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The design of a velocity converter for seeding machines

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Abstract

A velocity converter was designed to change the rotational speed of input shaft to different desirable rotational speed at output shaft continuously. The converter consists of three similar four-bar linkages, which the follower link oscillates by complete rotation of the crank link. A one-way cam clutch bearing was installed on the output shaft for each mechanism to transfer the rotation at one dirction. The three mechanisms are located on unique input and output shafts with 120 degree difference in phases to guarantee continuous rotation at output shaft. The length of the follower link is changeable while the lengths of the other links are fixed to vary the rotation ratio of input and output shafts. The relevant lengths of the links were found to be 2.5 cm for crank, 8 cm for coupler link and 10 cm for ground link while the length of rocker is variable from 5 to 12 cm. The converter locates between the career wheel and the metering system to provide different rotations of the metering system and change the distance of the seeds in seeders.

Keywords: Velocity converter, seeding machines, mechanism, four-bar linkage.

Abbreviations:			
a	Crank	O_2	Pivot of a and d
B	Pivot of a and b	O_4	Pivot of c and d
b	Coupler	ϕ_1, ϕ_2 and ϕ_3	Angle between a and d
C	Pivot of b and c	ψ_1, ψ_2 and ψ_3	Angle between c and d
c	Rocker or follower	θ	Position angle of b
d	Frame or ground	μ	Angle between c and b
K_1, K_2 and K_3	Coefficients		

Introduction

The seeding machines are used to place the seeds at certain distance and depth in the seedbed. Correct seeding helps to raise the field efficiency and productivity. However, crop productivity in developing world faces several constraints. One of the major crop productivity constraints in the third world is the unavailability of crop nutrients in appropriate amount and form to crops (Ali et al., 2008). The correct seeding also helps to have better control on weed management. The use of Integrated weed management has been indentified a viable weed control method in smallholder farms as it can lead to sustainable food production among other advantages (. Precision seeders locate the seeds at the optimum depth and distance to provide a better growing area for each seed. A seeder should set a seed in an environment in which the seed will reliably germinate and emerge (Ozmerzi et al., 2002). Precision sowing has been a major thrust of agricultural engineering research for many years; however, most of the research and development works have dealt with seeders for agronomic crops (Karayel et al., 2004). A seeding machine usually needs a system to adjust, control and change the distance between the seeds. The sowing of crop using contour method of sowing conserves soil moisture but it affects traditional practices for carrying farm operations (Khambalkar et al., 20100. Different kinds of gearbox have already been made and used to achieve this goal which most of them are capable to change the distance between seeds in steps. In theory of machines and mechanisms we concentrate on the movements of components that are often to all intents and purposes rigid; we are concerned with their shapes and positions, and how to measure them; and we study paths traced in the plane and in three dimensions by components (Hunt, 1976). In a specific design process, some initial design data are assigned, and then the design parameters are modified to improve the performance of the system. A design may undergo several iterations before reaching the optimal design based on experience or closed-form formulations (Yang and Abdel-Malek, 2007). Currently, several mechanical, pneumatic, hydraulic and magnetic mechanisms are used to transmit power between two intersecting shafts (Yaghoubi et al., 2010). Four-bar linkages have been widely used for path, function and motion generations. Adjustable four-bar linkages can greatly enhance the ability to generate various kinds of output motions. They provide not only flexibility that is required in many industrial applications, but also high operational speed, high load-bearing and high precision capabilities (Zhou and Cheung, 2004). Theoretically, a four-bar linkage with the same dimensions can generate infinite coupler curves. Therefore, to achieve a relative precise path synthesis, a very large number of coupler curves need to be stored in the coupler curves database, or the so-called ‘‘electron atlas’’ (Hongying et al., 2007). Another solution could be programming in a relevant computer program to evaluate a wide range of the possibilities and save time. Nowadays, computers play an important role in all engineering disciplines, including design synthesis and analysis. Analytical techniques which were too cumbersome to carry out by hand are now within grasp. Synthesis and analysis algorithms in kinematics and dynamics have received attention from researchers in mechanism design (Lee et al., 1999). Using a mechanism instead of gearbox in seeding machines helps to change continuously the distance between seeds. The objective of this research is to design a velocity converter including number of mechanisms which converts the rotation of input shaft and transfer a continuous and modified rotation to the output shaft.

Materials and methods

The synthesis and design of the mechanism carried out by a computer program using *Matlab* software. The process was done in three steps: I) selection of the kind of mechanism, II) determination of the number of mechanisms and III) calculation of the length of the links to provide a desirable motion.

Mechanism

A machine designer is generally aware of the input drive system or has the freedom to specify the drive system (Kimbrel, 1984). Different kinds of mechanisms were evaluated and the four-bar linkage mechanism was selected, which is one of the most useful and most common mechanisms (Martin 2002). A four-bar crank rocker mechanism (*Fig. 1*) was taken into account in this study where the links of the mechanism are defined as crank (*a*), coupler (*b*), rocker or follower (*c*) and frame or ground (*d*). The crank *a* rotates completely about pivot O_2 and by means of coupler *b* causes oscillation of the follower *c* about pivot O_4. Hence the mechanism converts rotation into oscillation motion. In order to have an operation for the crank rocker mechanism, the following conditions must exist otherwise the mechanism will be locked after some rotation (Martin, 2002).

$$a + b + c > d \quad (1)$$
$$a + d + c > b \quad (2)$$
$$a + b - c < d \quad (3)$$
$$b - a + c > d \quad (4)$$

This mechanism can realize transformation from rotation of the crank to swing of the rocker. The length of the crank link was taken much shorter than that of the coupler and rocker links in order to realize the transformation (Zhang and Xu, 2004).

Selection of the number of mechanisms

The complete rotation of the crank link causes oscillation movement of the rocker link so the rocker link will have active and passive phases. The sinus curve was taken into account to evaluate the approximate movement at different positions of the system with one, two and more mechanism-

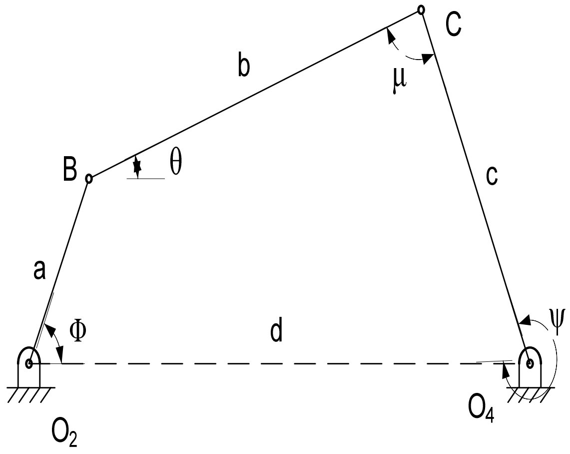

Fig 1. Schematic diagram of a four-bar linkage mechanism

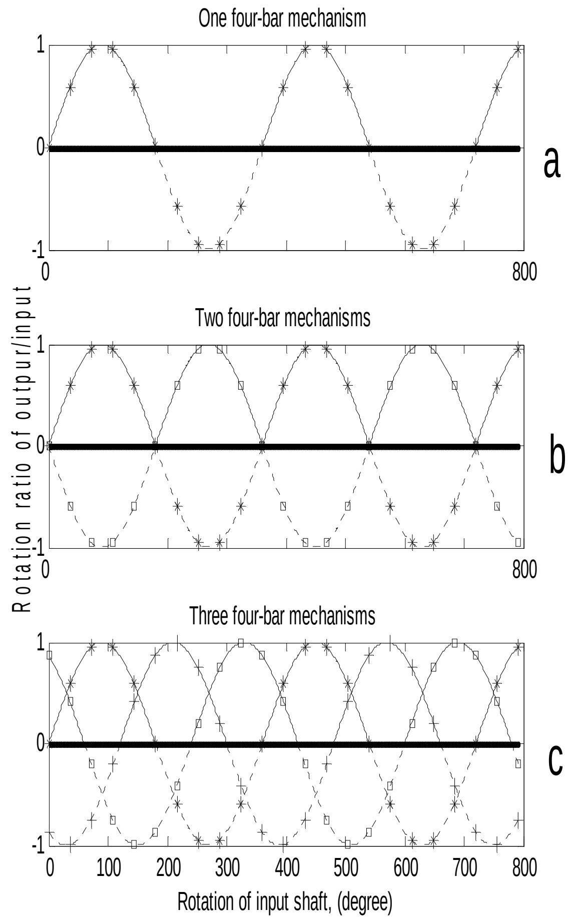

Fig 2. The approximate active and passive parts of the output shaft of the system with one (a), two (b) and three (c) four-bar linkage mechanism (s)

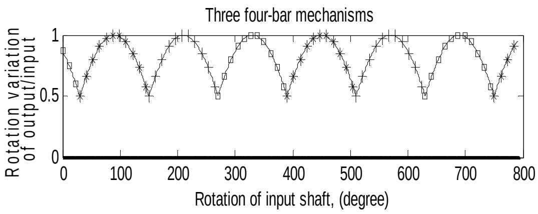

Fig 3. The approximate active part of the output shaft of the system with three four-bar linkage mechanisms

(s). Using one mechanism only gives rotation of output at about 50% of running time with a variation in rotation of output shaft (*Fig. 2a*). A system with two similar mechanisms with 180 degrees difference in phases between the mechanisms causes a rotation in output shaft almost all the time but the output will have again a big variation of rotational speed and a short time rest between the active phases of the mechanisms (*Fig. 2b*). Preparing a system with three mechanisms and 120 degrees difference in phases offers a non-stop rotation and less variation at output shaft (*Fig. 2c*). Adding more mechanism to the system causes less variation of rotation in output but the system will be bigger, heavier, more costly and more complicated. A system with three mechanisms will offer the required motion at output shaft. The approximate active part of the output shaft of the system with three four-bar linkage mechanisms with 120 degrees difference in phase is shown in *Fig. 3*.

Dynamic design

The *Freudenstein's method* was used to design the four-bar crank rocker mechanism in order to generate the desire function of the mechanism (Grosjean 1991). The relation between the lengths and the angles of the links in the mechanism can be developed as follow. In *Fig. 1* the links are considered as vectors and the sum of the *X* and *Y* components must be zero.

$$a\cos\phi + b\cos\theta = c\cos(\psi - 180) + d \quad (5)$$

$$a\sin\phi + b\sin\theta = c\sin(\psi - 180) \quad (6)$$

Rearranging and squaring both sides of *Eq.5* and *Eq.6* yields:

$$b^2\cos^2\theta = d - (c\cos\psi + a\cos\phi)^2 \quad (7)$$

$$b^2\sin^2\theta = (c\sin\psi + a\sin\phi)^2 \quad (8)$$

Expanding of *Eq.7* and *Eq.8* and then adding the right sides of the equations gives:

$$b^2 = a^2 + c^2 + d^2 + 2a\cos(\phi - \psi) - 2ab\cos\phi - 2cd\cos\psi \quad (9)$$

The *Eq.9* can be written as follow:

$$b^2 - (a^2 + c^2 + d^2) + d/c\cos\phi + d/a\cos\psi = \cos(\phi - \psi) \quad (10)$$

or

$$K_1\cos\phi + K_2\cos\psi - K_3 = \cos(\phi - \psi) \quad (11)$$

The *Eq.11* is called *Freudenstein's equation*, which relates to the crank angles and the lengths of the links for a four-bar linkage. The coefficients K_1, K_2 and K_3 in *Eq.11* are:

$$K_1 = \frac{d}{c} \quad (12)$$

$$K_2 = \frac{d}{a} \quad (13)$$

$$K_3 = \frac{(a^2 - b^2 + c^2 + d^2)}{2ac} \quad (14)$$

The coefficients K_1, K_2 and K_3 are computable by solving the equations 15, 16 and 17 when the positions of crank and rocker are defined by angles ϕ_1, ϕ_2, ϕ_3, ψ_1, ψ_2 and ψ_3 at three different locations. The angles introduce the rotation ratio between the input and output shafts.

$$K_1\cos\phi_1 + K_1\cos\psi_1 - K_3 = \cos(\phi_1 - \psi_1) \quad (15)$$

$$K_1\cos\phi_2 + K_2\cos\psi_2 - K_3 = \cos(\phi_2 - \psi_2) \quad (16)$$

$$K_1\cos\phi_3 + K_2\cos\psi_3 - K_3 = \cos(\phi_3 - \psi_3) \quad (17)$$

Assuming a value for one of the links *a*, *c* or *d*, the length of the other links can be calculated, using *Eq.12*, *Eq.13* and *Eq.14*. In this research the length of the crank was taken with certain length to have an optimum size and rotation for the converter.

Prototype converter

The velocity converter was constructed with three mechanisms. The size of the links was chosen based on the output of the program. A one-side clutch (*KK25*) was installed at the pivot O_4 for each mechanism to transfer the motion of the rocker to the output shaft only at one direction. When the rocker link moves to other direction no rotation will occur in the output shaft because of sliding the clutch. After construction the converter was tested for different lengths of the rocker link. For each length of the rocker, the input shaft was rotated 100 times and the rotation of the output shaft was measured. Each test replicated three times. Finally the results of the tests were compared to the results of the output of the program.

Results and discussion

A system with three four-bar crank rocker mechanisms with 120 degree differences in phase between the mechanisms was used to guarantee continuous motion of the output shaft (*Fig. 4*). The length of the rocker link is changeable while the lengths of the other links are fixed. At a constant rotation of input shaft, the rotation of output shaft is related to the length of the coupler link. Longer length for rocker *c* gives less rotation at output shaft. The length of crank was taken with a value of 2.5 cm regarding to the optimum size of the converter and conversion range of rotations. The computer program was run for a wide range of lengths for coupler, rocker and ground links. A range of 7 to 15 cm was applied for the length of coupler link. The output of the program

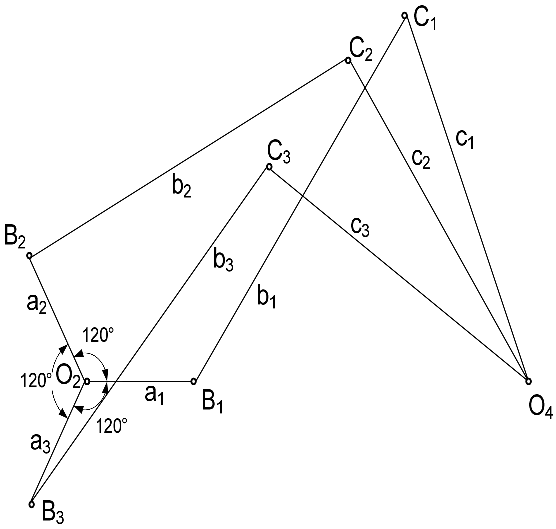

Fig 4. The situation of the links of three four-bar linkage mechanisms with 120 degree difference in phase.

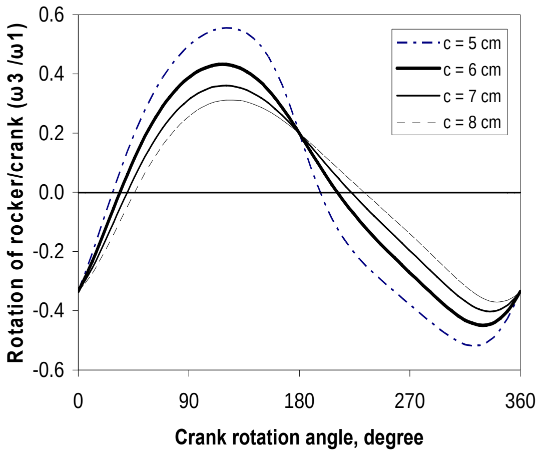

Fig 5. Rotation of rocker per crank at a complete rotation of crank about O_2 for four length rocker link (5, 6, 7 and 8 cm).

showed that the results are very close for the values of 8 to 11 cm for coupler link. The values above 11 cm cause a bigger size for converter and limitation in conversion range. The value of 8 cm was determined for the coupler link among a range of 8 to 11 cm to guarantee no lock for the mechanisms and having a small size for the converter. The length of ground link was fixed at 10 cm among the range of 3 to 11 cm to adapt the complete system, avoid conflict of the links and give the desirable conversions. Finally the rocker which is the variable link could have the lengths of 5 to 12 cm based on the defined conditions in the computer program. The computer program was run for all combinations of the selected lengths of each link. The results showed that in a complete revolution of the input shaft at a constant rotational speed, different rotational speeds could be achieved at output shafts rotates, even in the same direction of the input shaft or at opposite direction, so the velocity ratio vary in certain range regarding to the length of the links. As an example, the output of the program is shown in table 1 for the running with

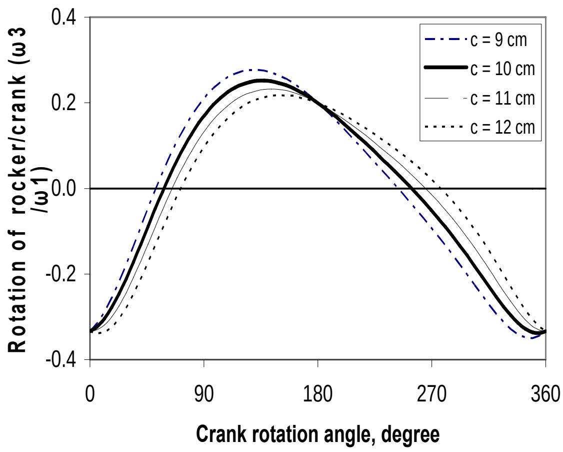

Fig 6. Rotation of rocker per crank at a complete rotation of crank about O_2 for four length rocker link (9, 10, 11 and 12 cm).

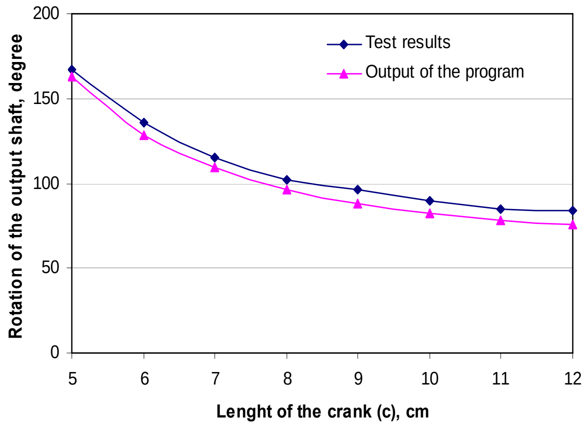

Fig 7. Rotation of output shaft for a complete rotation of the input shaft at different length of the crank.

given values of the links. The rotation ratios of the rocker per crank links are shown in *Fig. 5* and *Fig. 6* for different length (5 to 12 cm) of the rocker link. Figure 7 shows the comparison between the output of the program and the results of the tests. The figure shows almost a constant and systematic difference in all length of the crank link which could be related to the experimental errors or sliding of the clutches.

Conclusion

A system with three similar four-bar crank rocker mechanisms is applicable to change the rotation ratio in a velocity converter. Conversion at a desirable range is possible just by changing the length of the rocker link. The present converter gives rotation ratio at a range of 46.51 to 23.36% by changing the length of the rocker from 5 to 12 cm. The converter can be used in seeding machines to control the inter-row distance of the seeds in each line.

Table 1. The velocity ratio of the output shaft (rocker) and the position angle of links for a complete rotation of the input shaft (crank).

Rotation of crank (Φ), degree	Position angle of rocker (ψ), degree	Velocity Ratio (VR)	Rotational speed of rocker (ω_3), rev/min	Angle between coupler and rocker (μ), degree
0	297.92	-0.334	-3.34	55
10	294.83	-0.282	-2.82	56
20	292.35	-0.212	-2.12	57
30	290.62	-0.132	-1.32	59
40	289.73	-0.048	-0.48	62
50	289.66	0.032	0.32	65
60	290.35	0.104	1.04	68
70	291.71	0.166	1.66	72
80	293.64	0.217	2.17	76
90	296.02	0.256	2.56	80
100	298.74	0.285	2.85	84
110	301.70	0.303	3.03	87
120	304.78	0.312	3.12	91
130	307.91	0.311	3.11	94
140	310.98	0.302	3.02	97
150	313.93	0.285	2.85	99
160	316.68	0.262	2.62	101
170	319.16	0.233	2.33	102
180	321.33	0.2	2.00	102
190	323.16	0.164	1.64	102
200	324.61	0.126	1.26	101
210	325.68	0.087	0.87	99
220	326.37	0.049	0.49	97
230	326.67	0.011	0.11	94
240	326.60	-0.026	-0.26	91
250	326.16	-0.064	-0.64	87
260	325.34	-0.101	-1.01	84
270	324.14	-0.139	-1.39	80
280	322.56	-0.178	-1.78	76
290	320.58	-0.218	-2.18	72
300	318.20	-0.259	-2.59	68
310	315.42	-0.297	-2.97	65
320	312.28	-0.332	-3.32	62
330	308.82	-0.358	-3.58	59
340	305.17	-0.371	-3.71	57
350	301.48	-0.364	-3.64	56
360	297.98	-0.335	-3.35	55

The minus sign (-) means opposite rotation of the rocker

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References

Ali S, Riaz Khan A, Mairaj G, Arif M, Fida M, Bibi S (2008) Assessment of different crop nutrient management practices for yield improvement Australian Journal of Crop Science 2(3):150-157

Grosjean J (1991) Kinematics and Dynamics of Mechanisms. McGraw Hill

Hongying Y, Dewei T, Zhixing W (2007) Study on a new computer path synthesis method of a four-bar linkage. Mechanism and Machine Theory 42 (2007) 383–392

Hunt KH (1976) Geometry-The key to mechanical movements. Mechenism and Machine Theory. 11, 79-89

Karayel D, Barut ZB, Ozmerzi A (2004) Mathematical Modelling of Vacuum Pressure on a Precision Seeder. Biosystems Engineering 87(4): 437-444

Khambalkar vp, Nage SM, Rathod CM, Rahulkar PS, Dahatonde S (2010) Mechanical sowing of safflower on broad bed furrow. Australian Journal of Agricultural Engineering 1(5):184-187

Kimbrel JT (1984) Graphical synthesis of a 4-bar mechanism. Mechanism and Machine Theory. 19(1). 45-49

Lee MY, Erdman AG, Faik S (1999) A generalized performance sensitivity synthesis methodology for four-bar mechanisms. Mechanism and Machine Theory 34 (1999) 1127 -1139

Martin G (2002) Kinematics and Dynamics of Machines McGraw-Hill

Olorunmaiye PM (2010) Growth and yield in maize/cassava intercrop as affected by interactions of weed control methods. Australian Journal of Agricultural Engineering 1(3):106-111

Ozmerzi A, Karayel D, Topakci M (2002) Effect of Sowing Depth on Precision Seeder Uniformity. Biosystems Engineering 82(2): 227-230

Yaghoubi M, Jafary A, Mohtasebi SS (2010) Design, simulation and evaluation of a new universal joint with intersecting angle up to 100 degrees for farm machineries. Australian Journal of Agricultural Engineering 1(4):149-152

Yang J, Abdel-Malek K (2007) Design propagation in kinematics of mechanical systems. Mechanism and Machine Theory 42 807–824

Zhang JF, Xu QY (2004) Research on stability of periodic elastic motion of a flexible four-bar crank rocker mechanism. Journal of Sound and Vibration 274(1-2): 39-51

Zhou H, Cheung EHM (2004) Adjustable four-bar linkages for multi-phase motion generation. Mechanism and Machine Theory 39 (2004) 261–279