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Path Tracking Control of the Wheeled off-road Articulated Vehicle with Actuator Saturation

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Abstract
In order to study on the path tracking problem of the wheeled off-road articulated vehicle in autonomous motion, the path tracking error kinematics equation is derived. The backstepping method is presented to design the controllers for the path tracking systems in the presence of input saturation. The simulation results show that when the position deviation is 0.25m, the angle deviation along motion direction is 0.1rad, the angle between the front vehicle body and rear vehicle body is 0.1rad, the error curves fluctuate with time when there is no control input. After applying the anti-saturation control algorithm, the deviation error decreases rapidly and tends to zero, so it can implement the path tracking control of the wheeled off-road articulated vehicle.

Keywords: WHEELED OFF-ROAD ARTICULATED VEHICLE, PATH TRACKING; BACKSTEPPING, ACTUATOR SATURATION

Introduction

A new wheeled off-road articulated vehicle is designed whose front body and rear body are attached together by the universal joint, the steering of the vehicle is carried out by the hydraulic actuators between the front vehicle body and the rear vehicle body. The vehicle has excellent terrain adaptability and obstacle surmounting ability, because the front vehicle body and the rear vehicle body can passively adapt the terrain. It is shown in Fig.1.

Figure 1. The wheeled off-road articulated vehicle

However, when the wheeled off-road articulated vehicle is traveling in rough terrain, the actual path often generates deviation from the predetermined path. The reasons of deviation are following: First, the design error of the vehicle structure would lead to the steering instantaneous center of the rear vehicle body is not coincident with that of the front vehicle body; Second, the hydraulic actuators has flexibility, then the steering angle between the front and the rear vehicle body would be oscillating; Third, The effects of the terrain. The deviation error will lead to the vehicle difficulty to operate, sometimes even cause the vehicle rollover. In order to improve the vehicle for autonomous motion and work, so it is very important to study on the path tracking ability of the vehicle.

Many papers had discussed the path tracking control problem[1,2,3]. L. García-Pérez, M.C. García-Alegre etc.[4] studied the bi-directional communication between the farming vehicle and the human supervisor which used a wireless LAN. Tabatabaei O etc.[5] studied controller design of the articulated vehicle based on fuzzy logic theory; it was used to eliminate the articulated vehicle path tracking error. Ramon G, M. F, Teodoro A, etc.[6] study the motion control of mobile robot trajectory tracking through Lyapunov stability equation and adaptive feedback control method. B.J. Alshaer, T.T. Darabseh etc. [7] studied the highly scalable path-following controller for N-trailers with off-axle hitching. Although these papers put forward different control methods for path tracking, they did not consider the control input saturation. But actually the value of the vehicle steering torque which is produced by hydraulic actuators is bounded, and the steering angular velocity is limited. Therefore, the control input saturation should be taken into account when studying on the path tracking problem of the vehicle. The control input saturation is a common phenomenon in the engineering. Many papers research on the control input saturation problems recently, and gives out the different anti-saturation actuator design methods[8,9,10]. Many anti-saturation control algorithm has been used in control of ships and spacecraft[11,12].

This paper proposes the path tracking error model of the wheeled off-road articulated vehicle, analyzes the stability and controllability of the error system, and designs an anti-saturation control algorithm based on backstepping method.

1. The Path Tracking Error Model

The schematic diagram of the wheeled off-road articulated vehicle traveling in the motion plane is shown in Fig. 2. The inertial coordinate system $O_I x_I y_I z_I$ is located in inertial space. The reference coordinate system of the rear vehicle body is $O_b x_b y_b z_b$; The point P_r is the drive shaft center of rear vehicle body in the predetermined path L. The origin of the reference coordinate system is located in point P_r. x_R axis is along the tangential direction of path L. φ is the angle between the front vehicle body and the rear vehicle body of the vehicle. The angle between x_I axis and x_R axis is θ_r. The rear vehicle body velocity along the drive shaft is v, and the length of the front and the rear vehicle body is l.

The length of the front and rear body is l. The rear body velocity along the drive shaft is v, so the veloc-

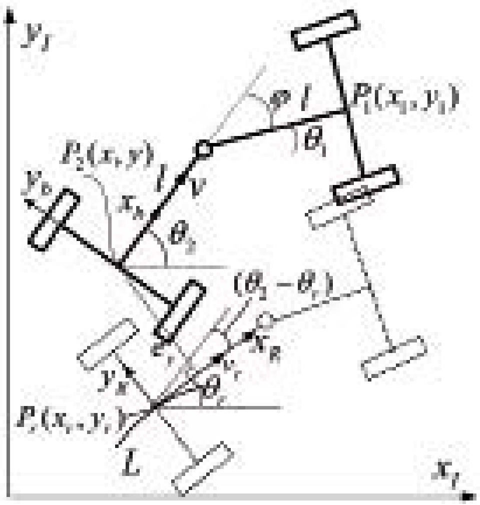

Figure 2. The coordinate system of the wheeled off-road articulated vehicle

ity of the point P_2 in the inertial coordinate system can be written as

$$\begin{aligned}\dot{x} &= v\cos\theta_2 \\ \dot{y} &= v\sin\theta_2\end{aligned} \tag{1}$$

With the assumption that there is no skid and slip when the articulated wheeled off-road vehicle is traveling, and the instantaneous center of the front body is in the extension line of front wheel axle, we can get

$$\dot{x}_1\sin\theta_1 - \dot{y}_1\cos\theta_1 = 0 \tag{2}$$

Based on the geometric relationship between the front and rear body, we can get

$$\begin{aligned}\theta_2 &= \theta_1 + \varphi \\ x_1 &= l\cos\theta_1 + l\cos\theta_2 + x \\ y_1 &= l\sin\theta_1 + l\sin\theta_2 + y\end{aligned} \tag{3}$$

Differentiating the Eq. (3) and using Eq. (1), Eq. (2), the equation is simplified as

$$\dot{\theta}_2 = \frac{v\sin\varphi - \dot{\varphi}l}{l(1+\cos\varphi)} \tag{4}$$

The kinematic equations of the wheeled off-road articulated vehicle can be written as

$$\begin{bmatrix}\dot{x}\\ \dot{y}\\ \dot{\theta}_2\\ \dot{\varphi}\end{bmatrix} = \begin{bmatrix}\cos\theta_2 & 0\\ \sin\theta_2 & 0\\ \dfrac{\sin\varphi}{l(1+\cos\varphi)} & -\dfrac{1}{(1+\cos\varphi)}\\ 0 & 1\end{bmatrix}\begin{bmatrix}v\\ \omega\end{bmatrix} \tag{5}$$

Where is the relative steering angle velocity of the front and the rear vehicle body.

The path tracking errors between the actual path and the predetermined path can be represented by e_y and e_θ. e_y is the vertical distance between the point P_2 and point P_r along the moving direction, and e_θ is the angle deviation along the moving direction [13, 14]. So the change rate of the path tracking error $\dot{e}_y$, $\dot{e}_\theta$ in reference coordinate system can be written as

$$\begin{aligned}\dot{e}_y &= v\sin e_\theta \\ \dot{e}_\theta &= \dot{\theta}_2 - \dot{\theta}_r\end{aligned} \tag{6}$$

The change rate of error between the actual path and the predetermined path along the moving direction can be written as

$$\dot{e}_x = e_y\dot{\theta}_r = v\cos e_\theta - v_r \tag{7}$$

Where v_r is the velocity of point P_r along the tangential direction of predetermined path, $v_r = \dot{\theta}_r / k_r$, k_r is the curvature of the predetermined path at point P_r. So, Eq. (7) can be rewritten as

$$\dot{\theta}_r = \frac{vk_r\cos e_\theta}{1+k_r e_y} \tag{8}$$

By combining Eq. (5), Eq. (6) and Eq. (8), $\dot{e}_\theta$ can be given as

$$\dot{e}_\theta = \frac{v\sin\varphi}{l(1+\cos\varphi)} - \frac{\dot{\varphi}}{(1+\cos\varphi)} - \frac{vk_r\cos e_\theta}{1+k_r e_y} \tag{9}$$

The path tracking problem can be solved by controlling the relative steering angle velocity ω, which makes the error of the actual path and the predetermined path $e_y \to 0$, $e_\theta \to 0$, so the path tracking error equation is written as

$$\begin{aligned}\dot{e}_y &= v\sin e_\theta \\ \dot{e}_\theta &= \frac{v\sin\varphi}{l(1+\cos\varphi)} - \frac{\dot{\varphi}}{(1+\cos\varphi)} - \frac{vk_r\cos e_\theta}{1+k_r e_y} \\ \dot{\varphi} &= \omega\end{aligned} \tag{10}$$

2. Stability and controllability analysis

When the value of control input ω is zero, it can be obtained by Eq. (10) that the equilibrium points of e_θ, e_y equal to zero. So, the second term of Eq. (10) is written as

$$0 = v\frac{\sin\varphi}{l(1+\cos\varphi)} - vk_r \tag{11}$$

The equilibrium point of φ is calculated as follows

$$\varphi_e = \arcsin\frac{k_r}{\sqrt{1+l^2k_r^2}} + \arcsin\frac{lk_r}{\sqrt{1+l^2k_r^2}} \tag{12}$$

We define $y = [e_y, e_\theta, \varphi - \varphi_e]^{\mathrm{T}} = [y_1, y_2, y_3]^{\mathrm{T}}$, so the path tracking error equation can be rewritten as

$$\begin{aligned}\dot{y}_1 &= v\sin y_2 \\ \dot{y}_2 &= v\frac{\sin(y_3+\varphi_e)}{l(1+\cos(y_3+\varphi_e))} - v\frac{k_r\cos y_2}{1+k_r y_1} - \frac{1}{1+\cos(y_3+\varphi_e)}\omega \\ \dot{y}_3 &= \omega\end{aligned} \tag{13}$$

The Jacobian matrix of the equilibrium point y is given as

$$A = \begin{bmatrix}0 & v\cos y_2 & 0\\ -v\dfrac{k_r{}^2\cos y_2}{(1+k_r y_1)^2} & v\dfrac{k_r\sin y_2}{1+k_r y_1} & v\dfrac{1}{l(1+\cos(y_3+\varphi_e))}\\ 0 & 0 & 0\end{bmatrix} \tag{14}$$

So, the characteristic roots at equilibrium point is calculated as

$$\lambda_1 = 0, \lambda_{2,3} = \pm vk_r i \tag{15}$$

Therefore, the system is critical stability based on the linear systems stability criterion. When the system parameters have slight deviation, the eigenvalues may be located in the right half complex plane, so the path tracking error system should be controlled. The controllability of system should be analyzed before designing control algorithm. The coefficient matrix of ω is given as

$$B=\begin{bmatrix} 0 \\ -\dfrac{1}{1+\cos(y_3+\varphi_e)} \\ 1 \end{bmatrix} \tag{16}$$

So, the controllability matrix of the system is written as

$$Q=[B, AB, A^2B] \tag{17}$$

The value of Q at the equilibrium point is calculated by Eq. (17) as follows

$$\det(Q)=-\frac{v^3(k_r^2+1/l)}{1+\cos\varphi_e}>0 \tag{18}$$

The rank of controllability matrix is 3, so the system is controllable.

3. Design the anti-saturation controller

We design controller for the purpose that make the state error parameters $[e_y \quad e_\theta \quad \varphi]$ tend to the equilibrium point $[0 \quad 0 \quad \varphi_e]$ by control algorithm. Because the control parameters which are provided by actuator are bounded in the engineering, the control input can be rewritten as

$$\omega=sat(\omega_c)=\begin{cases} sign(\omega_c)\omega_M & \omega_c>\omega_M \\ \omega_c & \omega_c<\omega_M \end{cases} \tag{19}$$

Where $sat()$ is saturation function, ω_c is the control input, ω_M is the upper limit of control input.

We define $x_1=y_1$, $x_2=[\sin y_2, y_3]^{\mathrm{T}}$, and Eq. (17) is rewritten as

$$\begin{aligned} \dot{x}_1 &= \phi_1 x_2 \\ \dot{x}_2 &= \phi_2(x_1,x_2)+B_0\omega \end{aligned} \tag{20}$$

Where

$$\phi_1=[v,0]$$

$$\phi_2=\begin{bmatrix} v\dfrac{\cos y_2\sin(y_3+\varphi_e)}{l(1+\cos(y_3+\varphi_e))}-v\dfrac{k_r\cos^2 y_2}{1+k_r y_1} \\ 0 \end{bmatrix}$$

$$B_0=\begin{bmatrix} -\dfrac{\cos y_2}{1+\cos(y_3+\varphi_e)} \\ 1 \end{bmatrix}$$

The backstepping method is used to design the controller[15,16]. In order to compensate the effect of the actuator saturation, the virtual state variables λ_1, λ_2 are constructed and given as

$$\begin{aligned} \dot{\lambda}_1 &= -\Gamma_1\lambda_1+\phi_1\lambda_2 \\ \dot{\lambda}_2 &= -\Gamma_2\lambda_2-\phi_1^{\mathrm{T}}\lambda_1+B_0\Delta\omega \end{aligned} \tag{21}$$

Where $\Delta\omega=\omega-\omega_c$, Γ_1, Γ_2 are positive definite matrix, the initial value of the virtual state variables are $\lambda_1(0)=0$, $\lambda_2(0)=0$. So we define the following change of coordinates

$$\begin{aligned} z_1 &= x_1-\lambda_1 \\ z_2 &= x_2-\lambda_2-\alpha \end{aligned} \tag{22}$$

Where α is the virtual control.

By combining Eq. (20), Eq. (21) and Eq. (22), we can get

$$\begin{aligned} \dot{z}_1 &= \phi_1 z_2-\Gamma_1 z_1+\phi_1\alpha+\Gamma_1 x_1 \\ \dot{z}_2 &= \phi_2+\phi_1^{\mathrm{T}}\lambda_1+\Gamma_2\lambda_2+B_0\omega_c-\dot{\alpha} \end{aligned} \tag{23}$$

The virtual control law is designed as follows

$$\alpha=-K_1\phi_1^{\mathrm{T}}x_1 \tag{24}$$

Where K_1 satisfies that $\Gamma_1=\phi_1 K_1\phi_1^{\mathrm{T}}$ is the positive definite matrix. Eq.(24) is substituted into Eq. (23), we obtain

$$\begin{aligned} \dot{z}_1 &= \phi_1 z_2-\Gamma_1 z_1 \\ \dot{z}_2 &= \phi_2+\phi_1^{\mathrm{T}}\lambda_1+\Gamma_2\lambda_2-\dot{\alpha}+B_0\omega_c \end{aligned} \tag{25}$$

The positive Lyapunov function is defined as

$$V_2=\frac{1}{2}z_1^{\mathrm{T}}z_1+\frac{1}{2}z_2^{\mathrm{T}}z_2 \tag{26}$$

The derivative of V_2 combines with Eq. (25) is given by

$$\dot{V}_2=-z_1^{\mathrm{T}}\Gamma_1 z_1+z_2^{\mathrm{T}}[\phi_1^{\mathrm{T}}z_1+\phi_2+\phi_1^{\mathrm{T}}\lambda_1+\Gamma_2\lambda_2-\dot{\alpha}+B_0\omega_c] \tag{27}$$

The control law ω_c is designed as follows

$$\omega_c=B_0^{\dagger}[-\phi_1^{\mathrm{T}}z_1-\phi_1^{\mathrm{T}}\lambda_1-\phi_2+\dot{\alpha}-\Gamma_2(x_2-\alpha)] \tag{28}$$

Where $B_0^{\dagger}$ is Moore - Penrose generalized inverse matrix of B_0. So Eq. (27) is rewritten as

$$\dot{V}_2=-z_1^{\mathrm{T}}\Gamma_1 z_1-z_2^{\mathrm{T}}\Gamma_2 z_2 \tag{29}$$

As Γ_1, Γ_2 are positive definite matrix, By applying the LaSalle-Yoshizawa theorem, it is obtained

$$\begin{aligned} \lim_{t\to\infty} z_1 &= \lim_{t\to\infty}(x_1-\lambda_1)=0 \\ \lim_{t\to\infty} z_2 &= \lim_{t\to\infty}(x_2-\lambda_2-\alpha)=0 \end{aligned} \tag{30}$$

So the positive Lyapunov function of λ is defined as

$$V_\lambda=\frac{1}{2}\lambda_1^{\mathrm{T}}\lambda_1+\frac{1}{2}\lambda_2^{\mathrm{T}}\lambda_2 \tag{31}$$

The derivative of V_λ is given by

$$\dot{V}_\lambda = -\lambda_1^{\mathrm{T}}\Gamma_1\lambda_1 - \lambda_2^{\mathrm{T}}\Gamma_2\lambda_2 + \lambda_2^{\mathrm{T}}B_0\Delta\omega \tag{32}$$

If the control input is not saturated, then $\Delta\omega = 0$, we can obtain that $\lim\limits_{t\to\infty}\lambda_1 = 0$, $\lim\limits_{t\to\infty}\lambda_2 = 0$ by applying the LaSalle-Yoshizawa theorem, and $\lim\limits_{t\to\infty}x_1 = 0$, $\lim\limits_{t\to\infty}x_2 = 0$, the asymptotically stable tracking of the system is implemented.

If the control input is saturated, then $\Delta\omega \neq 0$. In order to calculate the convergence of x_1, x_2, Eq. (28) can be rewritten as

$$\dot{V}_\lambda \le -a_1\lambda_1^{\mathrm{T}}\lambda_1 - (a_2 - 0.5)\lambda_2^{\mathrm{T}}\lambda_2 + \frac{1}{2}(B_0\Delta\omega)^{\mathrm{T}}(B_0\Delta\omega) \tag{33}$$

Where a_1, a_2 are the minimum eigenvalue of Γ_1, Γ_2, and $a_2 > 0.5$. Integrating both sides of Eq. (33), it can be given as

$$\begin{aligned} \|\lambda_1\|_2 &\le \frac{1}{\sqrt{2a_1}}\|B_0\Delta\omega\| \\ \|\lambda_2\|_2 &\le \frac{1}{\sqrt{2a_2 - 1}}\|B_0\Delta\omega\| \end{aligned} \tag{34}$$

The bound of z_1, z_2 is established by Eq. (29) as follows

$$\begin{aligned} \|z_1\|_2^2 &= \int_0^\infty z_1^{\mathrm{T}}z_1dt \le \frac{1}{a_1}V_2(0) \\ \|z_2\|_2^2 &= \int_0^\infty z_2^{\mathrm{T}}z_2dt \le \frac{1}{a_2}V_2(0) \end{aligned} \tag{35}$$

By combining Eq. (23) and Eq. (26), we obtain

$$\begin{aligned} z_1(0) &= x_1(0) \\ z_2(0) &= x_2(0) - K_1\phi_1^{\mathrm{T}}x_1(0) \\ V_2(0) &= \frac{1}{2}z_1(0)^{\mathrm{T}}z_1(0) + \frac{1}{2}z_2(0)^{\mathrm{T}}z_2(0) \end{aligned} \tag{36}$$

By combining Eq. (26), Eq. (34) and Eq. (35), the bound of x_1, x_2 is established as follows

$$\begin{aligned} \|x_1\|_2 &\le \frac{1}{\sqrt{2a_1}}\left(\sqrt{2V_2(0)} + \|B_0\Delta\omega\|\right) \\ \|x_2\|_2 &\le \frac{1}{\sqrt{a_2}}\sqrt{V_2(0)} + \frac{1}{\sqrt{2a_2}}\|B_0\Delta\omega\| + \|K_1\phi_1^{\mathrm{T}}\|\,\|x_1\| \end{aligned} \tag{37}$$

So, the tracking error can be reduced by increasing the value of a_1, a_2. If the value of a_1, a_2 is large enough, when $t \to \infty$, and then $x_1, x_2 \to 0$, the asymptotically stable path tracking for the system is ensured.

4 Simulation of the path tracking

In this section, the path tracking control algorithm in Section 3 is confirmed by simulation. The predetermined path is a circle, the radius is 10m, the velocity of the vehicle is 3m/s, the length of l is 1.5m. When the initial position deviation is 0.25m, motion direction angle deviation is 0.1rad, the angle between the front vehicle body and the rear vehicle body is 0.1rad, that is $y(0)=[0.25,0.1,0.1]^{\mathrm{T}}$. When the path tracking error equation has no control input, the path tracking error curves is shown in Fig.3. y_1, y_2 tends to fluctuate with time, only y_3 maintain the initial state. Because the eigenvalues of Jacobian matrix for the error system are a pair of imaginary roots and zero. The imaginary roots make y_1, y_2 fluctuate with time, and the zero root keeps y_3 in initial state.

When the anti-saturation control law is added into the path tracking error equation, the curves of error are shown in Fig.4. The error curves convergence to zero rapidly, so the anti-saturation control algorithm is correct and effective for the path tracking control of the wheeled off-road articulated vehicle.

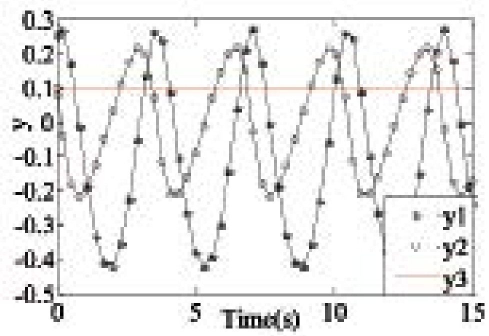

Figure 3. The error curve without control input

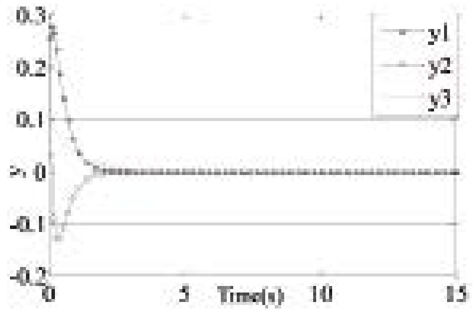

Figure 4. The error curve with the anti-saturation control algorithm

Conclusions

In this paper, we study the path tracking control problem of the wheeled off-road articulated vehicle, and analyze the stability and controllability of the path tracking error equation. The path tracking error system is critical stable when there is no control input. Taking into account the control input saturation, we design the ant-saturation controller based on the backstepping method; it makes the tracking error convergent with time. The results of the simulation are verified that the anti-saturation controller can decrease the path tracking error of the wheeled off-road articulated vehicle rapidly and effectively.

Acknowledgement

This article was supported by the National Natural Science Foundation of China (Grant No. 51405187).

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