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Mobile robot control in presence of sliding: Application to agricultural vehicle path tracking

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***Abstract*— Control design of mobile robots dedicated to path tracking tasks is generally based on the rolling without sliding assumption at the wheel/ground contact point. However, such an hypothesis is not relevant for off-road vehicle and the use of non sliding assumption in this case leads to important tracking errors. Moreover, the vehicles considered for off-road applications (such as agricultural tasks) have generally important mass and inertia and are equipped with low reactive actuators. As a result, delays are introduced into the control loop, which depreciate considerably the performances of autonomous path tracking, especially during a modification of path curvature. This paper proposes several developments, relying on adaptive and predictive observer-based control, dedicated to automatic guidance of off-road mobile robots. The relevancy of the proposed algorithm is investigated via full scale experiments, carried out with a farm tractor, as the considered application is agricultural work.**

Keywords : mobile robots, predictive control, adaptive control, observer, chained system, GPS, agriculture.

I. INTRODUCTION

The accuracy of vehicle guidance required during agricultural work (generally smaller than $\pm$15cm) needs an important concentration from the driver. Considering moreover the large range of work period and that the driver has also to pay attention to the implement supervision, such an accuracy is difficult to reach all day long. As a consequence, devices for driver assistance and especially automatic guidance systems have been developing more and more (see for instance [3] and [7] for general overviews on existing systems). The automatic devices currently marketed provide satisfactory results as long as rolling without sliding assumption is satisfied, but their efficiency is strongly depreciated since sliding occurs (e.g when following a curved path or when the vehicle is running on a slope). This paper deals with these difficulties. More precisely, an observer-based control law is introduced to compensate sliding effects while a predictive action is designed to face actuators and inertial delays.

In a first part (section II), incorporation of sliding phenomena in the vehicle modeling is investigated and an extended kinematic model is proposed. The attractive feature of this model with respect to dynamic ones lies in the reduction of the number of parameters to be known. An observer is then designed in section III, in order to estimate on-line these parameters, when the vehicle perception system is limited to a sole exteroceptive sensor (an RTK-GPS sensor). Non-linear control design is then investigated in section IV. It ensures a significant reduction of tracking errors in presence of sliding, as long as the curvature of the path to be followed is slow-varying. To deal with fast-varying curvature, a predictive algorithm is introduced into the control law in section V. This action enables to compensate for the overshoots observed in such situations and originated from the delays introduced by actuator features and vehicle inertia. The efficiency of the overall control scheme is finally validated through full scale experiments in section VI. The numerous tests demonstrate automatic path following accurate to within $\pm 15cm$ whatever the conditions of adherence and the shape of the path to be followed. Such results match the expectations in agricultural applications considered in this paper.

II. MODELING

A. Notations

Firstly, the classical description of a car-like mobile robot under rolling without sliding assumption (Ackermann's model) is recalled on figure 1. The vehicle is depicted as a bicycle model (one wheel for the front axle and one for the rear axle), and the following notations are used :

- $\mathcal{C}$ is the path to be followed,
- O is the center of the rear axle,
- M is the point of $\mathcal{C}$, which is the closest to O. M is considered as unique, which is realistic when the vehicle remains quite close to $\mathcal{C}$.
- s is the curvilinear abscissa of M along $\mathcal{C}$, and $c(s)$ is the curvature of $\mathcal{C}$ at this point.
- y and $\tilde{\theta}$ are respectively the lateral and angular deviation of the vehicle with respect to the reference path $\mathcal{C}$ (see figure 1).
- δ is the steering angle of the front axle and the sole control variable.
- v is the linear velocity of point O, considered as a measured parameter, whose value may be time-varying.
- L is the vehicle wheelbase.

Relying on these notations, the robot state vector can be constituted of $\begin{bmatrix} s & y & \tilde{\theta} \end{bmatrix}^T$ and the equations of motion

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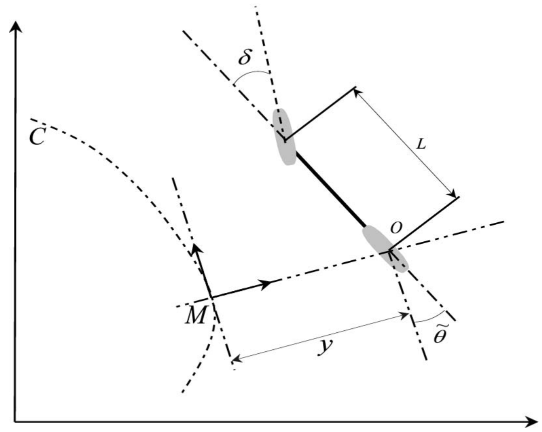


Fig. 1. Ackermann model of vehicle

under the previous hypothesis can be shown to be :

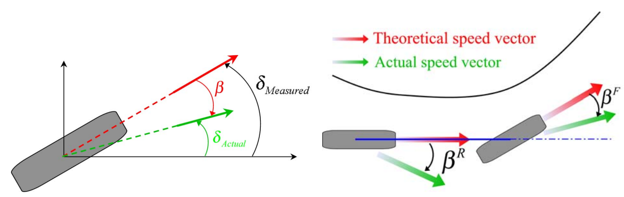
$$\left\{\begin{array}{lcl} \dot{s} & = & \frac{v\cos\tilde{\theta}}{1-c(s)y} \\ \dot{y} & = & v\sin\tilde{\theta} \\ \dot{\tilde{\theta}} & = & v\left(\frac{\tan\delta}{L}-\frac{c(s)\cos\tilde{\theta}}{1-yc(s)}\right) \end{array}\right. \tag{1}$$

It is known (see for instance [9]) that such a model can be turned into a "chained system". Using this property, path tracking can be addressed via non-linear control design. Satisfactory accuracy can then be obtained as long as no sliding occurs (see for instance [11] for road vehicle applications or [12] for off-road applications dedicated to straight line following on a flat ground).

B. Integration of sliding phenomenon

1) Description of sliding parameters: In the case of path tracking in off-road conditions, the rolling without sliding assumption is generally false. A control law based on model (1) will then not be able to ensure a null tracking error. To account for sliding effects, a first idea consists in taking part of dynamic approaches, including a tire/ground model (such as the celebrated Pacejka model, introduced in [2]). Unfortunately, the use of such models requires the knowledge of numerous parameters, which can be time-varying in off-road applications (as adherence conditions are not constant). As a consequence, the addition of new sensors is needed, increasing the price of the device. Moreover vehicle model becomes very large so that control design is more complex. An alternative solution is thus proposed in this paper. It preserves the use of a sole exteroceptive sensor and the simplicity of a kinematic approach with respect to control design.

In dynamic approach (such as described in [1]), the extraction of tire forces is achieved via a variable named side slip angle. This angle (called hereafter β and graphically depicted on figure 2(a)) represents the difference between the tire orientation and the actual speed vector orientation attached to this tire. Instead of using β as an intermediate variable to estimate the forces applied to the vehicle, we propose to make use of its kinematic meaning and to introduce these side slip angles for each wheel as parameters of a new kinematic model.



(a) Definition of a side slip angle

(b) Application to bicycle model

Fig. 2. Sliding parameters of extended kinematic model

As described on figure 2(b), two side slip angles are then integrated into the bicycle kinematic description (β^F for the front axle and β^R for the rear one). These two parameters are supposed to be suitably evaluated thanks to the observer defined in section III. The behavior of vehicle is then modified, as the speed vector orientations are no longer given by the wheel angle, but are supplied by the two side slip angles, representative of the sliding occurring on the vehicle.

2) Extended kinematic model: In view of figure 2(b), from a kinematic point of view, the vehicle in presence of sliding can be regarded as a vehicle with two steering axles moving without sliding : its front steering angle would be $\delta + \beta^F$ and the rear one would be β^R. Model of such vehicles are known (see [6] for instance) to be :

$$\left\{\begin{array}{lcl} \dot{s} & = & \frac{v\cos(\tilde{\theta}+\beta^R)}{1-c(s)y} \\ \dot{y} & = & v\sin(\tilde{\theta}+\beta^R) \\ \dot{\tilde{\theta}} & = & v\left[\cos\beta^R\frac{\tan(\delta+\beta^F)-\tan\beta^R}{L}-\frac{c(s)\cos(\tilde{\theta}+\beta^R)}{1-c(s)\,y}\right] \end{array}\right. \tag{2}$$

Model (2) is below named extended kinematic model. When side slip angles are null (β^F=0 and β^R=0), it can be checked that this model is equivalent to the classical model (1).

III. Estimation of sliding parameters

A. Observer principle

The perception system on-boarded is limited to one RTK-GPS, located on the top of the vehicle above the point O. It supplies a localization accurate to within $\pm 2cm$. This sensor allows to measure the state vector of model (2), i.e. $\begin{bmatrix} s & y & \tilde{\theta} \end{bmatrix}^T$, but not the two sliding parameters β^F and β^R. Since the adherence conditions as well as the configuration of vehicle are varying, it is necessary to estimate them online.

Classically, such an estimation is achieved using an observer (such as described in [10]). Unfortunately, no model for side slip angles variation can be used in view of our limited perception system. Therefore, standard observer cannot be used. An alternative is here proposed. It relies on the following consideration : the correct value for estimated sliding parameters (β^F and β^R) must ensure that the simulation of the model (2) supplies outputs ($\hat{y}$ and $\hat{\tilde{\theta}}$) converging to measured output (y and $\tilde{\theta}$ respectively). As a consequence, observation of ($\hat{\beta}^F$ and $\hat{\beta}^R$) can be viewed as a control problem. To highlight this assertion, let's introduce from model (2) the following system :

$$\dot{X}_{Obs} = f_{Obs}(X_{Obs}, u_{Obs})$$

$$X_{Obs} = \begin{bmatrix} \hat{y} \\ \hat{\tilde{\theta}} \end{bmatrix} \quad u_{Obs} = \begin{bmatrix} \hat{\beta}^R \\ \hat{\beta}^F \end{bmatrix} \quad \bar{Y} = \begin{bmatrix} y \\ \tilde{\theta} \end{bmatrix}$$

$$f_{Obs}\left[X_{Obs}, u_{Obs}\right) = \left\{ \begin{array}{l} v\sin(\hat{\tilde{\theta}} + \hat{\beta}^R) \\ v\left[\cos\hat{\beta}^R \frac{\tan(\delta+\hat{\beta}^F)-\tan\hat{\beta}^R}{L} \right. \\ \left. -\frac{c(s)\cos(\hat{\tilde{\theta}}+\hat{\beta}^R)}{1-c(s)\hat{y}}\right] \end{array} \right] \tag{3}$$

u_{Obs} is viewed as the control vector, X_{Obs} as the state vector and $\bar{Y}$ (the measurement supplied by the GPS) is viewed in this algorithm as the objective to be reached by the state X_{Obs}. Finally, δ is considered as a measured parameter. The observer design consists then in designing a control law for u_{Obs}, which ensures the convergence of X_{Obs} to the measurement $\bar{Y}$. If such a convergence is achieved, then, u_{Obs} constitutes clearly a relevant on-line estimation of sliding parameters.

This point of view is depicted on figure 3, where a second loop in the bottom of the figure achieving the observation of sliding parameters is added to the main control loop (detailed in section IV-B).

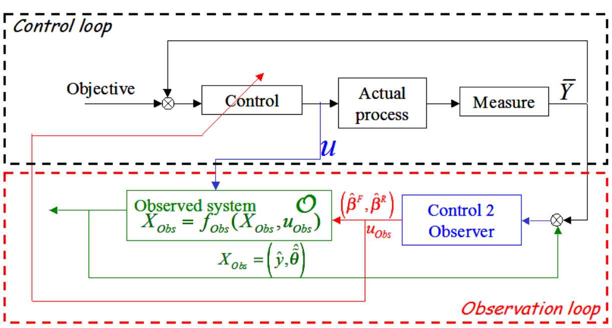


Fig. 3. Global scheme of observation algorithm

B. Estimation algorithm

This control problem can be addressed by considering the estimation error by ϵ defined by $\epsilon = X_{Obs} - \bar{Y}$. Using the notations introduced by (3), the variation of this error can be written as follows :

$$\dot{\epsilon} \;=\; f_{Obs}(X_{Obs}, u_{Obs}) - \dot{\bar{Y}} \tag{4}$$

where $\dot{\bar{Y}}$ is the numeric derivative of the measurements. Considering that in practice, the values for side slip angles stay close to zero, this equation can be linearized around a null control (β^F=0 and β^R=0). Equation (4) then becomes :

$$\dot{\epsilon} \;=\; f_{Obs}(X_{Obs}, 0) - \dot{\bar{Y}} + \frac{\partial f_{Obs}}{\partial u_{Obs}}(X_{Obs}, 0)u_{Obs} \tag{5}$$

As $\frac{\partial f_{Obs}}{\partial u_{Obs}}(X_{Obs}, 0)$ is invertible (as shown in [4]), u_{Obs} can then be designed as :

$$u_{Obs} = \left(\frac{\partial f_{Obs}}{\partial u_{Obs}}(X_{Obs}, 0)\right)^{-1}\left[G\cdot\epsilon - f_{Obs}(X_{Obs}, 0) + \dot{\bar{Y}}\right] \tag{6}$$

where G is an Hurwiz matrix. Reporting (6) in (5) indeed leads to :

$$\dot{\epsilon} \;=\; G\cdot\epsilon \tag{7}$$

This means that observation error converges to zero, as expected. The choice of G allows to specify a dynamic for the observer error. The sliding parameters observer used below is then control law (6).

IV. Control law in presence of sliding

As sliding parameters are properly estimated by the algorithm detailed previously, all the variables appearing in model (2) are now known. As a result, path tracking control design can be addressed.

A. Exact linearization - chained system

As pointed in section II, model (2) is equivalent to a model of vehicle with two steering wheels. According to [9], such a model can be turned into a chained system : indeed, model (2) can be converted into the chained form (8).

$$\left\{ \begin{array}{lcl} a_1' & = & 1 \\ a_2' & = & a_3 \\ a_3' & = & m_3 \end{array} \right. \tag{8}$$

by applying state transformation Φ and control transformation M defined by (9)

$$\begin{array}{l} \Phi: (s, y, \tilde{\theta}) \to (a_1, a_2, a_3)^T \\ \qquad \to \left(s, y, \tan\left(\tilde{\theta}+\beta^R\right)(1-c(s)y)\right)^T \\ M: \;\; \delta \;\; \to \;\; m_3 = \frac{m_2}{m_1} \\ \qquad m_2 \;\; = \;\; \frac{d}{dt}\left(\tan\left(\tilde{\theta}+\beta^R\right)[1-c(s)y]\right) \\ \qquad m_1 \;\; = \;\; v\frac{\cos\left(\tilde{\theta}+\beta^R\right)}{1-c(s)y} \end{array} \tag{9}$$

and then, by introducing a change in time-scale : in (8), $a'_{i=1..3}$ denotes the derivative of a_i with respect to curvilinear abscissa s.

B. Control law expression

The objective of the trajectory tracking problem is to ensure the convergence of the tracking error ($y \equiv a_2$) to zero. It can classically be obtained via the virtual control (10) :

$$m_3 \;=\; -K_d a_3 - K_p a_2 \quad (K_p, K_d) \in \Re^{+2} \tag{10}$$

where K_p and K_d can be viewed as the proportional and derivative gains of a PD controller, since injecting (10) into (8) leads to the differential equation (11).

$$a_2'' + K_d a_2' + K_p a_2 = 0 \tag{11}$$

This equation ensures both the convergence of a_2 and a_3 to zero. This means that the lateral deviation converges to zero, thanks to the compensation of rear side slip angle by a heading deviation ($a_3 \to 0 \Rightarrow \tilde{\theta} \to -\beta^R$).
Finally, by injecting the virtual control law (10) into the definition of m_3 given in (9), the non-linear control expression (12) can be calculated. It constitutes the control law for vehicle path tracking in presence of sliding. It can naturally be noticed that in absence of sliding, the classical control law proposed in [12] is recovered.

$$\begin{array}{rcl} \delta & = & \arctan\left\{\frac{L}{\cos\hat{\beta}^R}\left[c(s)\frac{\cos\tilde{\theta}_2}{\alpha}+A\frac{\cos^3\tilde{\theta}_2}{\alpha^2}\right]\dots \right. \\ & & \left. \dots+\tan\hat{\beta}^R\right\}-\hat{\beta}^F \end{array}$$

$$\left\{\begin{array}{rcl} \tilde{\theta}_2 & = & \tilde{\theta}+\hat{\beta}^R \\ \alpha & = & 1-c(s)y \\ A & = & -K_d\alpha\tan\tilde{\theta}_2-K_py+c(s)\alpha\tan^2\tilde{\theta}_2\dots \\ & & \dots+\frac{dc(s)}{ds}y\tan\tilde{\theta}_2 \end{array}\right. \tag{12}$$

Experiments demonstrate (see section VI) that control law (12) ensures accurate path tracking when the curvature of the path to be followed is constant. However, punctual overshoots can nevertheless be recorded at variations of path curvature (when entering or leaving a curve).

V. Predictive control

In order to avoid such overshoots at variations of path curvature, a predictive algorithm to be associated with the control law (12) is now designed. This algorithm is based on two important hypotheses : the path to be followed is known in advance and a model for the actuator can be extracted.

A. Control law separation

Values of sliding, as well as lateral and angular deviations, cannot be anticipated. Therefore, the prediction algorithm has to be designed exclusively with respect to the curvature of the reference path. As a consequence, the control law (12) is here decomposed into two additive terms, as follows :

$$\begin{array}{l} \delta=\delta_{Traj}+\delta_{Deviation} \\ \text{with}\ \left\{\begin{array}{rcl} \delta_{Traj} & = & \arctan(u) \\ \delta_{Deviation} & = & \arctan(\frac{v}{1+uv+u^2})-\hat{\beta}^F \end{array}\right. \\ \text{and}\ \left\{\begin{array}{rcl} u & = & \frac{L}{\cos\hat{\beta}^R}c(s)\frac{\cos\tilde{\theta}_2}{\alpha} \\ v & = & \frac{L}{\cos\hat{\beta}^R}A\frac{\cos^3\tilde{\theta}_2}{\alpha^2}+\tan\hat{\beta}^R \end{array}\right. \end{array} \tag{13}$$

Practically, these two terms ensure the two following tasks :

- $\delta_{Deviation}$ (null when there is no deviation nor sliding) : this term mainly relies on the deviations $(y, \tilde{\theta}_2)$ and on sliding parameters $(\hat{\beta}^R, \hat{\beta}^F)$. It ensures the convergence of deviations to 0. As these variables and parameters cannot be anticipated, this term will not be introduced into the predictive algorithm.
- δ_{Traj} (Non-null term when there is no deviation nor sliding) : this term ensures the convergence of vehicle curvature to the curvature of the reference path. As the future curvature of the path to be followed is known, the future objective attached to this term can be calculated. This term will be concerned by the predictive algorithm.

As a result, the predictive algorithm will be applied exclusively to δ_{Traj}. The proposed algorithm, detailed in [5], is summarized on figure 4.

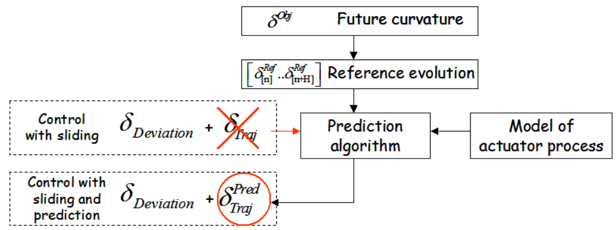


Fig. 4. Path tracking with sliding accounted and prediction

B. Functional Predictive Control

After the preliminary step consisting in the separation of the control law, a functional predictive control algorithm, see for instance [8], is applied in order to achieve predictive curvature control. The aim is to calculate the predictive term, denoted δ_{Traj}^{Pred} on figure 4, which has to replace the term δ_{Traj}. The prediction principle requires the following notations, illustrated on figure 5 :

- δ^C : *Control variable sent to the actuator.* In the current case of a separated control, this variable is only the trajectory part δ_{Traj} of the control law, defined in (13).
- δ^R : *Measured steering angle.* This is the output of the actuator process resulting from the action of the control δ^C, which is only the contribution of δ_{Traj}.
- H : *Horizon of prediction.* This is the time in the future, which will be used to determine the control value to be applied at the present time. In our case, H is chosen as a constant, equal to a multiple of the sampling time T_s, such as $H = n_H T_s$.
- δ^{Obj} : *Known future objective.* It represents the future desired actuator output value. In our case, this variable is linked to the future curvature of the reference path by the relation : $\delta^{Obj} = \arctan(L.c(s_H))$, where $c(s_H)$ is the curvature of the trajectory at the horizon H.
- δ^{Ref} : *Desired reference shape.* It is the shape expected to be followed by the actuator output δ^R to reach the future objective δ^{Obj}. Classically, a first order system is used :

$$\delta_{[n+i]}^{Ref} \quad = \quad \delta^{Obj}-\gamma^i\{\delta^{Obj}-\delta_{[n]}^{Ref}\} \tag{14}$$

where $i \in [0, .., n_H]$ and $\gamma \in [0, 1[$ is a parameter tuning a "settling time".
- $\hat{\delta}^R$: *Predicted output of the process.* This variable is the expected future response of the actuator (based on a model) to a given set of control δ^C.

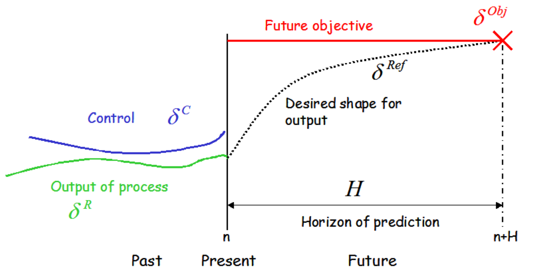


Fig. 5. Scheme of predictive control principle

Using these definitions, the predictive algorithm consists in computing the set of control values $\delta^C_{solution}$ to be applied during H to the actuator, which minimizes the quadratic error between the predicted output $\hat{\delta}^R$ and the desired reference shape δ^{Ref} (see equation (15)). This control set is based on a chosen control structure. The first value of the resulting control set constitutes the desired predictive term δ^{Pred}_{Traj}, actually applied to the actuator. Here, a longer horizon H has been chosen in order to increase the anticipative effect of the predictive algorithm, in such a way that the delays due to vehicle inertia could also be compensated, even if they are not explicitly entered into equation (15).

$$\begin{aligned} \delta^C_{solution} &= min_{\delta^C_{[n..n+n_H]}} \Sigma_{i=0}^{n_H} \left\{ \hat{\delta}^R_{[n+i]} - \delta^{Ref}_{[n+i]} \right\}^2 \\ \delta^{Pred}_{Traj} &= \delta^C_{solution,[n]} \end{aligned} \tag{15}$$

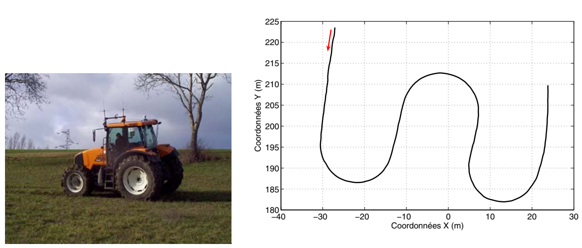
C. Global predictive control law

The solution extracted from the minimization problem is used instead of δ_{Traj}, as depicted on figure 4. As a result, the global control law, including compensation of sliding effects and predictive curvature control, is described as :

$$\delta = \delta^{Pred}_{Traj} + \delta_{Deviation} \tag{16}$$

VI. Actual path tracking results

A. Experimental equipment



(a) Farm tractor (b) Path with several half-turns

Fig. 6. Experimental vehicle and path to be followed on a level ground

The validation of the algorithm detailed in this paper has been achieved on a CLAAS Ares640 tractor, shown on figure 6(a). It is equipped with a Real Time Kinematic GPS (RTK-GPS), manufactured by Thales company. This sensor supplies a localization with respect to a reference station accurate to within $\pm$2cm. Using this material, two sets of trajectory tracking in a natural environment are reported below. The first test consists in the following of the path depicted on figure 6(b) on a level field at a speed of 8 km.H^{-1}. The second test is a straight line following on a sloping field (with a 15% slope) at the same speed.

B. Successive U-turns tracking

Control law (16) (with sliding compensation and prediction) has been implemented on the vehicle. The tracking error during the automatic following of path 6(b) has been recorded and is reported on figure 7 in dashed-dotted green line. This result is compared on the same figure with the results obtained with the control law (12) (with sliding accounted but without prediction, depicted in dotted red line) and with the control law designed under rolling without sliding assumption (without sliding accounted nor prediction, in black solid line).

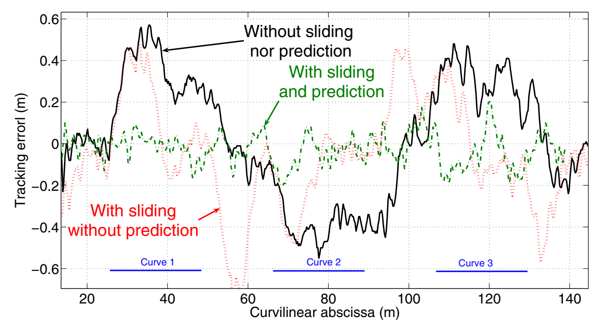


Fig. 7. Actual tracking results during successive U-turns

Firstly, this figure shows clearly the benefits of taking into account sliding effects : with control laws (12) and (16), the vehicle is able to reach a null tracking error, at least when sliding conditions are constant (during curves). It is not the case, with the control law neglecting sliding effects : the vehicle admits a 40cm lateral deviation during the curves.
Secondly, the interest in predictive action can be shown by comparing the red dotted line (without prediction) and the green dashed dotted line (with prediction). At each variation of the reference path curvature (beginning or end of a curve), the former admits important overshoots, when they have disappeared or been considerably reduced in the latter, thanks to the predictive action. As a result, during all path long, the vehicle with control law (16) is able to stay within a $\pm$15cm range, as expected in farm tasks. Moreover, the prediction algorithm enables to smooth the global behavior of the vehicle thanks to its anticipative effect (the variations in steering action are less sharp). Consequently, the comfort of the driver is increased and the solicitations of the mechanical components are decreased.

C. Straight line following on a slope

The second case where sliding cannot be neglected is the straight line following on a slope. The bad adherence conditions, associated to the weight of the vehicle, may lead to large errors. In this situation, the predictive algorithm has no use, since the curvature of the reference path is always null. As a consequence, only the results related to control law (12) are depicted on figure 8 in red dotted line and compared to the results obtained with classical control law (when sliding is not accounted) in black solid line.

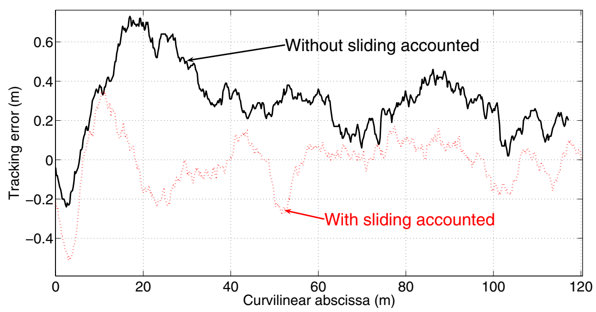


Fig. 8. Actual tracking results on a sloping field

It can be checked that the control law (12), which compensates sliding effects, is able to preserve a satisfactory accuracy, during almost all path long. On the contrary, with the classical control law, an important drift (roughly 30cm) is recorded. This lateral deviation is not exactly constant all path long. These variations are representative of the modifications of the adherence conditions due to punctual soil heterogeneities. The control law with sliding accounted is able to estimate these conditions on-line. Thanks to this adaptation, the vehicle can be kept close to the reference path, even if some residual overshoots can be observed, due to sharp adherence variations. Nevertheless, the value of these overshoots is quite limited. The results presented here have been confirmed by numerous tests, detailed more deeply in [4].

VII. Conclusion

This paper proposes a global approach for the path tracking problem dedicated to off-road vehicles, facing several phenomena (sliding of the wheels and actuator delays) induced by the motion on natural environments. The control strategy, based on the combination of several control principles, enables to significantly reduce, without any sensor addition, the loss of accuracy that can be recorded when a classical control law is implemented.

The problem of sliding effects has been explicitly taken into account by considering an extended kinematic model. An observer-like algorithm has first been designed to estimate on-line adherence conditions. Then, an adaptive non-linear control law, making use of these estimated parameters, has been proposed. Finally, the delays induced either by the actuators or the vehicle inertia have been compensated thanks to the addition of a Functional Predictive Control on the curvature servoing. The overall control law supplies the same accuracy than the one obtained when vehicles move on structured environment (such as car-like mobile robots moving on asphalt). This accuracy is moreover independent from the adherence conditions, the shape of the path to be followed or the soil configuration. The numerous experiments carried out with a farm tractor demonstrate that an $\pm 15cm$ accuracy can be reached in any situations. This meets the expectations in the considered application : automatic guidance of agricultural vehicles.

Currently, the weakest part of our global system is the perception device, consisting in a sole RTK-GPS sensor. This latter is settled on the top of the vehicle cabin, which is submitted to oscillations due to ground irregularities. This is penalizing for the accuracy of the measurement, and consequently depreciates the estimation algorithm. Furthermore, the use of an RTK-GPS is relatively expensive. We are currently working on the adaptation of these algorithms to another perception system, based on several sensors (Egnos GPS, gyrometer, inclinometer, etc). On one hand, it would be cheaper than the RTK-GPS on-boarded today, and on the other hand it could supply more information, improving the estimation of sliding effects.

Finally, the predictive algorithm can be improved by the explicit introduction of vehicle inertia inside the actuator model. The dynamic equations of the vehicle motion indeed allow to integrate the inertial effects into the predictive algorithm just as it has been done in this paper for the actuator model.

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