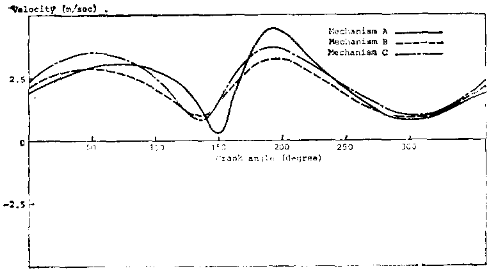


Fig. 14. Velocity of planting fork relative to machine frame.

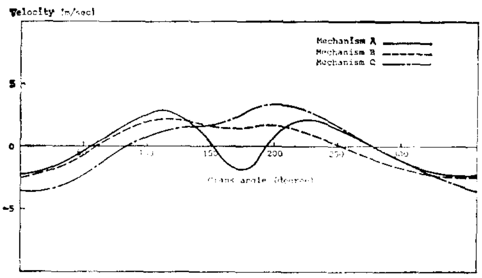


Fig. 15. Horizontal component of velocity of planting fork relative to machine frame.

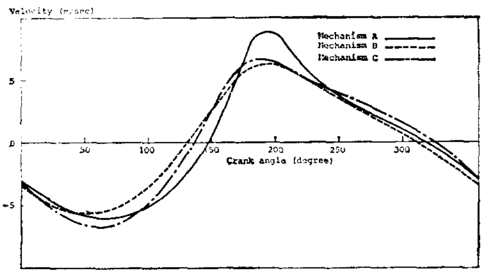


Fig. 16. Vertical component of velocity of planting fork relative to machine frame.

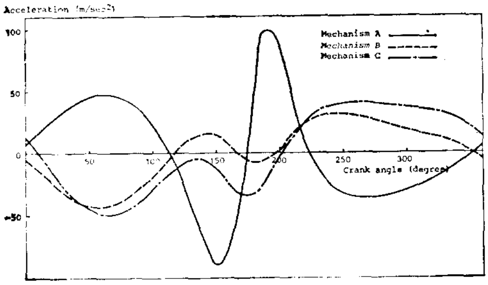


Fig. 18. Horizontal component of acceleration of planting fork relative to machine frame.

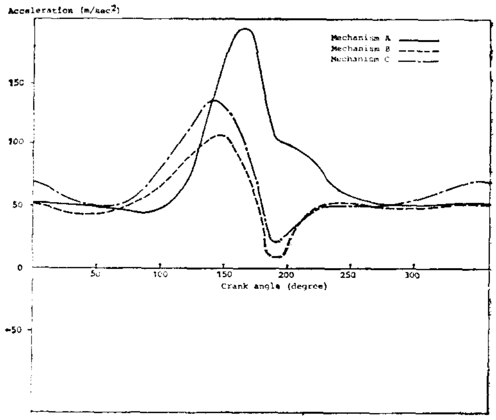


Fig. 17. Acceleration of planting fork relative to machine frame.

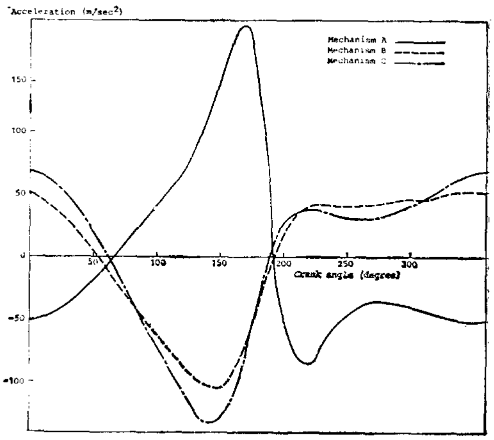


Fig. 19. Vertical component of acceleration of planting fork relative to machine frame.

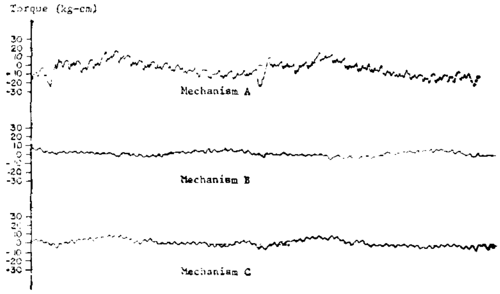


Fig. 20. Torque measurement at t00 rpm of driving link.

C. Torque analysis

The external torque requirements were measured when the driving link had constant angular velocities of 100, 160 and 240 revolutions per minute. In general, torque requirements increased with an increase in angular velocity. The torque measured are shown in Fig. 20 through Fig. 22.

Mechanism A required a higher torqu at a lower angular speed. However, torque requirement increased rapidly with an increase in angular speed. The greater torque requirement of mechanism A seemed to be caused by the heavier coupler link and a cam mechanism inside the planting fork which is used to release the seedlings in the soil. The mass moment of inertia of the coupler link was 47. 29 g-sec^2-cm in mechanism A, 37. 02 g-sec^2-cm in mechanism B and 16. 13 g-sec^2-cm in mechanism C.

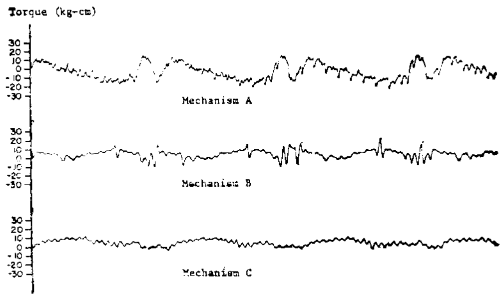


Fig. 21. Torque measuerment at 160 rpm of driving link.

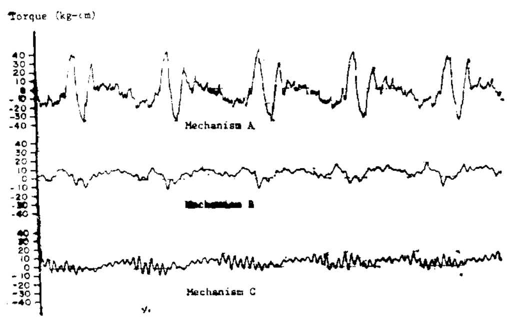


Fig. 22. Torque measurement at 240 rpm of driving link.

In mechanisms B and C, there was non-significant differences in torque requirements though they empoloyed quite different coupler link in the mass moment of inertia. It is understood that the cam mechanism may have the main effect to the higher torque occurrence in mechanism A.

Maximum torque obtained at three different velocities of the driving link are shown in Table 3 as follows:

Table 3. Maximum torque of the mechanism.

	Mechanism A (kg-cm)	Mechanism B (kg-cm)	Mechanism C (kg-cm)
100 rpm	23. 33	6. 67	6. 67
160 rpm	16. 67	13. 33	10
240 rpm	40	13. 33	13. 33

To measure the torque requirements when the planting fork traveled in the soil, the mechanism was pulled through the soil bin with a constant forward velocity of 60 cm per secound while, at the same time, the driving link of the mechanism rotated with a constant angular velocity of 240 revolutions per minute.

Fig. 25 shows the torque diagram so obtained. When the fork entered the soil the torque measured was 53. 33 kg-cm for mechanism A and 33. 33 kg-cm for mechanisms

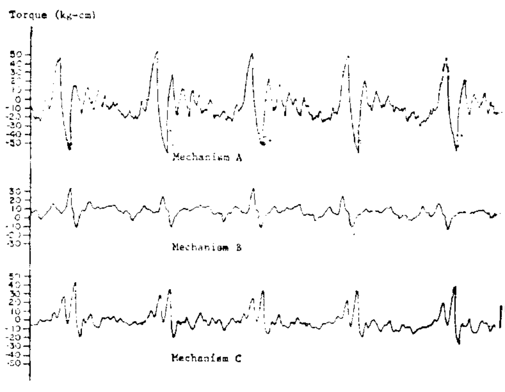


Fig. 23. Torque measurement in the soil at 240 rpm of driving link.

B and C respectively.

6. Summary and conclusion

An analytical method to determine the kinematic quantities of the crank and rocker mechanism was developed and applied to the analysis of the plnnting mechanism of a rice transplanting machine. The crank and rocker mechanism can be described completely by symbolic equation with kinematic notations. The equations derived herein are based upon the geometric transformations of the coordinate systems fixed to each linkage of the mechanism using 3×3 matrices.

The equations are:

1) Angular velocity of coupler link

$$\omega_3=\left[1+\frac{a_1\sin(\theta_2+\theta_3+\theta_4)}{a_3\sin\theta_4}\right]\omega_2$$

2) Angular velocity of rocker

$$\omega_4=\left[1-\frac{a_1\sin(\theta_2+\theta_3)}{a_4\sin\theta_4}\right]\omega_2$$

3) Angular acceleration of coupler link

$$\alpha_3=\frac{a_1\omega_2\{\cos(\theta_2+\theta_3+\theta_4)\sin\theta_4\omega_4-\sin(\theta_2+\theta_3+\theta_4)\cos\theta_4(\omega_4-\omega_3)\}}{a_3\sin^2\theta_4}+\frac{\omega_3}{\omega_2}\alpha_2$$

4) Angular acceleration of rocker

$$\alpha_4=\frac{-a_1\omega_2\{\cos(\theta_2+\theta_3)\sin\theta_4\omega_3-\sin(\theta_2+\theta_3)\cos\theta_4(\omega_4-\omega_3)\}}{a_4\sin^2\theta_4}+\frac{\omega_4}{\omega_2}\alpha_2$$

The position vector of any point on a link in the crank and rocker mechanism can be expressed with respect to the reference coordinate system using the following matrix equation:

$$\begin{vmatrix}1\\x\\y\end{vmatrix}=D\ A_2A_3\cdots A_n\begin{vmatrix}1\\x_n\\y_n\end{vmatrix}\quad n=2,3,4$$

where $\begin{vmatrix}1\\x\\y\end{vmatrix}$ is the position vector of a point on link n relative to the reference coordinate

$\begin{vmatrix}1\\x_n\\y_n\end{vmatrix}$ is the position vector of a point on link n relative to the coordinate system fixed in link n

D is the rotation displacement matrix

Ai is the displacement matrix

Successive differentiation of the matrix equation for the position of the point with respect to time will yield the equations for velocity and acceleration of that point. In addition, the external torque necessary to drive the mechanism was measured experimentally.

References

1. Cowie, Alexander. 1961. Kinematics and design of mechanism. International Text Book Company. Scranton, Pennsylvania.
2. Denavit, J. and R.S. Hartenberg. 1955. A kinematic notation for lower-pair mechanisms based on matrices. Trans. ASME, Vol. 77, pp. 215—221
3. Denavit, J., R.S. Hartenberg, R. Razi and J.J. Vicker Jr. 1965. Velocity, acceleration and static-force analyses of spatial linkages. Trans. ASME, Vol. 87, pp. 903—910.
4. Editorial Department, Shin-Norinsha Co., Ltd. 1971. The latest mechanization of rice transplanting in Japan. Agricultural Mechanization in Asia, Autumn.
5. Endo, Shunzo, Fukuji Miyazawa, Toshio Konaka and Hiroyuki Shinozaki. 1972. Studies on the rice transplanter for soil removed seedling. (in Japanese, English summary). Journal of Center of Agricultural Experiment Station Japan, No. 16,

pp. 87—129.

6. Frazer, R.A., W.J. Duncan and A.R. Collar. 1965. Elementary matrices. The Syndice of the Cambridge Universiiy Press. Bently House, 200 Euston Road, London, N.W.I.
7. Hinkel, Rolland T. 1960. Kinematics of machines. Prentice Hall Inc. Englewood Cliffs, N.J.
8. Hoshino, Seiji. 1974. Recent advances on rice transplanter. JARQ, Vol. 8, No. 4. pp. 209—213.
9. Ichikawa, Massauke. 1975. Kinematics of rice transplanter. (in Japanese, English summary). Journal of the Society of Agricultural Machinery, Japan, Vol. 34, No. 3, pp. 226.
10. Martin, G.H. and M.F. Spotts. 1957. An application of complex geometry to relative velocities and accelerations in mechanism. Trans. ASME, Vol. 79, pp. 687—693.
11. Namikawa, Kiyoshi. 1976. Link analysis of the rice transplanting mechanism. (in Japanese, English summary). Journal of the Society of Agricultural Machinery, Japan, Vol. 37, No. 3, pp. 295—299.
12. Okamura, S. 1969∼1970. Each rice transplanter-present condition and prbolem. (in Japanese). Farming Mechanization.
13. Raven, F.H. 1958. Velocity and acceleration analyses of plane and space mechanisms by means of independent position equation. Trans. ASME, Vol. 25, pp. 1—6.
14. Ryu, kwan Hee. 1972. Kinematic analysis of planting mechanism of rice transplanters. Department of Agricultural Engineering, Graduate School, Seoul National University. Suwon, Korea.
15. Shigley, Foseph Edward. 1961. Theory of machine. McGraw Hill Book Inc., N.Y.
16. Shu, C.H. and C.W. Radcliff. 1967. Synthesis of plane linkages with use of the displacemet matrix. ASME Paper No. 66-Mech 19, Trans. ASME, Series B, pp. 206—214.

APPENDIX I-1

Point P: Apex point of the planting fork.

XPA : X coordinate of point P relative to the reference coordinate system.

YPA : Y coordinate of point P relative to the reference coordinate system.

XPR : X coordinate of point P relative to the ground.

VXPA : X component of velocity of point P relative to the reference coordinate system. (cm/sec)

VYPA : Y component of velocity of point P relative to the reference coordinate system. (cm/sec)

VXPR : X component of velocity of point P relative to the ground. (cm/sec)

VP : Resultant velocity of point P relative to the reference coordinate system. (cm/sec)

VXC : Velocity of the center of mass of planting fork in X direction relative to the reference coordinate system. (cm/sec)

VYC : Velocity of the center of mass of planting fork in Y direction relative to the reference coordinate system. (cm/sec)

AXP : X component of accelaertion of point P relative to the reference coordinate system. (cm/sec^2)

AYP : Y component of acceleration of point P relative to the reference coordinate system. (cm/sec^2)

AP : Resultant acceleration of point P

relative to the reference coordinate system. (cm/sec²)

AXC : Acceleration of the center of mass of planting fork in X direction relative to the reference coordinate system. (cm/sec²)

AYC : Acceleration of the center of mass of planting fork in Y direction relative to the reference coordinate system. (cm/sec²)