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RESEARCH OF THE FORKLIFT POWER-ASSISTED STEERING SYSTEM BASED ON SAFETY STEERING SPEED CONTROL

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Abstract- Enhancing the safety of forklift power-assisted steering system is a problem urgently to be solved in practice. First of all, forklift power-assisted steering system model is established according to Lagrange dynamical equations, and three variable assistance characteristics curve fitted for reach trucks is designed combined with fuzzy control algorithm. Then sliding mode variable-structure control method based on motor current control is used tracking the target current and making contrast with the traditional PID in order to justify the validity of the algorithm. Finally, the expression of speed with tri-function is built. Optimization of safety speed is added on the basis of traditional power-assisted steering to ensure the stability and security of forklift steering.

Index terms: Dynamical characteristic curve, EPS of forklift, safety speed, sliding mode control based on reaching law.

I. INTRODUCTION

Automated material handling systems are used widely to transport products in warehouses, or distribute subparts between different assembly stations in a production line[1]. In recent years, vehicle electric power-assisted steering system and its control strategy have been researched both at home and abroad, and some intelligent algorithms have been applied to vehicle electric power-assisted steering system, including neural network, fuzzy control, H∞ control, LQR, road feel feedback, and optimization algorithm, etc[2-7]. Most of the research achievements mentioned above aimed at passenger cars, but forklifts have unique working properties as a kind of industrial trucks. It is used to load and unload indoors and outdoors. Heavy weight, complex environment, and changing load may lead to rollover accidents if the steering speed is too fast or the turning radius is too large. These factors make the stability and security of the forklift when steering become a problem worthy of attention. This paper finished theoretical research and simulation verification for dynamical characteristic, control strategy for electric power-assisted steering, safety speed of electric forklift power-assisted steering system.

The main content of this paper includes: (1) Combing with forklift operating characteristic and using the fuzzy control principle, three variable assistance characteristics curve based on steering wheel torque, reel-time speed, and load. It made the load variation be reflected in the computation process of target current, which improved the stability of the operation. (2) The study object is control strategy for electric power-assisted steering of 1.5t portable forklifts, and system model is established in the MATLAB. Making simulation comparison between PID control and sliding-mode control to track the target current, the sliding-mode control algorithm is adaptability to parameter perturbation, and can make the power-assisted control algorithm has strong robustness. (3) This paper theoretically analyzes safety steering of electric forklifts, shows the calculation method of safety speed which is affected by load, turning radius, steering real-time speed. Adding this method to the control method of the forklift power-assisted steering system made forklifts slow down when steering to prevent rollover and ensured a smooth steering.

II. THE WORKING PRINCIPLE AND STRUCTURAL ANALYSIS OF FORKLIFT EPS SYSTEM

As shown in figure. 1, forklift EPS system is mainly composed of torque sensor, input mechanism, controller, steering motor, retarding mechanism, meshing gear, and steering wheel etc [8, 9].

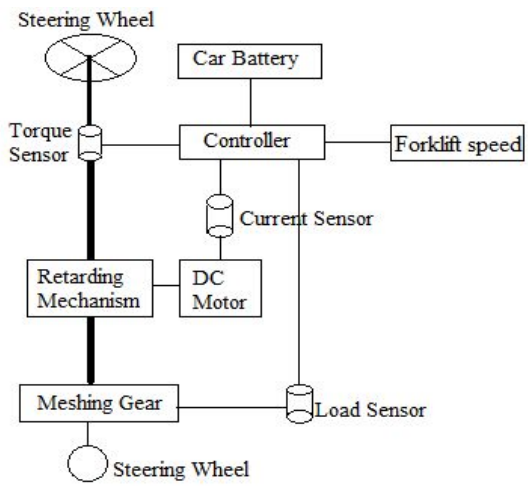

Figure 1. Forklift EPS system structure diagram

The working principle: When the forklift does not steering, the EPS does not work; when the driver input hand-force through steering wheel, the torque sensor to combine with steering shaft transforms relative rotation angular displacement generated by input shaft and output shaft under the force of torsion bar into electric signals, then sends turning signals to Electric Control Unit—ECU. According to sensor signals, ECU calculates assistance and direction of this time should be provided by a certain control algorithm, then electric motor outputs corresponding torque, and it was transferred to steering mechanism by transmission mechanism to generate assistance, so as to realize the real-time control.

III. THE DETERMINATION OF THE THREE VARIABLE ASSISTANCE CHARACTERISTICS CURVE

Assistance characteristics curve is the key to performance of EPS system. The law of assistance performance is determined by selecting the real time status and condition of vehicle [10-13]. Considering the relationship between and among forklift's speed, steering-wheel torque, assistance torque, assistance characteristics curve can be divided into linear type, broken-line type, curved type. Electric forklift operation properties make it wider range of load change. Under defining parameters and road conditions, positive torque load of steering wheel relates to its load. Therefore, steering wheel load signal—G is introduced in this paper. Combining speed signal—V with hand-force torque—Td, steering assistance characteristic curve of the forklift is determined, and its expression is shown in (1).

Assistance torque needed when steering:

$$M(V,G)=\begin{cases}0, & |Td|<Td0\\ f(V,G), & Td0\le|Td|\le Td\max\\ M\max, & |Td|>Td\max\end{cases} \qquad (1)$$

Taking 1.5t reach forklifts for example and measuring the front axle load and the biggest steering torque under different velocities, then be equivalent to the steering wheel, so motor steering torque is deduced by combining with fuzzy reasoning. By using the fuzzy control—FIS Editor function in MATLAB, and inputting speed—V, steering load—M, and outputting the biggest power-assisted steering torque, the three dimensional dynamical characteristics curve is plotted, as shown in figure. 2. It can illustrate that the size of power-assisted torque is closely related to the change of load. When speed is constant, the power-assisted torque is gradually increasing with the increase of load; when load is constant, the power-assisted torque is gradually decreasing with the increase of speed. Obviously, this power-assisted characteristic can meet requirements of steering performance. In addition, also considering steering road feel, the power-assisted torque needed is 0 when input hand torque is less than Td0; the power-assisted torque is Mmax when input hand torque is more than Tdmax. In this case, the Mmax is equal to 130N.

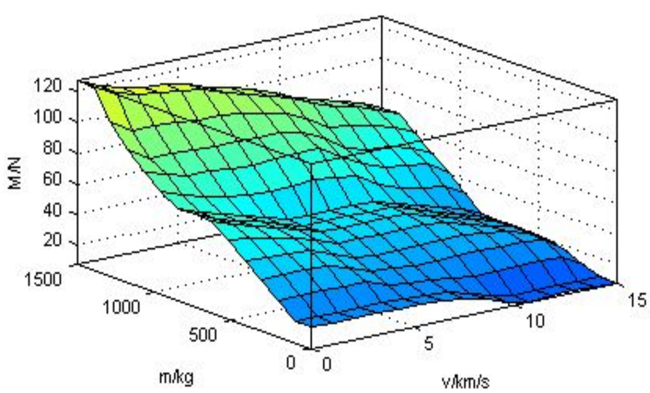

Figure 2. The three dimensional graph of the biggest power-assisted steering torque

According to the provision of handling portability in China automobile industry standard QC/T480-1999 (vehicle handling stability limit and evaluation method), steering wheel control force on average can't be greater than 50N and the biggest is less than 80N for vehicles whose maximum total mass is less than 2.5t . In this case, steering wheel hand torque—Td0 is equal to 5 N and T_{dmax} is 15 N. The relationship between power-assisted torque M and motor output torque is following as:

$$M = Tm \times gm \tag{2}$$

Combing the motor torque relationship: $Tm = ka \times I$, the target control current is following as：

$$I = \frac{M(Td - Td_0)}{ka \bullet gm(Td_{\max} - Td_0)} \tag{3}$$

In this equation， T_m is motor electromagnetic torque, and gm is motor reduction rate, and ka is electromagnetic torque coefficient.

IV. THE ESTABLISHMENT AND SIMULATION ANALYSIS OF THE FORKLIFT EPS SYSTEM MODEL

According to the Lagrange equations of the complete system under ideal constraints, the dynamical model of the EPS system is set up. The common form of Lagrange dynamics equation is:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_i}\right) - \frac{\partial L}{\partial q_i} = Q_{\mathrm{i}}, \quad i = 1, 2, \cdots, n \tag{4}$$

In this equation， q_i is generalized coordinate related to the system; Q_{i} is generalized active force corresponding to the generalized coordinate; n is degree of freedom of the system. Applying it to the forklift EPS system, the degree of freedom is to determine as 3. The corresponding generalized coordinate as followed: θ_{c} —angular displacement of steering input shaft angular displacement; θ_{m} —angular displacement of motor axis; θ_{g} —angular displacement of steering output shaft angular displacement.

The kinetic energy and potential energy are from input steering shaft, the motor, and steering gear:

$$T_M = T_{column} + T_{\text{motor}} + T_{\text{gear}} = \frac{1}{2} J_C \dot{\theta}_C^2 + \frac{1}{2} J_{\text{m}} \dot{\theta}_m^2 + \frac{1}{2} M_{\text{r}} r_{\text{p}}^2 \dot{\theta}_{\text{g}}^2 \tag{5}$$

$$V_M = V_{column} + V_{\text{motor}} + V_{\text{gear}} = \frac{1}{2} K_{\text{c}} \left(\theta_c - \theta_g\right)^2 + \frac{1}{2} K_{\text{m}} \left(\theta_m - g_m \theta_g\right)^2 + \frac{1}{2} K_{\text{g}} r_p^2 \theta_g^2 \tag{6}$$

So the Lagrange function is shown as followed:

$$L_M = T_M - V_{\text{M}} = \frac{1}{2} J_{\text{c}} \dot{\theta}_{\text{c}}^2 + \frac{1}{2} J_{\text{m}} \dot{\theta}_{\text{m}}^2 + \frac{1}{2} M_{\text{r}} r_p^2 \theta_g^2 - \frac{1}{2} K_{\text{c}} \left(\theta_c - \theta_g\right)^2 - \frac{1}{2} K_{\text{m}} \left(\theta_m - g_m \theta_g\right)^2 - \frac{1}{2} K_{\text{g}} r_p^2 \theta_g^2 \tag{7}$$

The EPS dynamics equation is shown as followed by differential computing:

$$\mathrm{J_c} \ddot{\theta}_{\text{c}} + \mathrm{K_c} (\theta_c - \theta_g) = \mathrm{Td} - \mathrm{B_c} \dot{\theta}_{\text{c}} \tag{8}$$

$$\mathrm{J_m} \ddot{\theta}_{\text{m}} + \mathrm{K_m} (\theta_m - g_m \theta_g) = \mathrm{T_m} - \mathrm{B_m} \dot{\theta}_{\text{m}} \tag{9}$$

$$\mathrm{M_r r_p} \ddot{\theta}_{\text{g}} - \frac{\mathrm{K_c}}{\mathrm{r_p}} (\theta_c - \theta_g) - \frac{g_m \mathrm{K_m}}{\mathrm{r_p}} (\theta_m - g_m \theta_g) + \mathrm{K_g r_p} \theta_g = -\mathrm{F_{re}} - \mathrm{B_g r_p} \dot{\theta}_{\text{g}} \tag{10}$$

The steering wheel torque equation is: $\mathrm{T_s} = \mathrm{K_c} (\theta_c - \theta_g)$

Voltage balance equation of the model of Dc power-assisted motor is:

$$\mathrm{U} = \mathrm{I_m} R + \mathrm{K_b} \dot{\theta}_{\text{m}} + L_m I_{\text{m}} \tag{11}$$

The meaning of parameters in equations and numerical value is shown in Table 1.

Table 1: System parameter list

symbol	**meaning**	**numerical value**	**symbol**	**meaning**	**numerical value**
Kc	Steering shaft stiffness coefficient	114.6Nm/rad	L	inductance of motor armature	2.34e-4H
Jc	moment of inertia of the input axle	0.089kgm^2	Jm	moment of inertia of motor	2.25e-4kgm^2
Bc	damping coefficient of the input axle	0.361Nm/s	Bm	damping coefficient of motor	0.0229Nm/s
gm	motor reduction rate	16.5	Jg	moment of inertia of output axle	0.023kgm2
Bg	damping coefficient of the output axle	0.023Nm/s	Ka	constant of motor electromagnetic torque	0.048N/A
R	resistance of motor armature	0.45Ω	Kb	back-EMF coefficient	0.0583V/rad/s

According to the above analysis, the model of forklift power-assisted steering system based on PID control in MATLAB is shown in figure. 3.

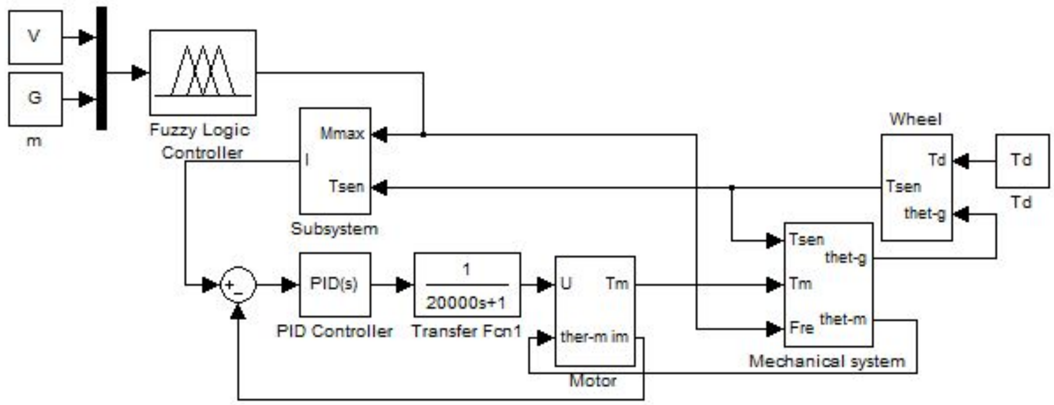

Figure 3. Simulation model diagram of Forklift EPS system PID control

At present, PID control algorithm is the common algorithm to track current in EPS system. EPS system is multiple-input multiple-output (MIMO) and has a certain degree of nonlinear and time-varying system containing structural perturbation and unstructured perturbation and so these uncertainty factors. While there is a change of system parameter or road agitation, PID algorithm on a certain optimal control parameters value cannot guarantee system to track the target current well. Control effect will become worse with the variation of parameters, whereas sliding mode controller can track the target current [14]. Sliding-mode control is popular with its robust and insensitive properties to matched disturbance and model uncertainty[15-17]. In this paper, the sliding control with reaching law is adopted. It can force system to move along the state trajectory of scheduled sliding mode.

Choosing the sliding mode control switching function s for current tracking error, it can be concluded that:

$$\dot{s} = \dot{I} - \dot{I}_m = \dot{I} - \frac{U - RI_{\mathrm{m}} - k_b\theta_m}{L} \qquad (12)$$

Choosing exponential reaching law with better reaching effect:

$$slaw = \dot{s} = -\varepsilon\,\mathrm{sgn}(s) - ks \qquad (13)$$

In this equation, if reaching coefficient $\varepsilon >$ max(d(t)) , then k > 0. d(t) is uncertainty of system and external disturbance and other factors. It suggests the following result according sliding equation: $\dot{s} = 0$.

$$U_e = L\dot{I} + RI_m + k_b\dot{\theta}_m \qquad (14)$$

Combing reaching law:

$$U_s = L[\varepsilon \mathrm{sgn}(s) + ks] \qquad (15)$$

So the sliding control strategy is as shown in the following equation:

$$U = U_e + U_s \qquad (16)$$

In the equation, Ue is equivalent control unit of sliding control, that is, certainty of Dc motor control system which is decided by the structure of system； Us is switching control of sliding control, that is, uncertainty of control system. By adjusting ε and k, steady state error can be under control, thus makes the system to be optimal. In order to reduce chattering when fast reaching, ε should be decreased and k increased at the same time. Using sliding mode variable structure control (SMVSC), the simulation model of forklift EPS system is shown as figure.4.

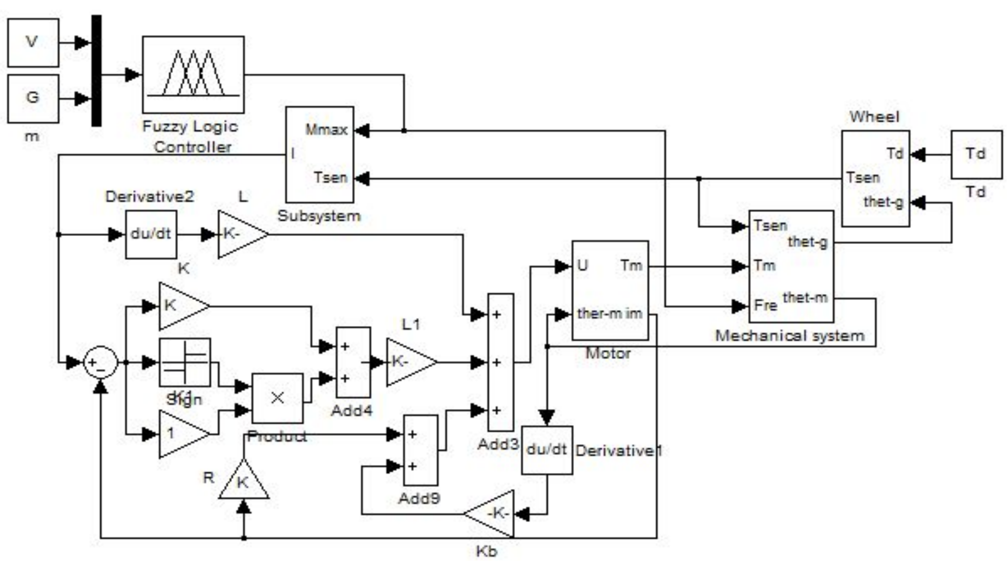

Figure 4. The simulation model diagram of forklift EPS system using SMVSC

In order to examine the effect of sliding mode control, simulation experiment is done. Setting the speed at 5Km/s, half load, and inputting hand torque step signal, there is no steady state error and overshoot in PID control and sliding mode control. Obviously, it can meet the demands at this time which is shown in figure. 5.

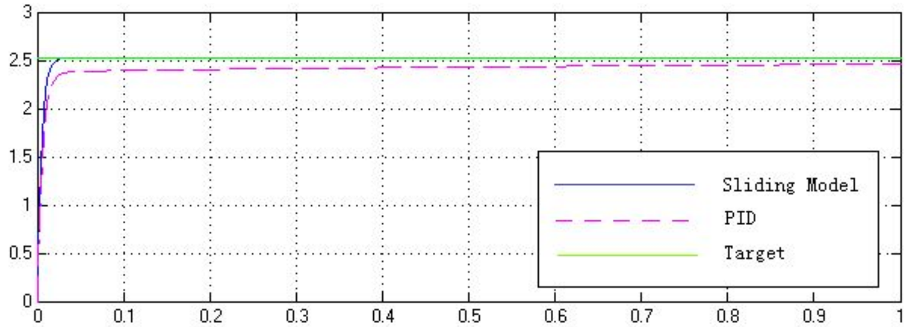

Figure 5. The comparison of tracking current of step power-assisted motors in different algorithms

Supposing the parameters are changing, and altering the value of motor armature resistance, the corresponding simulation diagram is shown in figure. 6. It is clear that there is no overshoot when the value of armature resistance is increased to 50%, but settling time becomes longer apparently. The state of system is unaffected and the situation of tracking target current is still good when using sliding mode control.

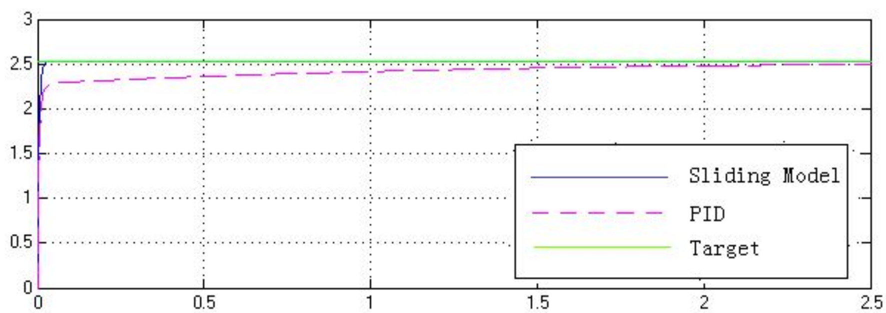

Figure 6. The comparison of tracking current of step power-assisted motors in different algorithms when changing parameters

In consideration of disturbance from ground when it runs, a continuous sine interference torque is added to tires in simulation. It turned out that the tracking effect of PID control algorithm is becoming worse and there are large fluctuations, yet sliding mode control can meet the demands though produced slight chattering when disturbed as is shown in figure. 7.

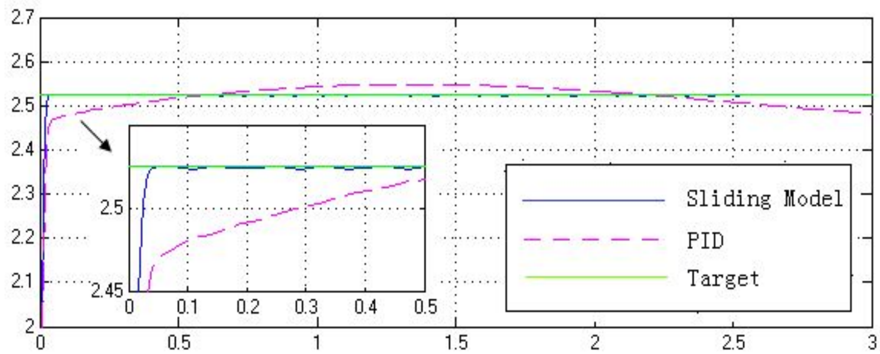

Figure 7. The simulation of tracking current of step power-assisted motors in different algorithms when disturbed

In conclusion, PID control can meet demands under a certain situation, but can't achieve satisfactory control effect once parameters are changed. It can't be used for uncertain system. For sliding mode control system, it has rapid response, and is not sensitive to parameters changes and disturbance, and is able to avoid system identification on line, but chattering effect needs to be eliminated.

V. CONTROL OF STEERING SPEED

When running on rapid steering condition, forklifts tend to be loss of stability so that it caused side slip and tumbling under centrifugal force. In order to avoiding this kind of traffic accidents, the value of steering acceleration should be limited. The driver should slow down to reduce centrifugal force which is proportional to lateral acceleration when taking curves or changing lanes. Only did like this, the security of rapid steering could be improved [18-20]. It can't be control very well only relying on experience. If the speed is not low enough, it would lead to accidents; if it is too low, it would be against transportation economy. So the problem of active slowdown can be solved satisfactorily if the max safety speed can be calculated in real time.

When the electric forklift is driving on a slope, if the angle of the slope is the largest overturning angle θ, that is to say, the electric forklift is in critical state, as is shown in figures 8 and 9.

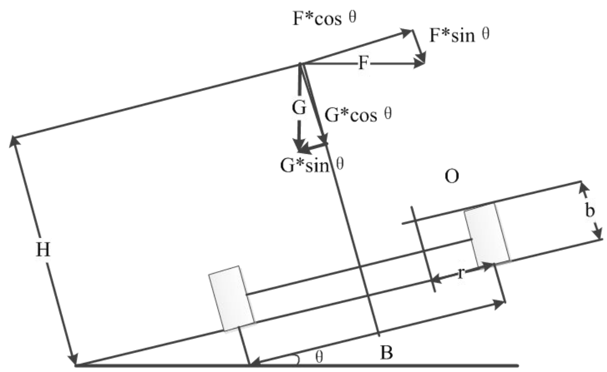

Figure. 8 The max safety speed

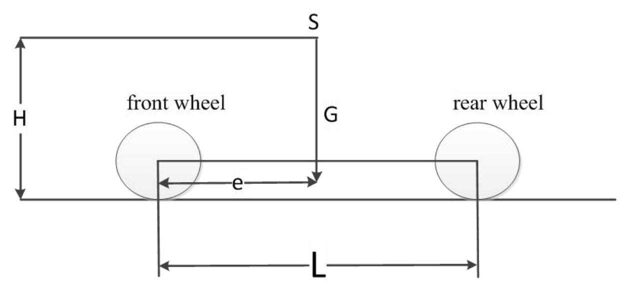

Figure. 9 The center of mass

In figure.8, the center of the overturning axis is at point O, and in figure. 9, the center of the vehicle gravity (c.g.) is at point S. So the forklift is affected by two kinds of force, one of which is resultant gravity G, the other is centrifugal force F. When the forklift is in the state as is shown in figure 8, the torque relative to the lateral overturning axis can be calculated as

$$(F\sin\theta + G\cos\theta)(\frac{B}{2} - \mathrm{r}) = (G\sin\theta + F\cos\theta)(H - b) \qquad (17)$$

As is known, the centrifugal force $F = mv^2 / R = ma$, the gravity G=mg. So equation (16) can be described as

$$(ma \cdot \sin\theta + mg \cdot \cos\theta)(\frac{B}{2} - \mathrm{r}) = (mg \cdot \sin\theta + ma \cdot \cos\theta)(H - b) \qquad (18)$$

Where b and r are the factors which can determine the location of the lateral overturning center. As can be known in figure. 8 and 9:

$$\begin{cases} \dfrac{r}{e} = \dfrac{B/2}{L} \\ \dfrac{\text{b}}{e} = \dfrac{H}{L} \end{cases} \tag{19}$$

The equation can be rewritten as follow:

$$\begin{cases} r = \dfrac{B/2}{L} \cdot e \\ b = \dfrac{H}{L} \cdot e \end{cases} \tag{20}$$

And it can be considered that $\tan\theta = \theta$ when θ is small enough. Besides, assume the distance between the center of mass and the front wheel is zero.

Combining equation (19) and (17), the acceleration can be derived

$$a = \frac{g\left[\theta(H-b) - (\frac{B}{2} - r)\right]}{\theta(\frac{B}{2} - r) - (H-b)} = \frac{g(\frac{B}{2} - \frac{B/2}{L} \cdot e)}{H - \frac{H}{L} \cdot e} = \frac{gB}{2H} \tag{21}$$

According to Newton's laws $F = ma = m\dfrac{v^2}{R}$ and equation (20), the max safety speed can be calculated as:

$$v_{\max} = \sqrt{\frac{gBR}{2H}} \tag{22}$$

Where the steering radius $R = L / \tan\theta$, as a result, the max safety speed can be rewritten as:

$$v_{\max} = \sqrt{\frac{gBL}{2H \cdot \tan\theta}} \tag{23}$$

According to the analysis above, if the forklift can reduce its speed actively when steering, it can avoid the roll-over accident and guarantee the steering stability and the steering security. Actually, in order to reduce the burden of the electronic control unit, it is general to use a fixed value deceleration project. In other word, firstly, list a table including angle range of the steering wheel and the deceleration values, secondly, determine deceleration range according to the angle of the steering wheel measured by angle sensor on electric forklift, and then look up the table to search corresponding deceleration value, lastly, subtract this fixed corresponding deceleration value from current real-time speed. Through this method is simple, it can't adapt to variable load characteristics of electric forklift very well due to sudden jump.

In this paper, a method which taking two curves to fit and optimize the speed is adopted. It can control speed actively to ensure safety steering. Three elements can lead to these accidents, such as road adhesion coefficient, load, turning radius, etc. Picking on three measurable elements, the optimized expression of speed is shown as followed:

$$V_{optimized} = F(v, T_{sen,} G) \qquad (24)$$

Refer to (23), v stands by speed; T_{sen} stands by hand torque detected by torque sensor; G stands by axle weight of steering wheel. Apparently, it is a complicated multiple element non-linear function. There is a need for designing or fitting to simplify. Supposing that three elements work independently, the expression can be transformed into the product of three separate unary function. That is:

$$V = F(T_{sen},)*f(G)*v \qquad (25)$$

Refer to (24), it is clear to see load size contributes to different steering trends. By the literature [21-24], load mass is a big reason to change motion trail. With the addition of mass, the trend or over-steering is increasing. When the load size adds up to a certain degree of quality, the trend became weaker. If it continues adding, the trend remained stable. Accordingly, the relationship between the over-steering factor and load can be obtained as is shown in Table 2.

Table 2: The relationship between the over-steering factor and load

G	0	100	200	300	400	500	600	700
f(G)	1	0.98	0.96	0.94	0.92	0.90	0.88	0.86
G	800	900	1000	1100	1200	1300	1400	1500
f(G)	0.86	0.88	0.85	0.82	0.79	0.76	0.75	0.75

The following expression can fit for the change of over-steering factor:

$$f(G) = A*G^2 + B*G + C \qquad (26)$$

With the increasing of the torque, the speed dropped gradually to ensure safe steering. The exponential function can fit for empirical data.

$$F(T_{sen}) = e^{aT_{sen}+b} \qquad (27)$$

After fitting by MATLAB,

$$V_{optimized} = \{(-1.535e-8)*G^2 + (-1.4023e-4)*G + 0.9984\}*\left(e^{-0.0465T_{sen}+0.1971}\right)*V \quad (28)$$

Adopting this control strategy in steering ECU, the optimal speed value can be calculated, and then was sent to the drive controller of steering which can control the motor speed to make the forklifts' speed approaching safety speed. For example, for the forklift with 8km/s, slope increasing torque was sent to steering wheel under no-load, one-half load, full load. So the speed when steering is shown in figure. 10. As you can see, with the increasing of load and steering torque, safety speed reduced smoothly to ensure steer safely.

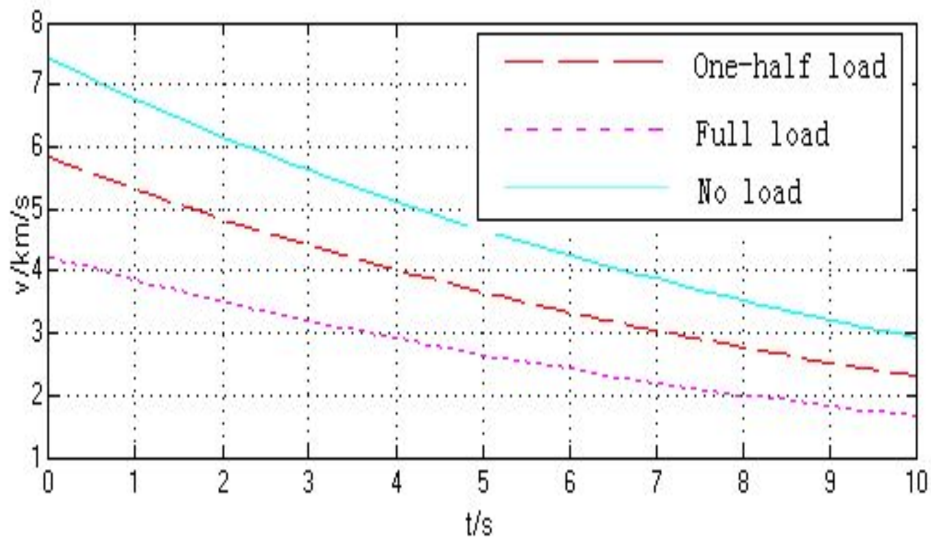

Figure 10. The safety speed changing curve under different steering conditions

VI. CONCLUSIONS

In recent years, the popularity rate of electric forklifts is growing as for main kind of transporting vehicles in the logistics system. The application area has changed into all walks of life from port and pier in the past. As one of the key components of forklifts, steering system decided whether it can work flexibly. This paper aimed at the simulation of dynamical characteristic curve, EPS of forklift, the assistance current control of EPS system, and safety steering speed. Sliding mode controller can track the target current and control effect will become better as the change of parameters. Adding the optimal speed to controller can ensure steer safely. It has obtained anticipatory effect and can meets actual requirements, and part of research results have been successfully used in the actual forklift EPS system.

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