

Design and Experiment of a Coordinated Longitudinal Height and Lateral Tilt Terrain-Following System for Combine Harvester Header Based on Smith-SMC

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ABSTRACT. *Accurate header terrain-following is essential for reducing harvest losses, yet many existing systems regulate only longitudinal height and adapt poorly to uneven terrain. This study developed a coordinated system for controlling the longitudinal height and lateral tilt of a combine-harvester header. A spherical contact-based terrain-sensing device was designed, and its return-spring parameters were optimized through ADAMS multibody dynamics simulations. To address pure time delay, model uncertainty, and external disturbances in the electro-hydraulic servo system, a Smith predictor-based sliding mode controller was formulated using a composite sliding surface and a continuous tanh reaching term. For a longitudinal height step from 50 to 100 mm, Smith-SMC reduced the settling time from 3.275 to 0.586 s relative to conventional PID control, an 82.1% reduction, while decreasing overshoot from 6.224% to 0.912% and maintaining a near-zero steady-state error. For lateral sinusoidal tracking, its RMSE relative to the delayed reference was 96.4% lower than that of Smith-PID. Simulations with model mismatch and equivalent input-pulse disturbances indicated stable operation within the tested uncertainty range, although performance remained limited by physical time delay and large delay mismatch. In field tests, 71.07% of header-height measurements and 75.06% of tilt-angle measurements remained within their permissible ranges. Across operating speeds of 4–10 km/h, Smith-SMC achieved an average stubble-height pass rate of 86.11%, 42.50 percentage points higher than manual control. The location-level mean absolute deviation remained below 11 mm in all groups, and the average soil-scooping rate was 3.06%. At 10 km/h, the pass rate remained 80.00%, with a mean absolute deviation of 10.90 mm. These results indicate that Smith-SMC can support coordinated and robust header terrain-following under the tested conditions.*

Keywords. *Combine harvester, Terrain-following, Coordinated control, Sliding mode control, Smith predictor, Field test.*

1 Introduction

With the continuous advancement of modern agricultural mechanization (Martin et al., 2022; Vougioukas, 2019; Wang et al., 2024), the trend of combine harvesters evolving toward larger scales and wider operating widths has become increasingly prominent, significantly enhancing agricultural production efficiency. As the most critical executing component of a combine harvester, the ground clearance of the header directly dictates the quality of crop harvesting. An excessively high header position leads to severe missed cuts and yield losses, whereas an excessively low position renders the header highly susceptible to soil scooping, inflicting irreversible transient damage upon large-inertia rigid mechanical structures (De Keyser et al., 2022; Huang, Jiang, Li, Guo, & Zhang, 2026; Wang et al., 2024). Consequently, achieving precise and robust header terrain-following in complex, unstructured agricultural landscapes has emerged as a core research challenge, breaking through the intelligent bottlenecks of modern large-scale combine harvesters.

The primary prerequisite for a header terrain-following system is the precise perception of topographical undulations. At present, academic discourse regarding header height detection technologies primarily revolves around two paradigms: non-contact and contact-type sensing. Some researchers have dedicated efforts to the

application of non-contact sensors, such as machine vision, LiDAR, and ultrasonics (Kise, Zhang, & Más, 2005; Oksanen & Visala, 2009; Plessen, 2026; Tan et al., 2023). However, a prevailing consensus indicates that in harsh field environments, such methodologies are fatally susceptible to interference from crop leaf obstruction, dust, and mud splashing, which substantially degrades their practical measurement reliability (Bechar & Vigneault, 2016). In contrast, profiling mechanisms based on physical contact remain the mainstream choice in the industry (Ni et al., 2021; Yang et al., 2022), owing to their high resilience to severe operational conditions. Nevertheless, contact-type detection is not without its challenges: under high-speed operations or bumpy excitations, the sensors are prone to high-frequency chattering, leading to feedback signal distortion. Addressing this limitation, recent studies have leaned towards utilizing multi-body dynamics software (Jin, Fuqiang, Zhenjie, Youliang, & Chengqian, 2024; Prasertkan, Usaborisut, Jindawong, Suwannakij, & Iamrursiri, 2024; Zhao, Xu, Zou, Shi, & Luo, 2026), such as ADAMS, to conduct fundamental kinematic optimization of mechanical components to suppress vibration. This offers robust theoretical support for designing novel contact-type contour-following devices equipped with active vibration-damping capabilities.

At the control strategy level, early research was largely confined to single-degree-of-freedom longitudinal height control, typically employing traditional PID or modified adaptive algorithms (Zhao et al., 2026). With the continuous expansion of header widths, a consensus has gradually formed among scholars: when confronted with complex sloping terrains, unitary height adjustment fails to prevent single-sided soil scooping. Thus, an evolution towards the coordinated control of both "longitudinal height and lateral tilt" is imperative (Peng, Ma, Wang, & Yuan, 2023; Xie & Alleyne, 2012). However, the coordinated control of large-inertia headers inherently relies on multi-cylinder electro-hydraulic servo systems. These systems are characterized not only by strong nonlinearities but also by non-negligible dead-time properties (Alleyne & Liu, 2000; Boaventura et al., 2012). Existing coordinated control studies generally face bottlenecks of sluggish response or insufficient disturbance rejection when managing such strongly coupled, large-delay systems.

Within the domain of nonlinear control, sliding mode control is highly esteemed for its robustness to bounded matched disturbances against parameter perturbations and external disturbances (Edwards & Spurgeon, 1998; Slotine & Li, 1991; Utkin, 2003). In practical applications, however, the presence of pure time delay severely undermines the reaching law of SMC, making it highly susceptible to inducing severe "high-frequency chattering," and potentially leading to the loss of control in electro-hydraulic systems (Briat, 2014; Gu, Chen, & Kharitonov, 2003; Karpenko & Sepehri, 2009). To overcome the time-delay dilemma, the Smith predictor is widely recognized as a classic methodology capable of effectively decoupling the dead-time term from the closed-loop characteristic equation (Astrom, Hang, & Lim, 2002; Normey-Rico & Camacho, 2007). In recent years, composite algorithms integrating the Smith predictor with SMC have demonstrated excellent control performance at the theoretical level (Sanz, García, & Albertos, 2018; Sindhuja, Vijayan, & Panda, 2023). It must be critically pointed out, however, that the majority of current research on advanced EHS algorithms remains confined to theoretical MATLAB simulations or idealized laboratory test benches. A massive research void exists in the current literature regarding the deep deployment of such composite algorithms in heavy-duty agricultural machinery subjected to extreme excitation, particularly studies substantiated by rigorous and authentic field experimental data, such as stubble qualification rates and soil scooping rates.

To bridge the gap between theoretical research and practical engineering application, this study developed a coordinated terrain-Following system for header longitudinal height and lateral tilt, utilizing the 4YZL-12F large-scale combine harvester as a platform. The core contributions of this study are: (1) A spherical terrain-following detection device was designed, and its return tension spring parameters were optimized via ADAMS simulation, effectively suppressing vibration induced by ground impact at the fundamental mechanical level. (2) A composite control strategy fusing the Smith predictor and SMC was proposed to compensate for the effect of the nominal time

delay on the closed-loop dynamics and unleash nonlinear robust stabilization capabilities. (3) Comprehensive comparative field experiments were conducted. This paper aims to provide a novel perspective for resolving the large-delay dilemmas in agricultural electro-hydraulic systems and to validate the engineering value of the proposed system in enhancing harvesting precision and safeguarding machinery security with conclusive field data.

2 System Composition and Working Principle

The coordinated header terrain-following system primarily comprises a spherical terrain-following detection mechanism, header lifting hydraulic cylinders and corresponding valve blocks, header leveling hydraulic cylinders and valve blocks, a PLC unit, a HMI display, a manual control module, and the header itself. The schematic diagram illustrating the composition of the terrain-following system is presented in Figure 1.

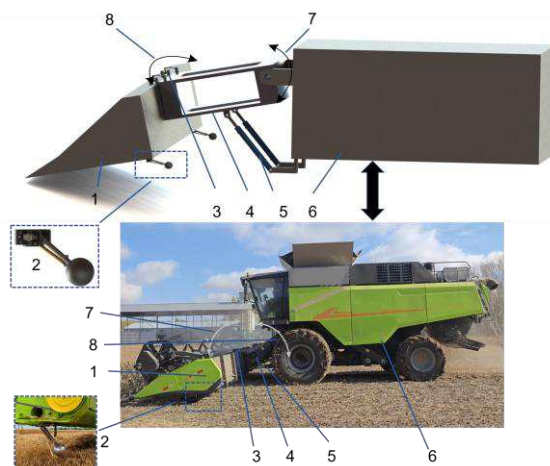


Fig. 1 Schematic diagram of the coordinated terrain-following system.(1. Header; 2. Spherical terrain-following detection mechanism; 3. Header leveling hydraulic cylinder; 4. Feeder house; 5. Header lifting hydraulic cylinder; 6. Combine chassis; 7. Direction of header lateral tilt; 8. Direction of header longitudinal lifting)

The spherical terrain-following detection mechanisms are symmetrically mounted on the left and right sides of the header. During the harvesting process, they maintain continuous contact with the ground and convert the terrain undulation signals into sensor output voltage signals ranging from 0.5 to 4.5 V. These signals are subsequently processed by the PLC control unit to determine the ground clearance on both sides of the header, which serves as the basis for evaluating the topographical conditions beneath the header. Based on the relatively parallel or tilted state between the header and the underlying terrain, the header height and tilt angle are adjusted either independently or jointly. Longitudinal adaptation is achieved through the lifting hydraulic cylinder, which dynamically pivots the integrated header and feeder house assembly relative to the chassis, offering an adjustment span of 50 to 1250 mm. To counter lateral slopes, the leveling hydraulic cylinder actuates the header, inducing a controlled lateral deflection relative to the feeder house.

Given their large cutting widths and high feed capacities, large-scale combine harvesters typically operate in plain areas characterized by gentle topography with slopes predominantly under 5° . To meet these specific operational conditions, the system's lateral deflection is constrained to $\pm 7.5^\circ$ (referenced from a 0° horizontal baseline on flat terrain). This capacity is fully adequate for the expected field conditions. Figure 2 delineates the working principle of the coordinated system, providing a schematic of the hydraulic actuation alongside the closed-loop feedback control framework for header elevation.

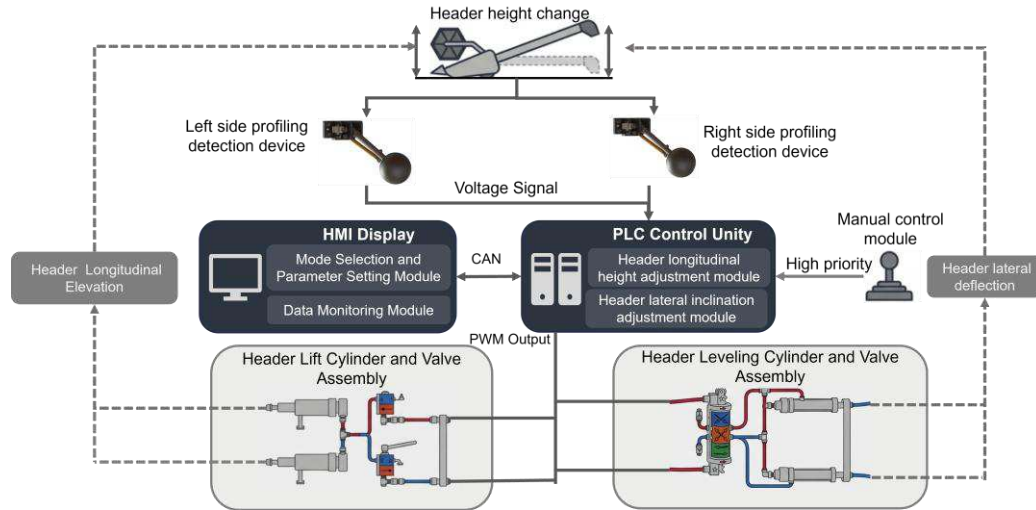


Fig. 2 Control principle diagram of profiling system.

3 Hardware Design of the Coordinated Terrain-Following System

3.1 Structural Design of the Terrain-Following Detection Mechanism

As shown in Figure 3, a spherical terrain-following detection device was designed to measure the height of the header. The detection range is 40–300 mm, the sensor arm is 350 mm long, the contour-following sphere has a diameter of 80 mm, and a wall thickness of 15 mm. Under the combined effect of its own weight and the tension of the return spring, the terrain-following sphere remains in close contact with the ground, ensuring the accuracy of the height measurement. The sensor arm drives the rotation of the rotary shaft, which in turn transmits ground undulation signals to the angle sensor. A Contelec Vert-X series $\pm 45^\circ$ angle sensor was selected to indirectly measure the header's height above the ground via the sensor's output signal.

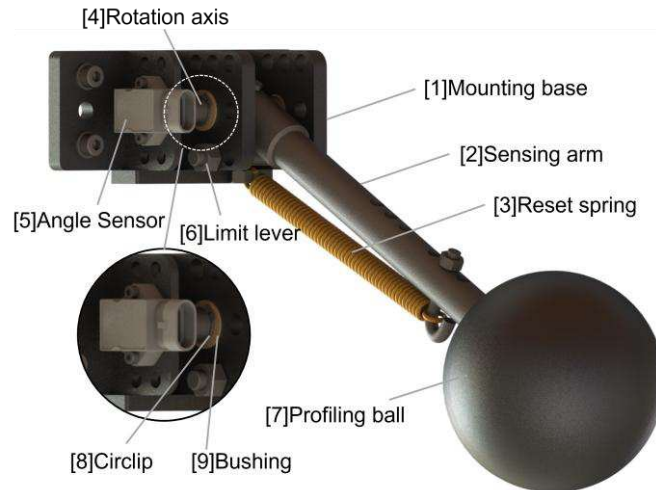


Fig. 3 Schematic diagram of the Spherical terrain-following mechanism

Contact-type terrain-following mechanisms are susceptible to vibration induced by impact loads during harvesting, which can potentially lead to substantial measurement errors. To address this, the kinematic process of the spherical terrain-following mechanism was simulated and analyzed using the ADAMS software platform to verify its terrain-following capability and optimize the parameters of the return tension spring, thereby mitigating the vibration issues encountered during the terrain-following process.

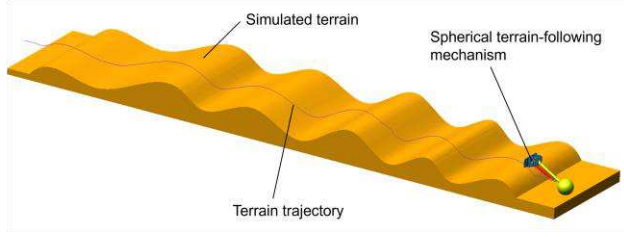


Fig. 4 Motion simulation of the Spherical terrain-following mechanism

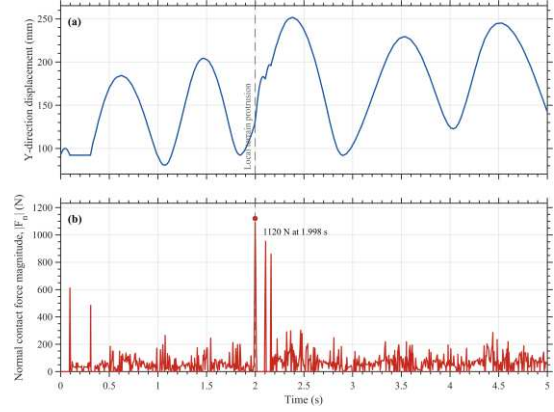


Fig. 5 Responses of the spherical terrain-following device.

(a) Y-direction displacement of the sensing-arm center of mass.

(b) Normal contact-force magnitude.

To improve the efficiency of simulation testing, the original terrain-following device was simplified when creating the simulation model. The simplified model consists of a mounting base, a rotating shaft, a sensing arm, a profiling ball, and a reset spring. A section of uneven terrain was simulated, with the material defined as soil, a density of $1.5 \times 10^{-6} \text{ kg/mm}^3$, a Poisson's ratio of 0.33, and a modulus of elasticity of 5 N/mm^2 . The tension spring is made of stainless steel wire, while the other components of the contour-following device are made of 45 steel. Since there is no relative motion between the rotating shaft, the sensing arm, and the terrain-following ball, these three components are treated as a single unit, referred to as the combined sensing arm. A fixed joint is established on the terrain; a rotational joint is added between the rotating shaft and the mounting base; a sliding joint is added between the mounting base and the ground; and the translational drive speed is set to 1 m/s . A contact constraint is added between the profiling ball and the terrain, and a flexible connection using a tension spring is added between the sensing arm and the mounting base to simulate the return spring. The simulation duration is set to 5 s with a step size of 0.001 . A schematic of the simulation model is shown in Figure 4.

Using the spring stiffness coefficient k and damping coefficient c as the influencing factors, simulations were conducted under different parameter combinations. The Y-direction displacement of the center of mass of the combined sensing arm and the normal contact force between the profiling ball and the simulated terrain were recorded. Based on the combined displacement and contact-force responses, $k=20 \text{ N/mm}$ and $c=0.5 \text{ N}\cdot\text{s/mm}$ were selected for subsequent analysis.

As shown in Fig. 5a, the combined sensing arm followed the main terrain undulations without sustained high-frequency oscillation. The corresponding normal contact-force magnitude is presented in Fig. 5b. During most of the traversal, the contact force fluctuated within a relatively low range. At $t=1.998 \text{ s}$, the profiling ball encountered a localized protrusion in the simulated terrain, producing a short-duration force peak of approximately 1.12 kN . The impact was followed by a brief separation and re-contact process, after which the contact force returned to its normal fluctuation range without sustained high-amplitude oscillation. These results indicate that the selected spring parameters allowed the profiling ball to recover contact rapidly after localized terrain excitation and provided acceptable terrain-following stability under the simulated conditions.

Based on the dimensions of the contour-following device and using the formula for selecting tension springs, the tension spring was ultimately determined to have a wire diameter of 3.5 mm , an outer diameter of 20 mm , and a length of 180 mm .

$$k = \frac{p}{\delta} = \frac{Gd^4}{8NaD^3} \quad (1)$$

where p denotes the load N ; δ represents the spring deflection; G is the shear modulus of elasticity $69 \times 10^3 \text{ N/mm}^2$;

d signifies the wire diameter; N_a stands for the number of active coils; and D refers to the mean coil diameter.

3.2 Header Height Calibration Experiment

To determine the header height, a mathematical relationship must be established between the header height and the output signal from the Spherical terrain-following mechanism's sensor, as shown in figure 6. By manually adjusting the header and recording the angular output value α ($0^\circ < \alpha < 90^\circ$) from the terrain-following device's angle sensor along with the header height H , the data is fitted and calibrated using the least squares method to derive the H - α relationship model.



Fig. 6 The Calibration of Header Height

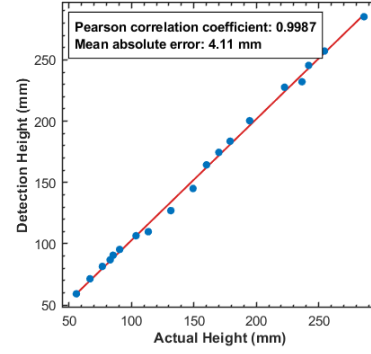


Fig. 7 Correlation Analysis between Detection and Actual Heights

$$H = -7.852\alpha + 483.2 \quad (2)$$

where H denotes the measured header height, and α represents the output angle of the sensor on the terrain-following detection mechanism.

After integrating the H - α relationship model into the control program, the header was adjusted randomly, and the terrain-following system's detected height was recorded at each position, while the actual height was measured with a tape measure. Three measurements were taken at each position, and the average was calculated. Measurements were stopped after collecting five sets of data in each of the three height ranges (50–100 mm, 100–200 mm, and 200–300 mm). The results are shown in Figure 7.

The measured height and the average detected height were 156.87 mm and 158.67 mm, respectively, with a Pearson correlation coefficient 0.99 and an average absolute error of 4.11 mm. The detected height values showed a strong correlation with the measured values, and the detection error was small, indicating that the spherical contouring device has good measurement accuracy.

4 Software Design of the Coordinated Terrain-Following System

4.1 Design of Control Strategy for Coordinated Terrain-Following System

During combine harvesting, terrain undulations, variations in crop feed rate and machine vibration jointly affect the header height in the longitudinal plane and its lateral tilt angle. (Dallabona, Blanke, & Papageorgiou, 2024; Feng & Yan, 2020; Fliess & Join, 2013; Sindhuja et al., 2023). The electrohydraulic actuators also exhibit high inertia, pure time delay and pronounced variations in operating load. To address these challenges, we develop a dual-channel control system that combines Smith feedback reconstruction, a continuous sliding-mode control law and explicit actuator saturation constraints. The longitudinal and lateral channels share the same control architecture but use distinct plant and controller parameters. To avoid mischaracterizing the control of a third-order plant as higher-order sliding-mode control, we hereafter refer to the scheme as the “Smith–SMC controller for third-order plants”.

4.1.1 Plant Models and Smith Feedback Reconstruction

Based on the system-identification results, the longitudinal height and lateral tilt channels are represented, respectively, as

$$G_H(s) = \frac{5e^{-0.2s}}{5.54 \times 10^{-7}s^3 + 1.55 \times 10^{-3}s^2 + 2s + 1} \quad (3)$$

$$G_T(s) = \frac{2.441e^{-0.2s}}{6.204 \times 10^{-7}s^3 + 1.92 \times 10^{-3}s^2 + 3.001s + 1} \quad (4)$$

Here, H and T denote the longitudinal height and lateral tilt channels, respectively. Both channels have a nominal pure time delay of $\tau_m = 0.2s$. The two transfer functions were assigned to their corresponding channels in the Simulink parameter file to prevent inadvertent interchange of the plant models.

The plant is expressed as

$$G_p(s) = G_m(s)e^{-\tau_p s} \quad (5)$$

where $G_m(s)$ is the nominal delay-free model. The Smith feedback reconstruction is then defined as

$$\Delta_y(t) = y(t) - y_m(t - \tau_m) \quad (6)$$

$$y_{\text{feed}}(t) = x_{m1}(t) + \Delta_y(t) \quad (7)$$

Here, Δ_y is the Smith residual between the actual output and the delayed output of the nominal model. It captures the combined effects of plant-model mismatch, external disturbances and measurement errors. Under exact model matching and in the absence of disturbances, Δ_y approaches zero, allowing the controller to act primarily on the delay-free model. When model mismatch occurs, the residual is reintroduced into the feedback path.

The Smith structure does not eliminate the physical time delay. Under nominal matching, it instead reduces the direct constraint imposed by the pure delay on the design of the closed-loop characteristic equation. Figure 8 shows the Smith–SMC implementation architecture shared by the longitudinal-height and lateral-tilt channels.

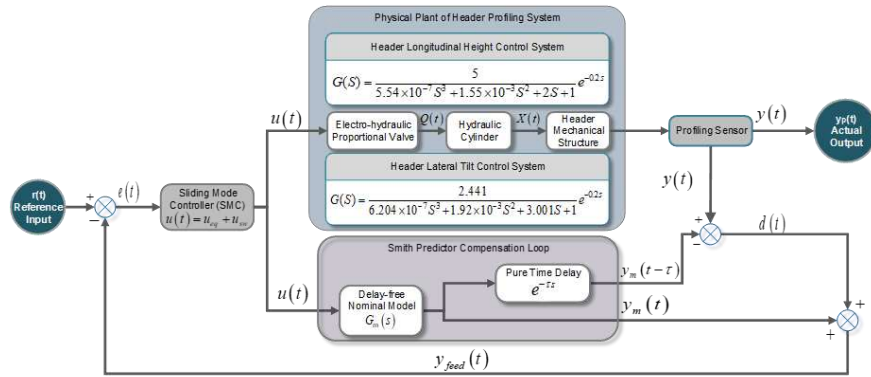


Fig. 8 Shared Smith–SMC implementation architecture for the longitudinal height and lateral tilt channels

4.1.2 Composite sliding surface, control law and stability analysis

The nominal delay-free model is expressed in controllable canonical form, with the state vector

$$\mathbf{x}_m = [x_{m1}, x_{m2}, x_{m3}]^T = [y_m, \dot{y}_m, \ddot{y}_m]^T \quad (8)$$

Its state equations are

$$\dot{x}_{m1} = x_{m2}, \quad \dot{x}_{m2} = x_{m3}, \quad \dot{x}_{m3} = -a_0x_{m1} - a_1x_{m2} - a_2x_{m3} + bu \quad (9)$$

To maintain consistency between the theoretical derivation and Simulink implementation, the position error is calculated using the Smith-reconstructed feedback. By contrast, the velocity and acceleration errors are calculated from the states of the nominal delay-free model. The composite error coordinates are defined as

$$e_0 = r - y_{\text{feed}}, \quad e_1 = \dot{r} - x_{m2}, \quad e_2 = \ddot{r} - x_{m3} \quad (10)$$

and the composite sliding surface is

$$s = c_1e_0 + c_2e_1 + e_2 \quad (11)$$

When $p^2 + c_2p + c_1$ is a Hurwitz polynomial, the coefficients are selected as $c_1 = \omega_n^2$ and $c_2 = 2\zeta\omega_n$. Because the feedback term in e_0 contains the Smith residual, differentiating the sliding surface introduces the term

227 $-c_1 \hat{\Delta}_y$. This term and the unmodelled dynamics are collectively represented by the lumped uncertainty $q(t)$,
 228 yielding

$$229 \quad \dot{s} = \sum_{i=0}^2 a_i x_{m,i+1} + c_1 e_1 + c_2 e_2 + r^{(3)} - bu + q(t) \quad (12)$$

230 Both Eqs. (12) and (13) use the third derivative of the reference trajectory, $r^{(3)}$, rather than its second derivative.
 231 Introducing a continuous quasi-sliding-mode reaching term gives the control law

$$232 \quad u = \frac{1}{b} \left[\sum_{i=0}^2 a_i x_{m,i+1} + c_1 e_1 + c_2 e_2 + r^{(3)} + \varepsilon \tanh\left(\frac{s}{\phi}\right) + ks \right] \quad (13)$$

233 Here, $\varepsilon > 0$ is the robust reaching gain, $k > 0$ is the linear convergence gain and $\phi > 0$ is the boundary-
 234 layer parameter. For step commands, the reference derivatives are set to zero except at the discontinuities. For
 235 sinusoidal commands, the analytical first, second and third derivatives are supplied explicitly. After saturation
 236 limiting, the same control input is applied simultaneously to the actual plant and the nominal delay-free model.
 237 Consider the Lyapunov candidate (Li, Yang, & Song, 2019)

$$238 \quad V = \frac{1}{2} s^2 \quad (14)$$

239 Assume that the actuator does not remain saturated and that the lumped uncertainty satisfies $|q(t)| \leq \bar{q}$.
 240 Substituting the control law in Eq. (13) into Eq. (12) gives

$$241 \quad \dot{V} \leq -ks^2 - |s| \left[\varepsilon \tanh\left(\frac{|s|}{\phi}\right) - \bar{q} \right] \quad (15)$$

242 When $\varepsilon > \bar{q}$, $|s|$ decreases outside the resulting quasi-sliding region and ultimately enters the bounded set

$$243 \quad |s| \leq \phi \operatorname{artanh}\left(\frac{\bar{q}}{\varepsilon}\right) \quad (16)$$

244 When $q(t) = 0$, s converges asymptotically to zero. Under bounded matched disturbances, s converges to a
 245 quasi-sliding region determined jointly by ϕ , ε and $\bar{q}$. Thus, the continuous $\tanh$ term reduces high-frequency
 246 chattering.

247 4.1.3 Simulation of Control Strategy for Coordinated Terrain-Following System

248 The Simulink model comprises two parallel channels for longitudinal height and lateral tilt control. Each channel
 249 contains a reference input, the Smith–SMC controller, a baseline controller, the nominal delay-free model, a physical
 250 pure-time-delay element, an equivalent input disturbance and a data-logging module. The model uses a fixed step
 251 size of 1×10^{-4} and an input sampling interval of 1×10^{-3} . The control input remains expressed in model-input
 252 units. Until it is calibrated to the proportional-valve current or voltage, it should not be interpreted as actual hydraulic
 253 pressure or valve-control current. The Simulink model and controller parameters are presented in Fig. 9 and Table 1
 254 and 2.

255 **Tab.1** Smith–SMC control parameters

Channel	ω_n / ζ	c_1 / c_2	$\varepsilon / k / \phi$	Control Limits
longitudinal height	13 / 0.80	169 / 20.8	300 / 40 / 50	± 250
lateral tilt	15 / 0.85	225 / 25.5	30 / 30 / 0.5	± 80

256 **Tab.2** Baseline controller parameters

Channel	Control	K_p	K_i	K_d	N
longitudinal height	PID	4.5856	1.6957	0.35761	21.816
lateral tilt	Smith-PID	12.6279	91.5955	0	100

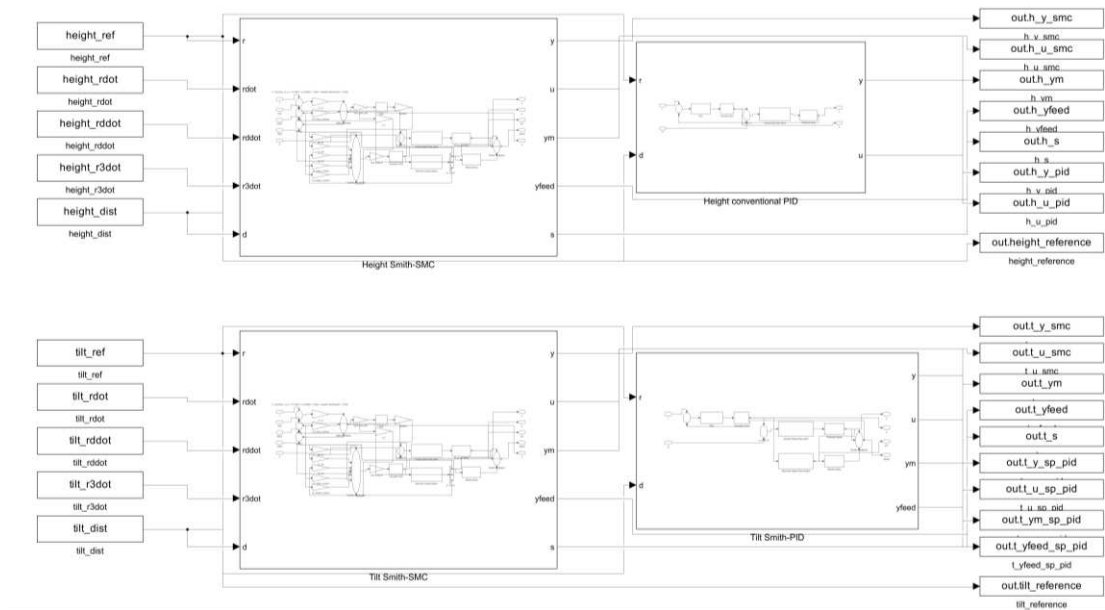


Fig. 9 Dual-channel Simulink model for longitudinal height and lateral tilt control

4.1.4 Simulation results and analysis

a. Control performance in the longitudinal height channel was evaluated using the second step change, from 50 to 100 mm. The 2% settling band was defined relative to the 50-mm step amplitude, giving a final-value band of 100 ± 1 mm. Settling time was measured from the command change at $t=3$ s until the output entered and remained within this band. The longitudinal-height step response and corresponding dynamic error are shown in Figure 10 and table 3.

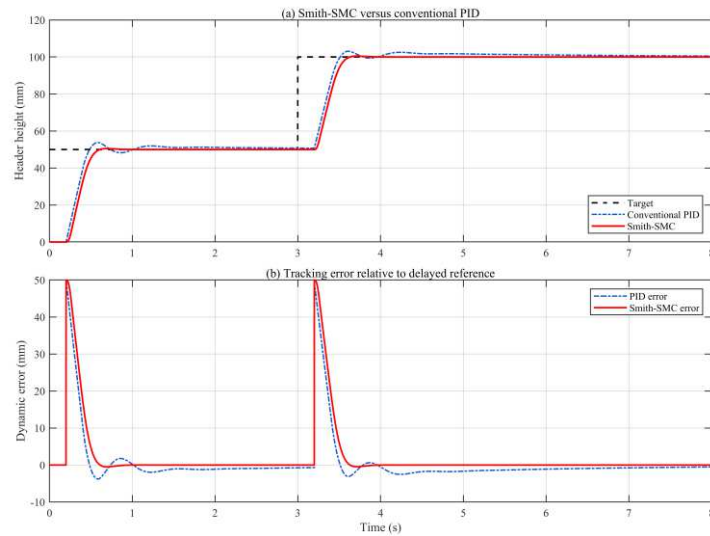


Fig. 10 Comparison of longitudinal-height step responses and dynamic errors

Tab.3 Performance metrics for the longitudinal-height step response

controller	Rise time (s)	Total settling time (s)	Net settling time (s)	Overshoot (%)	Steady-state error (mm)	Peak control input (mv)	Top control input(mv)
Smith-SMC	0.253	0.586	0.386	0.912	≈ 0	250	930.61
PID	0.232	3.275	3.075	6.224	-0.533	250	897.39

Smith-SMC reduced the settling time from 3.275 s with PID to 0.586 s, corresponding to a reduction of 82.1%. After subtracting the common physical time delay of 0.2 s, the net settling time decreased from 3.075 to 0.386 s, a reduction of 87.4%. Smith-SMC did not shorten the rise time relative to PID. Its principal advantage instead arose from suppressing overshoot and

subsequent oscillations. Overshoot decreased from 6.224% to 0.912%, while the steady-state error was nearly zero.

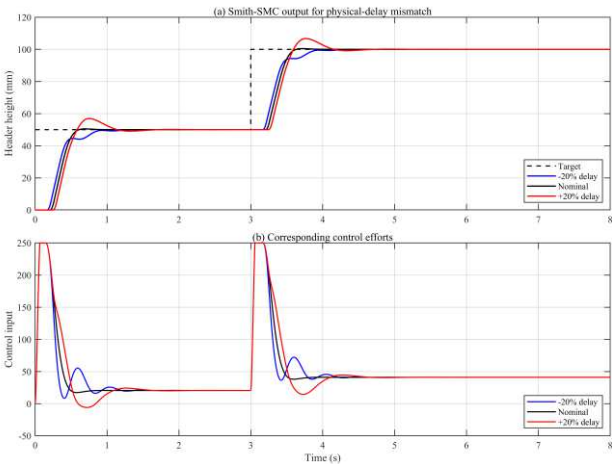


Fig.11 Longitudinal responses under $\pm 20\%$ mismatch in the actual pure time delay

Tab.4 Longitudinal responses under different actual pure time delays

Actual pure time delay (s)	Relative to nominal value	Settling time (s)	Overshoot (%)	Steady-state error (mm)
0.16	-20%	0.871	0.004	≈ 0
0.20	-	0.586	0.912	≈ 0
0.24	+20%	1.085	13.403	≈ 0

b .Across the tested variations around the nominal delay, the system continued to converge , as shown in figure 11 and table 4. However, a 20% increase in the actual delay increased the overshoot to 13.403% and the settling time to 1.085 s. These results indicate that the Smith predictor remains sensitive to time-delay mismatch. Therefore, practical deployment should prioritize identifying the maximum delay encountered during operation and retain sufficient stability margin to accommodate increases in delay.

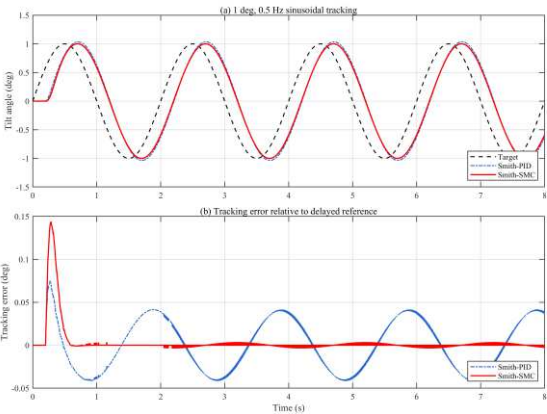


Fig.12 Comparison of lateral-tilt tracking for a 1° amplitude, 0.5-Hz sinusoidal command

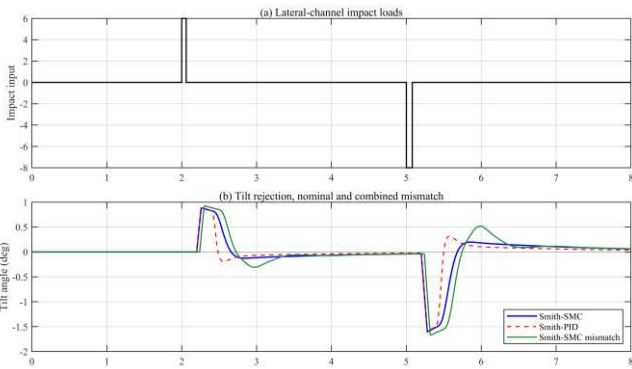


Fig.13 Lateral-tilt responses to positive and negative equivalent input pulses under the nominal model

Tab.5 Errors in lateral-tilt sinusoidal tracking

Controller	RMSE relative to delayed reference (°)	RMSE relative to original command (°)	MAE relative to delayed reference (°)	Peak error relative to delayed reference (°)
Smith-SMC	0.00105	0.43704	0.00045	0.00314
Smith-PID	0.02903	0.46036	0.02613	0.04142

c. To evaluate the dynamic tracking accuracy of the algorithm for the plant dynamics after accounting for the

pure time delay, a sinusoidal lateral-tilt tracking test was conducted. The results are presented in Fig. 12 and Table 5. Relative to the delayed reference $r(t - 0.2)$, Smith–SMC reduced the RMSE by 96.4% compared with Smith–PID, indicating high tracking accuracy for the plant dynamics after accounting for the pure delay. However, relative to the original command $r(t)$, the RMSE remained 0.43704° , only approximately 5.1% lower than that achieved by Smith–PID. This contrast indicates that the remaining tracking error was dominated by the physical time delay, which cannot be eliminated by the controller.

d. To evaluate disturbance-rejection and recovery performance, a positive pulse with an amplitude of 18 and a duration of 0.08 s was injected into the longitudinal-height channel at 2s, followed by a negative pulse with an amplitude of -22 and a duration of 0.10 s at 5 s. For the lateral-tilt channel, the corresponding pulses had amplitudes of 6 and -8 and durations of 0.06 and 0.08s, respectively. All disturbances were injected at the plant input. Their amplitudes are expressed in model input units and should not be interpreted directly as physical impact forces, The experimental results are shown in Figures. 13, 14 and 15.

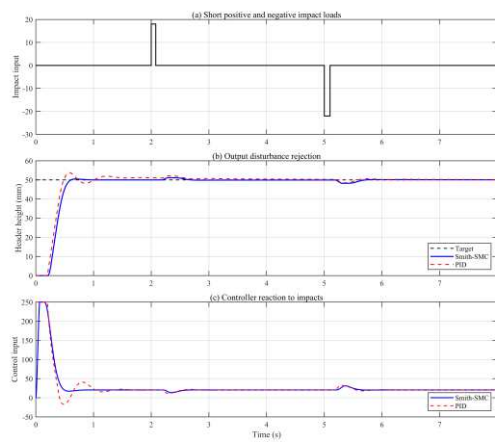


Fig.14 Longitudinal-height responses to positive and negative equivalent input pulses under the nominal model.

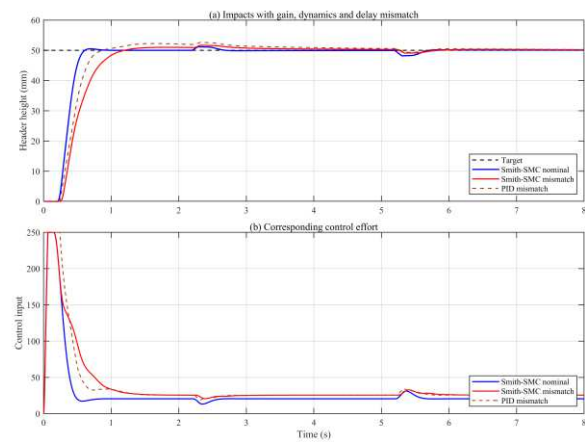


Fig.15 Longitudinal-height responses to input pulses under combined gain, dynamic and time-delay mismatch

Under nominal conditions in the longitudinal-height channel, Smith–SMC reduced the maximum deviation by 14.3% and the recovery time by 77.0% relative to Smith–PID. Under combined mismatch, Smith–SMC maintained a smaller maximum height deviation but required a longer recovery time than Smith–PID. Similarly, Smith–SMC did not outperform Smith–PID across all metrics under nominal conditions in the lateral-tilt channel. Its clearest advantage emerged under combined mismatch in this channel. Smith–SMC returned the tracking error to within 0.05° , whereas Smith–PID failed to recover within the 8-s simulation window.

Overall, after the identified plant models were correctly assigned to their respective longitudinal and lateral channels, Smith–SMC substantially reduced longitudinal settling time and overshoot. It also improved periodic lateral-tracking accuracy relative to the delayed reference. The continuous $\tanh$ reaching term maintained a continuous control signal, while simulations involving model mismatch and equivalent input pulses indicated stable operation within the tested uncertainty range. However, performance remained constrained by the physical time delay, actuator saturation and certain combinations of model mismatch. Within these limits, the proposed algorithm offers a favourable balance of rapid response, low overshoot and robustness to bounded uncertainty.

4.2 Program Design for Coordinated Header Terrain-Following

As shown in Figure 16 and Figure 17, terrain undulations can be classified into four typical scenarios based on the deviation between the heights of the header on both sides and the set target height: if the heights on both sides are within the allowable tolerance range, the ground is generally level; a unidirectional height deviation indicates that the ground on one side is elevated or sunken, while the other side is relatively level; a same-direction height deviation means that the ground on both sides is elevated or sunken simultaneously; and a opposite-direction height deviation

reflects that the ground on both sides is elevated or sunken in opposite directions.

This classification provides a basis for decision-making when the header performs contour-following maneuvers. When an opposite-direction height deviation exists, the ground slope β can be calculated based on the heights on both sides, and the header needs to adjust its lateral tilt angle α to adapt to this slope. In which

$$\alpha = \beta = \arcsin\left(\frac{HL - HR}{L}\right) \quad (17)$$

where HL and HR denote the detected heights on the left and right sides of the header, respectively; and L represents the lateral distance between the mounting points of the left and right terrain-following detection mechanisms, which is specified as 6000 mm for the 4YZL-12F experimental model.

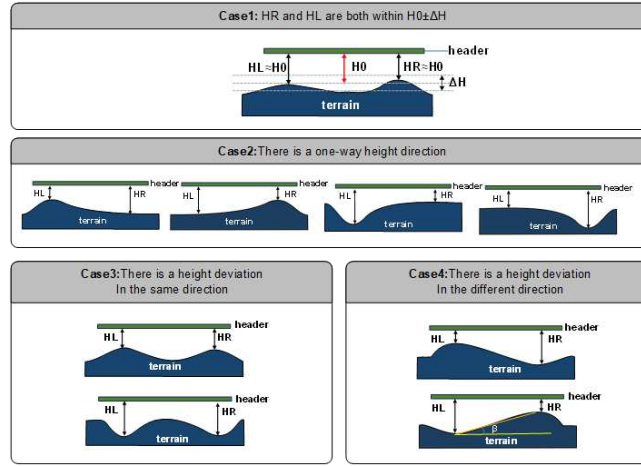


Fig. 16 Schematic diagram of terrain undulation(H_0 . Target height of header; HL . Left side height of header; HR . Right side height of header; ΔH . The floating range allowed for target header height is $[H_0 - \Delta H, H_0 + \Delta H]$. The shaded area between the two pairs of blue solid lines in the figure represents $\pm \Delta H$. 1. convex on the left; 2. convex on the right; 3. Concave on the left; 4. Concave on the right; 5. Concave on both sides; 6. Convex on both sides; 7. convex on the left, concave on the right; 8. Concave on the left, convex on the right.)

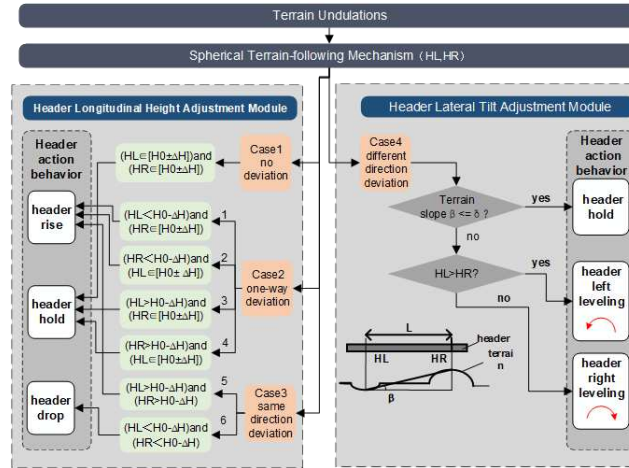


Fig. 17 Flowchart of the profiling action judgment program(δ . Slope gradient threshold)

When an elevation is detected on either one or both sides, the header must be promptly lifted to prevent it from digging into the soil. Given that the wheat header is a rigid structure and the header is incapable of local deformation, the header should maintain its current height without adjustment when a depression is encountered on only one side. Lowering the header under such circumstances would likely cause the opposite side to bulldoze into the soil. The header is lowered exclusively when depressions are simultaneously detected on both sides. In the presence of opposing height deviations, a header leveling action must be executed; relative to the forward direction of harvesting,

the header should be tilted towards the side registering a higher ground clearance.

5 Field Experiments

To verify the coordinated terrain-following capability of the system and the stubble height control precision, field experiments were conducted in Jiaozuo City, Henan Province, in June 2025. The experimental platform utilized was a 4YZL-12F large-scale combine harvester equipped with a 7.6 m wide wheat header. The experimental field featured natural undulations and slopes with varying gradients; the field experiment scenario is depicted in Figure 18.



Fig. 18 Field experiment scenario

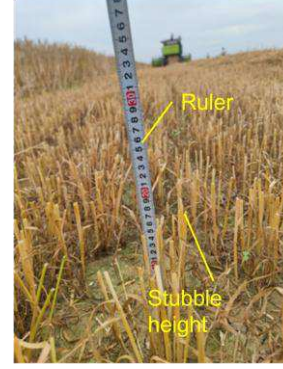


Fig. 19 Stubble height sampling

The coordinated terrain-following performance of the header was comprehensively evaluated based on the acquired real-time data of the average header height and tilt angle, in conjunction with the post-harvest stubble control precision.

The field experiment was conducted in two modes: manual and automatic. Comparative tests were performed for each mode, using stubble height and harvesting speed as experimental variables. Each test strip was 60 m long and 5 m wide. Twelve sampling locations were arranged at approximately 5-m intervals along the harvesting direction, as shown in figure 19. At each location, stubble height was measured at five individual points, yielding 60 raw measurements per experimental condition. The mean of the five measurements at each location was used for presentation in Tables 6 and 7, whereas the performance indices were calculated according to the data-processing procedures described below. In the automatic mode, six test groups were conducted first: to evaluate the adaptability of the cooperative contour-following system to different stubble height ranges, the first three groups were set at a uniform speed of 7 km/h, with stubble heights set at 60 mm, 100 mm, and 160 mm, respectively; To evaluate the contouring performance of the system at different speeds, the last three sets were conducted at speeds of 4 km/h, 8 km/h, and 10 km/h, respectively, with a uniform stubble height of 100 mm. In manual mode, six control tests were conducted under identical conditions. An individual stubble-height measurement was classified as acceptable when its absolute deviation from the target height did not exceed 20 mm. The pass rate (P), location-level mean absolute deviation (MAD_{loc}), and soil-scooping rate (SSR) were used as the principal performance indices.

$$\bar{h}_g = \frac{1}{5} \sum_{j=1}^5 h_{gj} \quad (18)$$

$$P = \frac{1}{60} \sum_{g=1}^{12} \sum_{j=1}^5 I(|h_{gj} - H_0| \leq 20) \times 100\% \quad (19)$$

$$MAD_{loc} = \frac{1}{12} \sum_{g=1}^{12} |\bar{h}_g - H_0| \quad (20)$$

$$SSR = \frac{n_s}{60} \times 100\% \quad (21)$$

where h_{jg} denotes the stubble height at the j th transverse sampling point within the g th longitudinal sampling location. $\bar{h}_j$ is the mean stubble height across the five transverse points at that location, and H_0 is the target stubble height. The indices are $g = 1, 2, \dots, 12$ and $j = 1, 2, \dots, 5$. I denotes the indicator function, and n_s is the number of soil-scooping events. For each operating condition, $n=12$ longitudinal sampling locations were evaluated, giving $N=60$ individual point measurements.

As shown in Figure 20, figure 21, the automatic terrain-following mode was evaluated at an operating speed of 7km/h and a target stubble height of 160 mm. The target lateral tilt angle was 0° , and the sampling period was 20 ms. The permissible header-height range was 140–180 mm, with a maximum height difference of 40 mm between the left and right sides. According to Eq. (17), these limits corresponded to a permissible header tilt range of $(\pm 0.5^\circ)$. The stubble-height results obtained under manual control and Smith–SMC-based automatic terrain-following control are presented in Tab.6 and Tab.7.

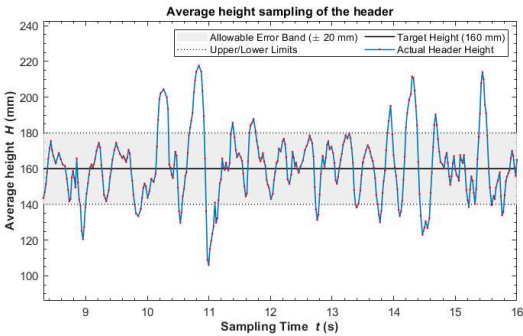


Fig. 20 Average height sampling of the header

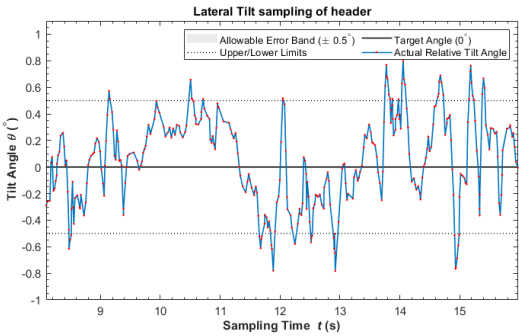


Fig. 21 Lateral Tilt sampling of header

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Tab. 6 Stubble height in automatic control mode using Smith-SMC control

Sampling location	Fixed harvest speed 7 Km/h			Fixed stubble height 100 mm		
	60mm	100mm	160mm	4 km/h	8 km/h	10 km/h
1	65.6	103.6	168.6	86.8	108.6	115.2
2	66.0	109.6	161.0	86.0	113.6	90.2
3	63.0	115.4	168.6	100.4	107.0	108.0
4	71.8	112.0	153.8	92.6	91.4	109.6
5	73.8	103.0	172.2	95.6	89.4	90.2
6	53.6	105.0	173.0	99.0	109.6	85.8
7	74.6	96.6	180.6	93.2	92.0	111.6
8	55.6	96.4	169.4	96.0	90.8	108.8
9	73.6	104.8	167.2	101.0	121.2	76.6
10	45.8	108.4	163.4	101.2	109.0	88.2
11	70.4	93.2	158.4	102.4	112.8	108.2
12	65.2	104.4	167.2	96.0	103.4	99.6
MAD_loc/mm	9.08	6.67	8.25	4.98	10.13	10.90
P/%	85.00	88.33	85.00	93.33	85.00	80.00
SSR(%)	8.33	3.33	0.00	0.00	1.67	5.00

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Tab. 7 Stubble height in manual control mode

Sampling location	Fixed harvest speed 7			Fixed stubble height 100		
	60mm	100mm	160mm	4 km/h	8 km/h	10 km/h
1	79.8	120.0	184.6	118.0	128.4	130.6
2	84.2	78.4	182.0	115.4	132.6	133.6

3	83.4	123.8	177.0	122.8	128.4	138.8
4	79.2	79.2	136.0	85.6	126.6	136.4
5	87.2	125.2	179.4	120.8	130.0	134.2
6	82.4	127.6	183.0	120.4	133.6	139.4
7	86.2	128.8	180.6	122.8	136.8	141.0
8	83.0	123.2	182.8	115.6	130.4	131.6
9	85.6	76.2	188.4	115.4	133.6	132.4
10	95.6	127.0	149.2	122.8	126.0	128.8
11	84.4	126.2	184.4	116.2	134.8	130.4
12	85.8	114.4	176.6	116.4	130.2	135.2
MAD_loc/mm	24.73	23.53	21.13	18.42	30.95	34.37
P/%	43.33	50.00	53.33	56.67	31.67	26.67
SSR/%	38.33	18.33	15.00	13.33	21.67	28.33

Note: Rows 1–12 report the mean of five individual point measurements at each longitudinal sampling location. For each operating condition, (P) was calculated from all 60 point measurements, (MAD_loc) from the 12 unrounded location-level means, and SSR from the number of soil-scooping events recorded across the 60 observations.

The experimental results led to the following conclusions:

(1) The measured header height and tilt angle remained close to their respective target values, with 71.07% and 75.06% of the measurements falling within the permissible tolerance ranges, respectively. These results provide preliminary evidence that the proposed header terrain-following system can coordinate the control of header height and tilt angle.

(2) Under the Smith-SMC-based automatic terrain-following mode, the stubble-height pass rate reached at least 85% for all target-height groups tested at a constant forward speed, while the mean absolute deviation remained below 10 mm. These results indicate that the proposed system can adapt effectively to different target stubble heights. At a forward speed of 8 km/h and a target stubble height of 100 mm, the pass rate under manual control was only approximately 31.67% because of operator response delays and limited control accuracy, making it difficult to track the target height effectively. In comparison, the proposed Smith-SMC control strategy maintained the stubble height close to the 100-mm target and achieved a pass rate of 85.00%. The strategy also reduced header bouncing caused by high-frequency terrain disturbances and exhibited good dynamic tracking accuracy.

(3) As the forward speed increased from 4 to 10 km/h, the stubble-height control performance of both manual control and Smith-SMC automatic control deteriorated to varying degrees because of increasingly pronounced high-frequency field disturbances. Nevertheless, the Smith-SMC strategy exhibited better robustness and disturbance rejection. At a forward speed of 10 km/h, the mean absolute deviation under manual control increased to 34.37 mm. By comparison, the Smith-SMC control system maintained a mean absolute deviation of 10.90 mm, a stubble-height pass rate of 80%, and a soil scooping occurrence of only 5%, indicating that severe deviations from the target height remained infrequent. Within the range of operating speeds and field conditions examined in this study, these results suggest that combining the delay-compensation capability of the Smith predictor with the robustness of sliding mode control can improve the stability and accuracy of combine-harvester header terrain-following control under high-speed and complex operating conditions.

6 Conclusions

To improve the terrain-following accuracy of large combine-harvester headers under complex terrain conditions, this study developed a coordinated control system for longitudinal header height and lateral tilt, optimized the spherical contact-based terrain-sensing device using ADAMS multibody dynamics simulations, and proposed a Smith-SMC strategy with a continuous tanh reaching term to address pure time delay, model uncertainty, and external

disturbances in the electro-hydraulic servo system. For a longitudinal height step from 50 to 100 mm, Smith-SMC reduced the total settling time from 3.275 s with conventional PID control to 0.586 s, corresponding to an 82.1% reduction, while decreasing the overshoot from 6.224% to 0.912% and maintaining a near-zero steady-state error. In lateral sinusoidal tracking, the RMSE relative to the delayed reference was 96.4% lower than that obtained with Smith-PID. Simulations involving model mismatch and equivalent input-pulse disturbances showed stable operation within the tested uncertainty range, although performance remained constrained by the physical time delay, actuator saturation, and large delay mismatch. In the field tests, 71.07% of the measured header-height data and 75.06% of the measured tilt-angle data fell within the permissible tolerance ranges. At a forward speed of 8 km/h and a target stubble height of 100 mm, Smith-SMC achieved a stubble-height pass rate of 85.00%, compared with 31.67% under manual control. At 10 km/h, it maintained a pass rate of 80.00% and a location-level mean absolute deviation of 10.90 mm, whereas the corresponding deviation under manual control was 34.37 mm. Across the tested speed range of 4-10 km/h, the system achieved an average stubble-height pass rate of approximately 86.11%; the location-level mean absolute deviation remained below 11 mm in all test groups, and the average soil-scooping rate was 3.06%. These results indicate that, within the tested ranges of model uncertainty, operating speed, and field conditions, the proposed Smith-SMC strategy provides a favorable balance among settling time, overshoot suppression, and disturbance rejection, thereby supporting coordinated automatic control of combine-harvester header height and lateral tilt.

Data availability

Data will be made available on request.

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