RESEARCH ON SPEED CONTROL OF AN ELECTRICALLY DRIVEN THRESHING CYLINDER IN A COMBINE HARVESTER BASED ON INERTIA PARAMETER IDENTIFICATION
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基于惯量参数辨识的联合收割机电驱动脱粒滚筒转速控制研究

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ABSTRACT
In response to the speed control requirements of the threshing cylinder in combine harvesters, this paper analyzes the working condition characteristics of the threshing cylinder and establishes a dynamic model. A mathematical model for the electrically driven threshing cylinder is established, and an adaptive speed control strategy is proposed based on the identification of moment of inertia. Experimental results indicate that, the control accuracy and robustness of the system are enhanced. During the start-up process, the overshoot in speed and convergence time are reduced by 11.1% and 31.3%, respectively; when fluctuations occur in the feed rate of the threshing cylinder, the maximum speed fluctuation and adjustment time are reduced by 55% and 33.3%, respectively; when the moisture content of crops changes, the shortest convergence time and maximum fluctuation amplitude of the threshing cylinder's speed are reduced by 57.7% and 18.8%, respectively. This research provides theoretical support and practical foundation for the intelligent control of electrically driven combine harvesters, demonstrating considerable application value.

摘要
针对联合收割机中脱粒滚筒的转速控制需求，本文分析了脱粒滚筒的工况特征并建立了动力学模型，构建了电驱动脱粒滚筒的数学模型，提出了一种基于转动惯量辨识的电驱动脱粒滚筒转速自适应控制策略。实验结果表明，系统的控制精度和鲁棒性得到增强。启动过程中的转速超调量和收敛时间分别减少了11.1%和31.3%；作物喂入量发生波动时，最大转速波动和调整时间分别减少了55%和33.3%；当作物的含水率变化时，脱粒滚筒的转速最短收敛时间和最大波动幅度分别减少了57.7%和18.8%。本研究为电驱动联合收割机的智能化控制提供了理论支持和实践基础，具备良好的应用价值。

INTRODUCTION
Under the dual challenges of skyrocketing global food demand and climate change, combine harvesters hold a central position in modern agricultural systems. Due to the seasonal nature and high labor intensity of the crop harvesting process, there is an urgent demand for combine harvesters that are efficient in production, provide good operational quality, are easy and comfortable to control, and possess intelligent features *(Liu et al., 2024; Ou et al., 2024)*.

The main working components of the combine harvester include header, threshing cylinder, separation and cleaning device and grain unloading auger. Among them, the threshing cylinder is one of the core components, with the main function of separating the grains from the rice heads. Threshing cylinder not only largely determines the threshing quality and production efficiency but also significantly impacts the subsequent processes of separation and cleaning in the combined harvester *(Wang et al., 2023; Tang et al., 2022; Yang et al., 2018; Miu et al., 2007)*. Traditional combine harvesters rely on mechanical transmission systems to transfer engine power to various working components, which limits the speed adjustment range and overload capacity. As a result, it is challenging for the speed of these working components to be adjusted in real-time according to fluctuations in the feed rate of the harvester. This can lead to issues such as significant crop losses due to carryover and clogging of the threshing cylinder, which in turn reduces the crop cleaning rate and overall production efficiency. Therefore, enhancing the dynamic responsiveness and overload capacity of the threshing cylinder's speed adjustment is crucial for improving the harvesting quality and efficiency of combine harvesters.

In response to issues such as the narrow adjustment range of the threshing cylinder speed of the combine harvester, high threshing loss rate, and frequent clogging, some scholars have conducted related research. The research on the operation quality and efficiency of the threshing cylinder of the combine harvester primarily focuses on improving the quality and efficiency of the threshing process through mechanical structural design and the collaborative optimization of operational parameters *(Ivan et al., 2015; Cristea et al., 2023; Vlădut et al., 2023)*. *Vlădut et al., (2022),* proposed a mathematical model describing the axial flow characteristics during the threshing and separation processes of the threshing cylinder. *Wang et al., (2022),* addressed the issue of clogging in the threshing cylinder by designing a telescopic rod tooth-ribbed rod roller type threshing device. This design improves the flow of crops through the synergistic operation of the internal structures. *Zewdie et al., (2024),* explored the performance of bean threshers, using different cylinder speeds, concave apertures, feed rates, and moisture levels as variables. They obtained relatively ideal operating parameters for the threshing cylinder, which improved threshing efficiency and reduced grain loss rates. In addition, research on the threshing cylinder mainly focuses on exploring the interactions between threshing quality, operating parameters, and mechanical structure, aiming for structural optimization design *(Wang et al., 2021; Song et al., 2022; Li et al., 2023; Gong et al., 2024; Tang et al., 2024)*. *Abdeen et al., (2025),* improved the operational parameters and structure of the rice threshing device, reducing costs and power consumption, and significantly enhancing the overall efficiency of the threshing cylinder. *Gou et al., (2025),* introduced the structural design and advancements in intelligent technology applications of the combine harvester's threshing device. However, the drawbacks of multi-stage mechanical transmission systems still exist, such as long transmission routes and low transmission efficiency, which restrict the independent control capacity for adjusting the speed of various operating components as needed. Additionally, the working environment of combine harvesters is complex, with issues such as crop lodging and uneven moisture content, leading to high harvesting loss rates and poor cleanliness.

With the global climate change and the increasing awareness of environmental protection, the rapid development of new energy technologies has become a key driver for the green transformation and upgrading of industries. This provides impetus and ideas for promoting the intelligent development of new energy agricultural equipment. The integration of smart technologies with new energy sources can enhance operational efficiency, reduce emissions, and optimize resource use in agricultural practices, ultimately contributing to sustainable farming and improved productivity *(Liu et al., 2024)*. However, research on electrically driven combine harvesters is still in the preliminary exploration stage *(Jiang, 2015; Han, 2017; Hou, 2020)*, and there is a notable lack of information regarding the electric drive research of core components like the threshing cylinder. This gap indicates a significant opportunity for further study and innovation in this area. Developing electric drive systems for the threshing cylinder could enhance efficiency, reduce fuel consumption, and support the overall shift toward sustainable agricultural practices. Addressing this research gap could lead to advancements in the performance and adaptability of electric combine harvesters in various working conditions.

In summary, this paper addresses the speed requirements of the threshing cylinder during the operation of combine harvesters and aims to improve the system's control accuracy and responsiveness, thereby enhancing harvesting quality and efficiency. Considering that the threshing cylinder is a special component with large inertia affected by its shape, size, and structure, a self-disturbance control method for the threshing cylinder's speed in rice and wheat combine harvesters based on moment of inertia identification is proposed. This research analyzes the working principle and operational needs of the threshing cylinder, develops a dynamic model for it, and establishes a torque equation. By directly driving the threshing cylinder with a Permanent Magnet Synchronous Motor (PMSM), the advantages of quick dynamic response, superior speed control, and strong overload capacity of the motor are utilized. This approach resolves issues related to fixed transmission ratios and the inability to individually control the threshing cylinder's speed. Furthermore, by combining the PMSM's motion model, the operational characteristics of the electrically driven threshing cylinder is analyzed. This allows the combine harvester to maintain the threshing cylinder's speed within optimal ranges under different working conditions, ultimately reducing threshing losses and improving operational efficiency and quality. The study supports the advancement of combine harvesters from automation to intelligent operation.

MATERIALS AND METHODS

Analysis of operating requirements for electrically driven threshing cylinder

This paper is based on the 4LZ-6.0ME tracked combine harvester from World Agricultural Machinery Co., Ltd. (Zhenjiang, Jiangsu Province, China). A schematic diagram of the motor-driven threshing cylinder is shown in Figure 1. In the diagram, T_e represents the electromagnetic torque of the PMSM, while $T_L+B\omega_m$ denotes the load torque on the threshing cylinder and external friction resistance. The PMSM drives the threshing cylinder to generate the rotational angular velocity ω_m. This setup illustrates the relationship between the motor's output and the load conditions on the threshing cylinder, emphasizing the importance of effectively managing these parameters to enhance operational performance.

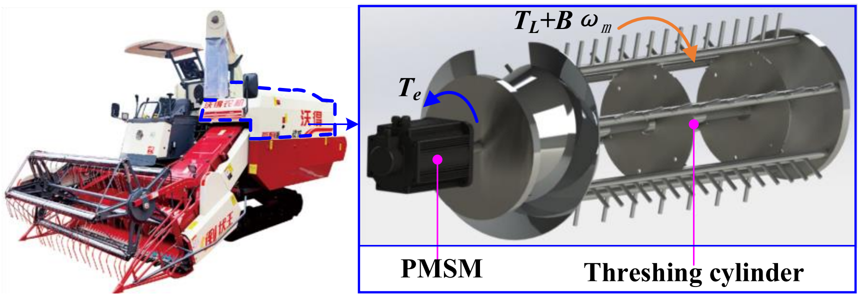


Fig. 1 - Schematic diagram of electrically driven threshing cylinder

The PMSM structure of this paper is shown in Figure 2 and its structural parameters are shown in Table 1 *(Xiang et al., 2024; Xiang et al., 2024)*:

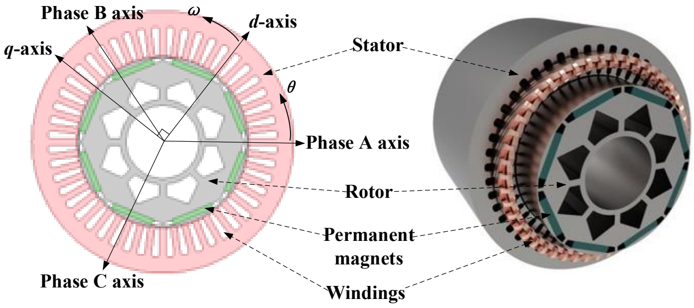


Fig. 2 - Structure diagram of PMSM (Permanent Magnet Synchronous Motor)

Table 1

Parameters of PMSM

Parameter	Value	Unit
Rated voltage	600	V
Rated power	11	kW
Rated speed	1500	rpm
Rated torque	70	Nm
Stator outer diameter	248	mm
Rotor outer diameter	160	mm
Rotor axial length	131	mm
Number of rotor poles	8	/

Relationship between threshing cylinder speed and threshing performance

There are many factors that influence the threshing quality of combine harvesters, including cylinder speed, feed rate, concave clearance, crop properties, field conditions, and the structural design of the cylinder. Among these, the rotational speed of the threshing cylinder is one of the key parameters affecting threshing performance, as the strength of the impact, brushing, and rubbing actions on the crops primarily depends on this speed *(Da, 2021)*. As shown in Figure 3, the relationship between the threshing cylinder speed and various performance indicators indicates that at lower speeds, the impact force is weakened, leading to reduced cleaning and separation rates, which in turn lowers production efficiency. Conversely, as the threshing speed increases, the brushing and rubbing actions become more pronounced, improving both the cleaning rate and operational efficiency *(Han, 2017)*. However, excessively high speeds can lead to increased damage to the grains and straw, as well as higher power consumption *(Qu et al., 2018)*. Additionally, the timing of crop harvesting and the moisture content of the crops impose certain requirements on the cylinder speed. When threshing wetter crops, it is advisable to increase the cylinder speed to enhance separation rates. Conversely, for drier crops, the cylinder speed should be reduced to ensure the integrity of the grains and minimize breakage *(Jiang, 2015)*.

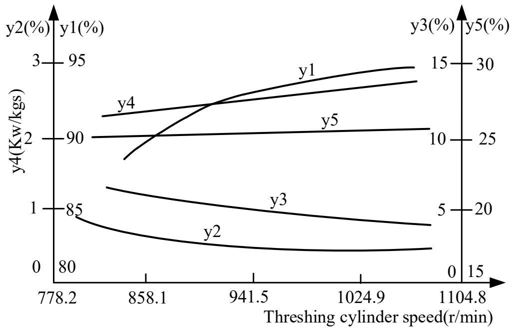


Fig. 3 - Relationship curves between threshing cylinder speed and various performance indicators
y1 – separation rate; y2– unthreshed rate; y3 – entrainment loss rate;
y4 – unit power consumption; y5 – trash content

In summary, it is essential to determine the most suitable threshing speed range for the threshing cylinder based on its real-time operational conditions. Additionally, the threshing speed should be adjustable according to factors such as the crop feed rate and growth status. By maintaining the cylinder speed within this optimal range, the harvesting quality and efficiency of the combine harvester can be significantly improved. This adaptive control approach not only enhances overall performance but also ensures better handling of varying crop conditions during operation.

Performance requirements of permanent magnet driven motors for threshing operations

For the electrically driven threshing cylinder, the performance requirements of the drive motor are shown in Figure 4. The performance requirements that the motor must meet include: (1) The motor should have a robust overload capability, able to output 2-3 times the peak torque for short durations. This is essential for overcoming the large inertia loads of the threshing cylinder, enabling acceleration and stable operation while maintaining the required rubbing intensity to reduce crop impurity and loss rates. (2) A higher torque-to-inertia ratio enhances the acceleration performance of the threshing cylinder. This ratio is critical for achieving quick response times during operation. (3) The motor should offer a broad range of speed adjustments to meet the threshing speed requirements for various crop types and operating conditions, thereby further reducing production costs. (4) The driving motor should operate efficiently across a wide range to accommodate random fluctuations in operational load, which helps increase the duration of effective operation for the electric drive system.

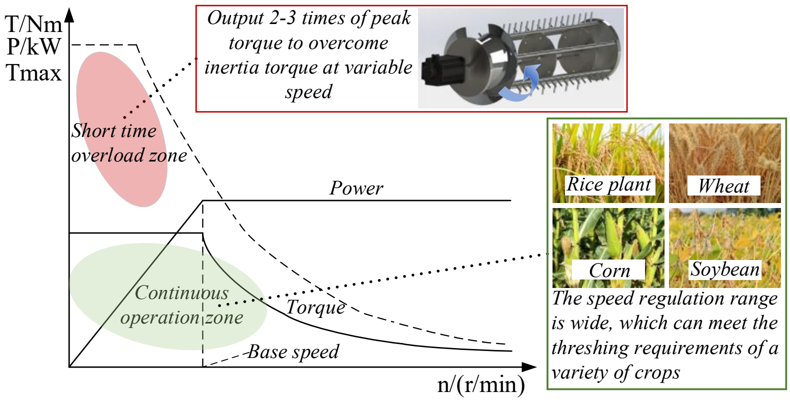


Fig. 4 - Performance output requirements of driving motor

By meeting these performance criteria, the electric motor can significantly enhance the operational efficiency and effectiveness of the electrically driven threshing cylinder in combine harvesters.

Mathematical model of electrically driven threshing cylinder

Mathematical model of PMSM

According to Figure 2, the mathematical model of PMSM in d-q axis system is established. The stator voltage equation for PMSM in the d-q coordinate system is as follows (eq. 1) *(Fu et al., 2024; Zhang et al., 2025)*:

$$\begin{cases} u_d = R_s i_d + L_d \frac{di_d}{dt} - \omega_e L_q i_q \\ u_q = R_s i_q + L_q \frac{di_q}{dt} + \omega_e (L_d i_d + \psi_f) \end{cases} \tag{1}$$

The electromagnetic torque equation is as follows (eq. 2) *(Fu et al., 2024; Zhang et al., 2025)*:

$$T_e = \frac{3}{2} P_n [\psi_f + (L_d - L_q) i_d] i_q \tag{2}$$

The mechanical motion equation is as follows (eq. 3) *(Fu et al., 2024; Zhang et al., 2025)*:

$$T_e = J \frac{d\omega_m}{dt} + B\omega_m + T_L \tag{3}$$

where: u_d, u_q = the stator voltage direct and cross-axis components [V]; i_d, i_q = the stator current direct and cross-axis components [A]; R = the stator winding resistance [Ω]; P = the number of pole pairs; ω_m = the mechanical angular velocity [rad/s]; L_d, L_q = the stator d, q axis inductance [H]; L_s = the stator inductance [H]; ψ_f = the magnetic chain of permanent magnets [Wb]; J = the rotational moment of inertia [kg·m^2]; T_e = the electromagnetic torque [N·m]; T_L = the load torque [N·m]; B = the viscous friction factor.

Mathematical model of threshing cylinder

When the combine harvester starts operation, the active torque required to drive the threshing cylinder rotor shaft mainly includes the torque M_k required to overcome its own idle running, the torque M_g to overcome its own moment of inertia to generate angular acceleration, and the torque M_w (working torque) that pushes the crop to overcome the resistance movement and make it gain acceleration *(Zhang et al., 2001)*. The force analysis is shown in Figure 5.

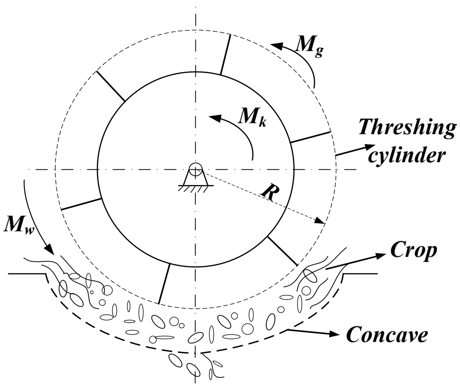


Fig. 5 - Force analysis diagram of threshing cylinder

M_k –idle running load torque of threshing cylinder, [N·m]; M_g – the inertia moment of threshing cylinder, [N·m];
M_w –the working torque of threshing cylinder, [N·m]

(1) Idle running load torque

The threshing cylinder has to overcome mechanical friction resistance and wind resistance during idle running, and according to Golikovich's cylinder theory *(Zhao,1984)*, the total idle running resistance torque is:

$$M_k = A + B\omega^2 \tag{4}$$

where: ω = the angular velocity of the cylinder[rad/s]; A = the torque generated by mechanical friction, taken as 0.004; B = the cylinder wind force torque coefficient, taken as 0.64 × 10-6.

(2) Moment of inertia of threshing cylinder

$$M_g = J \frac{d\omega}{dt} \tag{5}$$

where: J = moment of inertia of the threshing cylinder [kg·m^2].

(3) Working torque of threshing cylinder

The working torque of the threshing cylinder includes the torque required to accelerate the movement of the crops and the torque needed to overcome the friction between the crops and the cover plate, concave plate, and spike teeth during the movement. The research shows that *(Zhang et al., 2000)*, the tangential speed of crops at the threshing cylinder outlet is 1/2~1/5 of the linear speed of the cylinder rotation. The crop outlet speed ratio λ is introduced, which is defined as the ratio of the velocity of the stalks being ejected at the discharge opening to the rotational linear velocity, helps to analyze the process. The crop movement speed at the feeding inlet of the threshing cylinder is significantly lower than the average movement speed within the cylinder, so it can be neglected. Assuming that the crop flows continuously through the threshing cylinder without crop breakage or clogging, the amount of crop fed in at time Δt is essentially equal to the amount of crop flowing out. Neglecting the axial speed of motion of the stem draft output, the amount of change in the kinetic energy of crop in the threshing cylinder in time Δt is given as follows *(Zhang et al.,1999)*:

$$E_v = \frac{1}{2} q \Delta t \frac{\lambda+\Delta}{1+\Delta} R^2 \omega^2 \tag{6}$$

where: q = feed rate [kg/s]; λ = crop outlet speed ratio; Δ = grass-to-grain ratio; R = the effective radius of the spike teeth on cylinder [m].

Therefore, the torque required by the threshing cylinder to propel the crop in accelerated motion is:

$$M_v = \frac{1}{2} q \frac{\lambda+\Delta}{1+\Delta} R^2 \omega \tag{7}$$

Research has shown that the torque required to accelerate the crop always accounts for a certain proportion of the total working torque. Therefore, Golikovich introduced the friction coefficient f to represent the proportional relationship between the working resistance and the total working torque *(Kanavetsky,1983)*, as shown in equation (8):

$$M_w = \frac{M_v}{1-f} \tag{8}$$

where: f = the friction coefficient of threshing cylinder.

Combining equations (4), (5) and (8), the torque equation for the threshing cylinder can be obtained as shown in equation (9):

$$M_T = M_g + M_k + M_w = J\frac{d\omega}{dt} + (A + B\omega^2) + \frac{qR^2\omega(\Delta+\lambda)}{2(1-f)(\Delta+1)} \tag{9}$$

Combining equations (2), (3) and (9), the mathematical model of motor-driven threshing cylinder can be obtained as shown in equation (10):

$$\frac{3}{2} P_n i_q \left[\psi_f + \left(L_d - L_q\right) i_d\right] = J\frac{d\omega}{dt} + (A + B\omega^2) + \frac{qR^2\omega(\Delta+\lambda)}{2(1-f)(\Delta+1)} \tag{10}$$

Control strategy for electrically driven threshing cylinder

When the rotating shaft of the PMSM is coupled to the threshing cylinder, the moment of inertia of the motor consists of the moment of inertia of the motor rotor and the moment of inertia of the threshing cylinder. The fluctuation of crop feed rate makes the load of threshing cylinder produce random disturbances, which causes the change of moment of inertia, and the required torque of threshing cylinder is positively correlated with its own moment of inertia, when the moment of inertia of the threshing cylinder increases, the required torque of the cylinder increases accordingly, and vice versa *(Liu, 2013; Li, 2023)*. Therefore, the online identification strategy of moment of inertia is introduced to enhance the control accuracy and stability of the system.

The control system block diagram of the electrically driven threshing cylinder is shown in Figure 6, including speed control loop, current control loop, coordinate transformation and moment of inertia identification module.

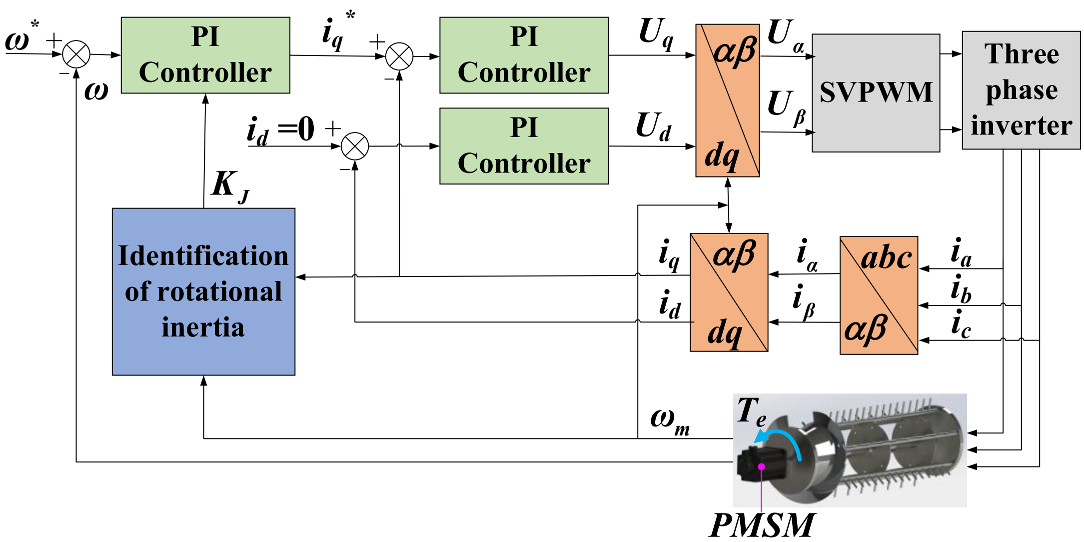


Fig. 6 - Control block diagram of electrically driven threshing cylinder

Identification of moment of inertia of electrically driven threshing cylinder system

Identification of moment of inertia of motor rotor

When the motor is undergoing acceleration or deceleration, the direction of the electromagnetic torque and the load is not fixed. Ignoring the motor's damping coefficient, let $\alpha=d\omega_m/dt$, then the mechanical motion equation of the motor can be expressed as:

(a) Acceleration process:

$$J\alpha_1 = T_e - T_L = K_T I_q - T_L \tag{11}$$

(b) Deceleration process:

$$J\alpha_2 = T_e + T_L = -(K_T I_q + T_L) \tag{12}$$

where: K_T = torque coefficient; I_q = q-axis current of motor(A).

In order to accurately identify the moment of inertia, it is necessary to take into account the effect of the load torque, and the load is equated as follows:

(a) Equation of motion for acceleration process:

$$K_T I_q = J\alpha_1 + T_L \triangleq J_{Acc}\alpha_1 \tag{13}$$

Then the equivalent moment of inertia during acceleration process is shown in equation (14):

$$J_{Acc} \triangleq J + \frac{T_L}{\alpha_1} \tag{14}$$

(b) Equation of motion for deceleration process:

$$K_T I_q = J\alpha_2 - T_L \triangleq J_{Dec}\alpha_2 \tag{15}$$

Then the equivalent moment of inertia during deceleration process is shown in equation (16):

$$J_{Dec} \triangleq J - \frac{T_L}{\alpha_2} \tag{16}$$

This leads to the estimation equation for the moment of inertia of the motor rotor as shown in equation (17):

$$\hat{J} = \frac{J_{Acc}+J_{Dec}}{2} = J + \frac{1}{2}\left(\frac{T_L}{\alpha_1} - \frac{T_L}{\alpha_2}\right) \approx J \tag{17}$$

The identification process of motor rotor moment of inertia is shown in Figure 7:

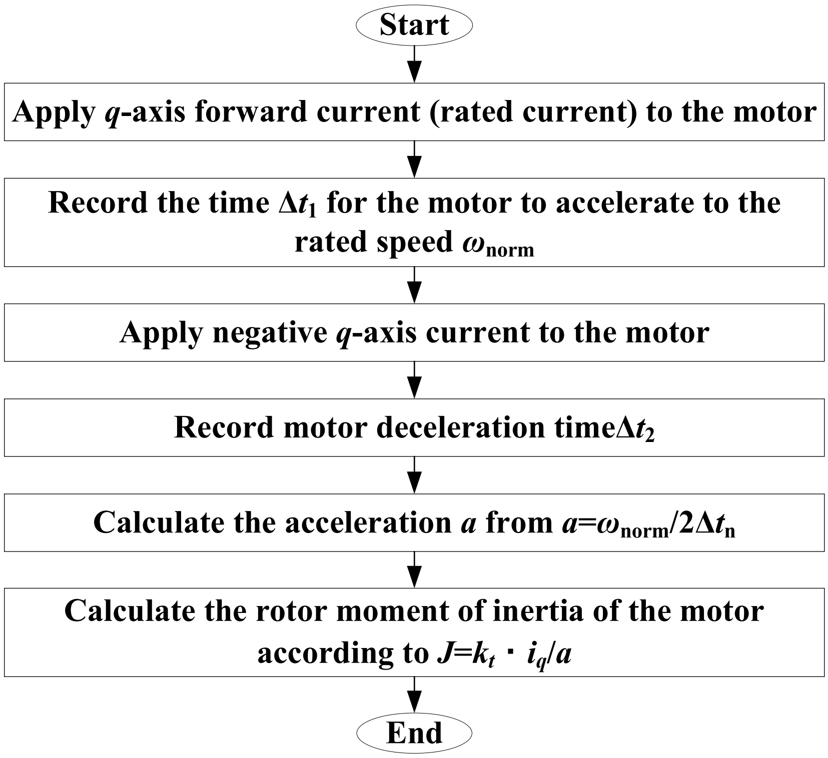


Fig. 7 - Flowchart for identification of motor rotor moment of inertia

Identification of moment of inertia of threshing cylinder

The moment of inertia identification of the threshing cylinder is implemented using the recursive least squares method with a forgetting factor *(He, 2020)*. The dynamic equations of the servo system are shown in equations (18) - (20):

$$T_m = JA_m \tag{18}$$

$$T_m = T_e - T_d - T_f \tag{19}$$

$$A_m = \frac{d\omega_m}{dt} \tag{20}$$

The performance index function equation (21) is constructed using T_m and A_m as input observations and the threshing cylinder moment of inertia J as the estimated quantity:

$$y = [T_m - JA_m]^2 \tag{21}$$

The criterion function is shown in equation (22):

$$V_N(k) = \frac{1}{N}\sum_{i=1}^{N}\left[T_m(k) - \hat{J}(k)A_m\right]^2 \tag{22}$$

The moment of inertia identification algorithm of the threshing cylinder is constructed based on the recursive least squares theory *(Song, 2019)*, as shown in equation (23):

$$\begin{cases} \hat{J}(k) = \hat{J}(k-1) + L(k)\left[T_m(k) - \hat{J}(k-1)A_m(k)\right] \\ L(k) = \frac{P(k-1)A_m(k)}{1 + A_m^2(k)P(k-1)} \\ P(k) = [1 - L(k)A_m(k)]P(k-1) \end{cases} \tag{23}$$

Based on equation (23), the least squares identification algorithm with the addition of a forgetting factor λ is shown in equation (24):

$$\begin{cases} \hat{J}(k) = \hat{J}(k-1) + L(k)\left[T_m(k) - \hat{J}(k-1)A_m(k)\right] \\ L(k) = \frac{P(k-1)A_m(k)}{\lambda + A_m^2(k)P(k-1)} \\ P(k) = [1 - L(k)A_m(k)]P(k-1)\lambda^{-1} \end{cases} \tag{24}$$

where: T_m = the torque for accelerating or decelerating the rotational inertia of the threshing cylinder [N·m]; T_e = the motor output torque [N·m]; T_d = the disturbance torque of threshing cylinder [N·m]; T_f = friction torque [N·m]; A_m = the motor acceleration [r/s^2]; J = the moment of inertia of threshing cylinder [kg·m^2].

The moment of inertia identification process for the threshing cylinder is illustrated in Figure 8.

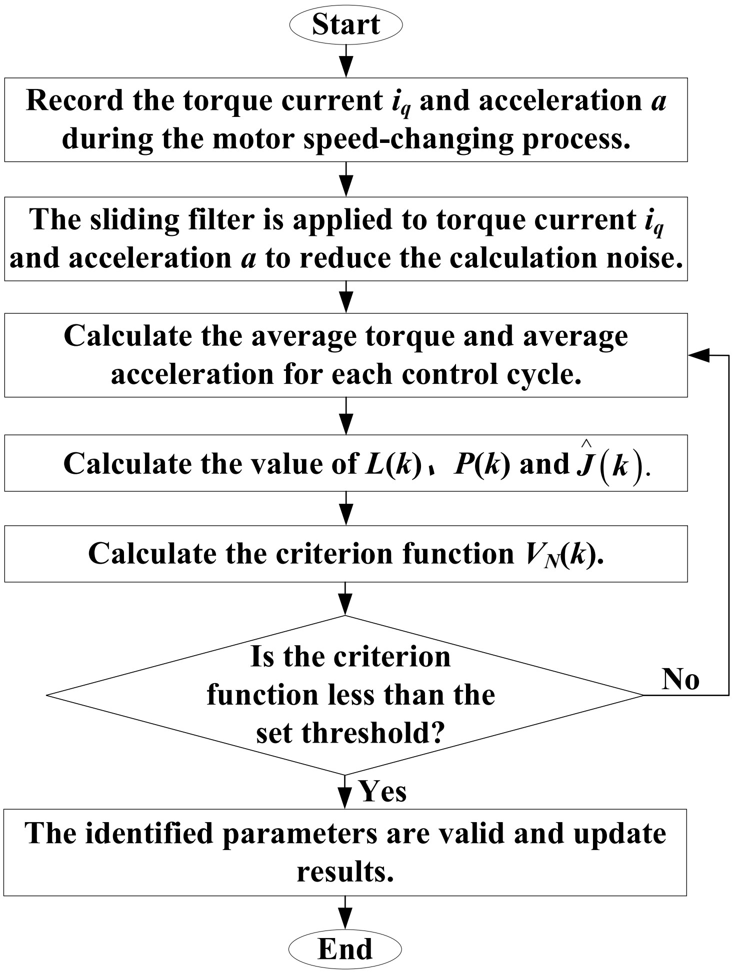


Fig. 8 - Flowchart for the identification of the moment of inertia of the threshing cylinder

Self-tuning of speed loop parameters in electrically driven threshing cylinder control system

The purpose of moment of inertia identification is to make real-time adjustments to the speed loop parameters based on changes in inertia, optimizing the conversion structure of the speed loop, and improving the responsiveness of speed conversion. The speed loop control block diagram is shown in Figure 9.

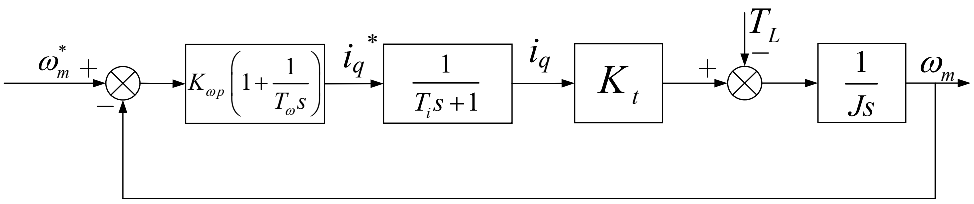


Fig. 9 - Speed loop control block diagram

In Figure 9, $\omega^*{}_m$ is the given speed; $K_{\omega p}$ is the proportional coefficient of the speed loop; T_ω is the integral time constant of the speed loop; T_i is the equivalent time constant of the current loop.

Based on the control block diagram in Figure 9, the open-loop transfer function of the system is derived as Equation (25):

$$G(s) = K_{\omega p}\left(1+\frac{1}{T_\omega s}\right)\left(\frac{1}{T_i s+1}\right)\frac{K_t}{Js} \tag{25}$$

The speed loop is a typical type II system *(Zhu et al., 2023)*, and the relationships of the system parameters are shown in equations (26) and (27):

$$T_\omega = hT_i \tag{26}$$

$$K = \frac{h+1}{2h}\cdot\frac{J}{T_i K_t} \tag{27}$$

where: K = the open-loop gain; h = the intermediate frequency bandwidth, according to the performance index of the system, take h = 5.

The calculation process of the PI parameters in the speed loop is shown in equations (28) and (29):

$$K_{\omega p} = \frac{J(h+1)}{2hT_iK_t} = \frac{J(h+1)R_s}{3hLp_n\psi_f} \tag{28}$$

$$K_{\omega i} = \frac{J(h+1)}{2h^2T_i^2K_t} = \frac{J(h+1)R_s^2}{3h^2L^2p_n\psi_f} \tag{29}$$

where: $K_{\omega i}$ = Integral coefficient of speed loop.

When the motor drives the threshing cylinder, the total moment of inertia J is shown in equation (30):

$$J = J_0 + J_L \tag{30}$$

where: J_0 = the moment of inertia of motor rotor [kg·m^2]; J_L = the moment of inertia of threshing cylinder [kg·m^2].

The ratio of the moment of inertia of the threshing cylinder to the motor rotor is defined as the load inertia ratio K_J, the expression of which is shown in equation (31):

$$K_J = \frac{J_L}{J_0} \tag{31}$$

According to equations (28), (29) and (31), the PI parameters can be transformed as shown in equations (32) and (33):

$$K_{\omega p} = \frac{(K_J+1)J_0(h+1)}{2hT_iK_t} = \frac{(K_J+1)J_0}{3hLp} \tag{32}$$

$$K_{\omega i} = \frac{(K_J+1)J_0(h+1)}{2h^2T_i^2K_t} = \frac{(K_J+1)J_0(h+1)R_s^2}{3h^2L^2p_n\psi_f} \tag{33}$$

By substituting the identified inertia ratio K_J into equations (32) and (33), the PI parameters of the speed loop are continuously revised, achieving the self-tuning capability of the PI parameters for the electrically driven threshing cylinder control system.

Experimental equipment

In this paper, based on the $dSPACE$1007 platform, a control system equivalent simulation test bench for an electrically driven threshing cylinder is constructed, as shown in Figure 10.

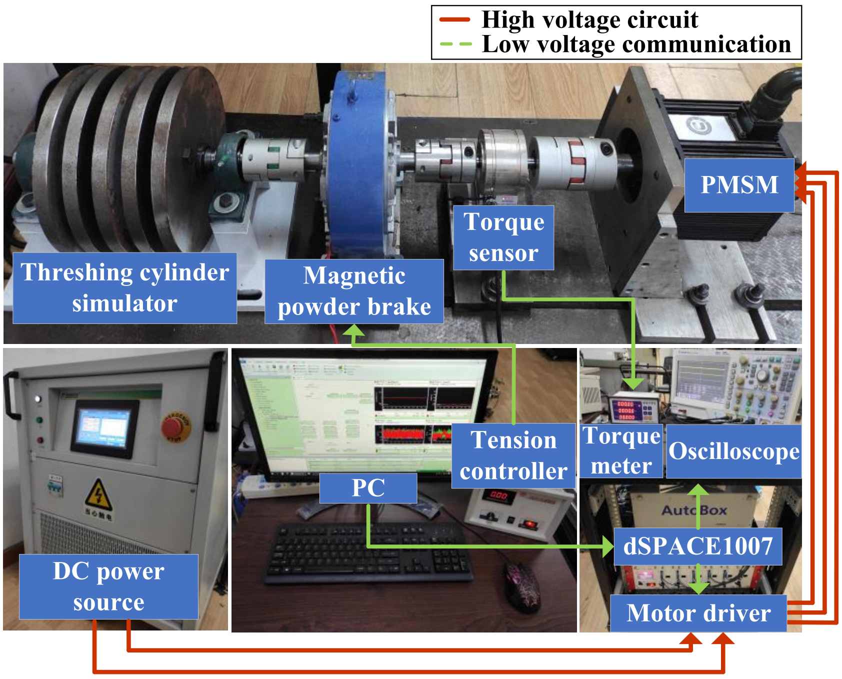


Fig. 10 - Equivalent simulation test bench for electrically driven threshing cylinder

Among them, the threshing cylinder simulator is used to simulate the threshing cylinder of the combine harvester, and the magnetic powder brake is used to simulate the load disturbance of the threshing cylinder caused by the crop feed rate of the combine harvester during the field operation. According to the field operation environment and operational requirements of combine harvester, three experimental conditions are designed to compare control effects and verify the superiority of the proposed algorithm.

RESULTS ANALYSIS AND DISCUSSION

Condition 1:

As shown in Figure 11, the threshing cylinder is considered a large inertia component due to its size, shape, structure and other factors, and the required torque is positively correlated with its own moment of inertia. Therefore, at the beginning of the harvesting operation the resistance of the threshing cylinder is larger, resulting in a higher starting current, which can lead to overshoot in the speed and a slower response time of the control system. Thus, it is necessary to enhance the responsiveness and robustness of the control system.

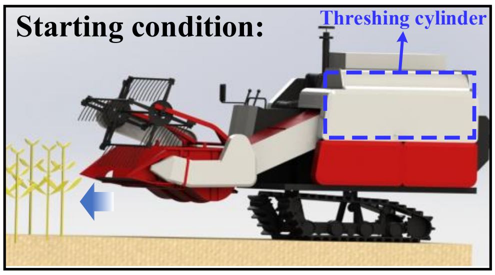
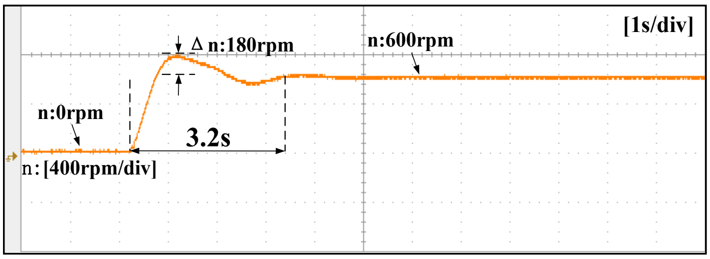


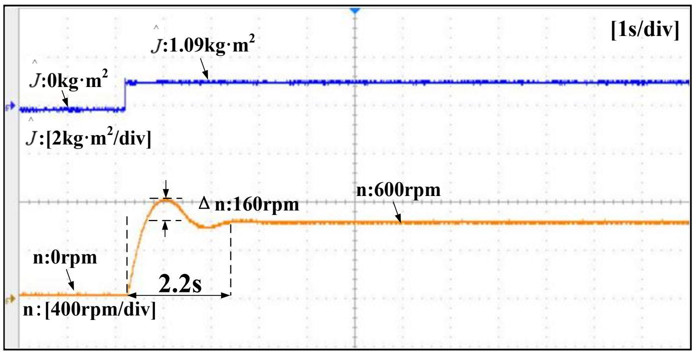
Fig. 11 - Starting condition of the threshing cylinder in the combine harvester

As shown in Figure 12, the start-up performance of the electrically driven threshing cylinder is compared with that of the traditional PI control method and the moment of inertia identification control method based on this paper, in which the sum of the moment of inertia of the motor and the threshing cylinder simulator is J=1.09 kg·m^2, and the target speed is 600 r/min.

It can be found that, in the start-up process of the electrically driven threshing cylinder, when the speed reaches the target speed, the value of the moment of inertia identified by the system in Figure 12(b) reaches the actual moment of inertia value, demonstrating high accuracy and good steady-state performance. After adaptive adjustment of the parameters of the speed loop, the start-up speed overshoot is 160 r/min, and the convergence time of speed oscillation is 2.2 s, which is 11.1% and 31.3% shorter than that of the speed overshoot amount of 180 r/min and the convergence time of 3.2 s in Figure 12(a), respectively.



(a) Traditional PI control method



(b) Control method based on moment of inertia identification

Fig. 12 - Comparison of start-up performance of electric drive system

The electrically driven threshing cylinder control method based on moment of inertia identification can effectively reduce the speed inertial shock and oscillation of the system during start-up process, which can effectively mitigate wear and tear on combine harvester.

Condition 2:

As shown in Figure 13, during harvesting operations, the combine harvester may encounter adverse weather conditions such as strong winds and heavy rain, which can cause crops to lodge. Additionally, the natural growth density of the crops may fluctuate, and the forward speed of the combine harvester can also vary due to factors like ground conditions and driver operating habits. These various external factors can lead to fluctuations in the feed rate of the combine harvester. Therefore, it is essential to enhance the stability of the threshing cylinder's speed to ensure a smooth operation process and improve the quality of threshing.

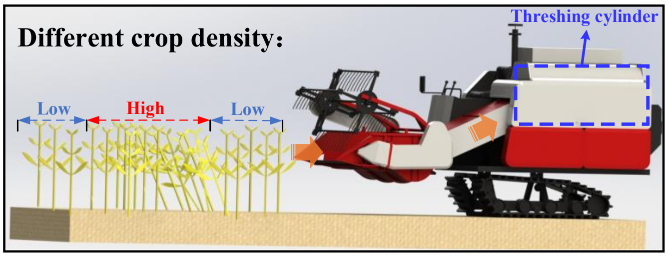
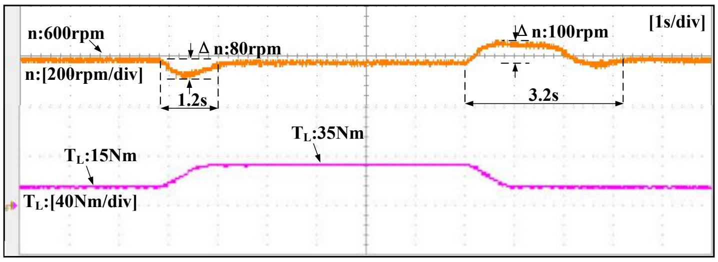
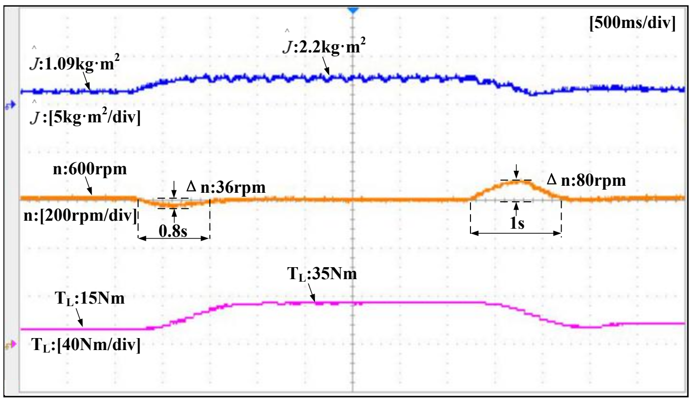


Fig. 13 - Fluctuation condition of feed rate to the threshing cylinder in combine harvester

The experimental results of the feed rate fluctuation tests for the two control strategies are given in Figure 14. In the feed rate fluctuation test, the electrically driven threshing cylinder runs at the rated speed of 600 r/min, the feed rate load changes abruptly from 15 Nm to 35 Nm, and then the load changes abruptly to 15 Nm after the speed stabilizes. As can be seen in Figure 14(b), when the feed rate load changes abruptly, the system can identify the actual value of moment of inertia and track the actual value, which has a better dynamic performance. The value of moment of inertia is fed back to the controller for real-time updating, and after self-tuning of the PI values of the speed loop, the maximum drop in speed and the maximum overshoot are 36 r/min and 80 r/min respectively, which are 55% and 20% less than those of 80 r/min and 100 r/min in Figure 14(a), and the dynamic adjustment time in Figure 14(b) during the process of abrupt increase and decrease of feed load are shortened by 33.3% and 68.8% respectively, indicating an enhanced disturbance rejection capability of the control system for the electrically driven threshing cylinder.



(a) Traditional PI control method



(b) Control method based on moment of inertia identification

Fig. 14 - Comparison of load sudden change experiment

The moment of inertia identification-based speed control method significantly enhances the load disturbance rejection performance of the electrically driven threshing cylinder, improving the system's robustness. This results in a smoother and more seamless threshing process for the crops, thereby further increasing the processing efficiency and the quality of threshing.

Condition 3:

As shown in Figure 15, during harvesting operations, the moisture content of crops in the same field can fluctuate due to external environmental influences. Therefore, the speed of the threshing cylinder needs to be adjusted in real-time. When the moisture content of the crops increases, the speed of the threshing cylinder should be appropriately increased; conversely, when the moisture content decreases, the speed should be appropriately reduced.

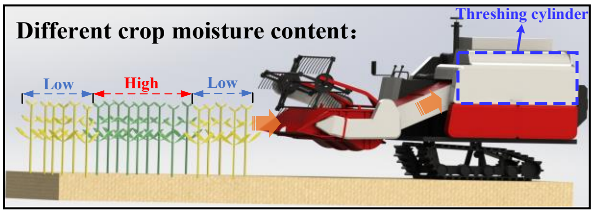
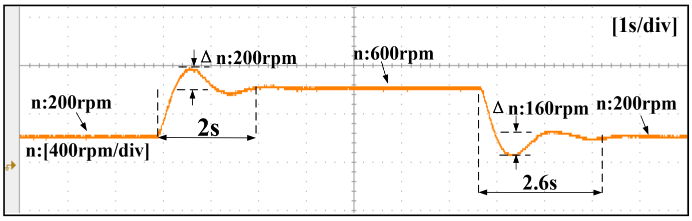


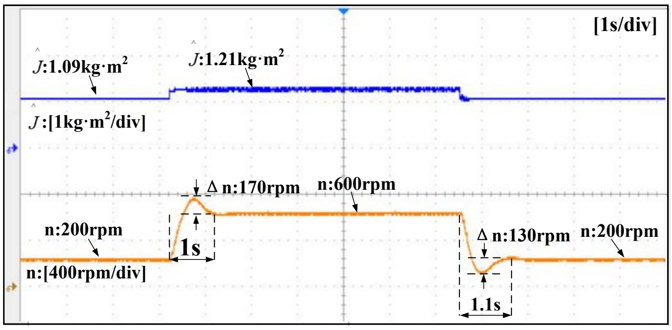
Fig. 15 - Fluctuating condition of crop moisture content

In order to verify the dynamic performance of the strategy proposed in this paper in the speed regulation process, the results of the comparison of the variable speed performance of the two control methods are given in Figure 16. In the experiment, the speed of the electrically driven threshing cylinder is increased from 200 r/min to 600 r/min, and then decreased back to 200 r/min after the speed stabilizes.

It can be seen that the system in Figure 16(b) can identify the actual value of moment of inertia during the speed change process and can track the actual value faster; the speed adjustment time during acceleration is 1 s, and the overshoot of the speed of the threshing cylinder is 170 r/min, which is 50% shorter and 15% smaller than the 2 s convergence time and 200 r/min speed overshoot of the traditional PI control strategy in Figure 16(a); the speed convergence time and the overshoot of the speed in Figure 16(b) are 57.7% shorter and 18.8% smaller, respectively, compared to Figure 16(a) during the speed deceleration process.



(a) Traditional PI control method



(b) Control method based on moment of inertia identification

Fig. 16 - Comparison of speed variation experiment

The speed control method based on moment of inertia identification effectively improves the speed regulation responsiveness and stability of the threshing cylinder in the combine harvester. For different types of crops or varying growth stages of the same crop, the adaptability of the threshing cylinder's speed is enhanced, which helps reduce harvesting losses and improve the crop's cleaning rate, further enhancing the quality of the harvest.

CONCLUSIONS

This paper studies the electrically driven speed control system for the threshing cylinder of a combine harvester during field operation, based on the working condition requirements. A dynamic model of the threshing cylinder is established, and a mathematical model of the electrically driven for the threshing cylinder is constructed by combining the motor motion model and characteristics of a permanent magnet synchronous motor. A load variation control strategy based on rotational inertia identification is proposed to meet the demands of multiple working points at different speeds under various operating conditions.

The experimental results indicate that during the startup process of the threshing cylinder, the control method based on rotational inertia identification reduces the overshoot and convergence time of the rotational speed by 11.1% and 31.3% respectively, compared to the traditional PI control method, effectively lowering the system's inertial impact and oscillation.

When there are fluctuations in the crop feed rate of the combine harvester, the control method based on rotational inertia identification decreases the speed drop and overshoot by 55% and 20%, respectively, while the dynamic adjustment time is shortened by 33.3% and 68.8%, enhancing the system's disturbance resistance. Additionally, under varying crop moisture content conditions, the convergence times of acceleration and deceleration during speed adjustments are reduced by 50% and 57.7%, while the overshoot of the rotational speed decreases by 15% and 18.8%. These improvements enhance the responsiveness and stability of the threshing cylinder speed regulation. This system can effectively adapt to the dynamic power demands of modern agriculture, providing a theoretical basis for the development of intelligent agricultural equipment.

Based on this study, follow-up research will focus on conducting collaborative research between the threshing cylinder and other working modules to explore the optimal speed matching ratios between the modules, thereby improving the overall operational efficiency of combine harvester. In addition, the research will also aim to further reduce energy consumption through advanced energy management strategies, achieving greater economic efficiency and environmental friendliness. Furthermore, the research will involve manufacturing a prototype for field experiments to further verify the superiority of the electric drive operating system.

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