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Trajectory Tracking Control of Farm Vehicles in Presence of Sliding

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***Abstract*— In automatic guidance of agriculture vehicles, lateral control is not the only requirement. Lots of research works have been focused on trajectory tracking control which can provide high longitudinal-lateral control accuracy. Satisfactory results have been reported as soon as vehicles move without sliding. But unfortunately pure rolling constraints are not always satisfied especially in agriculture applications where working conditions are rough and not expectable. In this paper the problem of trajectory tracking control of autonomous farm vehicles in presence of sliding is addressed. To take sliding effects into account, two variables which characterize sliding effects are introduced into the kinematic model based on geometric and velocity constrains in presence of sliding. With linearization approximation a refined kinematic model is obtained in which sliding appears as additive unknown parameters to the ideal kinematic model. By integrating parameter adaptation technique with *backstepping method*, a stepwise procedure is proposed to design a robust adaptive controller. It is theoretically proven that for the farm vehicles subjected to sliding, the longitudinal-lateral deviations can be stabilized near zero and the orientation errors converge into a neighborhood near the origin. To be more realistic for agriculture applications, an adaptive controller with projection mapping is also proposed. Simulation results show that the proposed (robust) adaptive controllers can guarantee high trajectory tracking accuracy regardless of sliding.**

***Index Terms*— Trajectory tracing control, nonholonomic systems, backstepping**

I. INTRODUCTION

Automatic guidance of farm vehicles develops with the requirement of modern agriculture. High-precision agriculture becomes a reality especially thanks to new localization technologies such as GPS, laser range scans, sonar. In agriculture fields it is quite common that several vehicles (including cropping, threshing, cleaning, seeding and spraying machines) compose a platoon for combined harvesting. In this case driving safety requiring constant longitudinal distances between the leading vehicle and following vehicles is an additional requirement along with the effort of improving lateral path-following performances. Since longitudinal-lateral control becomes more and more important, many research teams have paid their attention to trajectory tracking control, satisfactory results have been reported as soon as vehicles satisfy pure rolling constraints [1]-[4].

However due to various factors such as slipping of tires, deformability or flexibility of wheels, pure rolling constraints are never strictly satisfied. Especially in agriculture applications when farm vehicles are required to move on all-terrain grounds including slippery slopes, sloppy grass grounds, sandy and stony grounds, sliding inevitably occurs which deteriorates performances of automatic guidance and even system stability.

Until now there are very few papers dealing with sliding. [5] prevents cars from skidding by robust decoupling of car steering dynamics, but acceleration measurements are necessary and the steering angle is assumed small. [6] copes with the control of WMR (Wheeled Mobile Robot) not satisfying the ideal kinematic constraints by using slow manifold methods, but the parameter characterizing the sliding effects is assumed to be exactly known. Therefore [5][6] are not realistic for agriculture applications. In [7] a controller is designed based on the averaged model allowing the tracking errors to converge to a limit cycle near the origin. In [11] a general singular perturbation formulation is developed which leads to robust results for linearizing feedback laws ensuring trajectory tracking. But above two schemes only take into account sufficiently small skidding effects and they are too complicated for real-time practical implementation. In [8] [9] Variable Structure Control (VSC) is used to eliminate the harmful sliding effects when the bounds of the sliding effects have been known. The trajectory tracking problem of mobile robots in the presence of sliding is solved in [10] by using discrete-time sliding mode control. But the controllers [8]-[10] counteract sliding effects only relying on high-gain controllers which is not realistic because of limited bandwidth and low level delay introduced by steering systems of farm vehicles. In [12] sliding effects are rejected by rescheming desired paths adaptively based on steady control errors which are mainly caused by modeled sliding effects. Moreover a robust adaptive controller is designed in [13] which can compensate sliding by parameter adaptation and VSC. But [12] [13] only care about lateral control.

In the referred references most research works treated sliding as disturbances, but alternatively sliding can be also regarded specifically as time-varying parameters. On the other hand backstepping methods which are used widely in controller design have been proven powerful in controlling nonholonomic systems with uncertain parameters [14][16]. In our previous work [13] we have applied backstepping successfully to design a path following controller, so the purpose of this paper is to extend our lateral controller to design a practical longitudinal-lateral controller in presence of sliding.

The main idea of this paper is to introduce sliding effects as additive unknown parameters to the ideal kinematic model. Based on *backstepping method* a robust adaptive controller is designed. Furthermore to be of benefit to actual applications the robust adaptive controller is simplified into an adaptive controller with projection mapping. This paper is organized as follows, in section 2 a kinematic model considering sliding is constructed in the vehicle body frame. In section 3 a robust adaptive controller is designed by using backstepping methods. In section 4 the robust adaptive controller is simplified into an adaptive controller with projection mapping. In section 5, some comparative simulation results are presented to validate the proposed control laws.

II. Kinematic Model for Trajectory Tracking Control

A. Notation and Problem Description

In this paper the vehicle is simplified into a bicycle model, the kinematic model is expressed in the vehicle body frame (o, x', y') (see figure 1). Variables necessary in the kinematic model are denoted as follows:

- o (o_r) is the center of the (reference) vehicle virtual rear wheel.
- x' is the vector corresponding to the vehicle body axis
- y' is the vector vertical to x'
- (x_r, y_r) are the coordinates of the reference vehicle o_r with respect to the inertia frame.
- (x, y) are the coordinates of the vehicle o with respect to the inertia frame.
- (x_e, y_e) are the coordinates of the vector $\overrightarrow{oo_r}$ in the frame (o, x', y')
- $c(s)$ is the curvature of the path, s is the curvilinear coordinates (arc-length) of the point o_r along the reference path from an initial position.
- θ (θ_r) is the orientation of the (reference) vehicle centerline with respect to the inertia frame.
- $\theta_e = \theta_r - \theta$ is the orientation error.
- l is the vehicle wheelbase.
- v (v_r) is the linear velocity of the (reference) vehicle with respect to the inertia frame.
- v_x is the longitudinal velocity of the vehicle in the direction of ox' in the inertia frame. In this paper we assume that only lateral sliding occurs between tires and grounds, so v_x always equals to the wheel rotating velocity V_ω.
- δ is the steering angle of the virtual front wheel

So the trajectory tracking errors can be described by (x_e, y_e, θ_e). The aim of this paper is to design a controller (v_x, δ) which can guarantee the longitudinal-lateral errors x_e, y_e approach to zero and the orientation error θ_e is bounded in presence of sliding.

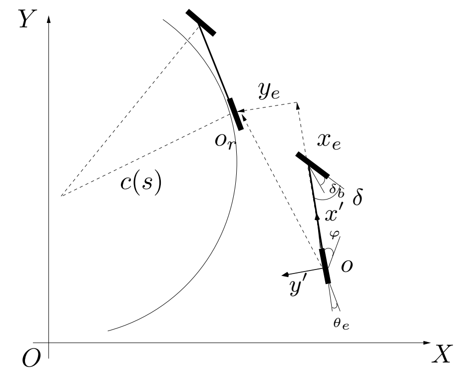

Fig. 1. Notations of the kinematic model

B. Kinematic Model

From figure 1, it is easy to obtain the following geometric relationship

$$\begin{pmatrix} x_e \\ y_e \\ \theta_e \end{pmatrix} = \begin{pmatrix} \cos\theta & \sin\theta & 0 \\ -\sin\theta & \cos\theta & 0 \\ 0 & 0 & 1 \end{pmatrix} \begin{pmatrix} x_r - x \\ y_r - y \\ \theta_r - \theta \end{pmatrix} \quad (1)$$

In this paper it is assumed that $|\theta_e| < \frac{\pi}{2}$. When vehicles move without sliding, the angular velocity can be expressed by

$$\dot{\theta} = \omega = \frac{v}{l}\tan\delta \quad (2)$$

The angular velocity of the reference vehicle is

$$\dot{\theta}_r = \frac{v_r}{\frac{1}{c(s)}} \quad (3)$$

The ideal kinematic model with respect to (o, x', y') can be developed directly by differentiating (1)

$$\begin{cases} \dot{x}_e = -v + v_r\cos\theta_e + \omega y_e \\ \dot{y}_e = v_r\sin\theta_e - \omega x_e \\ \dot{\theta}_e = v_r c(s) - \dfrac{v}{l}\tan\delta \end{cases} \quad (4)$$

But when vehicles move on a steep slope or the ground is slippery, sliding occurs inevitably, (4) is no longer valid. Since the longitudinal tire sliding is neglected, the violation of the pure rolling constraints is described by introducing the lateral sliding velocity v_y and bias of the steering angle δ_b. Therefore the velocity constraints become

$$\begin{cases} \dot{x} = v\cos(\theta + \varphi) \\ \dot{y} = v\sin(\theta + \varphi) \end{cases} \quad (5)$$

where

$$v = \sqrt{v_x^2 + v_y^2} \quad (6)$$

and φ is the side sliding angle defined by

$$\varphi = \arctan(\frac{v_y}{v_x}) \quad (7)$$

By using the similar method the kinematic model when sliding is taken into account is obtained

$$\begin{cases} \dot{x}_e = -v_x + v_r\cos\theta_e + \omega y_e \\ \dot{y}_e = -v_y + v_r\sin\theta_e - \omega x_e \\ \dot{\theta}_e = v_r c(s) - (\dfrac{v_x}{l}\tan(\delta + \delta_b) - \dfrac{v_y}{l}) \end{cases} \quad (8)$$

Remark that

$$v_x = v\cos\varphi \tag{9}$$

equals to the wheel rotating velocity which is the control law to be designed. In case no sliding occurs, $v_x = v$.

C. Kinematic Model with Linearization Approximation

In actual agriculture applications farm vehicles always move smoothly and most trajectories to be tracked are straight lines and circles, so the lateral sliding velocity and the steering bias vary not too greatly with time. Hence the sliding effects can be described exactly by

$$\begin{aligned} v_y &= \bar{v}_y + \varepsilon_1 \\ \delta_b &= \bar{\delta}_b + \varepsilon_2' \end{aligned} \tag{10}$$

where $\bar{v}_y$, $\bar{\delta}_b$ are time-invariant, ε_1, ε_2' are time-varying variables with zero mean value. Furthermore since the steering bias δ_b is quite small, the orientation kinematic equation in (8) can be linearized resulting in trivial errors. Therefore the kinematic model (8) is rewritten as

$$\dot{x}_e = -v_x + v_r\cos\theta_e + \omega y_e \tag{11a}$$

$$\dot{y}_e = v_r\sin\theta_e - \omega x_e - (\bar{v}_y + \varepsilon_1) \tag{11b}$$

$$\dot{\theta}_e = c(s)v_r - \frac{v_x}{l}\tan\delta + \frac{\bar{v}_y + \varepsilon_1}{l} - \frac{v_x}{l}(\tan\bar{\delta}_b + \varepsilon_2) \tag{11c}$$

where $\varepsilon_2 = \tan\varepsilon_2' + \epsilon$, ϵ is the error due to linearization approximation.

III. Backstepping-based Robust Adaptive Control Design

A. Trajectory Tracking Control for Ideal Kinematic Model

Notice that (4) is a 2-3 nonholonomic system in which y_e is not directly controlled. To overcome this problem the idea of backstepping is used: see [15] for details. Using backstepping we propose a stepwise design procedure for this 3-order nonholonomic system. Due to limited space, we do not present a detailed description of the design scheme. The resulting control law is

$$v_x = v_r\cos\theta_e + k_x x_e \tag{12}$$

$$\delta = \arctan(\frac{l\omega}{v_x}) \tag{13}$$

where

$$\omega = \frac{y_e v_r + \cos\theta_e v_r c(s) + k_y\sin\theta_e + k_u\tilde{u}_1}{\cos\theta_e + \frac{k_y x_e}{v_r}} \tag{14}$$

$$\tilde{u}_1 = \sin\theta_e + \frac{k_y y_e}{v_r} \tag{15}$$

We refer interested readers to [20] for details.

B. Robust Adaptive Control for Kinematic Model with Sliding

Consider the kinematic model (11). It is a 2-3 nonholonomic system with unknown constant parameters $\bar{v}_y$, $\bar{\delta}_b$ and time-varying disturbances ε_i. In this paper it is assumed that ε_i is bounded by

$$|\varepsilon_i| < \rho_i \tag{16}$$

So we are in the place to design a controller which not only can estimate and compensate unknown parameters but also is robust to ε_i.

step 1: Consider the sub-kinematic equations (11a) and (11b). The Lyapunov function candidate is chosen as

$$V_1 = \frac{1}{2}x_e^2 + \frac{1}{2}y_e^2 + \frac{1}{2}(\hat{v}_y - \bar{v}_y)^T\Gamma^{-1}(\hat{v}_y - \bar{v}_y) \tag{17}$$

where Γ is positive definite, $\hat{v}_y$ indicates the estimation of $\bar{v}_y$. The time derivative of V_1 along the kinematic model is

$$\begin{aligned} \dot{V}_1 = {} & x_e(-v_x + v_r\cos\theta_e) + y_e(v_r\sin\theta_e - \hat{v}_y - \varepsilon_1) \\ & +(\hat{v}_y - \bar{v}_y)\Gamma^{-1}(\dot{\hat{v}}_y + \Gamma y_e) \end{aligned} \tag{18}$$

Regard $u_1 = \sin\theta_e$ as the virtual control input of the first step. If choose u_1 as a variable structure controller

$$u_{1d} = \frac{-k_y y_e + \hat{v}_y - \rho_1 sign(y_e)}{v_r} \tag{19}$$

and

$$v_x = v_r\cos\theta_e + k_x x_e \tag{20}$$

$$\dot{\hat{v}}_y = -\Gamma y_e \tag{21}$$

then we have

$$\dot{V}_1 = -k_x x_e^2 - k_y y_e^2 - (\rho_1 - \varepsilon_1)|y_e| \tag{22}$$

So u_{1d} of (19) is the desired value of the virtual control input u_1 for the first step. If u_1 tracks (19) precisely, then the longitudinal and lateral deviations will converge to zero asymptotically.

Note that if the longitudinal sliding is not neglected, v_x can be designed easily by using variable structure control. Then the following step is similar.

Indeed in the closed loop system u_1 is not the actual control input, tracking u_{1d} with some errors, therefore $\tilde{u}_1$ is defined as

$$\tilde{u}_1 = u_1 - u_{1d} \tag{23}$$

In backstepping schemes the derivative of u_{1d} must appear in the following steps, but $sign()$ included in (19) is not differentiable, so $sign()$ is replaced by $\tanh()$ which is continuously differentiable. Therefore u_{1d} becomes

$$u_{1d} = \frac{-k_y y_e + \hat{v}_y - \rho_1\tanh(\frac{y_e}{\sigma_1})}{v_r} \tag{24}$$

where $\sigma_1 > 0$. Substituting (24) into (23) and computing time derivative yield to

$$\begin{aligned} \dot{\tilde{u}}_1 = {} & \cos\theta_e\big(c(s)v_r - \frac{v_x}{l}\tan\delta + \frac{\bar{v}_y + \varepsilon_1}{l} - \frac{v_x}{l}(\eta + \varepsilon_2)\big) \\ & + \frac{1}{v_r}(\varpi\dot{y}_e - \dot{\hat{v}}_y) \end{aligned} \tag{25}$$

where

$$\eta = \tan\bar{\delta}_b \tag{26}$$

$$\varpi = k_y + \big(1 - \tanh^2(\frac{y_e}{\sigma_1})\big)\frac{\rho_1}{\sigma_1} \tag{27}$$

Remark: For simplicity it is assumed that v_r is constant, in case v_r is time-varying, only variation is adding $\frac{\dot{v}_r}{v_r^2}\big(-k_y y_e + \hat{v}_y - \rho_1 \tanh\frac{y_e}{\sigma_1}\big)$ in (25).

step 2: consider the Lyapunov function as

$$V_2 = V_1 + \frac{1}{2}\tilde{u}_1^2 + \frac{1}{2}(\hat{\eta}-\eta)^T\gamma^{-1}(\hat{\eta}-\eta) \tag{28}$$

where γ is positive definite, $\hat{\eta}$ indicates the estimation of η. Regard $u_2 = \tan\delta$ as the virtual control input of the second step, then the time derivative of V_2 along (18) is

$$\begin{aligned}\dot{V}_2 &= x_e(-v_x + v_r\cos\theta_e) + y_e(v_r u_1 - \hat{v}_y - \varepsilon_1)\\ &+(\hat{v}_y-\bar{v}_y)^T\Gamma^{-1}(\dot{\hat{v}}_y + \Gamma y_e) + \tilde{u}_1\dot{\tilde{u}}_1 + (\hat{\eta}-\eta)^T\gamma^{-1}\dot{\hat{\eta}}\end{aligned} \tag{29}$$

Substituting (20)(24)(25) into (29), we have the following equation (see [20] for detail)

$$\begin{aligned}\dot{V}_2 &\le -k_x x_e^2 - k_y y_e^2 - (\rho_1-\varepsilon_1)|y_e| + y_e v_r\tilde{u}_1\\ &+(\hat{v}_y-\bar{v}_y)^T\Gamma^{-1}(\dot{\hat{v}}_y+\Gamma y_e)\\ &+\tilde{u}_1\Big(\cos\theta_e\big(c(s)v_r - \frac{v_x}{l}\tan\delta + \frac{\bar{v}_y+\varepsilon_1}{l} - \frac{v_x}{l}(\eta+\varepsilon_2)\big)\\ &+\frac{1}{v_r}(\varpi\dot{y}_e - \dot{\hat{v}}_y)\Big) + (\hat{\eta}-\eta)^T\gamma^{-1}\dot{\hat{\eta}} + \zeta_1\end{aligned} \tag{30}$$

where ζ_1 is a trivial variation due to the replacement of $sign()$ by $\tanh()$ in (24). From (30) the following equation can be obtained by algebraic transformation

$$\begin{aligned}\dot{V}_2 &\le -k_x x_e^2 - k_y y_e^2 - (\rho_1-\varepsilon_1)|y_e|\\ &+\tilde{u}_1(\lambda - \beta u_2 + \alpha - \frac{\varpi\varepsilon_1}{v_r} - \beta\hat{\eta} + \tau\varepsilon_1 - \beta\varepsilon_2)\\ &+(\hat{v}_y-\bar{v}_y)^T\Gamma^{-1}(\dot{\hat{v}}_y + \Gamma y_e - \Gamma\tilde{u}_1\tau + \Gamma\frac{\varpi}{v_r}\tilde{u}_1)\\ &+(\hat{\eta}-\eta)^T\gamma^{-1}(\dot{\hat{\eta}} + \gamma\tilde{u}_1\beta) + \zeta_1\end{aligned} \tag{31}$$

where

$$\alpha = \frac{\varpi(v_r\sin\theta_e - \hat{v}_y) - \dot{\hat{v}}_y}{v_r} \tag{32}$$

$$\tau = \frac{1}{l}(\cos\theta_e + \frac{\varpi x_e}{v_r}) \tag{33}$$

$$\beta = v_x\tau \tag{34}$$

$$\lambda = y_e v_r + \cos\theta_e c(s)v_r + \tau\hat{v}_y \tag{35}$$

In (31) let

$$\begin{aligned}\dot{\hat{\eta}} &= -\gamma\tilde{u}_1\beta\\ \dot{\hat{v}}_y &= -\Gamma y_e + \Gamma\tilde{u}_1\tau - \Gamma\frac{\varpi}{v_r}\tilde{u}_1\end{aligned} \tag{36}$$

and choose u_2 as

$$\begin{aligned}u_2 &= \frac{1}{\beta}\Big(k_u\tilde{u}_1 + \lambda + \alpha - \beta\hat{\eta}\\ &+\rho_1\big(\frac{\cos\theta_e}{l} + \frac{\varpi}{v_r}\Big|\frac{x_e-l}{l}\Big|\big)\tanh(\frac{\tilde{u}_1}{\sigma_2}) + |\beta|\rho_2\tanh(\frac{\tilde{u}_1}{\sigma_3})\Big)\end{aligned} \tag{37}$$

where $sign()$ has been substituted by $tanh()$ and $\sigma_i > 0$, then we get

$$\begin{aligned}\dot{V}_2 &\le -k_x x_e^2 - k_y y_e^2 - k_u\tilde{u}_1^2 - (\rho_1-\varepsilon_1)|y_e|\\ &-(\rho_2-\varepsilon_2)|\beta||\tilde{u}_1|\\ &-(\rho_1-\varepsilon_1)\big(\frac{\cos\theta_e}{l} + \frac{\varpi}{v_r}\Big|\frac{x_e-l}{l}\Big|\big)|\tilde{u}_1| + \zeta\end{aligned} \tag{38}$$

where $\zeta = \zeta_1 + \zeta_2$, ζ_2 is another trivial variation due to the substitution of $sign()$ by $tanh()$ in (37). (38) implies that the closed-loop system is uniformly bounded.

C. Stability Analysis

From (38) it is known that the longitudinal deviation x_e, lateral deviation y_e and $\tilde{u}_1$ are all bounded. Indeed all of them converge into a neighborhood of zero. The range of the neighborhood is determined by ζ which is linked to σ_i. The smaller σ_i is, the smaller the range of the neighborhood is, yielding higher accuracy.

When y_e and $\tilde{u}_1$ vary around zero, from (23) and (24) one gets that the orientation error θ_e converges into a neighborhood of

$$\bar{\theta}_e = \arcsin\big(\frac{\hat{v}_y}{v_r}\big) \tag{39}$$

IV. Simplified adaptive controller with projection mapping

The robust adaptive controller (37) with VSC can guarantee high tracking accuracy from academic point of view. But in actual applications due to limited bandwidth of agriculture vehicles and lag of hydraulic-drive steering systems, performances of the robust adaptive controller (37) may be deteriorated by significant "Chattering".

To be of more benefit to actual applications, the robust adaptive controller is simplified by setting ρ_i to zero, then we get $\varpi = k_y$ and the controller (37) is reduced into an ordinary adaptive controller without VSC components.

$$u_2 = \frac{1}{\beta}\Big(k_u\tilde{u}_1 + \lambda + \alpha - \beta\hat{\eta}\Big) \tag{40}$$

By using the similar Lyapunov's direct method, it is proven that the adaptive controller (40) leads to the following result

$$\begin{aligned}\dot{V}_2 &\le -k_x x_e^2 - k_y y_e^2 - k_u\tilde{u}_1^2 + \varepsilon_1(\frac{\tilde{u}_1\cos\theta_e}{l} - \frac{\tilde{u}_1 k_y}{v_r} - y_e)\\ &-\tilde{u}_1\cos\theta_e\frac{v_x}{l}\varepsilon_2\end{aligned} \tag{41}$$

(41) implies the closed-loop system is uniformly bounded. But comparing with (38) in which only ζ is a negligible disturbance, (41) is subjected to all the unmodeled sliding effects. (see [20] for detail)

To make the adaptive controller (40) more robust to the unmodelled sliding effects, projection mapping is used for the parameter adaptation procedure. The projection mapping $Proj_\xi(\bullet)$ is defined by [17], [18]

$$Proj_\xi(\bullet) = \begin{cases} 0 & if\ \ \hat{\xi} = \xi_{\max}\ \ and\ \ \bullet > 0\\ 0 & if\ \ \hat{\xi} = \xi_{\min}\ \ and\ \ \bullet < 0\\ \bullet & otherwise\end{cases} \tag{42}$$

By using projection mapping $Proj_\xi(\bullet)$, the robust adaptive laws become

$$\dot{\hat{v}}_y = Proj_{\bar{v}_y}\big(-\Gamma y_e + \Gamma\tilde{u}_1\tau - \Gamma\frac{\varpi}{v_r}\tilde{u}_1\big) \tag{43}$$

$$\dot{\hat{\eta}} = Proj_\eta\big(-\gamma\tilde{u}_1\beta\big) \tag{44}$$

The prior information on the bounds of the sliding effects $\bar{v}_y, \eta$ can be obtained off-line after performing large number of absolute coordinates measurements under different typical working conditions.

V. Simulation Results

First a classical "U" path with a perfect circular arc (path #1) is applied as the reference trajectory to test the proposed controllers. In the simulations, the gains used in (20) and (37) are set as $k_x = 0.6$, $k_y = 0.15$, $k_u = 1.14$. The gains of the adaptive laws (36) as set as $\Gamma = 0.2$, $\gamma = 0.05$. In actual implementations these gains should be tuned gradually to make an optimal compromise between transient characteristic and limited bandwidth of the steering system. The reference velocity is set as $v_r = 8.4km/h$ which is the normal velocity of agriculture vehicles in agriculture applications.

In the first simulation the constant sliding is introduced with $v_y = -0.1$, $\delta_b = -0.048$. The control law (12) without considering sliding is applied also with the same controller gains. The simulation results of the longitudinal, lateral and orientation errors are shown by figure 2-4. Since the vehicle velocity is initialized to zero, obvious longitudinal errors are noticed at the beginning of the simulations. The initial orientation errors are also nonzero. Those initial errors quite fit with the real working conditions. From the simulations it is clear that all the controllers can make the longitudinal-lateral errors approach to zero before sliding occurs. But when sliding appears, because the control law (12) does not take sliding effects into account, the longitudinal-lateral deviations (dashed line) become significant. While the robust adaptive controller (37) can compensate sliding effects through estimating them on line and counteract modeling inaccuracy by VSC, so the longitudinal-lateral deviations can converge to zero with a good transient response (solid line). Finally the adaptive controller (40) is simulated also. (40) can compensate time-invariant sliding, the effects of the time-varying sliding are moderated by projection mapping, hence its longitudinal-lateral deviations (dotted line) converge to zero with small offsets (due to linearization approximation in (11c)). The remarkable overshoots at the beginning and end of the curve are caused by "jump change" of the sliding effects and low level delay. The bounded orientation errors are shown by figure 4. As analyzed by section III-C the proposed controllers cannot make the orientation errors converge to zero, indeed they are bounded around (39). It is normal when sliding occurs known as "crab sliding". The evolution of the sliding parameters $\hat{v}_y$ (solid line), $\hat{\eta}$ (dashed line) is displayed by figure 5. At the beginning and end of the circle, $\hat{v}_y$ varies greatly which explains the overshoots of the lateral deviation, but as the vehicle follows the circle, $\hat{v}_y$, $\hat{\eta}$ evolve smoothly close to the real values.

Since the real sliding v_y and δ_b cannot be measured precisely. To simulate the actual working conditions, a set of real pre-measurement data is used to reconstruct the v_y and δ_b approximately. The longitudinal-lateral deviations are shown by 6, 7. The (robust) adaptive controllers yield small lateral deviations with zero mean value, while the

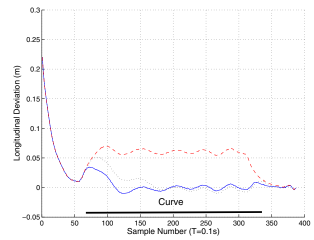

Fig. 2. Longitudinal deviation of path #1

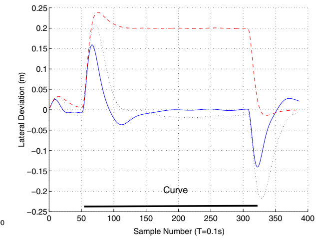

Fig. 3. Lateral deviation of path #1

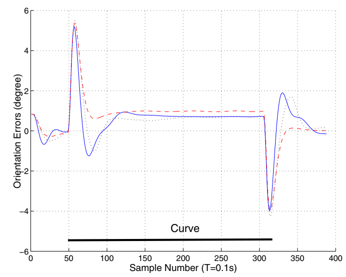

Fig. 4. Orientation errors of path #1

lateral deviation of the controller (12) is significant and has obvious bias. The longitudinal errors of the (robust) adaptive controller are also less than it of (12). It is because when the lateral sliding and steering bias are compensated by (robust) adaptive controllers, the negative influences of y_e and θ_e (due to sliding) on the longitudinal tracking accuracy is moderated.

In order to fully present the proposed controllers, another realistic reference trajectory #2 which is sampled in an actual agriculture application is tracked (see figure 8). The longitudinal and lateral deviations are displayed by figure 9, 10. The experimental data indicates that although the trajectory #2 is more complex than trajectory #1, the proposed controllers can still track it with high accuracy in presence of sliding. Furthermore the robust adaptive controller with VSC yields better transient performances at the expense of non-smooth movements (solid line) especially when low level delay is considered. While the adaptive controller (40) with projection mapping yields a movement with less oscillation (dotted line), but its bias is larger than VSC's. So in case when sliding is dominant, the robust adaptive controller with VSC is favorable. But for the vehicles whose bandwidth is limited, the adaptive controller with projection mapping is preferred.

VI. Conclusion

The problem of trajectory tracking control of autonomous agricultural vehicles in the presence of sliding is investigated in this paper. A kinematic model which integrates the sliding effects as additive unknown parameters is constructed. From this model, a robust adaptive controller is designed based on backstepping methods which can stabilize the longitudinal-lateral derivations into a neighborhood of zero and guarantees the orientation error converge into a neighborhood near the origin. In addition

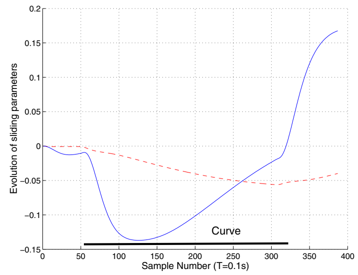

Fig. 5. Evolution of sliding parameters

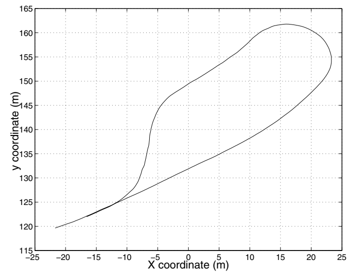

Fig. 8. Path #2 to be followed

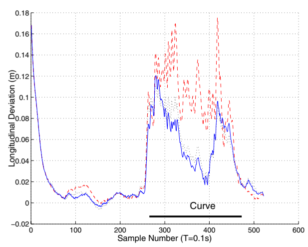

Fig. 6. Longitudinal deviation of path #1 with real measurements

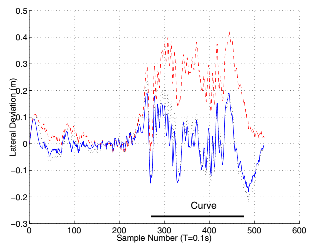

Fig. 7. Lateral deviation of path #1 with real measurements

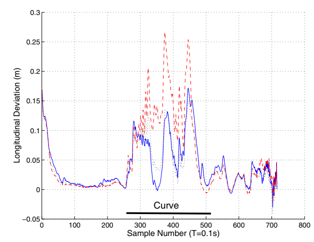

Fig. 9. Longitudinal deviation of path #2 with real measurements

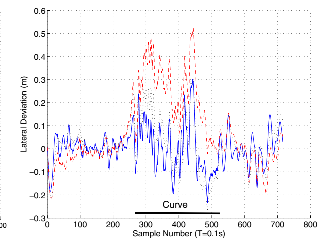

Fig. 10. Lateral deviation of path #2 with real measurements

a reduced adaptive controller with projection mapping is proposed for the purpose of smooth vehicle movements. Experimental comparative results show the effectiveness of the proposed control laws. The advantages of this scheme are that

- When no sliding occurs, the proposed controller can guarantee longitudinal-lateral deviations and orientation errors converge to zero.
- Integrating parameter adaptation with backstepping schemes yields a practical trajectory tracking controller for agriculture vehicles. Also it is applicable for platoon control.
- Backstepping procedures can be extended easily to high-order nonholonomic systems, for example trailer control.

The prospective works include extending backstepping methods to platoon control and using predictive control to decrease overshoots of lateral deviations [19].

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