Cutting speed follow-up adjusting system of bidirectional electric drive side cutter for rape combine harvester

Zhuohuai Guan¹,², Chongyou Wu¹, Ying Li¹, Senlin Mu¹ and Lan Jiang¹

¹Nanjing Research Institute for Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, Nanjing, China
²Synergistic Innovation Center of Jiangsu Modern Agricultural Equipment and Technology, Zhenjiang, China

Abstract

In rape combine harvester, side cutter must be equipped to cut off tangled rapeseed twigs. Inappropriate cutting speed would increase the repeated cutting and missing cutting of side cutter, which lead to serious header loss. In allusion to the problems mentioned above, bidirectional electric drive side cutter and a cutting speed follow-up adjusting system were proposed. The kinematic law of side cutter blades was analyzed. The trajectory, velocity, and acceleration of the two blades were the same, but the phase difference is $\pi$. Numerical simulation of cutting areas at different cutting speed ratios was carried out and the best cutting speed ratio was determined to be 1.1. Cutting speed follow-up adjusting system was designed based on matching relationship between combine harvester forward speed and side cutter cutting speed. Cutting speed follow-up adjusting system was designed with proportional–integral–derivative (PID) algorithm. The control parameters were determined to be $K_p = 1.3$, $K_i = 4.3$, $K_d = 0.007$. Simulation showed that the maximum overshoot of the system was 4.3%, steady-state error was 0.24%, and the rise time was 0.036 s. The cutting speed follow-up adjusting system was applied to the 4LZ-6T-type rape combine harvester. Experimental results showed that the side cutter cutting speed error was within 1.5%. When forward speed changed, the cutting speed response delay time was 1.5 s. The rape combine harvester header average loss was 2.96% and side cutter average loss was 0.81%. Compared to the fixed speed cutting, header loss was reduced by 14.05% and side cutter loss was reduced by 34.76%. The research can reduce the loss of rapeseed combine harvester and provide theoretical basis for the design of rapeseed combine harvester.

Corresponding author:

Senlin Mu, Nanjing Research Institute for Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, Nanjing 210014, China.

Email: 398764546@qq.com
Keywords
Rape combine harvester, header loss, side cutter, cutting speed, follow-up adjusting system

Introduction

Rape is one of the most important oil crops in the world. In China, rape planting area is about 7.3 million ha and the total output is over 14 million tons. The planting area and total output account for about 30% of the world. Rape is one of the most important oil crops in China and the world. The rape pods are fragile and its branches are twisted together at harvesting time. It is necessary to cut the tangled rapeseed twigs by the side cutter during harvest.\(^1\) However, side cutter would cut pod or pull rape twigs as well during branch cutting, which causes rapeseed falling and splash loss. At present, seed loss caused by header accounts for approximately 50% of the total loss of rapeseed harvesters, and the side cutter loss accounts for 40% of the header loss. Side cutter is one of the most important reasons for the high loss of rapeseed header.\(^2\)

The combine header includes multiple work parts; header loss is generally reduced by improving the structure and operating parameters of these work parts. Tofanica et al.\(^3\) tested the physical and mechanical properties of rapeseed stalks and analyzed the relationship between shear modulus and water content and cutting position. Research on the reeling speed and installation position, header conveyor belt, rotary hydraulic divider, header vibration, airflow assisted recovery structure, and other related header structures and parameters have also been carried out to achieve a better harvest effect.\(^4\)–\(^9\)

The rape side cutter generally achieves cutting function by the cooperation of the movable blade and the fixed blade. Movable blade driven by crank-linkage mechanism has a fixed cutting speed due to its mechanical transmission structure. The inertial force, generated in the side cutter under the work of the crank-linkage mechanism, increases the vibration of the header and the harvester.\(^10\) Bidirectional cutter can increase the relative speed between the cutter and the crop to balance the inertial force produced by the reciprocating motion of the cutter,\(^11\) leading to a widely application to rice, wheat, ramie, pasture, and bush harvesters.\(^12\)\(^,\)\(^13\) Xu et al.\(^14\) designed the hydraulic double-acting side cutter for the rapeseed header which increased the cutting efficiency and reduced the involvement of rape branches during harvest. However, the cutting speed cannot be adjusted during harvest.

At present, the cutting speed of conventional side cutter is fixed. The side cutter does not have enough time to cut the branches if the forward speed of the harvester is too fast; in contrast, the repeated cutting occurred if the forward speed is too slow. Thus, missing cutting of side cutter and repeated cutting will lead to harvest loss.

In this article, bidirectional electric drive side cutter for rape combine harvester and a cutting speed follow-up adjusting system were proposed. The forward speed of combine harvester was acquired based on GPS module. The bidirectional
electric drive side cutter was driven by a DC motor. Control system adjusted motor speed in real time to match the forward speed. In this way, side cutter can maintain the optimal cutting speed to reduce the harvest loss caused by the repeated cutting and missing cutting and improve the working performance of the rape combine harvester.

Materials and methods

Bidirectional electric drive side cutter

In order to achieve the real-time cutting speed control of the side cutter, the bidirectional electric drive side cutter consists of drive mechanism and blades. The drive mechanism was composed of motor, gear, eccentric wheel, and crank as shown in Figure 1(a). The motor drove the blades through the gear and the double eccentric mechanism. Two blades reciprocating moved in opposite directions to achieve bidirectional cutting and reduce the cutting inertia force at the same time. The blade structure is shown in Figure 1(b). Main structural parameters of side cutter are shown in Table 1.

Figure 1. Bidirectional electric drive side cutter: 1. Motor; 2. Gear; 3. Cabinet; 4. Crank; 5. Limit block; 6. Blade holder; 7. Eccentric wheel; 8. Eccentric shaft; 9. Upper blade; 10. Lower blade. (a) Structure scheme, (b) blade dimension, and (c) factual picture.
Cutting speed follow-up adjusting system

Cutting speed follow-up adjusting system is shown in Figure 2, mainly including transformer module, GPS module, main controller, motor drive module, and so on.

Power supply came from the car DC24V battery. The DC24V power was distributed by the transformer module to match different working parts. The working voltages of motor, GPS module, microcontroller, and motor driver module were 24V, 3.3V, and 5V and 5V, respectively. The motor drive module used a high-power MOS tube with optocoupler isolation, which mainly converted the 5V low-power PWM control signal generated by the I/O port of the single-chip into the

### Table 1. Main structural parameters of side cutter.

| Structural parameters           | Value |
|---------------------------------|-------|
| Transmission mechanism          |       |
| Eccentricity, \( e \) (mm)      | 30    |
| Crank length, \( l \) (mm)      | 80    |
| Blade                           |       |
| Length, \( L \) (mm)            | 1300  |
| Stroke, \( s \) (mm)            | 60    |
| Front axle width, \( a \) (mm)  | 10    |
| Center distance, \( b \) (mm)   | 60    |
| Bottom width, \( d \) (mm)      | 30    |
| Edge height, \( c \) (mm)       | 40    |
| Sliding angle, \( \alpha_b \) (°) | 19  |

![Figure 2. Cutting speed follow-up adjusting system: 1. Battery; 2. Transformer module; 3. GPS module; 4. Controller; 5. Motor drive; 6. Motor; 7. Side cutter.](image)

*Cutting speed follow-up adjusting system*

Cutting speed follow-up adjusting system is shown in Figure 2, mainly including transformer module, GPS module, main controller, motor drive module, and so on.

Power supply came from the car DC24V battery. The DC24V power was distributed by the transformer module to match different working parts. The working voltages of motor, GPS module, microcontroller, and motor driver module were 24V, 3.3V, and 5V and 5V, respectively. The motor drive module used a high-power MOS tube with optocoupler isolation, which mainly converted the 5V low-power PWM control signal generated by the I/O port of the single-chip into the
24V high-power PWM drive signal required by driving the motor. Optocoupler isolation prevented 24V motor operation (especially inrush current) from harming the microcontroller.

The STC15W4K60S4 MCU was adopted as main controller. The MCU has 2k SRAM and the highest operating frequency is 28 MHz. The main controller was developed based on keil uvision5 and programmed with C language. The speed accuracy of the GPS module is 0.1 m/s and the maximum signal update frequency is 10 Hz. The GPS module communicates with the MCU via USB. NMEA-0183 protocol is used to transfer the location data, which mainly includes GNGGA (GPS positioning information) frame, GNGSA (current satellite information) frame, GPGSV (visible GPS satellite information) frame, GNRMC (recommended positioning information) frame, GNVTG (ground speed information) frame, and GNGLL (geodetic information) frame. The speed information is contained in the GNVTG (ground speed information) frame; the basic format is as follows:

$\text{SGNVTG}, (1), T, (2), M, (3), N, (4), K, (5)*hh(CR)(LF)$

**Figure 3.** Program flow chart of velocity computing.
GNVTG: $ is the starting bit of the frame command, GN is the identifier, and VTG is the statement name;

By processing the data at (4) in the GNVTG frame, the forward speed of the current tractor can be obtained. The specific process is shown in Figure 3.

The optimum speed of drive motor was calculated as the input of the control system and the actual speed of the motor was taken as feedback. The system output is PWM wave to motor drive module and adjusted cutting speed of side cutter. Proportional–integral–derivative (PID) strategy was used to control motor speed in a closed loop. Based on the above processes, the follow-up adjusting of side cutter speed was realized. Due to the complicated working conditions in the field, the valid data of the forward speed would be cached and read when GPS signal was abnormal to ensure the normal work of side cutter even if GPS signal was interrupted. The control strategy is shown in Figure 4.

**Numerical simulation of blade trajectory**

The kinematic model of bidirectional electric drive side cutter is shown in Figure 5. The center of the first eccentric wheel is $O_1$, the center of the second eccentric wheel is $O_2$, and the eccentricity is $e$. The eccentric shaft is fixed at the midpoint of the line $O_1O_2$ connecting the centers of the two eccentric wheels, which drives the two
eccentric wheels to move synchronously and establishes a coordinate system with the eccentric shaft as the origin $O$. The crank is connected with the eccentric wheel and the blade, and the connection points with the two blades are $E$ and $F$. The lengths of both two crank is $l$. The blade moves along $y$ axis.

The motion of any point on the blade is the same in the $y$ axis direction. According to the geometric relationship, the ordinate of points $E$ and $F$ are, respectively

$$
y_E = l \cos \beta - e \cos \theta \quad \theta \in \left(-\frac{\pi}{2}, \frac{\pi}{2}\right)
$$
$$
y_F = l \cos \beta + e \cos \theta \quad \theta \in \left(-\frac{\pi}{2}, \frac{3\pi}{2}\right)
$$

where $y_E$ and $y_F$ were ordinate of points $E$ and $F$; $\theta$ was the angle between $O_1O_2$ and $y$ axis ($^\circ$); and $\beta$ was the angle between the crank and $y$ axis ($^\circ$).

According to equation (1), the motion trajectories of two blades are same but the phase difference is $\pi$. The displacement of the two blades relative to the starting point is equal in magnitude and opposite in direction. When $\theta = n\pi$ ($n = 0, 1, 2, ...$), $\beta = 0$, the displacement reaches an extreme value with the same magnitude as the eccentricity $e$.

The derivative of position equation is the expression of the cutting speed

---

**Figure 5.** Kinematics model of drive mechanism.

$O_1$ is the center of the first eccentric; $O_2$ is the center of the second eccentric; $O$ is the midpoint of $O_1O_2$; $\theta$ is the angle between $O_1O_2$ and the $y$ axis ($^\circ$); $\beta$ is the angle between the crank and $y$ axis ($^\circ$); $e$ is eccentric distance (mm); $l$ is the length of $O_2E$, $O_1F$ (mm); $E$, $F$ are the connection points between cranks.

---

Guan et al.
\[
\begin{align*}
  v_E &= -l \sin \beta \cdot \omega_1 + e \sin \theta \cdot \omega_2 \quad \theta \subset \left(-\frac{\pi}{2}, \frac{\pi}{2}\right) \\
  v_F &= -l \sin \beta \cdot \omega_1 - e \sin \theta \cdot \omega_2 \quad \theta \subset \left(-\frac{3\pi}{2}, \frac{3\pi}{2}\right)
\end{align*}
\]

(2)

where \(v_E\) is the speed of point \(E\) (m/s); \(v_F\) is the speed of point \(F\) (m/s); \(\omega_1\) is the crank swing angular velocity (rad/s); and \(\omega_2\) is the rotational angular velocity of the eccentric wheel (rad/s).

According to equation (2), the speed change curves of the two blades have the same shape and the phase difference is \(\pi\). The two blades have the same speed and opposite directions. When \(\theta = n\pi/2\) (\(n = 0, 1, 2, \ldots\)), \(\sin \beta = e/l\), the cutting speed of the blade reaches extreme value \(e|\omega_1 - \omega_2|\).

The derivative of speed equation is the expression of the cutting acceleration

\[
\begin{align*}
  a_E &= -l\omega_1^2 \cos \beta - l \sin \beta \alpha_1 + e\omega_2^2 \cos \theta + e \sin \theta \alpha_2 \quad \theta \subset \left(-\frac{\pi}{2}, \frac{\pi}{2}\right) \\
  a_F &= l\omega_1^2 \cos \beta + l \sin \beta \alpha_1 + e\omega_2^2 \cos \theta + e \sin \theta \alpha_2 \quad \theta \subset \left(-\frac{3\pi}{2}, \frac{3\pi}{2}\right)
\end{align*}
\]

(3)

where \(a_E\) is the acceleration of point \(E\) (m/s^2); \(a_F\) is the acceleration of point \(F\) (m/s^2); \(\alpha_1\) is the angular acceleration of crank \(O_2E\) (rad/s^2); and \(\alpha_2\) is the angular acceleration of crank \(O_1F\) (rad/s^2).

According to equation (3), the acceleration curves of the two blades have the same shape and the phase difference is \(\pi\). The accelerations of the two blades are equal in magnitude and opposite in direction. The inertial forces cancel each other, thus reducing the influence on the vibration of the header. When \(\theta = n\pi\) (\(n = 0, 1, 2, \ldots\)), \(\beta = 0\), the acceleration reaches extreme value \(|e\omega_2^2 - \omega_1^2|\).

According to the design principle of involute cylindrical gears, the number of teeth should be greater than minimum tooth undercutting and the two gears number should be mutual quality. So, one of the gear is 29 teeth and the other is 85, the reduction ratio of which is 3:1. And the diameter of the root circle of the large gear wheel is 104 mm. From equations (1)–(3), the key parameters affecting the kinematic characteristics of the drive mechanism are eccentric wheel eccentricity \(e\) and crank length \(l\). Eccentricity \(e\) is related to the movement range of the blade, which affects the effect of the blade in gripping the stem and cutting. Crank length \(l\) affects the stroke speed ratio of the reciprocating motion of the moving blade. According to the stroke distance and stroke speed ratio of the standard type II cutter, combined with the diameter of the root circle of the large gear, the eccentricity \(e\) is 30 mm and the crank length \(l\) is 80 mm.

In order to clarify the best cutting speed of the side cutter, numerical simulation of cutting area was carried out. The absolute trajectory of the side blade was a cosine curve synthesized by the forward motion of the harvester, and simple harmonic motion of the blade is relative to the harvester. The cutting area is shown in Figure 5.
The relationship between the forward distance $H_1$ in one cutting step and the forward area $S$ is obtained as follows

$$S = H_1 \cdot d = v_m T \cdot d = \frac{60dv_m}{2\pi}$$  \hspace{1cm} (4)

where $v_m$ is the forward speed of combine harvester (m/s); $H_1$ is the forward distance (mm); $T$ is the one cutting step time (s); $n'$ is the rotating speed of eccentric shaft (rad/min); and $d$ is the bottom width of blade.

Within one cutting step time $T$, the cutting areas of two single blades are $A_0B_0B_1A_1$ and $C_0D_0D_1C_1$, respectively, denoted as $S_1$; the combine cutting area of double blades is $PQMN$, denoted as $S_2$; the missing cutting area is $QEP'$, denoted as $S_3$; the repeated cutting areas are $A_2HE$, $C_2FG$, denoted as $S_4$. The expressions of each area are as follows

$$S_1 = \int_{x_{b_0}}^{x_{a_1}} (L_{a_0} - L_{a_1}) \, dx + \int_{x_{b_0}}^{x_{a_1}} (L_{b_0} - L_{b_1}) \, dx + \int_{x_{b_1}}^{x_{a_1}} (L_{b_0} - L_{b_1}) \, dx$$

$$S_2 = \int_{x_{m}}^{x_{e}} (L_{c_0} - L_{c_1}) \, dx + \int_{x_{m}}^{x_{p}} (L_{d_0} - L_{d_1}) \, dx$$

$$S_3 = \int_{x_{e}}^{x_{f}} (L_{a_2} - L_{a_3}) \, dx + \int_{x_{e}}^{x_{f}} (L_{c_2} - L_{c_3}) \, dx$$

$$S_4 = \int_{x_{t_2}}^{x_{t_1}} (L_{a_2} - L_{a_3}) \, dx + \int_{x_{e}}^{x_{f}} (L_{d_2} - L_{d_3}) \, dx$$

In order to quantify the cutting performance, all the cutting areas were calculated in MATLAB according to equation (5). The cutting speed ratio $K$ was defined as the ratio of side blade average speed $v_f$ to forward speed $v_m$ of the harvester

$$K = \frac{v_f}{v_m} = \frac{s}{H_1} = \frac{s}{H_1}$$  \hspace{1cm} (6)

where $s$ is the blade displacement (mm), and $s$ is equal to the forward distance in one cutting step time $T$ which was 60 mm.

All cutting areas can be expressed as a function of $K$

$$S_i = f(K)(i = 1, 2, 3, 4)$$  \hspace{1cm} (7)

The cutting rate $P_i$ is the ratio of cutting area $S_i$ to forward area $S$

$$P_i = \frac{S_i}{S}(i = 1, 2, 3, 4)$$  \hspace{1cm} (8)

The cutting areas and corresponding cutting rate in one cutting step were calculated at different $K$, the results of which are shown in Table 2.

According to numerical simulation results, the quadratic relationship between $P_3$ and $K$ can be fitted as $P_3 = 0.2851K^2 - 0.7656K + 0.5139$, $R^2$ was 0.998. When
Table 2. Numerical simulation results of cutting area.

| Cutting speed ratio, $K$ | Feeding distance, $H_1$ (mm) | Feeding area, $S$ (mm$^2$) | Single action blade cutting | Bidirectional action blades cutting | Missing cutting | Repeated cutting |
|-------------------------|-------------------------------|----------------------------|-----------------------------|------------------------------------|----------------|-----------------|
|                         |                               |                            | Area, $S_1$ (mm$^2$)       | Cutting rate, $P_1$ (%)            | Area, $S_3$ (mm$^2$) | Missing rate, $P_3$ (%) | Area, $S_4$ (mm$^2$) | Repeated rate, $P_4$ (%) |
| 0.80                    | 75                            | 4 500                       | 3 150                       | 70.0                               | 1 307           | 29.0            | 377             | 8.4             | 51              | 1.1             |
| 0.88                    | 68                            | 4 080                       | 3 080                       | 75.5                               | 1 370           | 33.6            | 247             | 6.0             | 45              | 1.1             |
| 1.00                    | 60                            | 3 600                       | 3 000                       | 83.3                               | 1 440           | 40.0            | 121             | 3.4             | 79              | 2.2             |
| 1.13                    | 53                            | 3 180                       | 2 930                       | 92.1                               | 1 491           | 46.9            | 39              | 1.2             | 125             | 3.9             |
| 1.33                    | 45                            | 2 700                       | 2 847                       | 105.4                              | 1 536           | 56.9            | 0               | 0               | 208             | 7.7             |
$K = 4/3$, the missing cutting area was 0. The quadratic relationship between $P_4$ and $K$ can be fitted as $P_4 = 0.204K^2 - 0.3092K + 0.127$, $R^2$ is 0.997. In the range of simulation, $P_4$ increased with the increase in $K$, leading to a larger repeated cutting area and higher cutting loss. Considering to achieve smallest missing cutting and repeated cutting areas, the cutting performance was best when the sum of the two functions was minimum as below:

$$P = P_3 + P_4 = 0.4891K^2 - 1.074K + 0.6409,$$

when $K$ was 1.1, $P$ took the minimum value. While $H_1 = 54.5$ mm. The relationship between $K$ and $P_3$, $P_4$, $P$ is shown in Figure 6.

According to equation (1), the speed of eccentric shaft $n'$ is

$$n' = 1000 \times \frac{60v_m}{2H_1} \tag{9}$$

According to the fact that the mechanical transmission ratio of is 3, the relationship between side cutter drive motor and eccentric shaft is

$$n = 3n' = 3 \times 1000 \times \frac{60v_m}{2H_1} \tag{10}$$

where $n$ is the rotating speed of side cutter drive motor (rad/min).

Equation (10) is the calculation method of drive motor expectation speed. The settlement result is the input of the control system.
Control parameter turning

PID control strategy was used to adjust motor speed. Parameter turning was the core content of PID control system design. The proportional coefficient, integral coefficient, and differential coefficient of the PID controller were determined according to the characteristics of the controlled process to obtain better control performance.

The basic parameters of side cutter drive motor were as follows: rated voltage $U_e = 24$ V, rated current $I_e = 9.19$ A, rated speed $n_e = 3600$ r/min, rated power $P_e = 180$ W, moment of inertia $J = 0.003$ kg m$^2$. According to empirical formula, motor rotating voltage coefficient $C_e = 0.058$, electric time constant $T_a = 0.012$ s, and mechanical time constant $T_m = 0.24$ s. The transfer function between the motor speed and the input voltage is as follows

$$H(s) = \frac{17.24}{0.0029s^2 + 0.24s + 1}$$

According to the expression of discrete PID

$$u_k = K_p e_k + K_i \sum_{j=0}^{k} e_j + K_d (e_k - e_{k-1})$$

where $k$ is the sampling sequence number, $k = 0, 1, 2, ...$; $u_k$ is the output value at the $k$ sampling time; $e_k$ is the deviation value input at the $k$ sampling time; $e_k$ is the deviation value input at the $k-1$ sampling time; $K_p$ is the proportional coefficient; $K_i$ is the integral coefficient; and $K_d$ is the differential coefficient.
Control parameter was turned based on the attenuation curve procedure, according to the 4:1 attenuation method. The step response under different control parameters is shown in Figure 7. The control parameters of response 1 were $K_p = 2.5$, $K_i = 5.9$, and $K_d = 0.013$; the control parameters of response 2 were $K_p = 1.3$, $K_i = 4.3$, and $K_d = 0.007$; the control parameters of response 3 were $K_p = 0.9$, $K_i = 4.3$, and $K_d = 0.007$; and the control parameters of response 4 were $K_p = 0.5$ and $K_i = 2$.

Control parameter was turned based on the attenuation curve procedure, according to the 4:1 attenuation method. The step response under different control parameters is shown in Figure 7. The control parameters of response 1 were $K_p = 2.5$, $K_i = 5.9$, and $K_d = 0.013$; the maximum overshoot of the system was 6.9%; steady-state error was 0.15%; and the rise time was 0.023 s. The control parameters of response 2 were $K_p = 1.3$, $K_i = 4.3$, and $K_d = 0.007$; the maximum overshoot of the system was 4.3%; steady-state error was 0.24%; and the rise time was 0.036 s. The control parameters of response 3 were $K_p = 0.9$, $K_i = 4.3$, and $K_d = 0.007$; the maximum overshoot of the system was 3.9%; steady-state error was 2.02%; and the rise time was 0.052 s. The control parameters of response 4 were $K_p = 0.5$, $K_i = 2$, and $K_d = 0.003$; the maximum overshoot of the system was 0.9%; steady-state error was 0.90%; and the rise time was 0.094 s. Considering maximum overshoot, steady-state error, and the rise time, the parameters of control system were defined as $K_p = 1.3$, $K_i = 4.3$, and $K_d = 0.007$.

Harvest experiment method

The field experiment was carried out on June 2019 in Dafeng, Jiangsu. The rape plant used in the experiment was Ningza 1818. The moisture content of seeds and stems was 17.2% and 49.7%, respectively. The rapeseed of 1000-grain weight was 3.81 g, height of the bottom pod was 68.0 cm, crown diameter was 34.6 mm, plant height was 134.6 cm, and yield was 3900 kg/ha.

The control system was evaluated by detecting the loss of header and side cutter. Trays were placed between adjacent rape plants and applied for collecting the Shatter. The length, width, and depth of each tray were 3, 0.16, and 0.1, respectively. The trays were placed in two directions. In the forward direction of harvester, the trays were
The trays placed in three positions, arranged parallel to each other. The both end of the line that placed trays exceeded the length of harvesting area by 0.2 m. The trays placed in this direction mainly collected the loss of header. In the normal direction of forward direction, the trays were placed in three positions too, arranged as a straight line under side cutter working areas and collect the loss caused by side cutter. Once the combine harvester finished harvesting, the trays were extracted. The rapeseeds inside the trays were separated and weighed. The actual losses could be calculated based on the ratio of harvested area to actual field area.

Cutting speed follow-up adjusting system of bidirectional electric drive side cutter was applied for 4LZ-6T rape combine harvester. As shown in Figure 8, the parameters of the combine harvester are shown in Table 3.

### Results and discussion

#### Control system experiment

Side cutter cutting speed adaptive adjustment control system test was carried out at an outdoor test site with a flat surface. The forward speed of the harvester kept constant; side cutter drive motor speed was continuously recorded. The variable

| Items                                      | Parameters       |
|--------------------------------------------|------------------|
| Dimensions (length × width × height) (mm)   | 6940 × 3130 × 3460 |
| Machine quality (kg)                       | 6450             |
| Header width (mm)                          | 2750             |
| Feed rate (kg/s)                           | ≤6               |
| Theoretical operating speed (km/h)         | 1.6–7.2          |
| Productivity (hm²/h)                       | 0.7–1.5          |

**Figure 9.** 4LZ-6T rape combine harvester and field harvest experiment.
motor speed was detected when forward speed was changed. The performance of the control system was verified by comparing with the theoretical values. The test result is shown in Figure 10.

Since the forward speed of the harvester was manually controlled, the actual forward speed of the combine harvester slightly fluctuated around the target value, 0.8 m/s. According to equation (6), the theoretical speed of the drive motor was 1321 rad/min. The actual average speed was 1342 rad/min with an error of 1.5%. When the forward speed became 1.2 m/s, the theoretical speed of the drive motor was 1982 rad/min. The maximum overshoot was 3.7% during regulation process and the rising time was 1.5 s. The actual average speed of motor was 1956 rad/min with an error of 1.3%. The experimental results show that motor speed could be adjusted according to the following up relationship when forward speed change.

Cutting speed ratio experiment

To test the performance of side cutter and the simulation of cutting speed ratio, experiments were carried out in five different cutting speed ratios. Each cutting speed ratio was tested three times. Header loss and side cutter loss were recorded and averaged. The results are shown in Table 4.

Header loss and side knife loss was minimum when cutting speed ratio $K$ was 1.08, close to simulation result 1.1. Compared with the other cutting speed ratios, side knife loss rate is reduced by 39.86%, 23.21%, 17.31%, and 28.93%, respectively. The tests indicate that header loss and side knife loss could be effectively reduced at the optimal cutting speed ratio. The average side cutter loss rate under different speed ratios was 1.13%, which accounts for 29% of header loss.

| Cutting speed ratio, $K$ | 0.76 | 0.91 | 1.08 | 1.63 | 2.4 |
|-------------------------|------|------|------|------|-----|
| Header loss (%)         | 3.52 | 3.35 | 3.02 | 3.37 | 3.41|
| Side cutter loss (%)    | 1.43 | 1.12 | 0.86 | 1.04 | 1.21|

*Figure 10. Motor speed.*

*Table 4. Loss in different cutting speed ratios.*

Guan et al. 15
The forward speed was considered to be variable factor. The performance of the cutting speed follow-up adjusting system was evaluated by comparing the loss rates with the system turning on or off in the harvester. Evaluation indicators included side cutter loss ratio to total header loss, reductions of header loss and side cutter loss. The results are shown in Table 5.

The header loss was 3.31%–3.59% and side cutter loss was 1.13%–1.34% when the cutting speed follow-up adjusting system was turned off. The minimum side cutter loss rate was 1.13% when forward speed was 0.47 m/s and the maximum loss rate was 1.34% when forward speed was 1.04 m/s. The average loss rate under different forward speed was 1.24% and increase with the forward speed. The header loss was 2.89%–3.06% and side cutter loss was 0.76%–0.86% when the cutting speed follow-up adjusting system was turned on. The minimum side cutter loss rate was 0.76% when forward speed was 0.47 m/s and the maximum loss rate was 0.86% when forward speed was 1.04 m/s. The average loss rate under different forward speed was 0.81% and increase with the forward speed. After using the cutting speed follow-up adjusting system, header loss reduction was 14.05%, and side cutter loss reduction was 34.76%.

According to the research by Ran and Wu, the combine harvester side cutter was in general driven by a crank link. The side cutter loss rate often accounts for more than 40% of header loss. From the test result in Table 4, the loss rate of header can be reduced by more than 27.5% when bidirectional electric drive side cutter was equipped. The side cutter driven by crank link is unable to adjust cutting speed during harvest; hence, inappropriate cutting speed would increase the repeated cutting and missing cutting of side cutter, which leads to serious header loss. From the test result in Table 4, the side cutter loss difference was no more than 2% in different forward speed when cutting speed follow-up adjusting system was turned on. Field experiment showed that the follow-up adjusting system can significantly reduce the header loss.

### Table 5. Result of field experiment.

| Serial number | 1   | 2   | 3   | Mean |
|---------------|-----|-----|-----|------|
| Forward speed (m/s) | 0.47 | 0.71 | 1.04 | 0.74 |
| System status | Off | On  | Off | On  | Off | On  | Off | On  |
| Header loss (%) | 3.31 | 2.89 | 3.42 | 2.92 | 3.59 | 3.06 | 3.44 | 2.96 |
| Side cutter loss (%) | 1.13 | 0.76 | 1.26 | 0.81 | 1.34 | 0.86 | 1.24 | 0.81 |
| Side cutter loss rate (%) | 34.14 | 26.30 | 36.84 | 27.74 | 37.33 | 28.10 | 36.10 | 27.38 |
| Header loss reduction (%) | 12.69 | 14.62 | 14.76 | 14.05 |
| Side cutter loss reduction (%) | 32.74 | 35.71 | 35.82 | 34.76 |

“Off” represents cutting speed follow-up adjusting system was turned off; “On” represents cutting speed follow-up adjusting system was turned on.

### Comprehensive test

The forward speed was considered to be variable factor. The performance of the cutting speed follow-up adjusting system was evaluated by comparing the loss rates with the system turning on or off in the harvester. Evaluation indicators included side cutter loss ratio to total header loss, reductions of header loss and side cutter loss. The results are shown in Table 5.

The header loss was 3.31%–3.59% and side cutter loss was 1.13%–1.34% when the cutting speed follow-up adjusting system was turned off. The minimum side cutter loss rate was 1.13% when forward speed was 0.47 m/s and the maximum loss rate was 1.34% when forward speed was 1.04 m/s. The average loss rate under different forward speed was 1.24% and increase with the forward speed. The header loss was 2.89%–3.06% and side cutter loss was 0.76%–0.86% when the cutting speed follow-up adjusting system was turned on. The minimum side cutter loss rate was 0.76% when forward speed was 0.47 m/s and the maximum loss rate was 0.86% when forward speed was 1.04 m/s. The average loss rate under different forward speed was 0.81% and increase with the forward speed. After using the cutting speed follow-up adjusting system, header loss reduction was 14.05%, and side cutter loss reduction was 34.76%.

According to the research by Ran and Wu, the combine harvester side cutter was in general driven by a crank link. The side cutter loss rate often accounts for more than 40% of header loss. From the test result in Table 4, the loss rate of header can be reduced by more than 27.5% when bidirectional electric drive side cutter was equipped. The side cutter driven by crank link is unable to adjust cutting speed during harvest; hence, inappropriate cutting speed would increase the repeated cutting and missing cutting of side cutter, which leads to serious header loss. From the test result in Table 4, the side cutter loss difference was no more than 2% in different forward speed when cutting speed follow-up adjusting system was turned on. Field experiment showed that the follow-up adjusting system can significantly reduce the header loss.
Conclusion

A new type side cutter for rape combine harvester was developed. The side cutter was driven by a pair of symmetrical eccentric wheels. The kinematic law of side cutter blades was analyzed. The trajectory, velocity, and acceleration of the two blades were the same, but the phase difference is $\pi$.

The cutting trajectory equation of side blade was established. Through numerical simulation, the relationship between cutting speed ratio and missing cutting rate was obtained, and repeated cutting rate was fitted. A minimum of the sum was taken as constraint, and the optimal cutting speed ratio was 1.1. According to the speed transfer relationship of bidirectional drive system, the mathematical expression between the forward speed and driving motor was determined.

Based on the research of optimal cutting speed ratio, speed follow-up adjusting system of bidirectional electric drive side cutter for rape combine harvester was developed. The GPS module was used to detect the harvester’s forward speed as input to the control system and the actual speed of the driving motor as feedback. The speed of side cutter was adjusted according to the mathematical expression between the forward speed and driving motor.

The cutting speed follow-up adjusting system of bidirectional electric drive side cutter was designed based on PID algorithm. The parameter setting method of control system was studied and determined to be $K_p = 1.3$, $K_i = 4.3$, and $K_d = 0.007$. Simulation showed that the maximum overshoot of the system was 4.3%, steady-state error was 0.24%, and the rise time was 0.036 s.

The cutting speed follow-up adjusting system was applied to the 4LZ-6T type rape combine harvester. Experimental results showed that the side cutter cutting speed error was within 1.5%. When forward speed changed the cutting speed, the response delay time was 1.5 s.

Field experiment showed that header loss and side knife loss could be effectively reduced at the optimal cutting speed ratio. Header loss of rape combine harvester was 2.89%–3.06% and side cutter loss was 0.76%–0.86% after using the cutting speed follow-up adjusting system. The header loss and side cutter loss were reduced to 14.05% and 34.76%, respectively.

The research can reduce the loss of rapeseed combine harvester and provide theoretical basis for the design of rapeseed combine harvester.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: We would like to express our appreciation to Program for National Key Research and Development Program of China (2016YFD0702101) and Synergistic Innovation Center of Jiangsu Modern Agricultural
Equipment and Technology (4091600002) and Program for and Nation Rape Industry System (CARS-13-10B).

**ORCID iD**
Zhuohuai Guan [https://orcid.org/0000-0002-6931-4345](https://orcid.org/0000-0002-6931-4345)

**References**

1. Alizadeh MR, Bagheri I and Payman M. Evaluation of a rice reaper used for rapeseed harvesting. *Am Eurasian J Agric Environ Sci* 2008; 2(4): 388–394.
2. Guan ZH, Wu CY, Wang G, et al. Design of bidirectional electric driven side vertical cutter for rape combine harvester. *T Chin Soc Agr Eng* 2019; 35(3): 1–8.
3. Tofanica BM, Cappelletto E, Gavrilescu D, et al. Properties of rapeseed stalks fibers. *J Nat Fibers* 2011; 8(4): 241–262.
4. Hirai Y, Inoue E and Mori K. Application of a quasi-static stalk bending analysis to the dynamic response of rice and wheat stalks gathered by a combine harvester reel. *Biosyst Eng* 2004; 88(3): 281–294.
5. Hobson RN and Bruce DM. PM—power and machinery: seed loss when cutting a standing crop of oilseed rape with two types of combine harvester header. *Biosyst Eng* 2002; 81(3): 281–286.
6. Morteza P and Raoufat K. Development and field evaluation of a rotary hydraulic divider for canola harvesting. *J Sci Technol Agr Nat Resour* 2009; 13(47): 181–194.
7. Li YM, Li YW, Xu LZ, et al. Structural parameter optimization of combine harvester cutting bench. *T Chin Soc Agr Eng* 2014; 30(18): 30–37.
8. Ma N, Zhang CL and Li J. Mechanical harvesting effects on seed yield loss, quality traits and profitability of winter oilseed rape. *J Integr Agr* 2012; 11(8): 1297–1304.
9. Huang XM, Zha XT, Zong WY, et al. Design and experiment of the cross-flow air pressure collecting device of rapeseed combined harvesting header. *T Chin Soc Agr Machin* 2016; 47(Suppl. 1): 227–233.
10. Maertens K, Baerdemaeker JD, Ramon H, et al. PH—power and machinery: an analytical grain flow model for a combine harvester, part I: design of the model. *J Agr Eng Res* 2001; 79(1): 55–63.
11. Wang SQ, Sun TK and Yu HJ. The analysis of the trajectories based on the basic parameters of the double eccentric wheel drive mechanism. *Mach Des Ma* 2013; 34(3): 38–40.
12. Liu ZG, Wang DC and Zhai GX. Design and experiment on reciprocating double knife shrub harvester. *T Chin Soc Agr Machi* 2013; 44(Suppl. 2): 102–106.
13. Chen N, Gong YJ and Chen JD. Double knife section reciprocating cutter and drive mechanism for combine. *T Chin Soc Agr Machi* 2008; 39(9): 60–63.
14. Xu LZ, Li YM and Ma CX. Design of main working parts of 4LYB1-2.0 rape combine harvester. *T Chin Soc Agr Machi* 2008; 39(8): 54–58.
15. Tamaoki T, Takezawa M, Kimoto M, et al. Study on online analysis of transfer function of variable-speed rolling mill motor with shaft torsional vibration systems. *IEEE T Ind Appl* 2011; 131(6): 844–857.
16. Kang M, Cheong J, Do HM, et al. A practical iterative PID tuning method for mechanical systems using parameter chart. *Int J Syst Sci* 2017; 48: 2887–2900.
17. Ran J and Wu C. Research status and prospect of cutting mechanism of grain harvester. *J Chin Agr Mech* 2019; 40(2): 25–34.

18. Xiang Y, Luo X, Zeng S, et al. Operation performance analysis of reciprocating cutter based on visual programming. *T Chin Soc Agr Machi* 2015; 31(18): 11–16.

19. Gharakhani H, Alimardani R and Jafari A. Design a new cutter-bar mechanism with flexible blades and its evaluation on harvesting of lentil. *Eng Agric Food Environ* 2017; 10(3): 198–207.

**Author biographies**

Zhuohuai Guan, Ph.D, Assistant Researcher at Nanjing Research Institute for Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, China. Research fields include design and intelligent technology of combine harvester.

Chongyou Wu, Ph.D, Professor at Nanjing Research Institute for Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, China. Research fields include planting and harvesting machinery.

Ying Li, Ph.D, Assistant Researcher at Nanjing Research Institute for Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, China. Research field is agricultural machinery.

Senlin Mu, B.A., Associate Professor at Nanjing Research Institute for Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, China. Research field is harvesting machinery.

Lan Jiang, M.A., Assistant Researcher at Nanjing Research Institute for Agricultural Mechanization, Ministry of Agriculture and Rural Affairs, China. Research field is harvesting machinery.