Dynamic analysis of pitman and wobble shaft drive mechanisms; consideration of the harvesting losses in the combine platform

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The main purpose of using a combine harvester is to obtain grain with minimum damage and losses. Furthermore, the grain must sufficiently clean the grain, without straw and stubble. An inseparable part of the combine harvester is the cut platform and its function is to cut and gather products and send them to the thresher unit. Conventional cutting platforms which are commonly used for harvesting products have a knife section and this section has several types of drive mechanism for movment, including the pitman drive mechanism, twine wheel drive and wobble shaft drive. The comparison of dynamic velocity and acceleration between pitman and wobble shaft drive mechanisms was investigated. Results showed the use of wobble shaft drive as a new combine harvester such as JD1165, JD1055, JD1450 cws, Class115CS and Class116CS could reduce vibration of the cutter bar and combine harvester. Harvesting losses are also reduced.

Key word: combine harvester, platform; losses, wobble shaft; velocity; acceleration

Introduction

Cutting and threshing of small grains by human power are tedious and time consuming tasks, hence mechanizing the grain harvesting had been the profound goal of farmers. There are modern combine harvesters with diverse design and construction. Nowadays, technology has provided the machine which is far compliant or flexible, i.e. a combine harvester which can harvest most crops within different farm conditions. During harvest operation the upright crop is pushed by reel against the cutter bar and onto the platform. Grain combine cutter bar is far akin to that of mower sicklebar but the knife

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section motion of former is slower than that of the latter and one side of it links to the reciprocating drive mechanism. A non uniform cutting causes the cutter bar to tears the crop stalks resulting in intense shattering which subsequently increases the grain losses. Drive mechanism types of cutter bar knives employed in grain combines are as follows:-

1. Pitman drive mechanism (slider crank mechanism): It is the oldest type of drive mechanism which is employed in combine (JD 955). Pitman is connected to the hub eccentrically and converts the rotational motion of crank wheel into reciprocating one, thereby ensures the reciprocating movement of knife section. It should be noted that the discussed drive mechanism nearly produces excessive vibrations (Behrouzi Lar, 2000 and Kepner *et al*., 1986).

2. Twin wheel drive mechanism: motion of such mechanisms tends to reduce vibration in cutter bar. In this type, a pair of short pitman has been incorporated that each of them is eccentrically linked to crank wheel (Twin wheel) and the wheels also are counter rotating. Knife is moved toward out when pitman length increases towards out and vice versa. For this mechanism, the velocity of knives is found to vary from 1800 to 2200 cycles (reciprocating motion) per minute (Mobli, 1988).

3. Wobble shaft drive mechanism: In such type a crank shaft (slider shaft) links to pivot yoke and produces an oscillating motion which is transmitted to crank arm through a vertical shaft connected to yoke, thereby a reciprocating motion is developed and transmitted to cutter bar. The cutter bar head has been connected to front end of drive arm and a counterbalance weight has been added to its rear end that it reduces vibrations. In wobble shaft drive unit, to drive the cutter bar, both a ball socket joint and a reciprocating arm equipped with counterweight have been employed. In this system, rotational motion is converted into reciprocating motion through an eccentric shaft connected to the yoke. This system also has compact size and produces less vibration. (Kepner *et al*., 1986). Such system is widely in used because of its compact size as well as its capability in reducing the grain losses. The aforementioned system has became common in several combine such as JD1165, Class (Models of CS115 and CS116) and other modern grain combines. Frequently, combine manufacturers claim that the latter system provides more uniform motion consequently reduces grain losses. However, the above suggestion was investigated by analyzing the stated mechanism motion (Akram, 1989 and Ayazi, 2005).

Materials and methods

Equations of velocity and acceleration in wobble drive mechanism:

In this mechanism, with fixing one of three components and by using spatial analysis based on inversion mechanism characteristic, velocity of any component has been computed.

At any instant each point belonged to wobble ball bearing, in addition to rotation about input axis shaft, its position changes relative to the axis. Since the motions stated above are interrelate, having velocity of one of them based on input velocity, it is possible to calculate another velocity. Let us consider plan Q which is perpendicular to the rotation axis, rotation of this plane is the same as that of the axis. In plane Q, variations of rotation angle relative to angle variations of point P can be computed By considering Fig. 1. θ is an angle formed between ball bearing and the axis perpendicular to input axis and α is its complementary angle (the angle formed between ball bearing and input axis). Note that both of them are interrelated (Mobli, 1988 and Ayazi, 2005).

It is assumed that point A in plane Q has rotated from point A to point B which is designated as φ. In the latter plane, if a straight line is drawn perpendicular to line $O'A$, common point of A' is obtained. From points A' and B two vertical lines are drawn such that point A'' and B' on plane P are obtained. The angle formed between $O'A''$ and $O'B'$ lines, is designated as β. To achieve a certain operating stroke, both crank throw (crank radius) and angle θ must be given as seen in Fig. 1., so that for this type of design by assumption $\theta = 15^{\circ}$ as follows:-

:

$$A'B = A''B'$$

$$\text{tag } \varphi = \frac{A'B}{O'A'}$$

$$tg\beta = \frac{A''B'}{O'A''}$$

$$O'A' = O'A'' \cos 15$$

$$tg\beta = \frac{A'B}{O'A'/\cos 15} \rightarrow \beta = \frac{A'B}{O'A'}\cos 15$$

$$tg\beta = tg\varphi \cos 15 \rightarrow \beta = tg^{-1}(\cos 15 tg\varphi)$$

From above equations and having φ, we can determine angle β at any instant. Angle β, i.e. rotation of a point in plane P, the position of the latter point with

respect to the axis can be calculated. For this purpose, plane P relative to axis can be determined.
In diagrams related to wobble drive, angular velocity of wobble output shaft ω_4 and its angular acceleration α_4 are derived by the following equations:-

$$\omega_4 = \frac{\omega_2 Cos15Cos75Sin\beta}{(Cos^2 + Cos^2 15Sin^2\varphi)\sqrt{1 - Cos^2 75Sin^2\beta}}$$

or (2)

$$\omega_4 = \frac{0.25\omega_2 Sin\beta}{(Cos^2\varphi + 0.93Sin^2\varphi)\sqrt{1 - 0.067Cos^2\beta}}$$

where

$\beta = tg^{-1}(\cos 15^0 tg\varphi)$

By differentiating the above expression with respect to time, output axis acceleration is obtained:

$$\alpha_4 = \frac{d\omega_4}{dt}$$

$$\alpha_4 = \frac{0.24\omega_2^2 Cos\beta + \omega_2 Sin2\varphi - 0.93\omega_2 Sin2\varphi - \frac{0.065\omega_2 Sin2\varphi}{2(1-0.067Cos^2\beta)^{\frac{1}{2}}}(0.25\omega_2 Sin\beta)}{(Cos^2\varphi + 0.93Sin^2\varphi)\sqrt[2]{1-0.067Cos^2\beta}} \quad (3)$$

Substituting φ and β magnitudes into equation 3, we have:

$\alpha_4 = 0 \Rightarrow \alpha^t_{J_6} = RJ_6 \times \alpha_4 = 0 \rightarrow \alpha^t_{J_6} = 0$

Zero acceleration reveals that the knife velocity is maximum in beginning of cutting operation in wobble shaft drive mechanism.

Velocity and acceleration equations in pitman drive mechanism:

Displacement, velocity and acceleration equations can be derived considering Fig.2. To assume that the crank rotates in the clockwise direction with an angular velocity ω, therefore we have:

$x = \{(L+R)^2 - S^2\}^{1/2} - R\cos\theta - L\cos\theta$

Also, we have:

$R\,sin\,\theta = (L\,sin\,\phi - S)$

$sin\,\phi = (R\,sin\,\theta - S)/L$

$cos\phi = \left(1 - sin^2\phi\right)^{1/2} = \left\{1 - 1/L^2\left(R\,sin\theta + S\right)^2\right\}^{1/2}$

To summarize of above expression, we can substitute the following approximation.

$\left(1 \pm B^2\right)^{1/2} = 1 \pm \frac{1}{2}B^2 - \frac{B^4}{2\cdot 4} \pm \frac{1\cdot 3B^6}{2\cdot 4\cdot 6} - \frac{1\cdot 3\cdot 5B^8}{2\cdot 4\cdot 6\cdot 8} \pm ....$

where $B = 1/L\left(R\,sin\theta + S\right)$

By using the first and second terms of above expression

$cos\phi = 1 - 1/2\left\{\left(R\,sin\theta + S\right)/L\right\}^2$

$x = \left\{\left(L+R\right)^2 - S^2\right\}^{1/2} - R\,cos\theta - L + 1/\left(2L\right)\left(R\,sin\theta + S\right)^2$

where $\theta = \omega t$ which by assumption ω is constant, velocity and acceleration can be expressed as:

$V = R\omega\,sin\theta + 1/L\left(R\omega\,cos\theta\right)\left(R\,sin\theta + S\right)$

$V = R\omega\sin\theta + R\omega/L\left(R/2\sin 2\theta + S\cos\theta\right)$

$A = R\omega^2\,cos\theta + R\,{}^{\omega}\!/_{L}\left(R\omega\,cos\,2\theta - S\omega\,sin\theta\right)$

$A = R\omega^2\left(cos\theta + {}^{R}\!/_{L}\,cos\,2\theta - {}^{S}\!/_{L}\,sin\theta\right)$

Generally, $1/25\langle R/L\langle 1/15$, hence velocity and acceleration can be calculated with appropriate approximation using the following relationship.

$V = R\omega\left(sin\theta + {}^{S}\!/_{L}\,cos\theta\right)$

$A = R\omega^2(\cos\theta - S/L\sin\theta)$

For non offset drive mechanism, S becomes zero and

$V = R\omega\,sin\theta$ و $A = R\omega^2\,cos\theta$

Then, displacement becomes: $x = R\left(1 - cos\theta\right)$

Motion analysis and applied softwares

Using the equation represented above earlier, dynamic analysis of velocity and acceleration in both pitman and wobble drive mechanism was accomplished by applying two softwares: Visual Basic and Visual Nastran.

It should be taken into consideration that the aforementioned softwares were employed due to the capability of them for accurate modeling of mechanisms and this fact that they are sufficiently reliable in calculating parameters related to mechanisms such as displacement, velocity and acceleration. At first, the velocity and acceleration of pitman drive mechanism were obtained using the comprehensive programs based on the derived equation written in VB. To establish the reliability of the software, the mechanism was modeled by applying Visual Nastran software and was observed the obtained diagrams are identical.

Since use of Visual Nastran software was more convenient than Visual Basic in terms of modeling three dimensional motions, therefore VN software that was used for both mechanisms for modeling and plotting the diagrams concerned with velocity and acceleration.

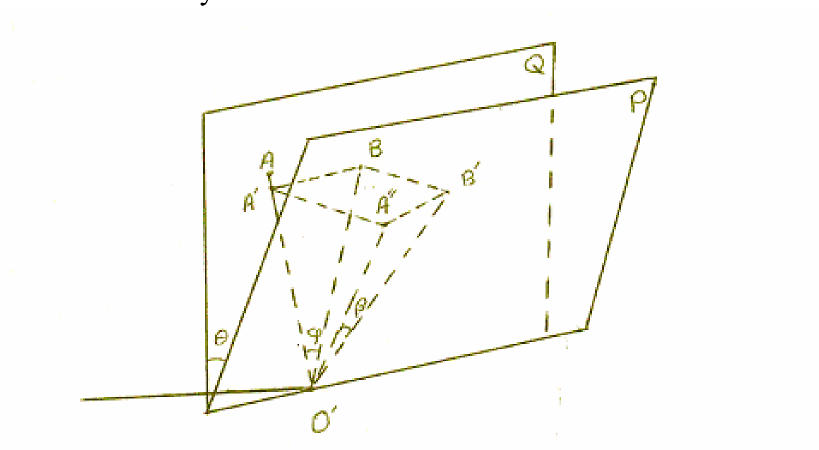

Fig 1. Rotation of a point located in plane P relative to plane Q.

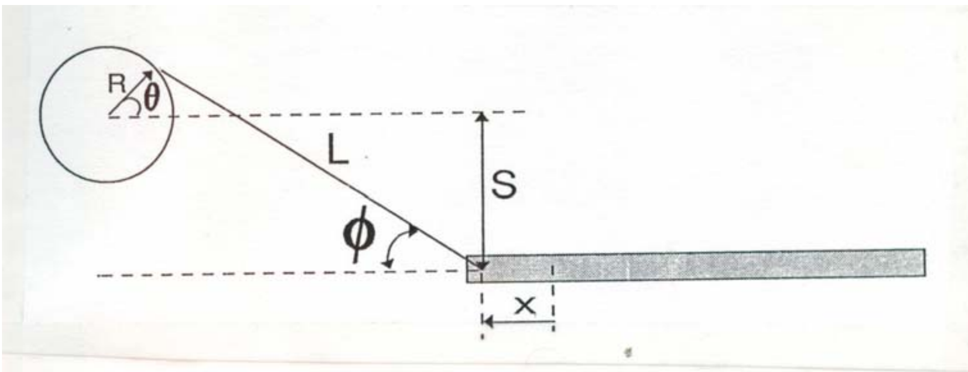

Fig 2. Pitman drive mechanism.

Results and discussion

Since motion and vibration are transmitted to cutter bar through a linkage arm in both under studied mechanisms, for analyzing the motion thus a point located on their linkage arms was considered (Rajabipour, 1993). Number of revolution of pitman and wobble shaft drive mechanism were 510 rpm (JD955

catalogue) and 542 rpm (Jd1165 combine catalogue), respectively and by using the applied software, motion analyzing were performed and diagrams were plotted. After assessing of obtained diagram for each mechanism, their results were compared as follows:-
The pitman drive mechanism diagram (Fig. 3) was indicated that velocity and acceleration had high oscillation amplitude causing the vibration to intensify in cutter bar. In the beginning of cutting operation, the magnitude of acceleration is not zero, thereby resistive inertia forces are developed which results in high vibration in cutter bar. There is an asymmetric state before reaching to peak point as seen in Fig. 3. It is due to the presence of dead points in motion path of pitman which such circumstances cause the irregular motion and vibration to be increased. But in velocity and acceleration diagrams of wobble shaft drive mechanism as shown in Fig. 4 and Fig. 5 can be indicated as in beginning of cutting task, magnitude of acceleration is zero and knife section velocity is maximum; such circumstances substantially tend to reduce unwanted inertia forces and diminish the vibrations in cutter bar. Oscillation amplitudes of velocity and acceleration for the mechanisms are inconsiderable and close to zero; which reduce the vibration in cutter bar. Velocity and acceleration diagrams of this drive mechanism are without dead point; consequently results a regular motion.

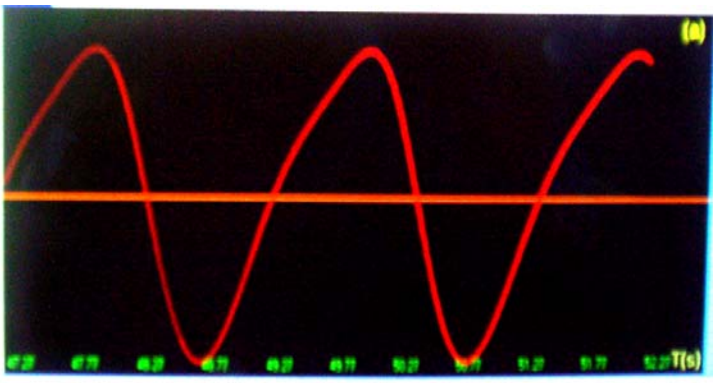

Fig 3. Acceleration diagram of pitman drive

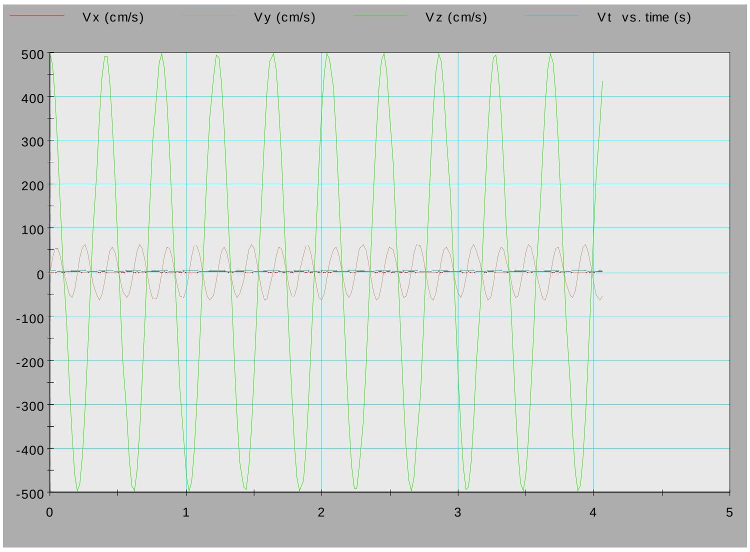

Fig 4. Velocity diagram of wobble shaft drive mechanism in three directions.

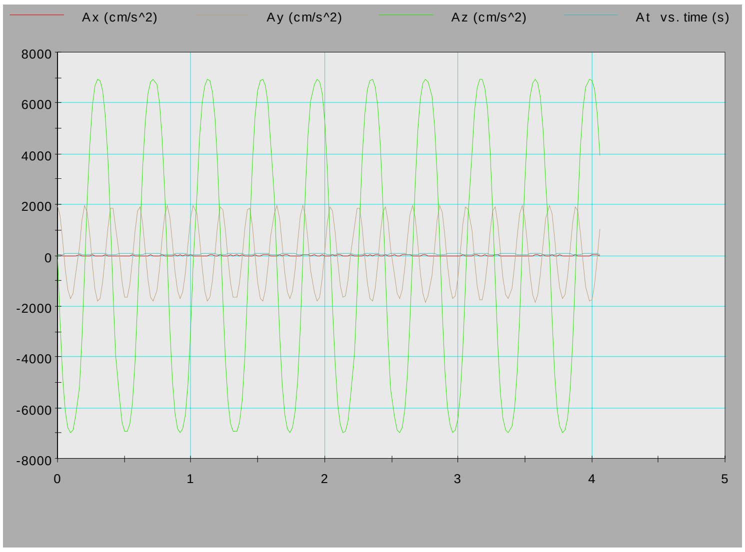

Fig 5. Acceleration diagram of wobble shaft drive mechanism in three directions.

Conclusions

In old cutting systems, transmitting the power to sickle bar have been accomplished by pitman drive mechanism while it is done by means of wobble drive mechanism in new systems (Anon, 2007). Converting the rotational motion into reciprocating one is satisfactorily carried out by the wobble shaft drive mechanism. The advantages of the latter mechanism are as follows:- low vibration, cutting process speed in wobble shaft drive mechanism is significantly higher than that of pitman mechanism; thereby harvesting operation is performed fast. Less velocity and acceleration amplitudes, more regular motion, and less vibration are the profits of this systems compared to pitman drive mechanism, therefore can be concluded the harvest losses would be reduced by using wobble shaft drive mechanism. Yoke only transmits the motion in horizontal direction and reject vertical motion because of restriction in two ends. In platform Seri 300, aforementioned operation is accomplished through an eccentric shaft in wobble system, which it considerably reduces the vibration. In wobble shaft mechanism, in the beginning of cutting process, the acceleration of the most components such as knife section becomes zero due to symmetry of knife stroke; therefore the inertia forces are inconsiderable. Contrary to pitman drive mechanism, there is not asymmetric state (irregularity) in the acceleration diagram related to wobble shaft drive mechanism. Such circumstances reduce the vibrations of cutter bar.

References

Akram, A. (1989). Design and development of sickle bar mower. Unpublished MSc Tesis. University of Tarbiate Modares.

Ayazi, M. (2005). Disigning and fabricating of platform of combine JS 1165. Unpublished MSc Tesis. University of Tehran.

Behroozi Lar, M. (2000). Engineering Principles of Agricultural Machines. Azad Uneversity Publication.

Kepner, R. A., Bainer, E., Barger, L. (1986). Principles of Farm MachineryMansouri Rad, D. 1999. Tractors and Farm Machinery. Bu Ali University Publication.

Mobli, H. (1988). Analysis of cutting mechanisms of existing combine in Iran in order to find a suitable program with regard to circumstances of Iran. Unpublished MSc Tesis. University of Tarbiate Modares.

Anonymous, (2007). Product range, 89/90. Class Combine Catalogue.

Rajabipour, A. (1993). Sickle bar motion project. Mc Gill University, Montral, Canada.

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