Design of four-bar mechanism for vibratory tillage cultivator using five precision position method for path generation problem

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Abstract: Agriculture is the backbone of the Indian economy. There are different agricultural operations that play a critical role in overall crop development. Farmers are in search of sustainable technologies that can contribute to their overall economy and improve efficiency. The paper deals to design a vibratory cultivator that saves the time and energy for the farming community. A four-bar mechanism is designed and developed to vibrate the cultivator. A proper analytical synthesis procedure is adopted, such as five precision position method for a particular vibratory tillage trajectory available from the literature. The dimensions of the four-bar mechanism are validated through MATLAB, and it is confirmed that the designed four-bar mechanism traces the path accurately for vibratory tillage operation.

Keywords: tillage; four bar mechanism; synthesis; precision position; trajectory.

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1 Introduction

Indian is an agricultural country, and around 54% of the Indian population is engaged in agriculture and related activities as their primary source of occupation and income (Bhatt et al., 2016). The key issues faced by Indian agriculture are inadequate agricultural storage, improper irrigation facilities, decreasing farm size, improper level of farm mechanisation, lack of soil nutrients, and dependence on monsoon (Singh et al., 2014).

Agriculture output has not been stable over the last couple of years, and there is a demand for improved farm agricultural implements that can perform the operations with fewer no of passes in the agricultural fields (Singh, 2016). In Indian more than 86% of the total land holding belongs to small and marginal farmers and this varied different agro condition forces the farmers to adopt more refined, efficient and improved farm implements and machinery that can perform the same level of operations in a single run as compared to traditional implements. Thus there is a need to work in these areas (Report, 2016).

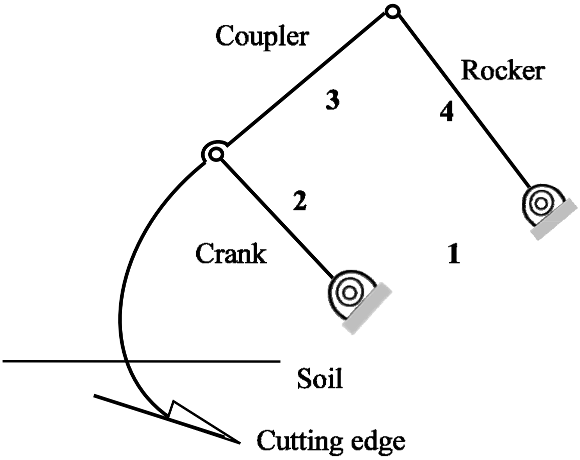
There are different tillage systems in agriculture. A combination tillage is one such option for improved tillage operation due to its many advantages. One of the reasons is the reduction of no passes of tillage equipment without compromising the quality of work. Another method in tillage systems is active tillage tools which have several advantages such as reduced draft requirements, improved soil properties, and power consumption of the system (Manian and Kathirvel, 2001).

Tillage is classified into two types, such as primary and secondary tillage. Nowadays, there is a shift among the farming community for efficient energy-saving tillage tools which can perform both primary and secondary tillage in one run. The initial soil depth of 20–25 cm is obtained in the primary tillage operation for better crop roots penetration in the soil while in secondary tillage operation it is around 10–15 cm for better soil aeration and is performed after the primary tillage operation. Different implements such as mouldboard plough, disc plough, subsoiler, cultivator, disc harrow are available for tillage operations (Singh, 2005). But the main limitation is the draft and power consumption in these implements. Thus, there is a need to work for the low draft tillage tools which can contribute to a farmer's economy by keeping the same level of work as performed by traditional tillage tools.

2 Active tillage tools

Active tillage plays an important role in agricultural operations (Gunn and Tramontini, 1955; Rao et al., 2019a, 2019b). There are different equipment available in active tillage systems that can perform the same and improved the level of operations as traditional tools. Tillage tools can be classified into two types one is an active tillage tool, and another is a passive tillage tool. Active tillage tools are multipowered tools in which the power is supplied to the tillage tool through power take-off (PTO) shaft and other transmission systems. Active tillage tools play an important role in agricultural operations (Upadhyay and Raheman, 2019; Singh and Chaudhary, 2019). There are several advantages in developing an active tillage tool that aims to achieve the same level of operation and energy saving. Oscillations are provided to the tillage tool with a certain amplitude and frequency, and this is called a vibratory tillage concept. The crank mechanism is provided to provide vibrations to the tillage tool, as shown in Figure 1. Through the gearbox assembly, the PTO rpm is reduced for the required crank rpm (Xirui et al., 2016; Shahgoli et al., 2009).

Figure 1 Vibratory mechanism in tillage tool



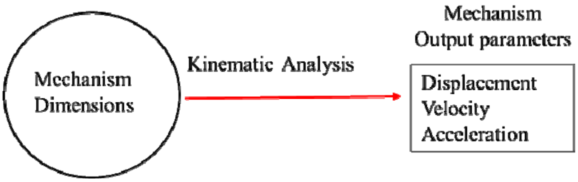
Source: Bernacki et al. (1972)

The concept of vibration in tillage implements came long back during 1955 by Gunn and Tramontini. There are several various advantages of oscillation mode tools such as reduced draft consumption, improved soil properties, increased soil crumbling efficiency. But there is a conflict regarding power requirement that it may increase, decrease, and remain the same during the tillage operation.

3 Synthesis of mechanism

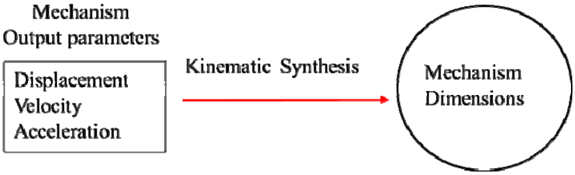
Planar linkages are classified into three types, such as graphical, analytical, and optimal methods (Sandor and Erdman, 1984). Kinematic synthesis is used to calculate the mechanism dimensions for the given output parameters. Three different categories of kinematic synthesis are classified as path motion and function generation (Russell et al., 2013). Path refers to the achieving of the path points as output parameters. Motion generation refers to the design of the mechanism for reaching the rigid body positions as an output parameter (Norton, 2009). Figure 2 and Figure 3 described the difference between analysis and synthesis.

Figure 2 Kinematic analysis (see online version for colours)



Source: Russell et al. (2013)

Figure 3 Kinematic synthesis (see online version for colours)



Source: Russell et al. (2013)

There are different methods to solve through kinematic synthesis. Mainly two methods of kinematic synthesis are classified one is analytical, and another one is graphical technique. In a graphical method, it does not include the computational effort to calculate the problem while in case of an analytical method different mathematical models can be used to solve it analytically and numerically through different optimisation algorithm.

The graphical method is simple and easy for the designers to fast and accurate design. There is a lot of repetitiveness and complexity involved during the graphical synthesis procedure. The error also is involved during this process. A lot of geometrical constructions are involved in achieving a solution that passes through the prescribed precision position. There are three methods for graphical synthesis one is two precision positions, three precision positions, and four precision positions. The analytical synthesis also covers the two, three, four, and five precision position synthesis techniques (Sandor and Erdman, 1984). The paper consists of designing a four-bar mechanism through the synthesis procedure for providing vibratory motion to the tillage tool. Different methods are explored but in this paper, only five precision point method is concentrated which gives the better dimensions of four-bar mechanism for the following tillage operation.

4 Related works

The concept of vibratory tillage is studied and applied and for mechanical manipulation of the soil by Johnson and Buchele (1969). The objective of the study is to evaluate vibratory tillage tool systems on clod size distribution, the total energy required for clod size distribution under different soil condition. Results concluded that soil particle size is greatly influenced by the vibratory tillage tool. The energy input for clod size distribution is affected through vibratory tillage. A digger blade for potato harvesting is developed by Johnson (1974). Different objectives are studied and implemented such as minimum damage to potato tuber, reduced draft for harvester, material feeding unit into the harvester, enhance soil separation which resulted in the improved efficiency of the machine. Thus the working efficiency of the machine is enhanced by considering all the points while digging the potato in the field.

A vibrating furrow opening tool is designed for a minimum tillage operation in the planting system by Tompkins and Bledsoe (1979) and through field tests, the system performance is assessed in the field. The impact of the vibratory tool is compared to the fluted coulter type opener based upon the performance. It is observed that the vibratory tool performed better than the non-vibratory system and more power is required by the vibratory tool as compared to non-vibratory one. A series of soil tank experiments are performed by Butson and Macintyre (1981) to measure the draught and power requirement for soil cutting. Sinusoidal vibrations are applied in the direction of the travel. Different frequencies and amplitudes up to 50 Hz and 8 mm are examined for two different speeds. The draught reduction is found to be reduced by 50 % when the velocity ratio is greater than 1. But power reduction is increased. Thus from the study, it is concluded that vibrations play an essential role in high draft equipment such as subsoiler, chisel plough.

A mathematical model is developed by Butson and Rackham (1981) for draught prediction and compared with the existing models for prediction of draught than any other model. Critical factors are considered taking into consideration the several parameters for effective prediction of draught than other available models. A rotary chisel is designed and developed by Hendrick (1980). Field trials are performed in the soil bin and it is observed that rotary chisel disturbed more amount of soil than a rigid type chisel.

A mechanism is designed by Narayanarao and Verma (1982) for providing the oscillations to the tillage tool. A quick return mechanism is designed and installed for providing simple harmonic motion to the tillage tool. The work also consisted to develop mathematical modes for the prediction of draft and power consumption for vibratory soil working tool. The developed system is tested for performance parameters in the soil bin measuring the draft, vertical force, and tool velocity. Thus in this study theoretical values are compared with the predicted values.

A vibratory potato digger is developed using the orbital vibration theory by Al-Jubouri and Mcnulty (1984). The motion of the digging blade and soil forces are considered to design the machine. In this, the velocity ratio is a key factor for the performance of the machine and it is observed that as the velocity ratio increases the draught force decreases, and the power requirement is increased. After theoretical validation, experimental tests are conducted using a prototype of vibratory potato digger at a speed of 3 km/h, depth of 200 mm, the amplitude of 10–25 mm, and frequency of 7.5–18 Hz respectively. Thus there is a reasonable agreement between the experimental and predicted draught requirements and it is observed that potato damage and losses decreased as compared to conventional non-vibratory potato diggers.

A series of experiments conducted by Niyamapa and Salokhe (2000) on vibratory tillage tool in sandy loam soils evaluated the various performance parameters. It is observed that the draft consumption increases initially with an increase in forward speed but later it decreases with the increase in the forward speed. The power requirement is also more as compared to non-vibrating one around 41–45 % more. Soil pulverisation is better as compared to the non-oscillating one.

A two-row deep oscillating ripper is developed by Shahgoli et al. (2009) for Australian vineyards to break the compacted hardpan soil. System parameters such as the oscillation angle played a vital role in the performance. Optimum oscillation angles are identified for a combination of six different angles in sandy loam soil. The frequency and amplitude are found to be 4.9 Hz and 60–69 mm, respectively. Draft reduced by 50 % for negative oscillation angles. It is observed that power also significantly effected with changing oscillation angles.

A two tine oscillatory subsoiler for small tractors is developed by Shahgoli et al. (2010a) and a dynamic simulation model is studied to see the unbalanced forces produced. Three different phases such as soil cutting, backing off, and catching up are considered for modelling the dynamic simulation model. Various parameters are taken into the account such as positive and negative oscillation angles, the inertia of the tine, tyre stiffness, and damping parameters while modelling the model. The final results are compared with field measured parameters and it is observed that there is a good correlation between them.

A deep working oscillatory subsoiler is developed and tested in sandy loam soil by Shahgoli et al. (2010a). The most efficient setting is identified, and the comparison between straight and bent leg tines are made. Vibrational parameters such as frequency are varied and it is observed that the draft reduced from 25.8 kN to 9.3 kN. The optimum frequency setting is proposed for operating the subsoiler at the optimal draft.

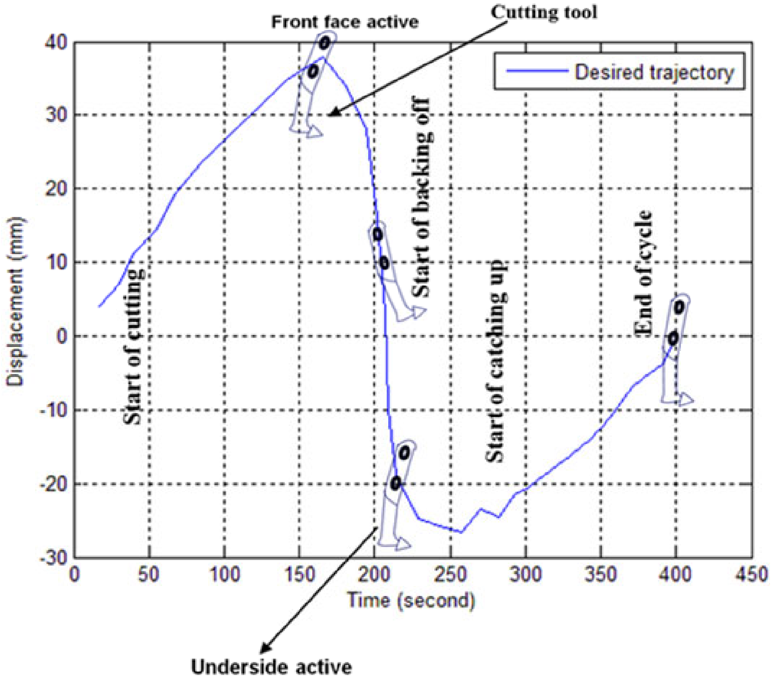
A three-row vibratory subsoiler is evaluated in the field by Shchukin et al. (2015). A comparative evaluation of the machine is performed with the non-vibratory tool. It is observed that the vibration of the tool reduced the lumpiness and rigidness of the tilled soil. Soil structure is also improved in the soil by the developed vibratory subsoiler.

Three experimental trials with three treatments such as rotary tillage, vibrating subsoiling, and non-vibrating subsoiling are conducted by Tang et al. (2015). The vibrating subsoiler showed decreased bulk density and penetration resistance by 9.58 % at 30–45 cm and 10.21 % at 15–30 cm as compared to the rotary and non-vibrating subsoiler. Thus from the study, it is concluded that the vibrating subsoiling system improved the soil properties effectively with decreased traction resistance.

An innovative method is proposed by Razzaghi and Sohrabi (2016) to analyse the displacement, velocity, and force on vibrating tillage tools based on the polar coordinates system. The design parameters are also analysed for the performance of the tillage tool.

Thus from the literature survey, it is observed that there is a need to design an active tillage machine which plays an essential role in agricultural operations. Energy consumption during the field operation is a vast area of concern to farmers and scientists, and all the field operations require several passes of farm machinery, which makes the soil compact and hard. Thus combination or active tillage equipment can solve this problem.

Figure 4 Tool trajectory in the soil (see online version for colours)



Source: Shahgoli et al. (2010a, 2010b)

5 Tool trajectory explanation

There are different phases in vibratory tillage according to Shahgoli et al. (2010a, 2010b), that the tool has to follow the path to complete the operation and the phases are explained is as follows and as shown in Figure 4:

1 Cutting: The tool tine tip penetrates in the uncut soil and tool is oriented at the rake angle of 36° normally.

2 Backing off: The cycle in which the tine front phase disengages from the uncut soil and goes backward to the loosened soil.

3 Catching up: This phase is for negative or zero oscillation angles and the tool moves towards the loosened soil and continues till it reaches the uncut soil part of the oscillation phase.

4 End of the cycle: This is the end of the phase and the tool completes the process. Thus the mechanism should follow the different stages during the operation in the soil.

Through different methods available for the coordinate extraction the precision points are extracted for the tillage tool trajectory available as shown in the Figure 4. P_{xd} and P_{yd} are the horizontal and vertical coordinates of the tillage tool trajectory extracted from the process. The values of the X and Y extracted for the trajectory are shown in the Table 1.

Table 1 Desired points for the tool trajectory

Desired precision points	P_{xd}	P_{yd}
1	16.741	4.089
2	30.134	7.101
3	40.179	11.214
4	54.688	14.292
5	68.080	19.363
6	84.821	23.452
7	104.911	27.560
8	126.116	31.673
9	141.741	34.756
10	165.179	38.881
11	180.804	33.964
12	194.196	28.036
13	202.009	16.077

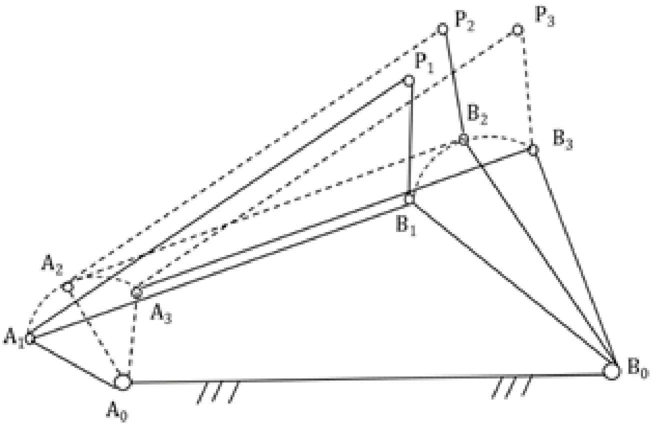
Source: Shahgoli et al. (2010a, 2010b)

These are the values of the coordinates for the trajectory explained for the vibratory tillage operation.

6 Graphical synthesis of linkages

Graphical method is a method adopted by the designers for mechanism design. There are different types of graphical synthesis procedures for mechanism design. One among them is three, four and five precision position synthesis procedure (Sandor and Erdman, 1984).

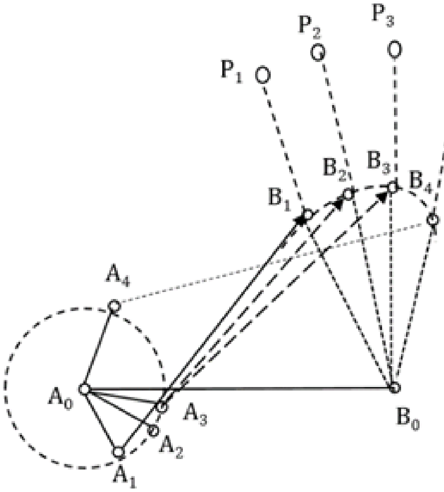
Figure 5 Completed mechanism design three precision position path generation problem



Source: Sandor and Erdman (1984)

It is a simple geometric procedure to design a four bar path generator mechanism for three precision positions (Rao et al., 2018). The selected precision points P_1, P_2 and P_3 the coupler point P on the coupler link should pass and that is called three precision point mechanism design as shown in Figure 5. Similarly in four precision position the steps are defined and the mechanism is designed as shown in Figure 6.

Figure 6 Four precision position design



Source: Sandor and Erdman (1984)

7 Five precision position analytical synthesis

The four-bar linkage can be synthesised using three and four precision points as a path generation problem. Thus in Burmester curve theory there are infinite pivoted points for any four arbitrarily prescribed positions and any two of these pair can form a four-bar mechanism whose coupler point will match these prescribed precision points. Thus for five precision points, there are no free choices according to the theory, and the number of real equations and real unknowns are equal, concluding that these equations can be solved.

The second concept came from further investigating the four precision position theory and centre and circle points. Suppose that the circle and centre-point curves are plotted for prescribed positions 1, 2, 3, and 4, a new set of curves are obtained for the first three precision positions and the fifth position is superimposed with the first set. The intersection of the curves yields to a common solution, and a Burmester pair is identified that will be able to guide the coupler that will pass through all the five prescribed positions.

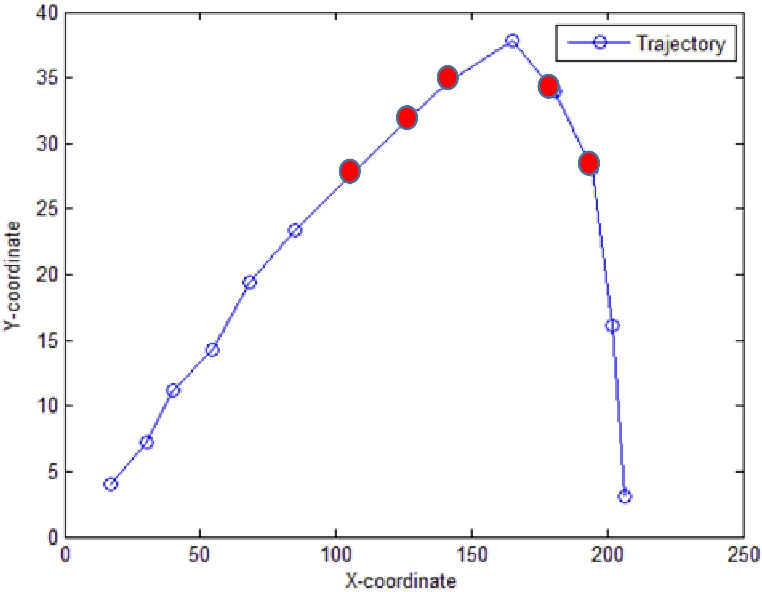
In this work, four different cases are considered, and the circle and centre-point curves are plotted. Four different combinations of four positions are considered to improve the accuracy of the synthesis. In the first case position, 1-2-3-4 is considered other cases include the position namely 1-2-3-5, 1-2-4-5, and 1-3-4-5. It is being noted that position 1 is common in all four cases. The centre-point and circle point curves of these four cases are generated using the Burmester curve theory. The common intersection point of centre-point curves for these four cases yields fixed pivots' locations of the four-bar mechanism for all five positions (1-2-3-4-5). These Burmester curves are generated using MATLAB software. In this section five precision points were selected from the tool trajectory as mentioned in above section. Five precision points are taken and shown in Table 2.

Table 2 Desired five precision points for tool trajectory

Desired points(mm)	*1*	*2*	*3*	*4*	*5*
Pxd	104.911	126.116	141.741	180.804	194.196
Pyd	27.560	31.673	34.756	33.964	28.036

Source: Shahgoli et al. (2010a, 2010b)

Figure 7 Actual trajectory and prescribed precision points (see online version for colours)

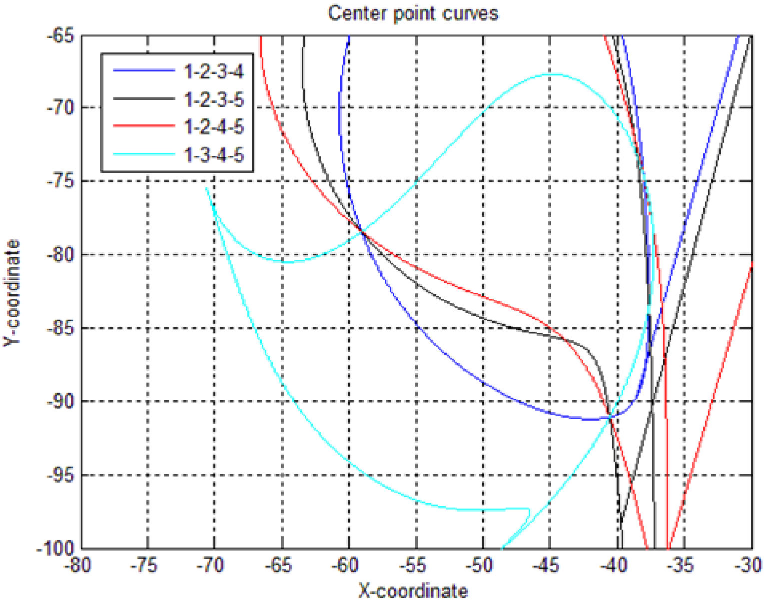


Coordinate transformation process is applied to the precision points given in Table 3 and the desired precision point 5 is made as (0, 0) and with respect to that other precision points are identified and given in Table 3.

Table 3 Desired five precision points for tool trajectory after coordinate transformation

Desired points (mm)	*1*	*2*	*3*	*4*	*5*
Pxd	0	–13.4	–53.5	–68	–89.3
Pyd	0	5.8	6.7	3.6	–0.5

Figure 8 Centre point curves for position 1-2-3-4, 1-2-3-5, 1-2-4-5, 1-3-4-5 (see online version for colours)



The parameters for Burmester curve generation are calculated and shown in Table 4.

Table 4 Parameters for Burmester curve generation

1	*2*	*3*	*4*
P_{21}	14.6	Delta2	156.589^0
P_{31}	53.92	Delta3	172.86^0
P_{41}	68.10	Delta4	176.97^0
P_{51}	89.30	Delta5	180.34^o

Position vector for the given prescribed coordinates for position 1, 2, 3, 4 and 5 can be calculated from the following formula.

$$P_{\delta j} = R_1 - R_j$$

where

R_1 vector with reference to the 1st precision point

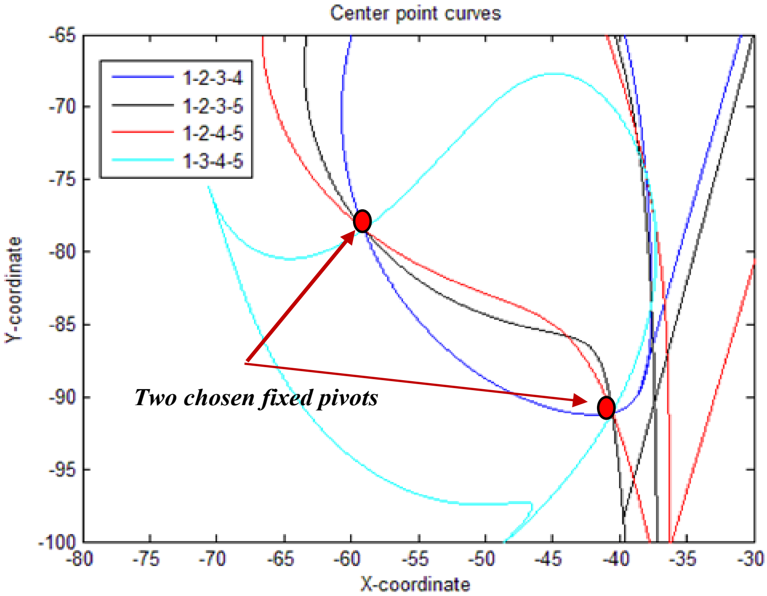
R_j vector with reference to the *j*th precision point

$P_{\delta j}$ position vector with respect to final and initial precision points.

Delta angle is calculated with respect to the initial vector R_1 (0, 0) and displaced position.

The chosen pair for centre points for the four bar mechanism is indicated and shown in Figure 9.

Figure 9 Common intersection points of centre point curves (see online version for colours)



The coordinates of the chosen fixed pivots are shown in Table 5.

Table 5 Coordinates of centre points

S. no.	*X*	*Y*
1	–40.56128	–91.0973
2	–59.0478	–78.473

The value of β_2 taken in the Burmester curve generation is varying from 0 to 3,600 with a step size of 0.001. Therefore a total of 360,001 iterations of centre points are present in the Burmester curve. It can be observed that the common intersection point is approximately at the same iterations for the four cases separately and is tabulated in Table 6.

Table 6 Iteration comparison for the four cases of positions

Positions	*Iterations (X)*	*Iterations (Y)*
1-2-3-4	47828	236169
1-2-3-5	47828	236169
1-2-4-5	47829	236159
1-3-4-5	40293	108289

Figure 10 Circle point curves for position 1-2-3-4, 1-2-3-5, 1-2-4-5, 1-3-4-5 (see online version for colours)

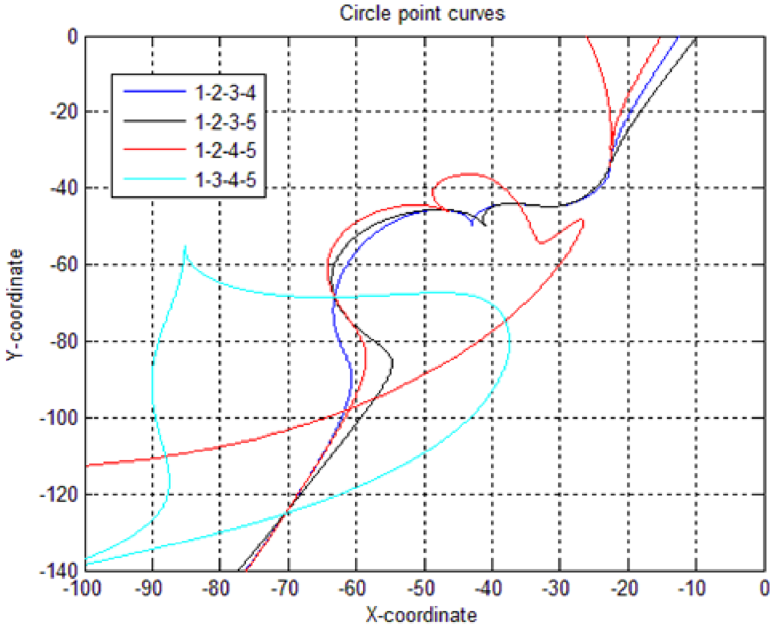
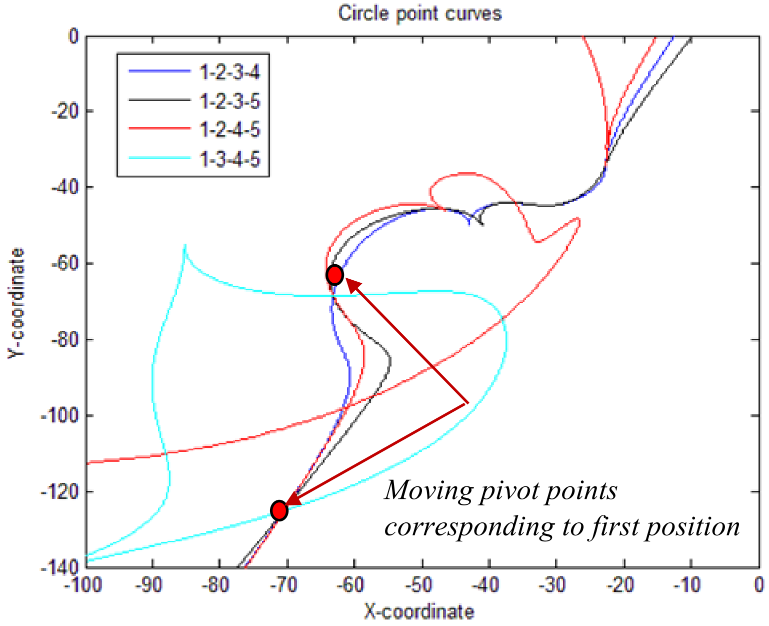
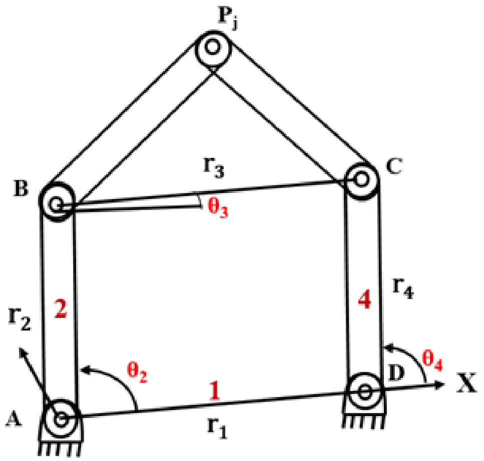


Figure 11 Common intersection points of circle point curves (see online version for colours)



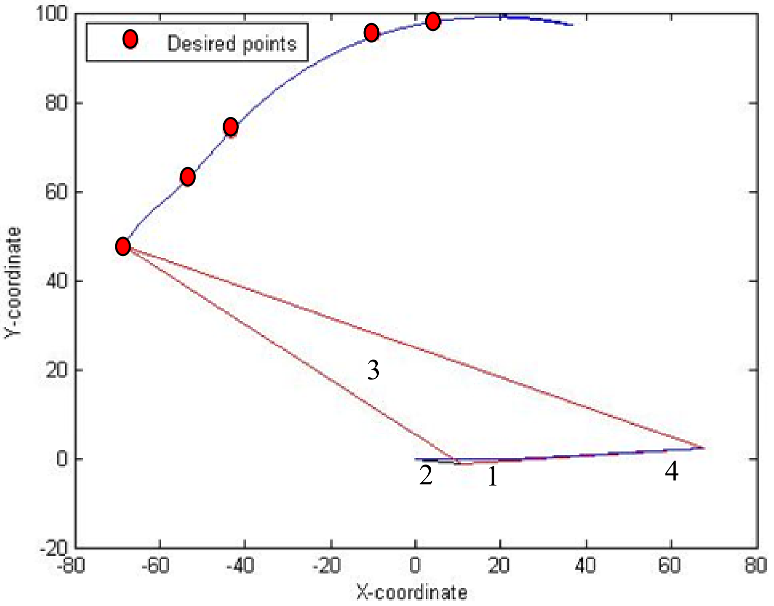
It can be observed that there is a deviation in iterations for fourth data, but the error is very less when the coordinates of fixed pivots are calculated. Similarly, the plots of moving pivots or circle point curves are also generated using the same above principle as shown in Figures 10 and 11 respectively.

Figure 12 Four bar mechanism with dimensions for the prescribed five precision point trajectory (see online version for colours)



Notes: AD = r_1 = 220 mm; AB = r_2 = 101 mm; BC = r_3 = 561 mm; CD = r_4 = 450 mm.

Figure 13 Synthesised four bar linkage mechanism for five precision point (see online version for colours)



The common intersection point chosen is indicated in Figure 11. The circle point curves intersect each other corresponding to the same iteration as in case of centre points curves, i.e., 47828 and 236159. Thus by knowing the positions of fixed and moving pivots corresponding to the first position as shown in Table 5 we can easily draw and synthesise a four-bar mechanism which is shown in Figure 12.

Through analytical synthesis procedure mechanism, dimensions are obtained and simulated in MATLAB to observe whether the designed mechanism passes through the desired trajectory or not. It is observed that the synthesised four bar mechanism can easily track the desired five prescribed position points accurately as shown in Figure 13. Optimal synthesis is performed for more accuracy in path tracking.

$$R_1 = 104.911 + 27.560i;\ R_2 = 126.116 + 31.673i;\ R_3 = 141.741 + 34.756i;$$
$$R_4 = 180.804 + 33.964i; R_5 = 190.196 + 28.0361i.$$

$$P_{21} = -21.205 - 4.113i;\ P_{31} = -36.83 - 7.196i;$$
$$P_{41} = -75.893 - 6.404i;\ P_{51} = -85.285 - 0.4761i$$

8 Conclusions

The paper proposes a four bar mechanism with dimensions of crank as 101 mm, ground length of 220 mm, coupler of 561 mm and follower as 450 mm for vibratory tillage operation using five precision position method. The different phases for tool trajectory are adopted from the literature and synthesised for tracking the tillage tool trajectory. It is observed that the tillage tool tracks the path accurately through selected five precision position method as explained in the previous section. Thus the dimensions obtained through the process can be used for providing the oscillations to the cultivator in tillage operation. Further the work is to fabricate and develop a machine for vibratory tillage operation with mechanism designed through five precision point synthesis method and perform the field evaluation of the developed machine with different parameters. Preliminary trials observed that there is a reduction of draft and power with the developed vibratory tillage cultivator.

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