

Steering Angle Balance Control Method for Rider-Less Bicycle Based on ADAMS

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Abstract Bicycle is a typical unstable system. Although it is partly similar to the inverted pendulum systems, the complex mechanical characteristics cause that it is difficult to be analyzed and controlled. In this paper, Constrained Lagrangian method is used to building a rider-less bicycle dynamic model. Through the analysis of the dynamic model, the control system is designed to be divided into two parts: the forward control and the balance control. The rear wheel gets a stable speed by controlling the torque of the motor. Steering Angle Balance Control method is designed to keep the system balance. Use ADAMS to simulate a rider-less bicycle control system. The simulation shows that using Steering Angle Balance Control method can balance the bicycle to run forward and swerve stably.

Keywords Rider-less bicycle · Constrained lagrangian method · Steering angle balance control · ADAMS

1 Introduction

The rider-less bicycle has advantages like small size, flexibility, and environment friendly etc. It draws a plenty of attention because it has a bright future of applying to forest patrol, field rescue, entertainment and many other respects. However, since the system of bicycle is non-linear and non-holonomic system, there are lots of precarious and dubious factors. Thus, using mechanism to control bicycle and make it run steadily is not easy.

Through the neoteric research that aimed at riding bicycle, Getz [1] set the torque of handelbar and the driving torque of rear wheel as the input of a system,

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and built a simple forepart bicycle-robot-dynamics model with an inner balance control machine. Iuchi [2] remodeled and redesigned bicycles through input the control handelbar, driving rear and motor driving which can control roll single pendulum angle. And they successfully realized that bicycle running in a static speed at 2.5 km/h. Tanaka [3, 4] built the non-linear dynamics model of bicycle system, besides, they used partial linear theory to simplify the model. They brought up a design concept that use position controller to stabilize the bicycle body and implement track control using track controller. Domestic professor Wei Shi Min and his research team [5] are working at the analysis of a front-wheel drive bicycle robot.

This article builds up a rider-less bicycle model, and using Constrained Lagrangian method to deduce the dynamic equation. Through the analysis of the dynamic equation, design a bicycle model which the driving control and balance control are separated. According to the different running-speed commands, the bicycle gets a stable forward speed by adjusting the torque of the rear driving motor. In the condition of having a stable speed, the body of the bicycle can keep balance through actualize the feedback of bicycle body roll angle to adjust the steering angle by using the Steering Angle Balance Control method.

2 Rider-Less Bicycle Dynamic Model

2.1 Model Building

The rider-less bicycle is considered as a two-part platform: a rear frame and a steering mechanism. Before analyzing the bicycle mode, a few consume conditions are settled as following: (1) the contact of the tread and the ground is point-contact, thus the thickness of the tire are supposed to be ignored; (2) the frame of the bicycle is regard as a point mass; (3) the ground is horizontal; (4) there is no side sliding when the bicycle is running [6–8].

Like what is shown in Fig. 1, using three coordinate systems: fixed coordinate system (X, Y, Z) , this coordinate system is fixed on the ground; bicycle body moved coordinate system (x, y, z) , the axis x of this coordinate system is in the direction of the connect between point C_2 which is the contact point of rear wheel and ground and point C_1 which is the contact point of front wheel and ground. The coordinate origin O is in the position of C_1 . The axis x and the axis y are orthogonal, and parallel with the ground; bicycle body coordinate system $B(x_B, y_B, z_B)$. The x_B axis and z_B axis of this coordinate system are fixed on the bicycle frame plane, the origin O_B is the centroid. As the bicycle body coordinate system relative to the fixed coordinate system, ψ is the roll angle, φ is the list angle. If define the unit vectors of these three coordinate systems are: $(\mathbf{I}, \mathbf{J}, \mathbf{K})$, $(\mathbf{i}, \mathbf{j}, \mathbf{k})$, $(\mathbf{i}_B, \mathbf{j}_B, \mathbf{k}_B)$, the conversion of these three coordinate system is:

Fig. 1 The rider-less bicycle system model

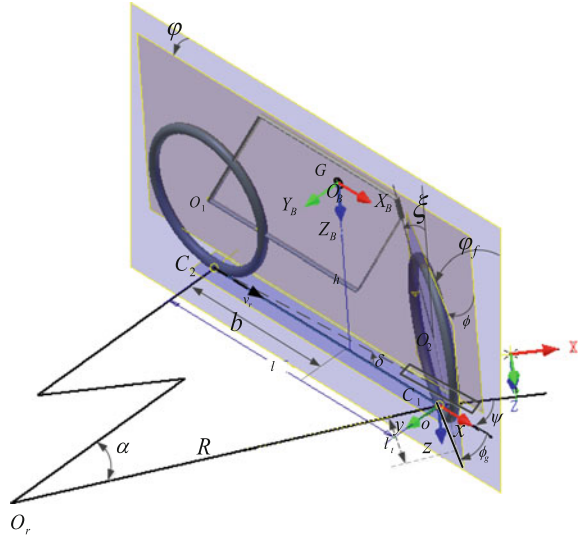
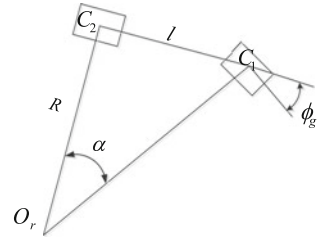


Fig. 2 Rider-less bicycle model overlook-diagram



$$\begin{bmatrix} \mathbf{i}_B \\ \mathbf{j}_B \\ \mathbf{k}_B \end{bmatrix} = \begin{bmatrix} \cos \psi & \sin \psi & 0 \\ -\cos \phi \sin \psi & \cos \phi \cos \psi & \sin \phi \\ \sin \phi \sin \psi & -\sin \phi \cos \psi & \cos \phi \end{bmatrix} \begin{bmatrix} \mathbf{I} \\ \mathbf{J} \\ \mathbf{K} \end{bmatrix} \quad (1)$$

Let O_r denote the instantaneous rotation center R is represent the rotation radius. ϕ_g is the projection of the steering angle on the ground. According to the geometry connection between these, like the diagram which is shown in Fig. 2:

$$\alpha = \phi_g. \quad (2)$$

Other parameters in the model:

v_r Speed of the rear wheel and the ground contact point. Since tire has no side sliding, the direction of the v_r is along the axis x_B .

ϕ Angle of the steering.

m Mass of the bicycle system.

- J_s Moment of inertia of the steering fork (with the front wheel set) about its rotation axis.
- l Distance between C_1 and C_2 .
- l_t Horizontal distance between the line which contains the rotation axis and the contact-point of the front wheel.
- h Height of the model's centroid.
- r Radius of front and rear tires.
- δ Rear frame rotation angle.
- ξ Front fork angle of the steering.
- b Horizontal distance between the centroid and the center of the rear wheel.

In order to clear and simplify the writing, this article use x , C_x and S_x to present the arbitrary angle, $\cos x$ and $\sin x$.

2.2 State Equation

Aiming at the non-holonomic system which stated as above, this article imposes the Constrained Lagrangian method to get the dynamic equation conveniently. The first step gets separately the kinetic energy of the rider-less bicycle system, the rotation kinetic energy of the steering fork, and the potential energy of the system. The derivation processes are as following:

when the rider-less bicycle system is running forward, the speed of the centroid is:

$$\mathbf{v}_G = \left(v_r + h\dot{\psi}S_\varphi \right) \mathbf{i} + \left(b\dot{\psi} + h\dot{\phi}C_\varphi \right) \mathbf{j} + h\dot{\phi}S_\varphi \quad (3)$$

When turning the steering, the steering fork produces the rotation kinetic energy. The ground projection of the steering angle ϕ is ϕ_g . And give the definition of the dynamic steering variable σ [9]:

$$\sigma := \tan \phi_g = \frac{l}{R} \quad (4)$$

Through the geometry shape of the steering mechanism get the geometry connection as follow:

$$\tan \phi_g C_\varphi = \tan \phi C_\xi \quad (5)$$

When the roll angle and steering angle are small, Eq. (5) can be replaced by Eq. (6) approximately.

$$\dot{\sigma}C_\varphi = \dot{\phi}C_\xi \quad (6)$$

Besides, the rotation of the steering also affects the potential energy of the system. When the steering angle is zero, the projection of the bicycle body likes the dotted line which is shown in Fig. 1 because of the height of the body roll centroid is $h \cos \varphi$. And when the steering rotated an angle ϕ , the projection of the bicycle body rotated an angle of δ , meanwhile, the height of the centroid get short by Δh_G . The δ can be approximate to:

$$\delta = \frac{l_t \phi_g C_{\xi}}{l} \quad (7)$$

Using an approximate of small angle $\sigma \approx \phi_g$, Δh_G can be calculated as:

$$\Delta h_G = \delta b S_{\varphi} \approx \frac{b l_t \sigma C_{\xi}}{l} S_{\varphi} \quad (8)$$

As what has been listed above, the Lagrangian L of the system is:

$$L = \frac{1}{2} J_s \dot{\phi}^2 + \frac{1}{2} m \mathbf{v}_G^2 - mg(h \cos \varphi - \Delta h_G) \quad (9)$$

According to the Euler-Lagrange equation:

$$\frac{d}{dt} \left(\frac{\partial L}{\partial \dot{\mathbf{q}}} \right) - \frac{\partial L}{\partial \mathbf{q}} = \mathbf{u} \quad (10)$$

In the equation:

$$\mathbf{q} = [\varphi \quad s \quad \sigma \quad \psi]^T; \dot{\mathbf{q}} = [\dot{\varphi} \quad v_r \quad \dot{\sigma} \quad \dot{\psi}]^T; \mathbf{u} = \begin{bmatrix} 0 \\ \tau \\ \tau_{\sigma} \\ 0 \end{bmatrix} \text{ and } s \text{ is the remove}$$

of the bicycle. τ is the torque which throw on the rear wheel's axle and make the bicycle to move forward. τ_{σ} is a turning torque of handlebar.

When bicycle is running forward, the rear wheel's speed and the roll angle have a relation as the following equation, which immediate the non-holonomic constraint of the system:

$$v_r = R \dot{\psi} = \frac{l}{\sigma} \dot{\psi} \quad (11)$$

The non-holonomic constraint can be simplified as follows:

$$\mathbf{A}_c \dot{\mathbf{q}} = 0 \quad (12)$$

where $\mathbf{A}_c = \begin{bmatrix} 0 & 1 & 0 & -\frac{l}{\sigma} \end{bmatrix}$

Using non-holonomic constraint combine with Lagrange multipliers to resettle the dynamic Eq. (10), the equation will be in the format as follows:

$$\mathbf{H}(\mathbf{q}) \ddot{\mathbf{q}} + \mathbf{V}(\dot{\mathbf{q}}, \mathbf{q}) = \mathbf{u} + \mathbf{A}_c^T \lambda \quad (13)$$

The $\mathbf{H}(\mathbf{q})$ in the equation is a 4×4 matrix, and $\mathbf{V}(\dot{\mathbf{q}}, \mathbf{q})$ is a 4×1 matrix. The λ in the equation is the Lagrange multipliers. Then the equation can be farther trimmed into a new format like following:

$$\dot{\mathbf{q}} = \mathbf{C}(\mathbf{q}_r) \dot{\mathbf{q}}_r \quad (14)$$

where $\dot{\mathbf{q}}_r = [\dot{\varphi} \quad v_r \quad \dot{\sigma}]^T$. The $\mathbf{C}(\mathbf{q}_r)$ is a 4×3 matrix that each line of which is an new space vector of $\mathbf{A}_c$.

Calculi Eq. (14), and there is a new equation as follows:

$$\ddot{\mathbf{q}} = \mathbf{C}(\mathbf{q}_r) \ddot{\mathbf{q}}_r + \dot{\mathbf{C}}(\mathbf{q}_r) \dot{\mathbf{q}}_r \quad (15)$$

The dynamic equation with three variables can be obtained by putting Eq. (15) into Eq. (13) and multiplies with $\mathbf{C}^T(\mathbf{q}_r)$:

$$\begin{aligned} & \begin{bmatrix} mh^2 & \frac{mbh\sigma}{l} C_\varphi & 0 \\ \frac{mbh\sigma}{l} C_\varphi & (1 - \frac{h\sigma}{l} S_\varphi)^2 + \frac{b^2 \sigma^2}{l^2} & 0 \\ 0 & 0 & \frac{J_s C_\varphi^2}{C_\xi^2} \end{bmatrix} \ddot{\mathbf{q}}_r \\ & - \begin{bmatrix} -m(1 - \frac{h\sigma}{l} S_\varphi) \frac{h\sigma C_\varphi}{l} v_r^2 + mg(hS_\varphi + \frac{l_r b C_\xi}{l} \sigma C_\varphi) - \frac{bh}{l} C_\varphi v_r \dot{\sigma} \\ 2m(1 - \frac{h\sigma}{l} S_\varphi) \frac{h\sigma}{l} C_\varphi \dot{\varphi} v_r + \frac{mbh\sigma}{l} S_\varphi \dot{\varphi}^2 + B_\omega \dot{\sigma} \\ \frac{2J_s}{C_\xi^2} S_\varphi C_\varphi \dot{\sigma} \dot{\varphi} + \frac{mbgl_r C_\xi}{l} S_\varphi \end{bmatrix} \\ & = \begin{bmatrix} 0 \\ \tau \\ \tau_\sigma \end{bmatrix} \end{aligned} \quad (16)$$

And in Eq. (16): $B_\omega = 2 \left[(1 - \frac{h\sigma}{l} S_\varphi) \frac{h}{l} S_\varphi - \frac{b^2 \sigma}{l^2} \right] v_r - \frac{b}{l} v_r - \frac{bh}{l} C_\varphi \dot{\varphi}$

When the J_s is small and smooth enough, get $\dot{\sigma}$ as one of the input capacity. Thus the equation can be simplified as follows:

$$\mathbf{M} \ddot{\mathbf{q}}_1 = \mathbf{K}_m + \mathbf{B}_m \begin{bmatrix} \dot{\sigma} \\ \tau \end{bmatrix} \quad (17)$$

And in Eq. (17):

$$\dot{\mathbf{q}}_1 = [\dot{\varphi} \quad v_r]^T;$$

$$\mathbf{M} = \begin{bmatrix} \mathbf{M}_{11} & \mathbf{M}_{12} \\ \mathbf{M}_{21} & \mathbf{M}_{22} \end{bmatrix} = \begin{bmatrix} h^2 & \frac{bh\sigma}{l} C_\varphi \\ \frac{bh\sigma}{l} C_\varphi & (1 - \frac{h\sigma}{l} S_\varphi)^2 + \frac{b^2\sigma^2}{l^2} \end{bmatrix};$$

$$\mathbf{K}_m = \begin{bmatrix} -(1 - \frac{h\sigma}{l} S_\varphi) \frac{h\sigma C_\varphi}{l} v_r^2 + g(hS_\varphi + \frac{l_b C_\xi}{l} \sigma C_\varphi) \\ 2(1 - \frac{h\sigma}{l} S_\varphi) \frac{h\sigma}{l} C_\varphi \dot{\varphi} v_r + \frac{bh\sigma}{l} S_\varphi \dot{\varphi}^2 \end{bmatrix};$$

$$\mathbf{B}_m = \begin{bmatrix} -\frac{bh}{l} C_\varphi v_r & 0 \\ B_\omega & \frac{1}{m} \end{bmatrix}.$$

3 Control System Design

Equation (17) indicates that the torque τ of the rear wheel has no effect on the bicycle body's rolling dynamic. Thus design to separately control the bicycle running forward and the bicycle balance. In the forward control, the running speed can get stable through adjusting the torque of the motor and stabilize the rotational speed of the rear wheel according to the different command of forward running speed. When balance controlling the system, using the Steering Angle Balance Control method. When the bicycle body is incline, the angle sensor can sense the roll angle as well as the angular velocity. Using the signal processing to get the accurate roll angle, then use the PI controller to adjust the steering angle which can keep the balance of the bicycle. After the system gets balanced, if the reference angle φ_d of the bicycle body is changed, the bicycle curve can be achieved using the Steering Angle Balance Control method which stated above (Fig. 3).

When the forward running speed is an invariable, which means $\dot{v}_r = 0$, the dynamic equation of the system can be simplified as:

$$\ddot{\varphi} - \frac{g}{h} S_\varphi = \frac{\sigma^2 S_\varphi C_\varphi}{l^2} v_r^2 + (\frac{gl_b C_\xi}{lh^2} C_\varphi + \frac{v_r^2 C_\varphi}{lh}) \sigma - \frac{b C_\varphi v_r}{lh} \dot{\sigma} \quad (18)$$

When φ is a small angle, the linearization of the system make the transfer function of the system is:

Fig. 3 The block diagram of control system

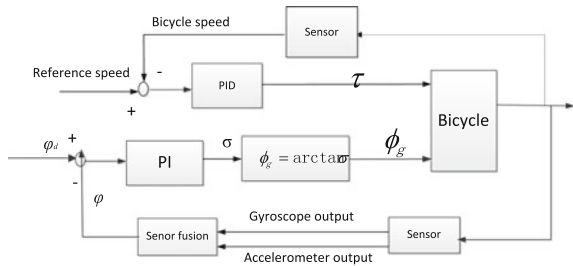


Table 1 The parameters of rider-less bicycle

Mass of tires (m_W)	0.5 kg
Mass of bicycle M	3.1 kg
Moment of inertia of the steering fork (J_s)	3.675E-002 kg/m ²
Steering front fork angle ξ	15°
Horizontal distance between the line which contains the rotation axis and the contact-point of the front wheel (l_f)	0.018 m
The horizontal distance between the centroid and the center of the rear wheel (b)	0.15 m
Radius of tires(r)	0.07 m
Distance between the center of two tires (l)	0.29 m
Height of the bicycle centroid (h)	0.25 m
The horizontal distance between the centroid and the center of the rear wheel (b)	0.15 m

$$G_{\phi\sigma}(s) = -\frac{\frac{bv_r}{lh}s + \frac{hv_r^2 + gbl_f C_\xi}{lh^2}}{s^2 - g/h} \quad (19)$$

Design and make the small rider-less bicycle for laboratory use, the detail parameters are as Table 1:

The Open loop transfer function is:

$$G_k s = \frac{-2.069v_r s - (1.41 + 13.793v_r^2)}{s^2 - 39.2}$$

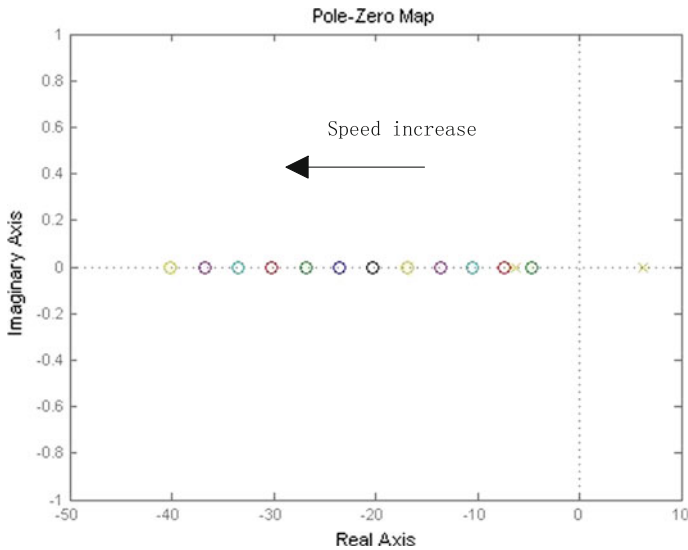
**Fig. 4** The zeros and poles while v_r is increasing

Figure 4 shows that with the increasing of v_r , a pole of the system is always stays in right half plane. This is an unstable system, therefore, through the feedback of the real-time roll angle φ , the excogitation that using PI controller to control the motor rotation angle of the steering can maintain the steady of the system. The angular amount of the steering angle can be expressed as follows:

$$\sigma = -K_P(\varphi - \varphi_d) - K_I \int (\varphi - \varphi_d) dt \quad (20)$$

Using Laplace transform can get the expression of the controller's transfer function D:

$$D(s) = K_P + \frac{K_I}{s} = \frac{sK_P + K_I}{s} \quad (21)$$

And the Closed loop transfer function of the system can be ensured by Eq. (20):

$$Gs = \frac{G_k s D(s)}{1 + G_k s D(s)} \quad (22)$$

As an example, the preform of the reference roll angle φ_d is a square wave. By final-value theorem, the state of the roll angle can be obtained after the system becomes stable.

$$\lim_{t \rightarrow \infty} \varphi(t) = \lim_{s \rightarrow 0} s G(s) \varphi_d(s) = 0 \quad (23)$$

where $\varphi_d(s) = \delta \left(\frac{1 - e^{-\varepsilon s}}{s} \right)$, δ is the peak value of the square wave, ε is the duration time of the square wave.

The result indicated that Steering Angle Balance Control method can make the roll angle to be convergent, and that a square wave preform can be used as the reference roll angle φ_d to realize the bicycle's steering.

4 Simulation by ADAMS

This article uses the ADAMS (Automatic Dynamic Analysis of Mechanical Systems) software to imitate the control method which is indicated above. Through building a motor model of rider-less bicycle in the ADAMS software environment (Fig. 5) to observe the actual effect of the Steering Angle Balance Control method that indicated in this article.

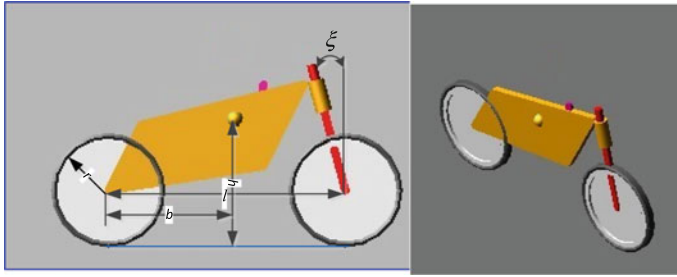


Fig. 5 The front elevation and the three-view drawing of ADAMS model

4.1 The Simulation of Balance Control Under Different Target Speed

Put different torques to the rear wheel as the impetus for the bicycle to move forward. The rotational speed of the rear wheel is increased from zero to 410, 820, 1240 degree/s, and let the eventually theoretical forward speed be 0.5, 1, 1.5 m/s. After the speed being stable, adding a disturbing torque at centroid. Design the value of K_I, K_P of the balance controller separately to make sure that the bicycle steadily moving forward (Figs. 6, 7, 8, 9, 10, 11 and 12; Table 2).

From the simulation data which listed above, some conclusions can be attained as follows:

1. Because the bicycle tires will have small distortion when the bicycle is running forward, the radius of gyration will be smaller than the actual radius. Therefore, the actual speed of running steadily forward will be less than the value that rotational speed multiplies with r .

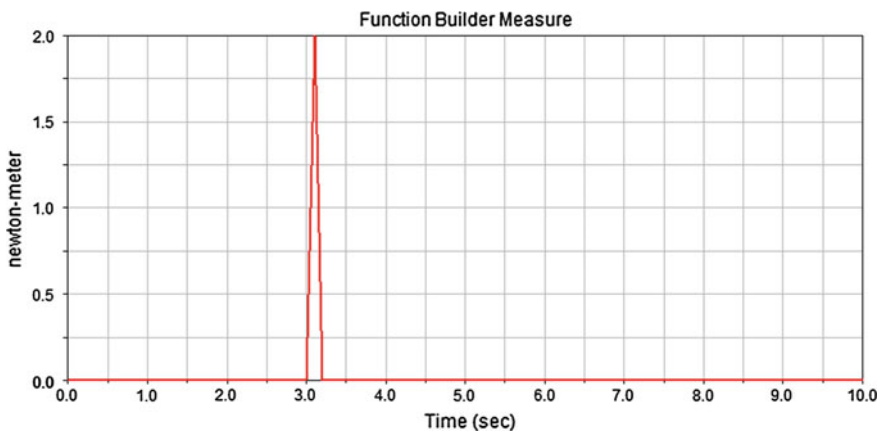


Fig. 6 The disturbing torque which added to the centroid after the system become stable

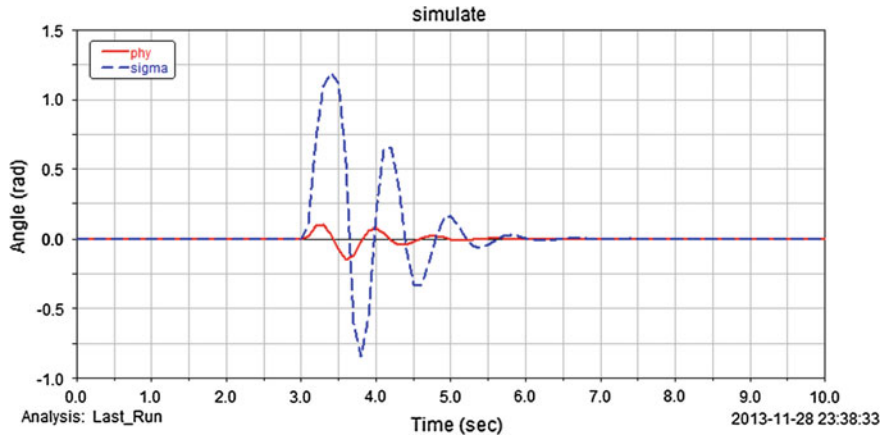


Fig. 7 The result of ϕ and σ when the target rotational speed of ADAMS model is 410 degree/s

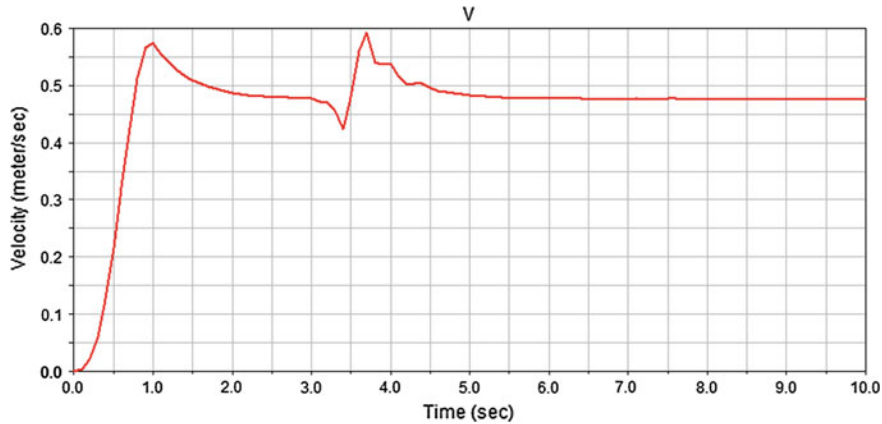


Fig. 8 The speed of the bicycle when the target rotational speed of ADAMS model is 410 degree/s

2. When the bicycle body incline to the right, the steering will turn right. It will make the weight of the body shift to the left and engender a gravity torque. And the gravity torque will pull the system back to the balance point again, Vice versa.
3. The speed v_r of the system has negative correlation with K_P and K_I of the balance controller: when the speed v_r increases gradually, the control coefficient $K_P(3 - 2.5)$ and $K_I(100 - 30)$ that make the system stable and the stable time is always 3 s (from the third sec to the sixth sec) are both decrease.
4. The smaller target speed that given to the rear wheel, the easier the running forward of the bicycle will be affected by the torque of the steering; according to

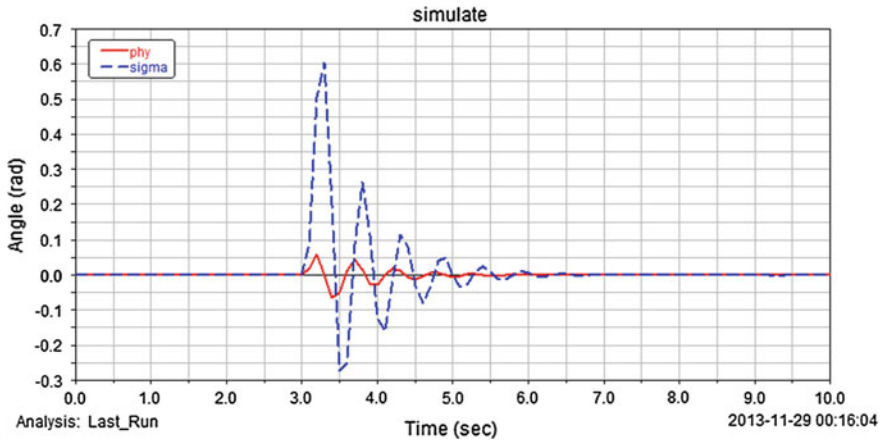


Fig. 9 The result of ϕ and σ when the target rotational speed of ADAMS model is 820 degree/s

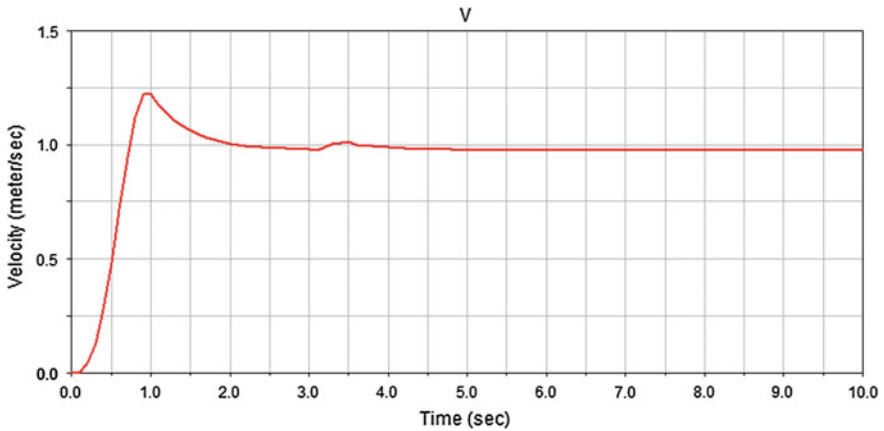


Fig. 10 The speed of the bicycle when the target rotational speed of ADAMS model is 820 degree/s

this assumption, this is one of the reason why the balance control is difficult when the speed approaches to zero.

5. When the command of the rotational speed which given to the rear wheel accumulates from zero to the stable speed, the speedup-static-friction that tires get from the ground is heavier than the uniform velocity-static-friction. When the command suddenly changed from accumulate speed state to the stable speed, however, the bicycle still get a strong effect due to the speedup-static-friction which makes the overshoot phenomenon exists in the bicycle's speed.

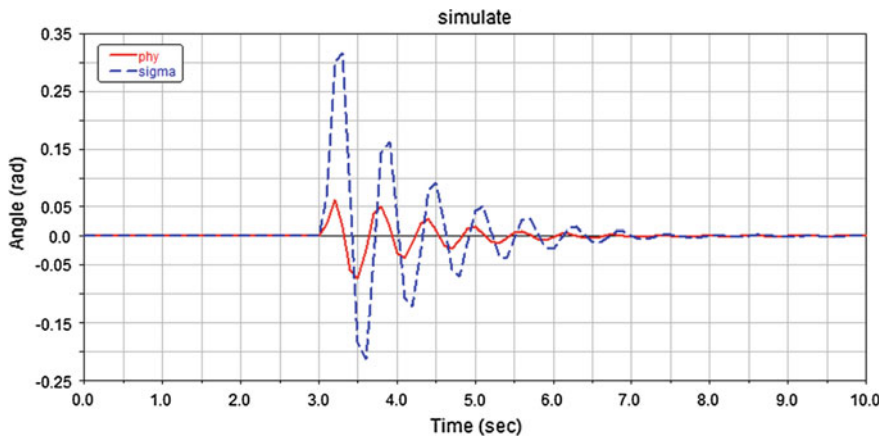


Fig. 11 The result of ϕ and σ when the target rotational speed of ADAMS model is 1240 degree/s

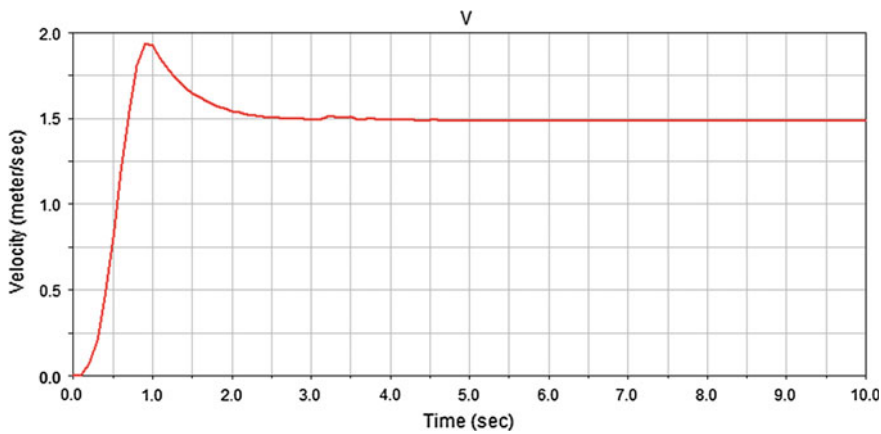


Fig. 12 The speed of the bicycle when the target rotational speed of ADAMS model is 1240 degree/s

Table 2 The relationship between PI parameter and target speed in the balance controller

v_r	K_P	K_I
0.5	3	100
1	2.8	80
1.5	2.5	30

4.2 Steering Simulation

In order to realize the simulation of the Steering of the rider-less bicycle, three different reference angles φ_d are designed after the forward speed of the bicycle keeps stable. Then the right turning will be engendered according to φ_d (Fig. 13).

Here cites an example of the bicycle-body speed stabilize at 1.5 m/s, the K_P, K_I of the PI controller choose the control coefficient that keep the bicycle body stable when the rotational speed is 1240 degree/s. Which means $K_P = 2.5$, $K_I = 30$ (Figs. 14, 15 and 16).

The result of simulation indicates that all of the variables will converge to a fixed value and get in the steady-state. What is worth paying attention to is the under-shoot phenomenon will appears in the dynamic of σ . This phenomenon can be explained as, in order to achieve the goal of turning right, the steering will turn left a little at first, which makes the weight center engenders a right incline gravity torque.

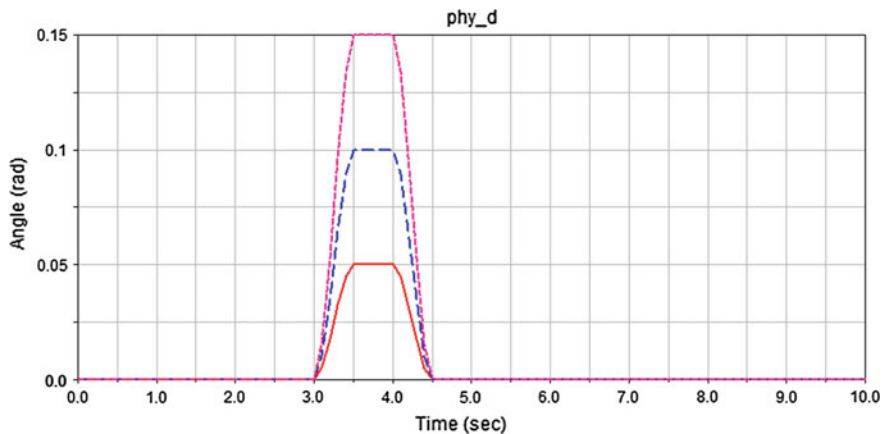


Fig. 13 Three different reference angles φ_d of ADAMS

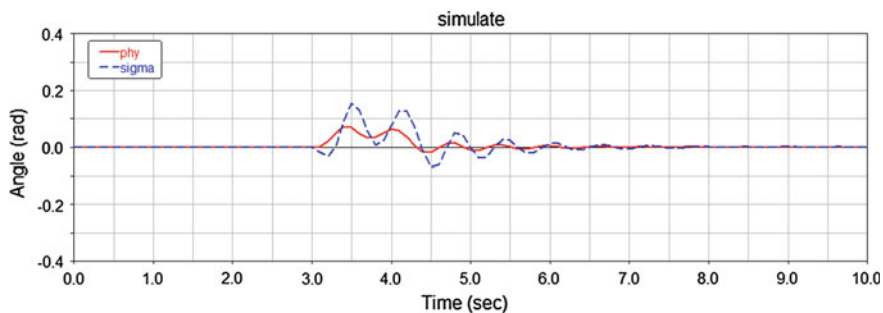


Fig. 14 The result of ϕ and σ when the reference angle φ_d of ADAMS is 0.05 rad

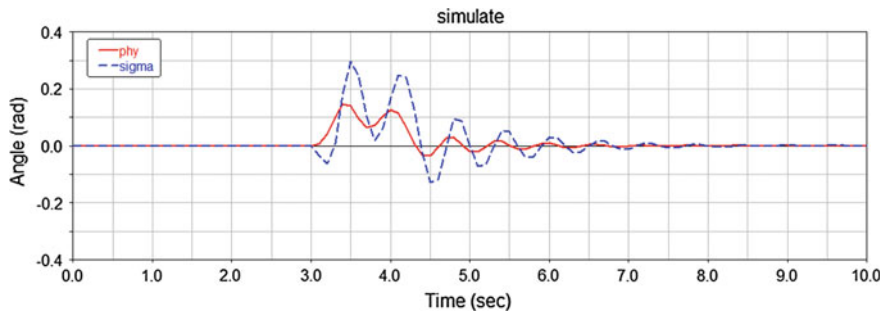


Fig. 15 The result of ϕ and σ when the reference angle φ_d of ADAMS is 0.1 rad

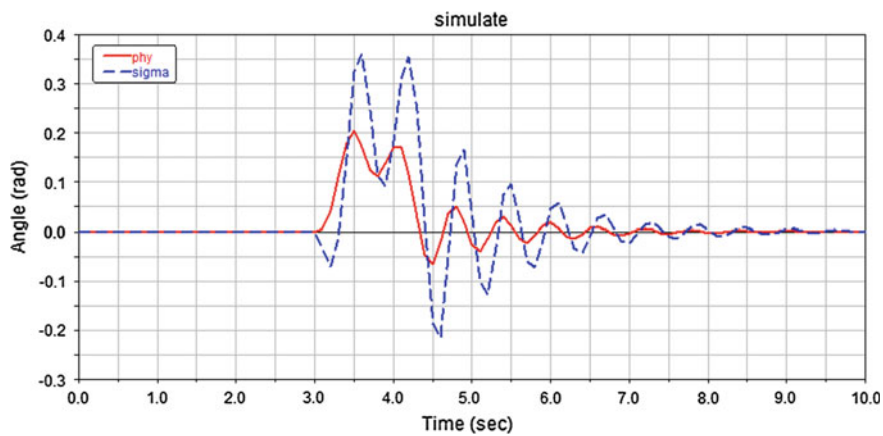


Fig. 16 The result of ϕ and σ when the reference angle φ_d of ADAMS is 0.15 rad

Then the steering will turn right immediately for keeping balance. Until the reference angle return-to-zero, the balance controller finally pushed the bicycle back into vertical and running along a straight line.

Figure 17 contains the bicycle tracks under three different reference angle, the original point is the beginning of the bicycle, the red line is the turning centroid's track while the reference roll angle is 0.05 rad, the blue line is the turning centroid's track while the reference roll angle is 0.1 rad, and the pink one is the turning centroid's track while the reference roll angle is 0.15 rad. These lines show that the bigger the bicycle body's reference angle, in the same time, the smaller the turning radius of the bicycle; the bigger the angle that the bicycle turns, also the bigger the angle which the steering is supposed to turn.

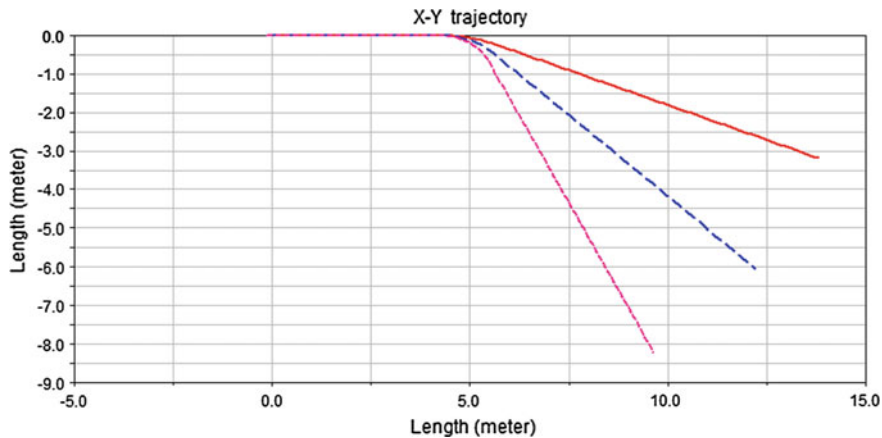


Fig. 17 The bicycle tracks which simulate by ADAMS under there different reference angles φ_d (the red line stands for 0.05 rad, the blue line stands for 0.11 rad, and the pink one represents 0.15 rad)

5 Conclusion

Use the Constrained Lagaragian method to build a dynamic model of the rider-less bicycle.

Through the observation of the dynamic model, discovering that the torque τ which set on the rear wheel that impetus the bicycle has no straight effect on the incline dynamic of the bicycle. Thus design to separately control the bicycle running forward and the bicycle balance. In the forward control, the rear wheel get a stable speed through adjust the torque of the motor under different forward speed commands. In the condition of the bicycle's speed is stable, design a steering-angle-balance-control method to keep the bicycle system in balance. Through feedback the real-time roll angle of the bicycle and control the curve of the steering by controller.

Use the simulation of the ADAMS, simulating the balance control and steering of the rider-less bicycle separately. The simulation indicates that the Steering Angle Balance Control method can realize the bicycle balance control and the curve control effectively. Meanwhile, the bigger the speed of the bicycle, the smaller the control coefficient (K_P, K_I) of the balance controller PI. Thus the bicycle body is easier to control; besides, after the bicycle get to the target speed.

References

1. Getz, N.H., Marsden, J.E.: Control for an autonomous bicycle. In: Proceedings of 1995 IEEE International Conference on Robotics and Automation, vol. 2, pp. 1397–1402 (1995)
2. Iuchi, K., Niki, H., Murakami, T.: Attitude control of bicycle motion by steering angle and variable COG control. In: 31st Annual Conference of IEEE (IECON 2005), 6 pp. Industrial Electronics Society (2005)
3. Tanaka, Y., Murakami, T.: A study on straight-line tracking and posture control in electric bicycle. *Industr. Electron. IEEE Trans.* **56**(1), 159–168 (2009)
4. Tanaka, Y., Murakami, T.: Self sustaining bicycle robot with steering controller. In: The 8th IEEE International Workshop on Advanced Motion Control, 2004 (AMC'04), pp. 193–197 (2004)
5. Yonghua, H., Qizheng, L., Shimin, W., Lei, G.: Control system design for a bicycle robot based on double DSP. *Comput. Meas. Control*, **19**(12), 2965–2968, 2988 (2011) (in Chinese)
6. Guo, L., Huang, Y., Liao, Q., Wei, S.M.: Kinemics analysis for self-balancing bicycle robot. *J. Beijing Univ. Posts Telecommun.* **34**(6), 99–102 (2011) (in Chinese)
7. Zhong W.: Design of embedded control system for the riderless bicycle. Beijing University of Posts and Telecommunications (2010) (in Chinese)
8. Yu X.: A study on the control and realization for bicycle robot nonlinear system. Beijing University of Posts and Telecommunications (2010) (in Chinese)
9. Jingang, Y., Yizhai, Z., Dezhen, S.: Autonomous motorcycles for agile maneuvers, part I: dynamic modeling. In: Proceedings of the 48th IEEE Conference on Decision and Control, 2009 held jointly with the 2009 28th Chinese Control Conference. CDC/CCC 2009, pp. 4613–4618 (2009)

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