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Front Wheel Skidding Compensation System in Snow Ploughing

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Abstract:

Tractor based snow ploughing with front mounted blade is steered by using front steering wheels of the tractor, but in slippery conditions the front wheels tend to slide sideways remarkably. Normally the driver compensates the skidding by steering the front wheels to opposite direction. If the skidding is not compensated, the tractor eventually drifts to the oncoming lane or spins around.

In this paper, an automated method to estimate and to compensate the slipping is proposed. The estimation is based on a simple kinematic model of tractor-plough system with some additions. The estimation relies on an angular velocity measurement (Fiber Optic Gyro) of the heading that is able to recognize small changes of the driving direction that indicates the presence of slipping. The indicated slipping is compensated by a feedback law in a drive-by-wire steering system.

The results show that the proposed system is able to measure fairly realistic values for the slipping angle and to compensate the slipping. Driver experiences confirm that less steering corrections are needed to keep the tractor on the desired trajectory.

Keywords: Electronic Stability Control, Drive-By-Wire, State Feedback Control, Yaw rate control

1. INTRODUCTION

In northern Europe winter road maintenance is carried out by using either trucks equipped with blades or snow plough to remove snow from the road or by using agricultural tractors. Winter time road maintenance is a common seasonal business operation for many farmers in rural area, as the tractors are not used for field operations. This research is focused on tractor based road maintenance, carried out by a tractor with a front mounted snow plough, typically ploughing the snow to the right in driving direction.

The snow ploughing in slippery conditions is naturally unstable process. The ploughed snow produces forces to the plough that cause turning moment of the tractor and eventually skidding of the wheels, if there is not enough friction. At the present, the driver compensates the skidding of the front wheel manually. If the skidding is not compensated, the tractor may drift to the oncoming lane or spin around. Amount of slip depends not only road surface but also on rubber material of front wheels and if spikes and/or tire chains are used.

The slipping or the lateral motion of the vehicle can be automatically controlled either by directly controlling the steer angle or by controlling the longitudinal force generated at each wheel. The latter is the most common one in commercial vehicle stability control systems (Rajamani, 2006). However, many studies representing the former are reported, e.g Lee (1990) and Hosaka and Murakami (2004).

Before the control, the undesired lateral motion due to slipping has to be estimated. In literature (Wong, 2008), the tyre slipping is usually modelled by introducing a slip angle

which is added to the steer angle. Wang and Low (2008) presented models for different kinds of wheeled mobile robots with slipping angle introduced. Lenain et al. (2005) have studied slipping especially with tractors in field conditions. Daily and Bevely (2004) used GPS to estimate the lateral movement and tyre slipping.

Despite a large number of studies done in this field, commercial tractors still does not have stability control systems. Stability control in tractor is challenging due to varying parameters in dynamic behaviour. Especially in snow ploughing, the properties of the road surface can vary significantly within a short distance. One wheel may be on the ice and the other on firm snow and the condition can change to opposite quickly. Also the force produced by snow can vary widely. There are also other forces that affect the stability of the vehicle.



Figure 1. Snow plough was mounted into the front hitch of the tractor and the positioning devices was mounted on the roof of the tractor.

2. TEST CONFIGURATION

The test configuration consisted of a standard tractor, a snow plough and positioning devices (Figure 1). The tractor was Valtra T132 that was equipped with an ISO 11783 compatible Tractor ECU implementing Class 3 commands with the guidance option. The snow plough was a swivelling front-mounted plough manufactured by Arctic Machine Oy. In the research, a prototype ISO 11783 implement ECU for the plough was developed with ISO 11783 development tool chain proposed by Oksanen et al. (2011).

The positioning devices consisted of a Fiber Optic Gyro (FOG), a Inertial Measurement Unit (IMU) and a Real Time Kinematic (RTK) – GPS. The FOG was KVH DSP-3000, IMU was Inertial-Link 3DM-GX2 and RTK-GPS was Trimble 5700. The devices were connected together and the information was merged with an embedded controller. The merged information was sent to the CAN-bus using standard NMEA 2000 messages.

Normally commercial Electronic Stability Control (ESC) systems use the wheel brakes to control the under and over steering of the vehicle. The ISO 11783 T-ECU with Class 3 allows implements to control the hydraulic valves, PTO, rear-hitch, steering and speed (ISO 11783-9:2002). Unfortunately, it is not possible to control the breaks through a standard ISO 11783 network. Therefore in this project it was chosen to compensate the skidding by controlling the steering automatically by drive-by-wire manner. For the safety reasons, the original steering wheel of the tractor was not modified. Instead, an additional smaller steering wheel was added that was used in drive-by-wire system.

3. METHODS

The ploughed snow produces forces to the plough that cause turning moment (Figure 2). The difference of the forces on different sides of the plough can be measured from the pressure of the hydraulic cylinders. However, from these measurements it is not possible to directly calculate the skidding of the front or rear wheels due to changes of road surface and other external forces.

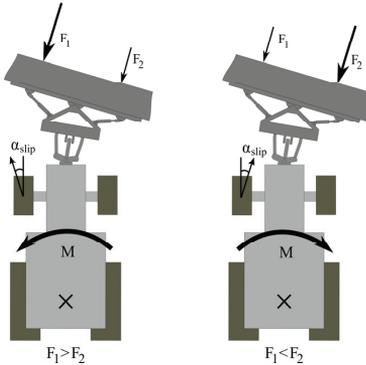


Figure 2. Forces on the plough causes turning moment in either direction.

In this research the undesired skidding behaviour is assumed to occur in front wheels. The skidding is estimated indirectly based on that assumption by using an Extended Kalman Filter (EKF). Finally the skidding is compensated by using state feedback law.

3.1 State estimation

The model in the state estimation is kept as simple as possible in order to limit number of estimated parameters online.

The skidding of the front wheels is estimated by using an additional slipping angle of the front wheels:

$$\alpha_{eff}(t_k) = \alpha_t(t_k) + \alpha_{slip}(t_k) \quad (1)$$

where α_{eff} is the effective steering angle, α_t is the true steering angle and α_{slip} is the slipping angle at the time t_k (Figure 3).

Also, the dynamic behaviour of the front wheel steering system is modelled by first order low-pass filter:

$$\alpha_t(t_{k+1}) = k_\alpha \alpha_t(t_k) + (1 - k_\alpha) \alpha_d(t_k) \quad (2)$$

where α_d is the commanded steering angle and k_α is a dynamic parameter.

Otherwise the kinematic model is a standard bicycle model:

$$\begin{bmatrix} x_R(t_{k+1}) \\ y_R(t_{k+1}) \\ \theta(t_{k+1}) \end{bmatrix} = \begin{bmatrix} x_R(t_k) + v_t(t_k) \cos \theta(t_k) T \\ y_R(t_k) + v_t(t_k) \sin \theta(t_k) T \\ \theta(t_k) + v_t(t_k) \frac{\tan \alpha_{eff}(t_k) T}{a} \end{bmatrix} \quad (3)$$

where (x_R, y_R) is the position of the rear axle, θ is the heading of the tractor, v_t is the speed of the tractor, a is the wheelbase and T is the discretization time.

In addition to the slipping angle, the estimated states of the system are the tractor position, heading, speed and the true steering angle:

$$x(t_k) = [x_R \ y_R \ \theta \ v_t \ \alpha_t \ \alpha_{slip}]^T \quad (4)$$

Because the speed is manually controlled and the behaviour of the slipping angle is unknown, these are assumed to be unknown constant random variables disturbed by white noise in the state estimation.

The measurements in the state estimation are the same as the states with the exception that the slipping angle is not directly measurable and the measurements are delayed:

$$h(\hat{x}(t_k)) = \begin{bmatrix} \hat{x}_R(t_{k-\tau(x_R)}) \\ \hat{y}_R(t_{k-\tau(y_R)}) \\ \hat{\theta}(t_{k-\tau(\theta)}) \\ \hat{v}_t(t_{k-\tau(v_t)}) \\ \hat{\alpha}_t(t_{k-\tau(\alpha_t)}) \end{bmatrix} \quad (5)$$

where $\tau(\cdot)$ is the delay time of the measurement .

In this research, the discretization time T of the state estimator was 10 ms, but the measurements' intervals were longer and the measurements were not synchronized. For that reason, the measurement function was modified in every time step so that only the measurements that were read from the CAN-bus after the previous estimation step were used.

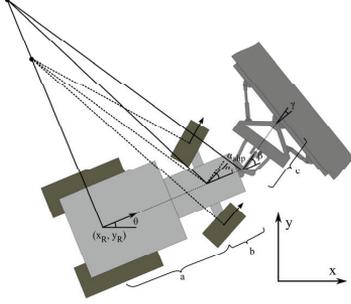


Figure 3. States and parameters of the estimation model.

3.2 Heading estimation

The heading is the most important measurement in the slipping estimation. For the heading measurement, the Fiber Optic Gyro (FOG) is used. The raw measurement of the Gyro is first processed before using it in the EKF.

The FOG that is used, measures the rotation around the z-axis of the vehicle. Because the vehicle moves on uneven surfaces, the measured rotation is not the same as the rotation around the z-axis of the ground. The *Yaw* angle or the *Course* that is wanted to be measured in general is the rotation around the z-axis of the ground. The accuracy of the Gyro also necessitates that the earth rotation must be compensated.

The earth's rotation rate, which interferes with the measurement, is first calculated. The dot product between earth rotation axis and the z-axis of the vehicle is calculated:

$$Z_{earth} \cdot Z_{vehicle} = \begin{bmatrix} \cos(Lat) \\ 0 \\ \sin(Lon) \end{bmatrix} \cdot \begin{bmatrix} cR * cY * sP + sR * sY \\ cR * sY * sP - sR * cY \\ cP * cR \end{bmatrix} \quad (6)$$

, where

$$\begin{aligned} cP &= \cos(Pitch), sP = \sin(Pitch), \\ cR &= \cos(Roll), sR = \sin(Roll), \\ cY &= \cos(Yaw), sY = \sin(Yaw). \end{aligned} \quad (7)$$

The *Yaw* is the rotation around the ground z-axis, with the positive direction up. The *Roll* and the *Pitch* are the inclination angles around the x- and y-axis of the Vehicle respectively. Now, the earth rotation that the Gyro measures can be calculated according to equation:

$$\omega_{earth} = -7.29212 * 10^{-9} * (Z_{earth} \cdot Z_{vehicle}) \quad (8)$$

The second step is to correct the measurement error due to the inclination. The rotation rate that the Gyro measure is projected to the ground after the removing of the measured earth rotation and the measured pitch rotation:

$$\omega_{Yaw} = \frac{\omega_{gyro} - \omega_{earth} + \omega_{Pitch} * sR}{cR * cP} \quad (9)$$

where ω_{Yaw} is the projected measurement of the rotation rate around the z-axis of the ground, ω_{gyro} is the raw rotation rate measured by the gyro and ω_{Pitch} is the Pitch angle rate.

The ω_{Yaw} is now the rotation rate around the ground z-axis, from where the Yaw angle can be calculated:

$$Y = \tilde{Y} + Y_{bias} = \int \omega_{Yaw} + Y_{bias} \quad (10)$$

where Y_{bias} corrects the zero heading to the north and \tilde{Y} is the uncorrected Yaw angle calculated by integrating the ω_{Yaw} .

The bias Y_{bias} is found by using the *Compass* measurement. Since there is no magnetometer included, the compass angle is obtained from the RTK-GPS as a course. The GPS course measurement is not accurate for slow driving speeds. For that reason, there are three different methods for different driving speeds (the driving speed is read from the CAN-bus).

When the driving speed is above 0.5 m/s, the heading measurement from the RTK-GPS is considered to be sufficiently accurate. First, the momentary difference between the uncorrected Yaw angle and the measured Compass angle is calculated:

$$\Delta\tilde{Y} = \text{mod}(\tilde{Y}(t_{k-\tau(\tilde{Y})}) - Y_{compass}, 2\pi) \quad (11)$$

where $\tilde{Y}(t_{k-\tau(\tilde{Y})})$ is the delayed Yaw angle, because the Compass angle is 250 ms delayed due to the calculation method of the RTK-GPS. After that, the bias is found by filtering the momentary difference values:

$$Y_{bias}(t_{k+1}) = \text{mod}(Y_{bias}(t_k) + W * \Delta Y_{bias}, 2\pi) \quad (12)$$

where W is the filter weight and ΔY_{bias} is calculated according to the equation:

$$\Delta Y_{bias} = \text{mod}(\Delta\tilde{Y} - Y_{bias}(t_k) + \pi, 2\pi) - \pi \quad (13)$$

The weight W is modified on every measurement update step according to the equation:

$$W = \max\left(\frac{1}{1/W + 1}, W_{min}\right) \quad (14)$$

where the W_{min} is the minimum weight of the filter.

When the driving speed is below 0.5 m/s but above 0.05 m/s, there is no accurate absolute heading measurement. In this case, only the filter weight is modified according to equation:

$$W = \min\left(\frac{1}{1/W - 1}, 0.5\right) \quad (15)$$

When the driving speed is below 0.05 m/s, the heading is considered to be constant. In this case, the bias is found by keeping the heading constant:

$$Y_{bias}(t_{k+1}) = \tilde{Y}(t_k) - Y(t_k) \quad (16)$$

With the adaptive filtering, an approximation for the bias is obtained from the first measurements but the noise of the GPS is reduced on later measurements.

3.3 Control law

In the automatic control system, the user gives the actual effective steering angle, which is the sum of the steering angle of the wheels and the slipping angle:

$$\alpha_{steer}(t_k) = \alpha_{eff}(t_k) = \alpha_t(t_k) + \alpha_{slip}(t_k) \quad (17)$$

where α_{steer} is the user controlled steering angle. According to system equations (Eq. 2) the controlled steering angle of the wheels should then be direct state feedback:

$$\alpha_d(t_k) = \alpha_{steer}(t_k) - k_1 \alpha_{slip}(t_k) \quad (18)$$

where k_1 is a tuning parameter.

If the slipping cannot be compensated by turning the wheels over the desired steering angle i.e. the steering angle is limited or the slipping angle is over the threshold, the force caused by the ploughing is attempted to reduce by lifting the plough and reducing the ploughing angle.

The user controls also the plough by setting the desired sideshift from the centre driving line and by setting the desired ploughing angle. The rotation angle of the front hitch is calculated from the desired sideshift of the plough ζ and the steering angle α_{steer} according to the equation:

$$\beta(t_k) = \text{asin}\left(\frac{\zeta(t_k)}{c}\right) + k_2 \alpha_{steer}(t_k) \quad (19)$$

where k_2 is a tuning parameter and c is the distance between the front hitch rotation point and the plough. Parameter k_2 can be calculated so that the lateral movement caused by the curve is approximately compensated in small angles:

$$k_2 = \frac{a + b + c}{a} \quad (20)$$

where b is the distance between the front axle and the front hitch rotation point.

Equally, the ploughing angle is calculated from the desired ploughing angle γ_{steer} compensated by the sideshift:

$$\gamma(t_k) = \gamma_{steer}(t_k) - \text{asin}\left(\frac{\zeta(t_k)}{c}\right) \quad (21)$$

4. RESULTS AND DISCUSSION

The system was initially tested and tuned in the simulator, where the slipping and other parameters are given based on early identification tests. Finally, testing and tuning was performed on the test road with actual equipment.

The discretization time of the models and the time step of the control system was:

$$T = 10 \text{ [ms]}$$

The dynamic parameter of the steering system was:

$$k_\alpha = 0.90$$

which is equal to 90 ms time constant within continuous time first order system. The dynamic parameter depends not only on the steering mechanics and the control system related to that, but also on the friction of the road surface and tires.

The standard deviations of the states (the diagonal of the Q matrix) in the EKF were:

$$\begin{aligned} \sigma(x_R) &= \sigma(y_R) = 2 * 10^{-4} \text{ [m]} \\ \sigma(\theta) &= 2 * 10^{-5} \text{ [rad]} \\ \sigma(v_t) &= 0.1 \text{ [m/s]} \\ \sigma(\alpha_t) &= 9 * 10^{-3} \text{ [rad]} \\ \sigma(\alpha_{slip}) &= 5 * 10^{-5} \text{ [rad]} \end{aligned}$$

The crosscovariances of the states were set to zero.

The standard deviations of the measurements were:

$$\begin{aligned} \sigma(\theta_{meas}) &= 3 * 10^{-3} \text{ [rad]} \\ \sigma(v_{meas}) &= 0.022 \text{ [m/s]} \\ \sigma(\alpha_{meas}) &= 0.05 \text{ [rad]} \end{aligned}$$

The covariance and the crosscovariance of the measured position were obtained directly from the accuracy estimation of the RTK-GPS device. All other crosscovariances of the measurements were also set to zero.

The delay times of the measurements were:

$$\begin{aligned} \tau(x_R) &= \tau(y_R) = 200 \text{ [ms]} \\ \tau(\theta) &= 310 \text{ [ms]} \\ \tau(v_t) &= \tau(\alpha_t) = 160 \text{ [ms]} \end{aligned}$$

The functionality of the system was first tested in the simulator. In the simulator test, the wanted effective steering angle of the front wheels was a sine wave with 14 deg amplitude and 13 sec wave length. The wanted effective, the measured and the estimated steering angles together with the estimated slipping angle and the deviation estimate of the slipping angle are depicted in Figure 4 and in Figure 5.

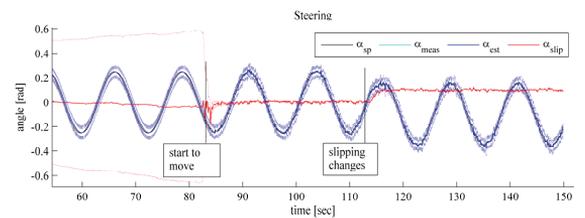


Figure 4. Slipping estimation in simulator. Tractor started to move at the time position 82 and the slipping changes from 0 to 0.1 rad at the time 112. Thin lines represent the covariance estimates of the corresponding states.

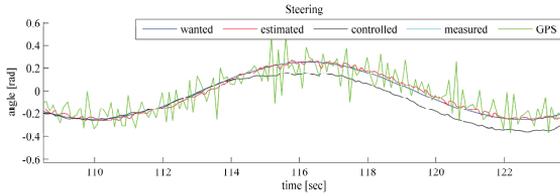


Figure 5. The desired, estimated and controlled steering angles when the slipping changes.

In the first 82 seconds of the test, the tractor was stationary and the filter was stabilized. After that, the speed of the tractor was kept constant 10 km/h (2.8 m/s). After 112 seconds, the slipping angle was changed from 0 to 5.7 deg simulating stepwise change in road conditions. As it can be seen, the slipping estimation is uncertain before the tractor moves. Quickly after the tractor starts to move, the deviation estimate decreases significantly. After the simulated slipping changes, also the estimate follows the true slipping. The estimated effective steering angle continues to follow the wanted effective steering angle because of the feedback law.

After the functionality was proven in the simulator, the system was tested in the real environment. Several different tests were performed of which two are reported in here. The test road that used in the tests was covered with solid ice and no spikes were used in the front wheels.

The first test corresponds to normal snow ploughing conditions (Figure 6 and Figure 7). After 70 seconds, the snow is pushed against the snow windrow on the right side of the road. There are also two sharper turns at the time between 88 to 90 seconds and at the time between 95 to 110 seconds. During the test, the speed varied between 3 m/s to 4 m/s (10 km/h to 15 km/h). As it can be seen from the Figure 6, the effect of the slipping is quite significant. The difference between the wanted effective steering angle and the true controlled steering angle, i.e. slipping, is 12 degrees at most and about 5 degrees on average.

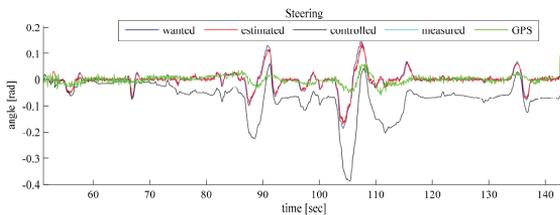


Figure 6. The wanted, estimated and controlled steering angles in normal ploughing situation.

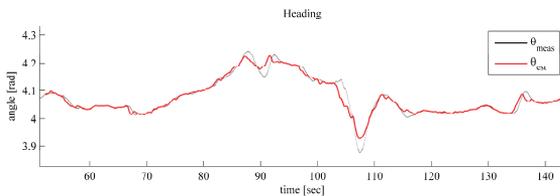


Figure 7. The estimated and measurement heading angle in normal ploughing situation.

The second reported test in the real environment corresponds to the case where different sizes of snowdrifts are situated on the road. The tractor is wanted to go straight, but the force caused by the snowdrift turns it. The time instant when the plough hits the snowdrift is depicted in Figure 8 and in Figure 9. In Figure 8 are the steering angle and the estimated slipping angle. In Figure 9 is the moment of the force about the turning point of the plough calculated from the pressure measurements of the hydraulic cylinders. As can be seen, the force produced by the snow corresponds to the slipping. The speed was kept approximately constant 4 m/s (15 km/h) during the test.

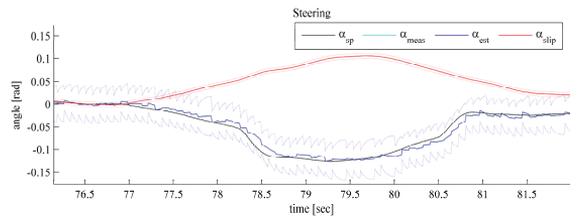


Figure 8. The steering angle and the slipping estimate when the plough hits the snowdrift. Thin lines represent the covariance estimates of the corresponding states.

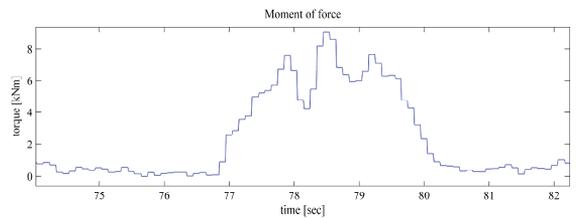


Figure 9. The moment of force about the rotation point of the plough when the plough hits the snowdrift.

These results show that the system is able to measure quite realistic slipping angles. It is hard to say whether the estimated slipping angles are absolutely correct in the real environment as no other measurement system was used as a reference. However, the functionality of the estimator is proven by using the simulator where exactly same model of the system is used. Moreover, the user experiences with the system confirm that the system is able to compensate the slipping and less steering correction is needed as can be seen from Figure 6.

The speed used in the test was well below the speeds that could be used in snow ploughing on highways. There are two reasons for that. The first one is that the steering actuator is not allowed to work in steer-by-wire mode with high speeds i.e. ISO 11783 Class 3 steering commands could not be used with higher speeds. Another one is that the kinematic model is not any more valid with higher speeds than 4.5 m/s (Werner et.al. 2012). Dynamic model should be derived and more experiments should be done with higher speeds. Also, the feedback to the driver within steer-by-wire system should be investigated.

5. CONCLUSIONS

In this research, an automated method to estimate and compensate the slipping in snow ploughing is proposed. The estimation is based on a simple kinematic model with added slipping and steering dynamics. The state of the model is estimated from RTK-GPS and Fiber Optic Gyro measurements by using an Extended Kalman Filter. The indicated slipping is compensated by a feedback law in a drive-by-wire steering system.

The results show that the proposed system is able to estimate plausible slipping angles and to compensate the slipping. The simulator results show that a rather simple model is sufficient if the slipping is occurring only in front wheels. Driver experiences confirm that less steering corrections are needed to keep the tractor on the desired trajectory.

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