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Research on Steering Balance Control of Two-wheeled Vehicles Under Large Time Delay

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Abstract: Time lag is very common in real control systems and is more important in driverless intelligent single rutted two-wheeled vehicles. Time lag can lead to slower response of the control system to external changes, which can reduce the stability or even imbalance of the vehicle. Traditional control methods may not be directly applicable to systems with time lag, so special control strategies are needed to deal with the effects of time lag. In this paper, we build a non-time-lagged steering balance controller by analysing the dynamics model of an unmanned intelligent single rutted two-wheeled vehicle, and then control the time lag by using the improved Smith predictor, and finally verify the steering balance of the method under large time lag by MATLAB simulation.

Keywords: Steering balance; Improved Smith; Time-delay control; Single rutted two-wheeler.

1. Introduction

Autonomous driving will be a trend in the future of driving and is what we have been working towards now. Compared with driverless cars, two-wheelers are more flexible, and can carry out transport, reconnaissance and other work through narrow spaces such as alleyways. And the first and foremost groundwork for intelligent driving of two-wheeled vehicles is the balance problem. In fact, the current balance control methods are roughly divided into two categories, steering balance [1-2] and gyroscope balance. Gyroscope balance has the characteristics of strong anti-interference and strong stability. However, according to the current research status, there is no latest research result that can solve the large angle steering problem of gyrobalance. The steering balance is not such a problem, compared with the gyro balance, the steering balance is more flexible, and can fully play the mobility and flexibility of the two-wheeled vehicle. As we all know, steering balance depends on the reflection speed of the steering mechanism, in high speed or complex road surface, the negative impact of the time lag of the steering mechanism will be doubled with the speed and complexity of the road surface, and may even lead to the loss of control of the vehicle body and destruction.

There are fewer studies on the control of time lag systems for single rutted two-wheeled vehicles, but the phenomenon of time lag in control exists in various fields of application, where transmission delays may occur when signals are transmitted from one place to another. Sensors are used to sense the environment or acquire input signals. However, the sensors themselves may have response delays. Classic examples are heating of extruded plastics in an extruder, temperature control in a boiler room, process control of chemical reactions in a chemical plant, and so on.

Consider a system with a time lag where the time lag is τ . The transfer function $G(s)$ can represent the effect of time lag by introducing an additional factor $e^{-\tau s}$ For the analysis and design of control systems, the time lag transfer function can be used to analyse the stability, frequency response and control performance of the system. However, the presence of time lag makes the poles of the controlled system to be in the right half plane of the complex plane, which leads to system divergence and loss of control.

Smith prognostic control is widely used in time-lag systems, and its principle is mainly to eliminate the feedback path time lag and reduce the negative effects of the time lag on the whole system, such as oscillation and divergence. However, in actual time-lag systems, the condition that traditional Smith control requires an accurate model is not always achievable. The index for dealing with time lag decreases with the external disturbance or the error of building the model itself, which in turn generates some methods to improve the Smith control, such as anti-interference type, filtering type, adaptive type, and so on. By combining with the grey wolf algorithm, it is proposed to improve the Smith predictor to make the anti-disturbance enhancement [3]. Feedforward and feedback controllers are designed to improve the tracking expectation without overshooting as well as eliminating unwanted disturbances [4]. The Adding controllers and filters to the feedback path is used to improve model parameter mismatch [5]. Fuzzy PID with Smith's Using fuzzy PID in combination with Smith predictors to increase the adaptivity and robustness of the control system[6] .The EHA is optimised according to a genetic algorithm to obtain the EHA. The optimal time lag feedback coefficients and time lags of the EHA suspension system are obtained by genetic algorithm optimisation, which is used as the optimal time lag feedback model to design the adaptive Smith feedback time lag control strategy[7].The adaptive Smith feedback time lag control strategy is designed using the optimal time lag feedback model. The Smith predictor is improved to eliminate the dead time and prevent overshoot [8, 9]. The PD is combined with Smith to design an adaptive Smith feedback time lag control strategy. The feedforward and feedback paths are controlled using a combination of PD and Smith, which improves the response time and ensures a robust closed-loop performance [10] . In this paper, a conventional controller is designed by analysing a second-order nonlinear single rutted two-wheeler dynamics model, which is then improved to a steering balance controller under large time lag conditions by using a filter in combination with Smith's method.

In Section II of this paper, we describe in detail the modelling process of the self-balancing single rutted two-wheeler. Section III designs the steering balance controller for the non-time-lag system. Section IV designs the steering balance controller for the time-lag model in conjunction with the improved Smith predictor. Section V will verify the effectiveness and performance advantages of the proposed method through MATLAB simulation experiments. Finally, Section VI concludes and outlooks the whole study.

2. Two-wheeled Vehicle Dynamics Modelling

2.1. Modelling two-wheeled vehicles

The information about the parameters such as mass, centre of mass height, wheel radius etc. as shown in the later chapters are selected for this study for the two wheelers. are shown in Table 1 below. All the data in this section is collected through an actual two-wheeler platform.

Table 1. Parameters of two-wheeled vehicles

Parameter name	notation	magnitude	unit
total mass	m	256	kg
wheelbases	l	1.6	m
Distance from the centre of mass to the rear axle	l_r	0.625	m
centre of gravity	h	0.46	m
Moment of inertia around the cross roller	I	35	$kg \cdot m^2$
Angle between the front fork and the normal plane of the front wheel axis	r	29.3	$^\circ$
Fork Offset Distance	d_1	0.025	m
Front steering mass	m_f	30	kg
Front steering centre of mass height	h_f	0.540	m
Front steering centre of gravity to steering axis distance	d_f	0.02	m
Front wheel rolling radius	r_f	0.28	m
Rear wheel rolling radius	r_r	0.29	m
gravitational acceleration	g	9.8	m/s^2
Roll angle/roll angle	θ	-	$^\circ$
steering angle	δ	-	$^\circ$
Motion steering angle	δ_R	-	$^\circ$
Front wheel speed	v_f	-	m/s
Rear wheel speed	v_r	-	m/s
Radius of rotation of the front wheel around the instantaneous centre	R_f	-	m
Radius of rotation of the rear wheel around the instantaneous centre	R_r	-	m
Centrifugal acceleration of the front wheel around the instantaneous centre	a_{cf}	-	m/s^2
Centrifugal acceleration of the rear wheel around the instantaneous centre	a_{cr}	-	m/s^2

A conventional two-wheeler is a vehicle structure in which the front wheels control the steering and the rear wheels act as the drive. In this model it is assumed that the front and rear wheels are in contact with the ground as a single point and that the wheels do not slip sideways and that all angles are small. The ground coordinate system O-XYZ is established and the direction of the arrows on the schematic indicates the positive direction of these angles. Points P1 and P2 are the contact points between the centre of the front wheel and the centre of the rear wheel and the ground, respectively, and contact point P3 is the point where the extension of the fork steering intersects the ground. As shown in Fig. 1, the roll angle and steering angle are generated by the turning of the vehicle body, as well as by the ground road conditions and other conditions.

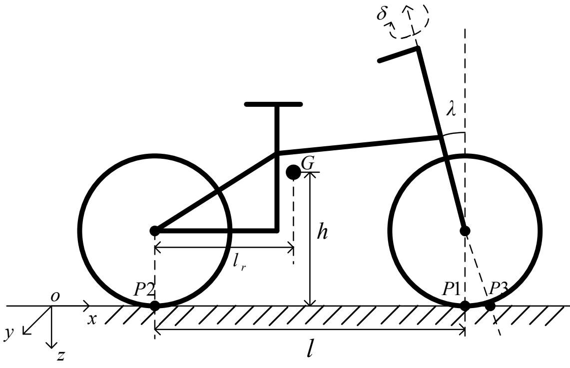

Figure 1. Side view of a two-wheeled vehicle

Since the fork swivel does not pass through the centre of the front wheel, it generates a transverse roll moment when steering, as shown in Fig. 2. When the tyre rotates around the steering axis during steering, the handlebar's point of contact with the ground, P1, moves and generates a moment in the direction of the transverse roll, which is in the same direction as the gravitational moment.

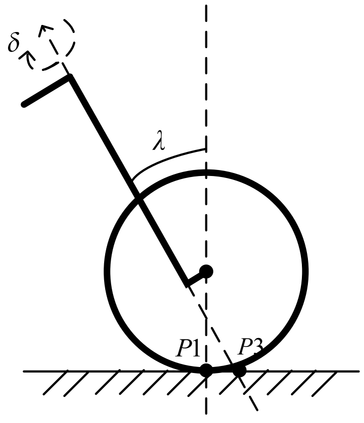

Figure 2. Front steering system structure

In this paper, the moment force analysis and dynamic modelling of a single rutted two-wheeled vehicle is carried out by the Newton-Euler method, and the following equation for the total moment of the two-wheeled vehicle around the x-axis is obtained:

$$M_{all} = M_G + M_{steering} \tag{1}$$

The moment of inertia with respect to point O is obtained from Newton's second law:

$$M_{all} = \left(I_O + mh^2\right)\ddot{\theta} \tag{2}$$

When the vehicle body is tilted, the change in the cross-roll angle generates a gravitational moment:

$$M_G = mgh\sin\theta \tag{3}$$

The vehicle body generates centrifugal acceleration and hence centrifugal moments due to cornering:

$$M_{steering} = \frac{1}{l}mv_r^2 h\tan\delta_R\cos\theta \tag{4}$$

The cross-roll moment generated by the fork pivot not passing through the centre of the front wheel:

$$M_f = mg\frac{l_r}{l}d_1\delta_R \tag{5}$$

Then the total moment equation:

$$\left(I_O + mh^2\right)\ddot{\theta} = mgh\sin\theta - \frac{1}{l}mv_r^2 h\tan\delta_R\cos\theta + mg\frac{l_r}{l}d_1\delta_R \tag{6}$$

3. PID Controller Design

Design a traditional PID controller, two-wheeled vehicle desired tilt angle as the target value set by the control system, the actual vehicle state information obtained through the sensor as the system model state feedback, the control quantity is the steering angle, design the following steering balance controller to achieve steering balance. The error between the roll angle feedback and the target roll angle is adjusted by the PID controller, at this time the controller is

$$u = K_P e + K_I \int e dt + K_D \dot{e} \tag{7}$$

Where, e is the roll angle error, $e = \theta_{target} - \theta$, θ_{target} is the desired roll angle, u is the control quantity output to the steering mechanism by the controller, K_P K_I K_D is the proportional, integral and differential link.

4. Improved Smith Predictive Compensation Control

The Smith's prognostic control algorithm proposes a solution to the problem of a closed-loop time-lag system by compensating a prognostic model in parallel to eliminate the effect of time lag on the regulation process. However, in practice, the system model is often affected by factors such as parameter uncertainty, external disturbances and measurement noise, leading to discrepancies between the predictive model and the true model. Such model errors may lead to fluctuations or instability in the output of the predictor. In order to solve this problem, this paper uses a filter in combination with the Smith predictor to regulate the mismatch between the reference model and the real model. The control block diagram of the Smith system is shown in Fig. 3.

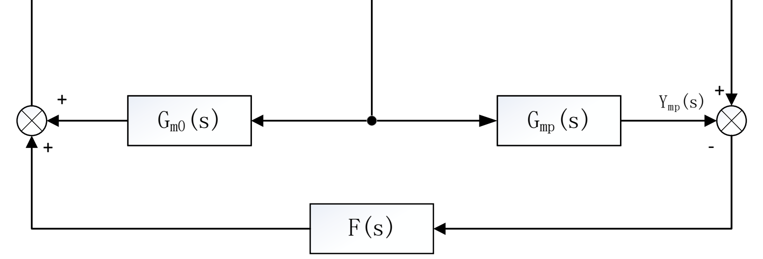

Figure 3. Smith control block diagram

where $G_c(s)$ is the controller, $G_p(s)$ is the generalised controlled object and $G_{mp}(s)=G_m(s)e^{\tau s}$.

The closed-loop transfer function of the control system is:

$$\frac{Y(s)}{R(s)}=\frac{G_c(s)G_p(s)e^{-\tau s}}{1+G_c(s)G_m(s)+G_c(s)(G_p(s)e^{-\tau s}-G_m(s)e^{-\tau s})F(s)} \tag{8}$$

$F(s)$ can be regarded as a low-pass filter, which has a buffering effect on the mismatched prediction model. The error between the actual output of the control system and the output of the predictor is buffered by the inertial link and input to the controller, which reduces the effect of the mismatched prediction model on the stability of the system.

5. System Simulation and Result Analysis

5.1. Simulation parameters

In order to analyse the stability of the self-balancing motorbike, the roll dynamics model obtained from Eq. (6) is linearised by a Taylor series expansion near the equilibrium point, which can be approximated by $\sin\theta=\theta$, , and $\cos\theta=1$ $\tan\theta=\theta$ since the roll angle θ and steering angle δ of the vehicle are relatively small at this point.

The single rutted two-wheeler transfer function can be obtained by Laplace transform, taking the steering angle as the input to the system and the roll angle as the output of the system, and its mathematical model is represented as:

$$G(s)=\frac{\theta(s)}{\delta(s)}=\frac{1}{(I_o+mh^2)s^2-mgh}\left(-\frac{1}{l}mv^2h\cos\lambda+mg\frac{l_r}{l}d_1\right) \tag{9}$$

According to the terminal value theorem one can get its steady state value when s→0

$$K_0=\frac{\theta_0}{\delta_0}=\lim_{s\to 0}G(s)=\frac{v^2h\cos\lambda-gl_rd_1}{ghl} \tag{10}$$

If the forward speed v is constant, then K_0 is also constant. The steering control method only complies with the physical property that the centrifugal and gravitational moments of the vehicle are in opposite directions if $K_0>0$ is satisfied, which means that the forward speed v should be satisfied:

$$v>\sqrt{\frac{gl_rd_1}{h\cos\lambda}} \tag{11}$$

In order to achieve a stable turn, the speed should be made as large as possible above this critical value, substituting the parameter values, the steering control should be balanced and the speed $v>0.64m/s$.

5.2. Simulation results

For numerical simulation, PID controller, PID combined with Smith predictive control are designed and second order two-wheeler system model is loaded in the simulation environment respectively. Then, comparisons are made by setting the same working conditions and reference signals. The initial parameter vehicle speed is 1m/s when the initial state cross roll angle is 2 degrees, and the expectation is to reach 0 degrees.

Figure 4 shows that the conventional PID feedback steering balance control system in the process of adjusting the steering angle to reach the desired equilibrium point, due to the pure time delay characteristics are not compensated for, the roll angle in about 6s before reaching the equilibrium, the curve there is an obvious oscillation. The conventional Smith feedback steering balance control system can compensate for the pure time delay characteristic in the rolling process, and in the ideal state, the reference model and the actual model error is 0, to reduce the oscillation of the curve, and the desired roll angle is reached after 5 s. The conventional Smith feedback steering balance control system can compensate for the pure time delay characteristic in the rolling process.

In the previous scenario, when the time lag of the system is 0.025 sec, the effect of time lag on PID control is relatively limited. However, when the time lag of the system increases to 0.18 s, the traditional PID control strategy cannot achieve stable convergence of the system, as shown in Fig. 5. In this case, the control strategy presented in this paper will have more significant advantages and can cope with the effect of large time lag on the system performance more effectively.

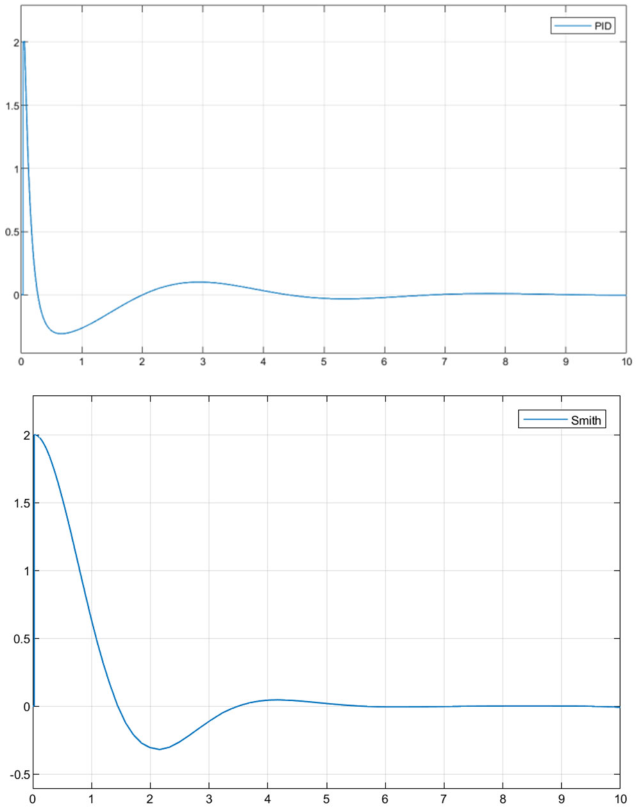

Figure 4. PID, Smith control strategies under time delay condition of 0.025s respectively

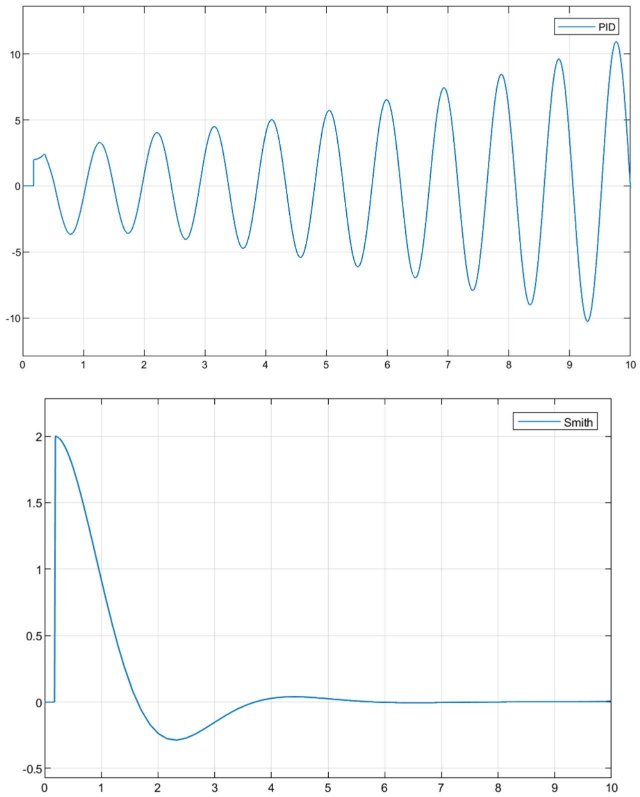

Figure 5. PID, Smith control strategies under time delay condition of 0.18s respectively

6. Summary

In the study of this paper, simulation experiments are used to further verify the effectiveness of the filter combined with the Smith predictor in solving the two-wheeled vehicle time lag problem. Through the reasonable design of the control strategy, the effect of large time lag on the system performance is successfully mitigated, resulting in a significant improvement in the steering balance of the two-wheeled vehicle. However, it is worth pointing out that, in terms of the steering balance algorithm, although a certain effect has been achieved, there is still room for further improvement. When facing nonlinear second-order systems, we can consider more complex control strategies and algorithms to better adapt to the complex characteristics of real systems.

In summary, this study provides a useful exploration and experimental basis for the steering balance control of two-wheeled vehicles, however, there are still some challenges and directions that need further in-depth research. It is believed that with the continuous progress of technology and the continuous improvement of theory, these problems will be better solved, and a solid foundation will be laid for the realisation of a safer, stable and efficient vehicle control system.

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