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Design of an Agricultural Cutter Using the Theory of a Four-Bar Mechanism

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Abstract: Cutting green or dry crops requires the use of cutter bars or cutter bars, minimizing fruit loss as much as possible and also adapts with the geometry, in particular the terrain of the potatoes during the weeding phase. Currently, two types of cutter bars are fitted to mowers, combine harvesters, hedge trimmers, etc. Among these two types, single blade cutter bars are the oldest and most common while the more recent double-blade cutter bars remain the most efficient despite all the technological complications which more or less affect their commissioning. Thus during the operation of these machines a very agitated kinetodynamic behavior (presence of fluctuation, shock and vibration), was found compromising the conduct of the cutting operation, the premature wear of the moving parts, the efficiency of the machine and even its survival. The choice of the theme of this work was dictated by general considerations on the project to improve the cutting operation, essentially aimed at reducing the risk of jamming, calming the noisy operation of the unit, eliminating the vibrations which the system is the seat and thus improve the performance of the machine through the design and introduction of a new architecture of the mechanism controlling the movable cutting members.

Keywords: Harvesting Machines, Cutter Bar, Control Mechanism, Kinetodynamics

1. Introduction

Obtaining green fodder by mowing or dry harvesting is one of the most important and vital operations on most farms [1, 2]. It turns out that the good execution and the continuation in cutting and work in mowing as in harvesting pose a certain number of challenges [3-5], it must know the mechanical characteristics of the stems of the plant to be cutting [6], like as removing weeds from the soil of fruit trees and mechanically weeding potatoes before harvest, wheat and barley harvest [7]. Among those facing these challenges are mainly agricultural machine builders, mechanical engineers and all users among farmers. In practice, not only are these challenges linked to conditions such as the type of land and the product cut, also climatic variables, the intrinsic nature of crops and the different modes of operation of users, but they essentially depend on the structural characteristics and recommended by the mechanisms controlling the cutting members within the machines themselves [8-10].

Of the same type or by using mechanisms with articulated bars or levers and swings (plane mechanisms mounted in tandem called Robert and Evans) illustrated in Figure 3. However, in most specialized agricultural machines reserved for cutting, the mobile elements are controlled using regular mechanisms ($e=0$) of the crank type, by eccentric mechanisms ($e\neq0$) of the same; or using articulated bar mechanisms or levers and swings (plane mechanisms mounted in tandem, said by Robert and Evans). And it is by virtue of the dynamic use of these mechanisms used to control the cutters that very agitated kinetodynamic behavior has been observed, studied and quantified [11-15]. This very agitated and fluctuating behavior compromises the conduct of the cutting operation and therefore the yield of the harvest, multiplies the rate of premature wear of the moving parts and thus increases the risks of jamming following the incessant growth of the game. Functional existing at the interface of moving parts. For several decades and thanks to major design attempts [12]; production and tests carried out on prototypes of this agricultural equipment; we are working to introduce new models with a view to: [11].

Totally or partially cancel the contact forces in order to reduce wear and tear on the moving parts.

Reduce totally or - in most practical cases - partially the harmful growth of functional play in order to minimize the risks of jamming, snagging and stuffing.

Destroy the vibrations of which the system is the seat in order to manage to calm the noisy operation of the machine during the cutting operation.

For these reasons, the choice of a new architecture among the mechanisms controlling the cutting members seems to require an innovation study as precise and direct as possible; study that is the subject of this work.

2. Materials and Methods

2.1. Position of the Problem

In the cutter bars with a single blade, the cutting of the fodder is obtained by the covering several times per second of a fixed branch (counter blade fixed on the fingers) by a movable branch (blade properly itself) [16]. One of these two branches, acting like a shear, is animated by alternating movement created; in most of the cases; by an irregular mechanism $e \neq 0$ rod-crank type (Figure 1). In contrast to this and on the cutter bars with two blades, the slicing is obtained by the overlapping of the sections of two blades animated by opposite reciprocating movements coming from a double crank-rod system.

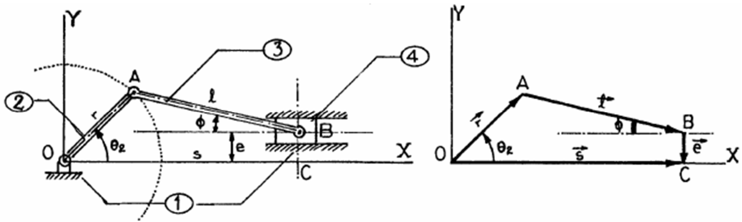

Figure 1. *Single blade and double blade cutter bar controlled by mechanisms rod-crank type plans (1: Fixed link, 2: Crank 3: Connecting rod, 4: Slider or Piston).*

by a vector writing, the equation (1) summarizes the parameters of the figure 1 or e≠ 0: (if e=0 it disappears from the equation)

$$\vec{r} + \vec{l} + \vec{e} = \vec{s} \tag{1}$$

by projection we get the equation (2)

$$\begin{cases} rcos\theta 2 + lcos\emptyset + 0 = s \\ rsin\theta 2 - l\ sin\emptyset - e = 0 \end{cases} \tag{2}$$

From expression (2), it follows that we can deduce the relation which expresses the variation of the angle (Φ) as a function of the generalized coordinate (θ_2)., Or then equation (3):

$$\emptyset = \arcsin\left(\frac{rsin\theta 2 - e}{l}\right) \tag{3}$$

Although these control systems are more or less based on the structural skeleton of the irregular crank mechanism; a kinetodynamic analysis summarized in Table 1 shows the following facts:

Table 1. *Kinetodynamics of single and double-blade cutter bars.*

Single blade cutter bar	**Double blade cutter bar**
Kinematic law:(4)	Kinematic law: (5)
$\vec{V}(\theta) = -rw\left[sin\theta + \left(\frac{l}{r}\right)K_1 sin\emptyset\right]\vec{\imath} + 0\vec{\jmath} + 0\vec{k}$	
$\vec{A}(\theta) = -rw^2[cos\theta + (\frac{l}{r})K_3 sin\emptyset + (\frac{l}{r})K_2^2 cos\emptyset]\vec{\imath} + 0\vec{\jmath} + 0\vec{k}$	
Avec:$\emptyset = \arcsin[\frac{r\ sin\theta - e}{l}]$	$\vec{V}(\theta) = -rw\left[sin\theta + \left(\frac{r}{2l}\right)sin2\theta\right]\vec{\imath} + 0\vec{\jmath} + 0\vec{k}$
$K_1 = -(r\ sin\theta + lK_2 sin\emptyset)$	$\vec{A}(\theta) = -rw^2[cos\theta + \left(\frac{r}{l}\right)cos2\theta]\vec{\imath} + 0\vec{\jmath} + 0\vec{k}$
$K_2 = \frac{r}{l}(\frac{cos\theta}{cos\emptyset})$	
$K_3 = -(r\ cos\theta + lK_4 sin\emptyset + lK_2^2 cos\emptyset)$	
$K_4 = -\frac{r}{l}\left(\frac{sin\theta}{cos\emptyset}\right) + K_2^2 tg\emptyset$	

With the following indication of the geometric and kinematic parameters that these sets recommend:

$V_{(\theta)}$, $A_{(\theta)}$=instantaneous linear speed and acceleration of the cutting blade,

θ=angular position of the connecting rod: r=radius of the crank, l=length of the connecting rod, ω=angular speed of the connecting rod

(i). Starting from the single cutter bar until the introduction of the double-cutter bar, some advantages are clearly apparent in favor of the use of the latter but nothing has been accomplished in view of the suppression of shocks and vibrations due to the opposite movements of the two blades.

(ii). It should also be mentioned that for these systems, the effects of the rotating masses (mt) can be offset by counterweights mounted on the crank; on the other hand the effects of the sliding masses (mc) are not.

(iii). Again in this respect, even the polar and Cartesian representations of the vibration forces $F_v(\theta)$ for each of the aforementioned systems (Figure 2), illustrate, however, that the desired balancing of the moving masses (sliding masses) remains quite incomplete.

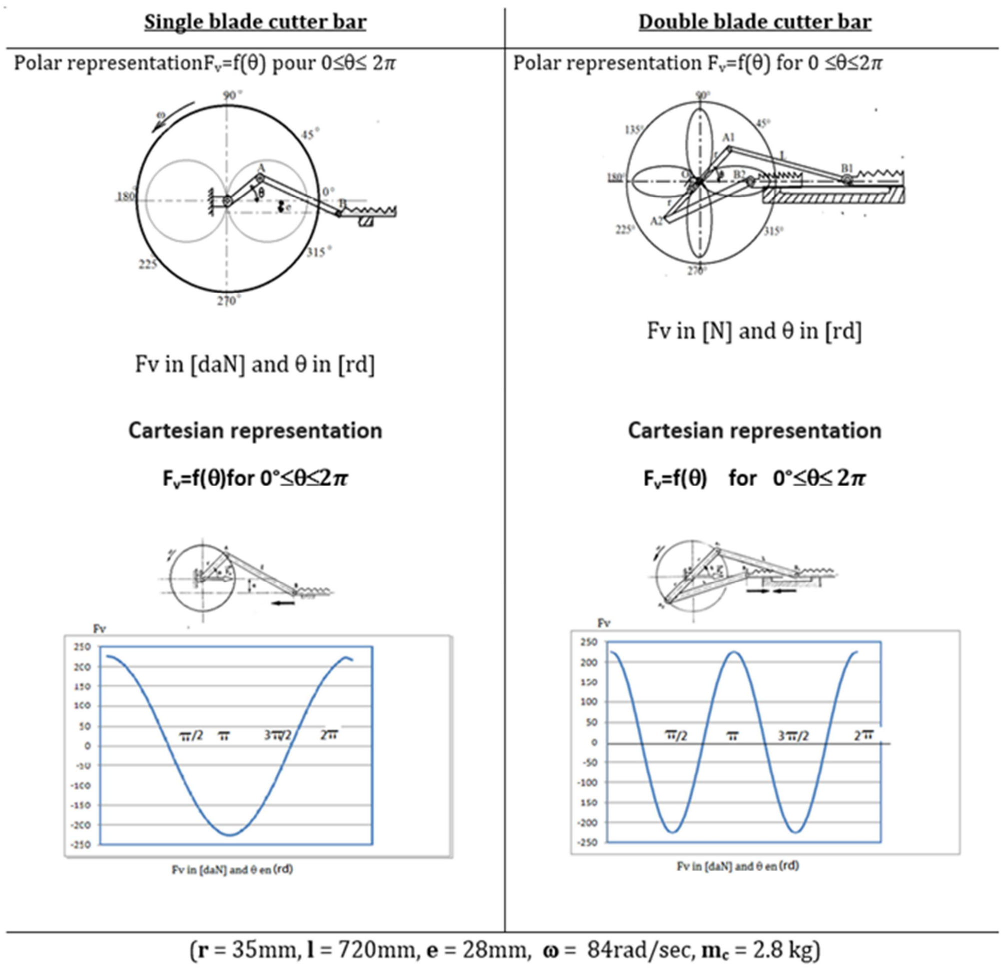

Figure 2. *Representations of the vibration force (Fv) for single and double blade cutter bars.*

2.2. Methodology for Improving Balancing

Among the few works currently existing, one could note the studies in which, by applying the synthesis of mechanisms according to the new concept based on the notion of centers instantaneous rotation (CIR) relative to both fixed and mobile links, the centroids recognized as geometric foci of (CIR) and the kinematic properties of the latter, a completely new architecture of the crank-crank mechanism can be envisaged by view of improved balancing [17-19].

Search for centroids of the crank-rod mechanism.

2.3. Graphic Determination of the CIR

We know from the course of theoretical mechanics that the planar movement of a solid body (a link in the mechanism) can be assimilated at every moment, to a rotation around a point called instantaneous center of rotation or rotopole. For the connecting rod-crank mechanism, the (CIR) noted here by (P) of the connecting rod supposed to be moving relative to the chassis, will be located graphically according to Burmester rules, at the meeting of the two perpendicular to the velocity vectors that recommend the ends A and B of the connecting rod (Figure 3).

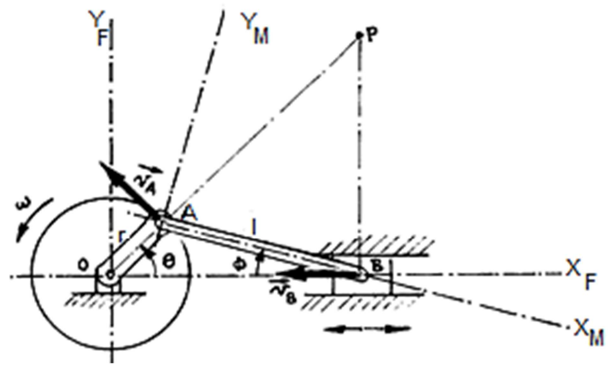

Figure 3. *Graphical determination of the CIR for a crank-rod mechanism.*

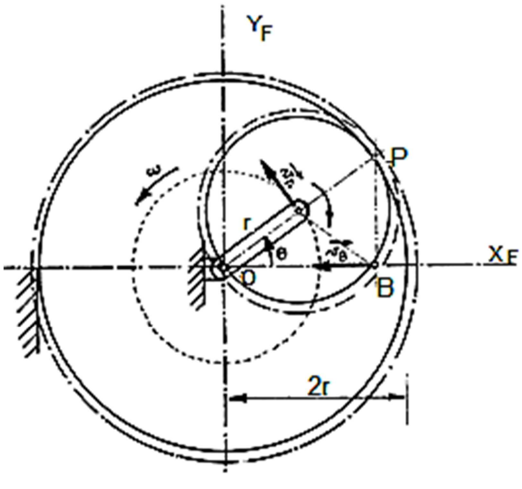

Figure 4. Arrangement of centroids of the isosceles mechanism (r=l) with connecting rod-crank.

2.4. Analytical Determination of CIR

From theoretical mechanics, we also know that the place or the geometric focus of the (CIR), following the continuous rotation animation of the crank is indeed a curve called Centroid. This centroid can be fixed and it is called base, as it can be mobile and it is called rolling; depending on whether we invert the role of fixed and mobile links in the system. If we take the whole of Figure 3, the parametric equations of the centroids: fixed (base) and mobile (rolling) defined respectively in the fixed references (X_fOY_f) and (X_mAY_m) are easily described by the following relationships (equation 4, 5):

$$\begin{cases} X_F = rcos\theta + lcos\emptyset \\ Y_F = rsin\theta + ltg\theta cos\emptyset \end{cases} \tag{4}$$

and

$$\begin{cases} X_M = l - rsin\theta sin\emptyset - ltg\theta cos\emptyset sin\emptyset \\ Y_M = rsin\theta cos\emptyset + ltg\theta cos^2\emptyset \end{cases} \tag{5}$$

with: $cos\emptyset = \sqrt{1 - (\frac{r}{l} sin\theta)^2}$

In the particular case where the plane crank-rod mechanism is of the isosceles type for which (r=I) the parametric equations of the fixed (base) and mobile (rolling) centroids are simply reduced to the following expressions equation (6, 7)

$$\begin{cases} X_F = 2rcos\theta \\ Y_F = 2rsin\theta Y_M \end{cases} \tag{6}$$

$$\begin{cases} X_M = rcos2\theta \\ Y_M = rsin2\theta \end{cases} \tag{7}$$

Thus the fixed centroid (base) taken in the fixed reference (X_fOY_f) is the big circle described by the Cartesian equation: $x^2f + y^2f = 4r^2$ and of radius equal to 2r; while the moving centroid (rolling) caught in the moving frame (X_mAY_m) is the small circle described by the Cartesian equation:

$x^2M + y^2M = r^2$ and of radius equal to r, (Figure 4).

3. Result and Discussion

At the end of the application of this concept based on the concept of (CIR) and fixed and mobile centroids, the transmission of movement between the crank (OA) and the drive point (B) can be reproduced in full by two cogwheels: one, representing the fixed centroid, with internal toothing and large radius equal to 2r, in engagement with another mobile (representing the mobile centroid), with external toothing and radius equal to r. The structural diagram of this new mechanism is presented in Figure 5. Although the crank-link mechanism has been completely replaced by its conjug 'conjugate' 'known as the central mechanism; it has been confirmed from mathematical and simulation analyzes [20, 21] that the laws of movement remain unchanged.

The advantages Of the New Design, If we want to ensure the balance of rotating masses (mt) and sliding masses (mc); this is achieved by juxtaposing two mechanisms with elements of equal mass and arranged symmetrically; whereby there is self-balancing of the whole [22].

It is easy to point out that when this condition of symmetry is ensured, the barycenter of all the sliding (mc) and rotating (mt) masses will be brought to point (O); that is to say on the axis of rotation of the carrier link of the two toothed wheels whatever the angular positions (θ) or (θ+π) of these. Indeed and following the rotation around the axis (O) of the gear wheels which are engaged with the large fixed gear wheel; the rotating masses (m_t) automatically develop the following vibration force:

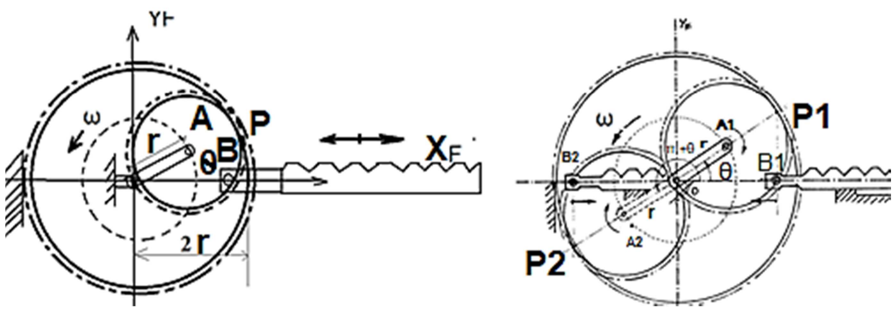

Figure 5. Central mechanism and arrangement for self-balancing.

$$\vec{F}_{V/m_t} = -m_t r\omega^2[(cos\theta + \cos(\theta+\pi))\vec{\imath} + (sin\theta + \sin(\theta+\pi))\vec{\jmath}] + 0\vec{k} \quad (8)$$

While the sliding masses (mc) in turn develop the following vibration force:

$$\vec{F}_{V/m_c} = -2m_c r\omega^2(cos\theta + \cos(\theta+\pi))\vec{\imath} + 0\vec{\jmath} + 0\vec{k} \quad (9)$$

The balance of these forces is indeed zero and consequently by eliminating all the vibration (Fv(mt, mc)=0); we thus succeed in completely balancing the fluctuations due to the moving masses and in calming the operation of the whole.

Finally, the exclusion of dynamic contact forces at the interface of the sliding masses and their guides is made possible by the fact that the points (B1) and (B2) are "real" training points. The simplified kinematics (speed and unidirectional acceleration) that these points recommend is self-supporting since:

$$\vec{V}_{B_1}(\theta) = -\vec{V}_{B_2}(\theta+\pi) = -2\, r\omega sin\theta\vec{\imath} + 0\vec{\jmath} + 0\vec{k} \quad (10)$$

$$\vec{A}_{B_1}(\theta) = -\vec{A}_{B_2}(\theta+\pi) = -2\, r\omega^2 cos\theta\vec{\imath} + 0\vec{\jmath} + 0\vec{k} \quad (11)$$

Thus the cancellation of the dynamic forces of contact between the rubbing parts reduces the risks of wear by friction, and consequently, slows the growth of the functional play and thus avoids the appearance of the phenomena of jamming, catching and stuffing [23].

4. Conclusions

The importance of production and increasing the yield of agricultural materials, especially crops with the invention of the cutting system with the reduction of vibrations as much as possible when cutting crops. The design and production of the prototype of a guide mechanism to be used in particular on harvesting machines with a double blade cutter bar, shows that the operation is made quieter and less noisy (total absence of shock, vibration and fluctuation) because the mechanism - being more compact - consists of a fixed crown of radius equal to 2r engaged with two identical toothed wheels of the same radius equal to r. Analysis of the simplified mathematical model of this mechanism clearly shows that the solution, based on a centroid architecture, also remains promising for other technological and industrial applications affecting other fields of science and technology.

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