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RESEARCH ON POWER CHARACTERISTIC OF THE ELECTRIC FORKLIFT EPS SYSTEM

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Abstract - This paper has given the structure, operating principle, and force analysis of electric power steering (EPS) system aimed at a type of forklift. Combing with forklift operating characteristic, three variable power characteristics curve based on steering wheel torque, real-time speed, and load is designed, so is the three dimensional diagram of forklift power gradient by using fuzzy rule. The dynamic model of EPS system and two-degree-of-freedom linear model of forklift dynamics are established. This paper presents a simulation on the basis of three variable power characteristics and dynamic model. The results show that forklift EPS system can provide appropriate power according to the changes of speed and load. It also can meet the demands of coordination between steering portability and road sense.

Index terms: EPS system of forklift, power gradient, three variable power characteristics curve, modeling, simulation.

I. INTRODUCTION

The application of forklift is changed from ports and wharfs in the past to every aspect of life now, and is widely used to load and unload indoors and outdoors. Being a key part of forklift, steering system performance determined its safety operating.

With the continuous renewal of automotive electronic technology, there are different degrees of breakthrough in terms of EPS system research in every way, and the scope of EPS application also expanded gradually. In recent years, achievements have been obtained in EPS and forklift control system both in China and abroad[1-5].

Nowadays, forklift EPS research focus on building more precise EPS model, descripting power characteristics curve, improving control strategy and fault diagnosis, and the reliability of controller[6-12].

This paper content and innovation mainly includes: (1) give the basic structure and operating principle of electric power steering (EPS) system aimed at a type of forklift. (2) analyze basic requirements of the design of power characteristics curve. Considering load changes have a greater influence on forklift steering, this paper introduces weight signal. Combining speed and hand force of steering handwheel, the power gradient is deduced and target current is calculated by using fuzzy inference, thus a more reasonable power characteristics curve is obtained. The advantage of fuzzy inference is to improve the power characteristics by adjusting the membership functions and the control rules. (3) MATLAB simulation of steering portability and road sense.

II. THE STRUCTURE AND OPERATING PRINCIPLE OF FORKLIFT EPS SYSTEM

Taking a type of forklift as an example, EPS system of a steering axle-type is adopted in the mechanical structure. Its system structure is as shown in figure 1. It mainly includes car battery (DC48V), steering pile pillar assembly (including steering handwheel, steering axle of input and output), power motor assembly (including motor, retarding mechanism, and electromagnetic clutch), electronic control unit (including all kinds of sensor and controller) and so on[13].

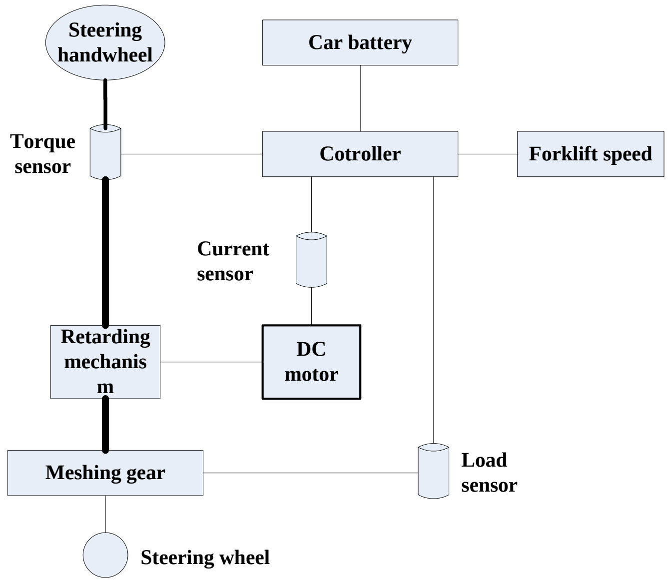


Figure 1. Forklift EPS system structure

(1) The operating principle

When the forklift does not steer, the EPS does not work; when has any steering actions, electric signal is entered into electric control unit (ECU) to provide reasonable signal fitted for real-time forklift condition[14]. ECU calculates power target current of this time based-on a certain control algorithm, then power motor outputs corresponding torque, and it is transferred to steering mechanism by transmission mechanism to generate power, so as to realize the real-time power control.

(2) Force analysis of electric forklift EPS system

EPS system mainly includes the hand torque from driver, power torque from motor, and the total steering resistance torque. The sum of hand torque and power torque keeps balance with resistance torque. According to steering signal, power motor provides appropriate power torque. The sum of hand and power torque acts on steering axle to overcome resistance torque from itself and outside, so achieves steering.

III. THE DESIGN OF POWER CHARACTERRISTIC CURVE FOR FORKLIFT EPS SYSTEM

a. Forklift power characteristic

The determination of dynamical characteristics is one of the core issues of EPS system. It means the relationship between the power (provided by expected system) and real-time steering conditions. The main involved problem is the balance between steering portability and road sense. Common dynamical characteristics curve is usually expressed as the relationship among motor output torque and steering wheel input torque, speed and other variables. The design's purpose is to determine present control target and determines the power magnitude provided by power motor, that is the control rule of EPS system[15].

Combining with forklift operating features, we can infer that ideal power characteristic curve should have the following features:

(1) If hand input torque by the driver is in a small range, the power motor doesn't work in order to keep a certain degree of road sense. As the speed increases, the scope of no working becomes bigger correspondingly.

(2) If steering in situ, power mechanism provides maximum power to help the driver reduce energy consumption.

(3) If at lower speed, power mechanism provides appropriate power.

(4) The power characteristic cure reflects changes of forklift load.

(5) If speed is bigger than the pre-set limit value, the electromagnetic clutch is separated, so the power steering system is switched to purely mechanical steering system[16].

b. The determination of forklift power characteristic curve

Electric forklift is different from other common passenger vehicles. Its bigger self-weight and load changes influence on power characteristic. That is to say, keeping the same system parameters and road surface conditions, there is a corresponding relationship between resistance force and load weight. Therefore, steering wheel load signal—*G* is introduced in this paper. Combining speed signal—*V* with hand-force torque—T_d, steering assistance characteristic curve of the forklift is determined.

Power torque needed when steering is shown in (1):

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

Because $f(V,G,T_d)$ is a function of three variables, it is difficult to fit and design. To simplify, put this function into a product of a binary function and a unary function. It is shown in (2).

$$M(V,G,T_{\mathrm{d}})=\begin{cases} 0, & |T_d|\le T_{d0} \\ K(V,G)*f_1(T_d-T_{\mathrm{d}0}), & T_{d0}<|T_d|\le T_{d\max} \\ M_{\max}, & |T_d|>T_{d\max} \end{cases} \qquad (2)$$

In this equation, f_1 is function about $T_d-T_{\mathrm{d}0}$. Define k as power gradient, $k=K(V,G)$. It is binary function of axle load and steering speed. It can get the value of power torque which should be provided from power gradient at any time.

Normally, the table of steering power torque in specific axle load and speed can be list by experimental values, so the corresponding power gradient can be calculated and the model of power gradient can be fitted. In an actual situation, it is unrealistic and unnecessary to divide input variable—speed. Firstly, drivers usually judge speed and load by their experience and habit. The second, electric forklift EPS system is complex, nonlinear and uncertain. Because it is influenced by road friction and outside disturbance, so it is difficult to establish a precise model to present the relationship among motor power, speed, steering wheel torque, and load. These factors suggest that it is a good choice to introduce fuzzy relation into the design of power steering characteristic curve.

Taking a type of forklift with 1.5 tons rated load as an example, its biggest speed is 15Km/h. We can set the hand torque value: T_{d0}=3Nm, $T_{d\max}$=15Nm. The table of the biggest torque ($M_{\max}$) needed with different load and speed when steering is shown in Table 1.

Table 1: The biggest torque needed when steering

Speed (Km/h)	Non-load (0Kg)	Half load (750Kg)	Full load (1500Kg)
0	30	62	133
5	25	52	105
10	20	42	85
15	16	32	65

We can get the gradient between steering torque and power torque under different speed and load. Its expression is shown in (3).

$$k = \frac{M_{\max} - T_{\mathrm{dmax}}}{T_{\mathrm{dmax}} - T_{\mathrm{d0}}} \quad (3)$$

According to (3), the value of gradient can be inferred as shown in Table 2.

Table 2: Typical value of gradient

Speed (Km/h)	Non-load (0Kg)	Half load (750Kg)	Full load (1500Kg)
0	1.250	3.917	9.833
5	0.833	3.083	7.500
10	0.417	2.250	5.833
15	0.083	1.417	4.167

Input "Fuzzy" in the command window of MATLAB to callout fuzzy control toolbox. There are two input variables and one output variables. We can get a fuzzy control module to be edit, which is with load (*G*) and speed (*V*) for input. After fuzzification of input variables, process of fuzzy reasoning, and defuzzification of output variable, the gradient (*k*) of motor power can be got. The advantage of fuzzy reasoning is to use adjustable membership function and control rule to perfect power characteristic.

The membership function of each fuzzy controller parameter in this example is shown in figure 2, 3, 4.

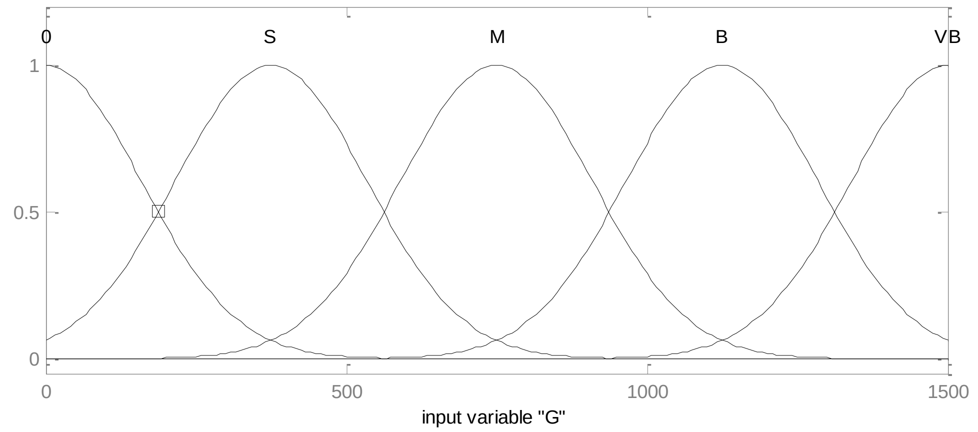


Figure 2. The membership function of input variable *G*

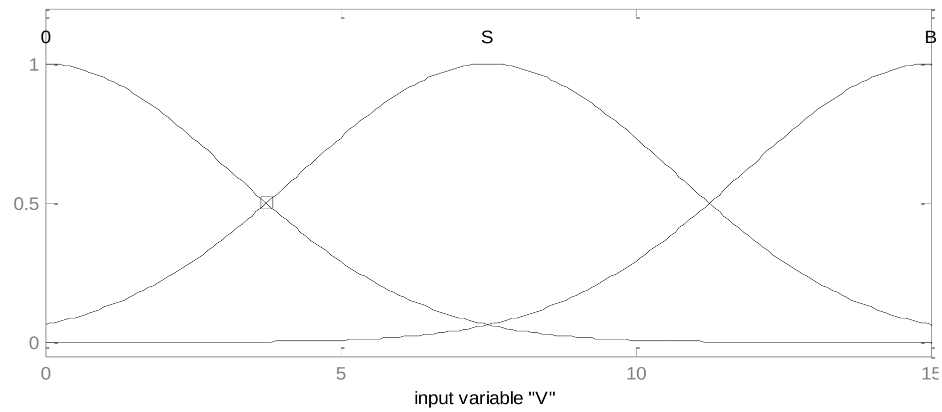


Figure 3. The membership function of input variable *V*

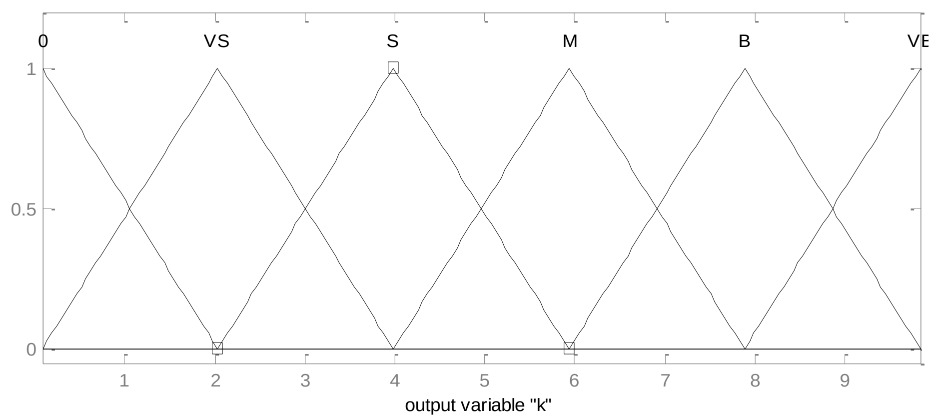


Figure 4. The membership function of output variable *k*

The editing interface of fuzzy controller is shown in figure 5.

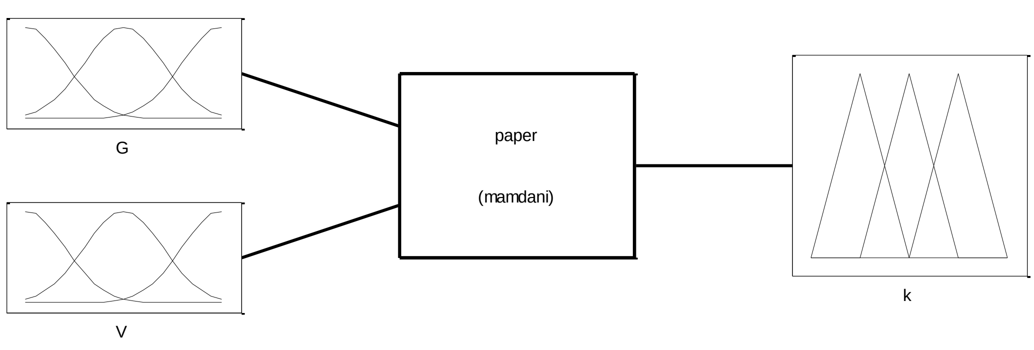


Figure 5. The editing interface of fuzzy controller

The fuzzy control rule of this example is shown in Table 3.

Table 3: The fuzzy control rule

v \ k \ G	0	S	M	B	VB
0	VS	S	M	B	VB
S	0	VS	S	M	B
B	0	0	VS	S	M

The three-dimensional diagram of power gradient can be described by the above fuzzy control relationship as shown in figure 6. This figure is a three-variables surface based on real-time speed, steering wheel torque, forklift load.

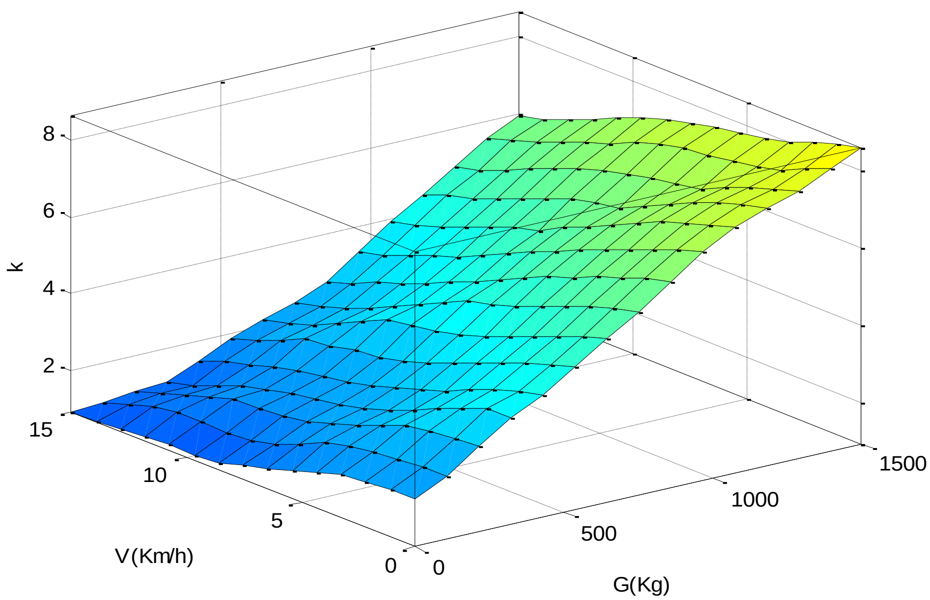


Figure 6. The three-dimensional diagram of power gradient

It can be inferred that: when keeping the invariable speed, power gradient and torque are increased with the increasing load, when keeping the invariable load, power gradient and torque are increased with the decreasing speed. It turns out that this power characteristic can satisfy the demand of steering portability. Considering steering road sense, if hand torque is less than T_{d0}, motor doesn't provide power torque, if hand torque becomes bigger, power torque is increased, if hand torque is bigger than $T_{d\max}$, power torque achieves the biggest value and doesn't increase.

IV. THE MODEL ESTABLISHMENT FOR FOERKLIST EPS SYSTEM

a. Dynamic model establishment for EPS system

Based on structure of forklift in figure 1, let it be supposed that: where J_k, J_c, J_m *are* moment of inertia of steering wheel, steering mechanism, motor, and B_k, B_c, B_m are damping coefficient of them respectively. T_k, T_f, T_m, T_a, T_n are steering wheel input torque, equivalent steering resistance torque of output axle, motor electromagnetic torque, motor output torque, and output torque of torque sensor. θ_k, θ_c, θ_m are steering wheel angle, which is equivalent to the output axle from steering wheel, motor angle. g is transmission ratio of retarding mechanism (from motor to output axle). K_s is stiffness of torque sensor.

a.i The establishment of power motor model

The equation of permanent magnet DC power motor is as following in (4)[17].

$$U = E + RI + L\,{}^{\mathrm{d}I}\!/_{\mathrm{d}t} = K_b\omega + RI + L\,{}^{\mathrm{d}I}\!/_{\mathrm{d}t} \qquad (4)$$

Where U is motor voltage, E is back EMF produced by motor, R is equivalent resistance in motor loop, L is motor inductance, K_b is back EMF coefficient, ω is motor speed.
Equation (5) and (6) can be inferred by the analysis for the mechanical part of motor. Where K_a is coefficient of motor electromagnetic torque.

$$T_m - \frac{T_\mathrm{a}}{g} = J_m \frac{\mathrm{d}\theta_m^2}{dt} + B_m \frac{\mathrm{d}\theta_\mathrm{m}}{dt} \qquad (5)$$

$$T_m = K_\mathrm{a} I \qquad (6)$$

a.ii The establishment of steering system model

Ignoring motor stiffness, the relationship between motor angle and output axle angle can be expressed as equation (7).

$$\theta_\mathrm{m} = g\theta_c \qquad (7)$$

By modeling for steering wheel dynamics:

$$T_k - T_n = J_k \frac{\mathrm{d}\theta_k^2}{dt} + B_k \frac{\mathrm{d}\theta_\mathrm{k}}{dt} \qquad (8)$$

By modeling for torque sensor dynamics:

$$T_n = K_s\,(\theta_\mathrm{k} - \theta_c) \qquad (9)$$

By modeling for output axle dynamics:

$$T_\mathrm{n} + T_\mathrm{a} - T_f = J_c \frac{\mathrm{d}\theta_c^2}{dt} + B_c \frac{\mathrm{d}\theta_\mathrm{c}}{dt} \qquad (10)$$

If forklift's tire is turning in a small angle, its lateral deviation characteristic appears to be a linear relationship, Where K_c is equivalent output axle stiffness of lateral deviation.

$$T_\mathrm{f} = K_c \theta_c \delta \qquad (11)$$

The relationship between practical power torque and motor output torque is shown in (12).

$$M = T_m g \qquad (12)$$

b. Dynamic model establishment for forklift

Considering that the two-degree-of-freedom linear model of forklift dynamics is in common use, in this model, lateral acceleration is less than 0.4g. Suppose some secondary causes can be ignored to highlight features of lateral movement in simplified figure. If the vehicle is driving at constant speed, only movements on horizontal plane will be considered[13,18,19], the characteristic of tires keeps constant, and vehicle's load is ignored, two-degree-of-freedom linear model of forklift dynamics can be established as shown in figure 7.

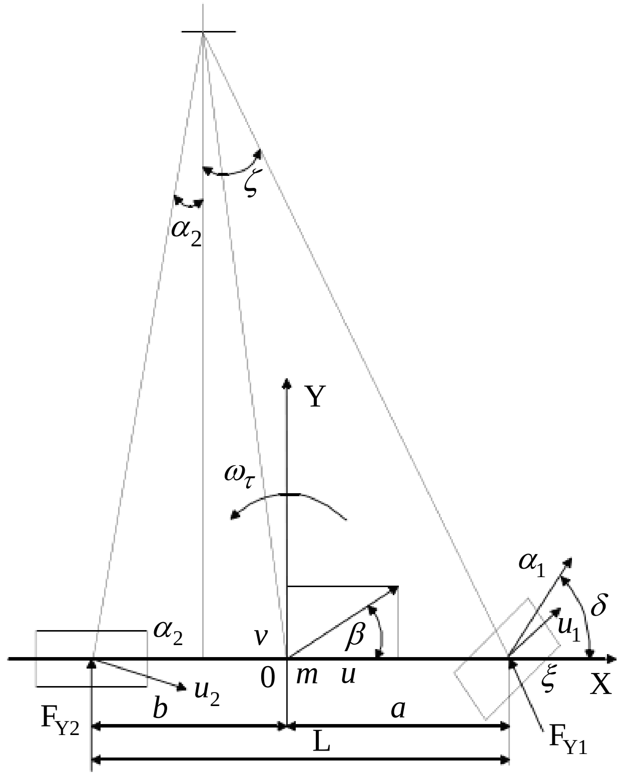


Figure 7. The dynamical forklift model based on two-degree-of-freedom

In figure 7, the resultant force along with Y axle from external force can be expressed as:

$$\sum F_Y = F_{Y1}\cos\delta + F_{Y2} = ma_y \tag{13}$$

The sum of torque around the center of mass:

$$\sum M_Z = aF_{Y1}\cos\delta - bF_{Y2} = I_z\dot{\omega}_z \tag{14}$$

Where F_{Y1}, F_{Y2} are the cornering forces of front and back wheels. v, u are speed component along with X axle and Y axle. k_1, k_2 are stiffness of lateral deviation of front and back wheels. δ is angle of steering wheel, a is distance from the center of front wheel to the center of mass, b is distance from the center of back wheel to the center of mass, m is mass of forklift, I_z is moment of inertia around the Z axle, ω_{r} is yaw rate, $\dot{\omega}_z$ is yaw acceleration.

Considering δ is small enough, make this equation true: $\cos\delta = 1$. By taking the limit and ignoring second differential parts, the following two equations can be inferred.

$$\sum F_Y = k_1\alpha_1 + k_2\alpha_2 \tag{15}$$

$$\sum M_Z = ak_1\alpha_1 - bk_2\alpha_2 \tag{16}$$

Supposing the speed of front axle and back axle are u_1, u_2, their side slip angle is α_1, α_2. The side slip angle of the center of mass is $\beta, (\beta = v/u)$. ξ is the included angle between u and X axle. Its expression is as follows:

$$\xi = \frac{v + a\omega_r}{u} = \beta + \frac{a\omega_r}{u} \tag{17}$$

The side slip angle of front axle and back axle can be inferred by composition principle.

$$\alpha_1 = -(\delta - \xi) = \beta + \frac{a\omega_r}{u} - \delta \tag{18}$$

$$\alpha_2 = \frac{\mathrm{v} - b\omega_r}{u} = \beta - \frac{b\omega_r}{u} \tag{19}$$

Combing all equations, the expression of two-degree-of-freedom movement can be inferred as follows:

$$\begin{cases} (k_1 + k_2)\beta + \frac{1}{u}(ak_1 - bk_2)\omega_r - \mathrm{k}_1\delta = m(\dot{v} + uw_r) \\ (ak_1 - bk_2)\beta + \frac{1}{u}(a^2k_1 + b^2k_2) - ak_1\delta = I_z\dot{\omega}_z \end{cases} \tag{20}$$

V. THE SIMULATION OF POWER CHARACTERISTIC FOR FORKLIFT EPS SYSTEM

a. Determination of simulation parameters

Taking a type of forklift with 1.5 tons rated load as an example, its simulation parameters is shown in the following Table 4.

Table 4. Simulation parameters

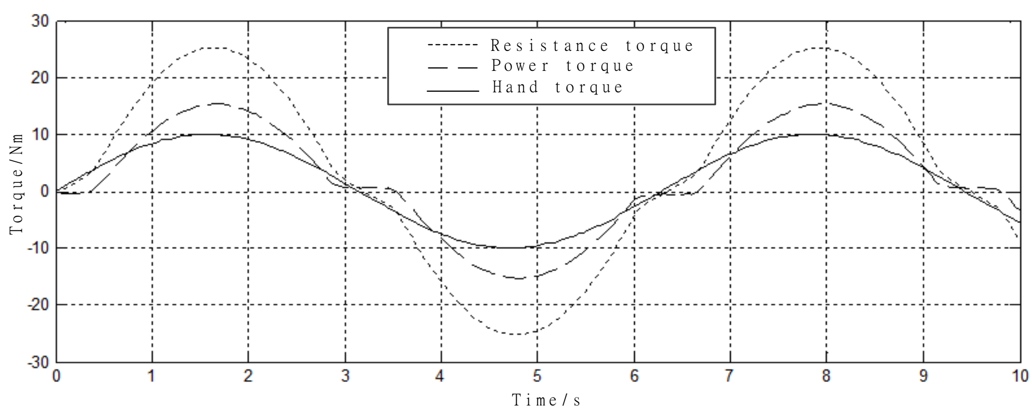
symbol/(unit)	meaning	numerical value	symbol/(unit)	meaning	numerical value
K_c/(N m rad^{-1})	Steering axle stiffness coefficient	114.600	J_g/(kg m^2)	moment of inertia of output axle	0.023
J_c/(kg m^2)	moment of inertia of the input axle	0.089	K_b/(V s rad^{-1})	back-EMF coefficient	0.0583
B_c/(N m s^{-1})	damping coefficient of the input axle	0.361	K_a/(N m A^{-1})	constant of motor electromagnetic torque	0.180
G	motor reduction rate	16.500	m/(kg)	mass of forklift	2340

$B_g/(N\ m\ s^{-1})$	damping coefficient of the output axle	0.023	$a/(m)$	distance from the center of front wheel to the center of mass	1.352
R/Ω	resistance of motor armature	0.450	$b/(m)$	distance from the center of back wheel to the center of mass	1.485
L/H	inductance of motor armature	2.339	$k_1/(N\ rad^{-1})$	stiffness of lateral deviation of front wheels	-62618
$J_m/(kg\ m^2)$	moment of inertia of motor	2.250/10 4	$k_2/(N\ rad^{-1})$	stiffness of lateral deviation of back wheels	-110245
$B_m/(N\ m\ s^{-1})$	damping coefficient of motor	0.029	$I_z/(kg\ m^2)$	moment of inertia around the center of mass	3895

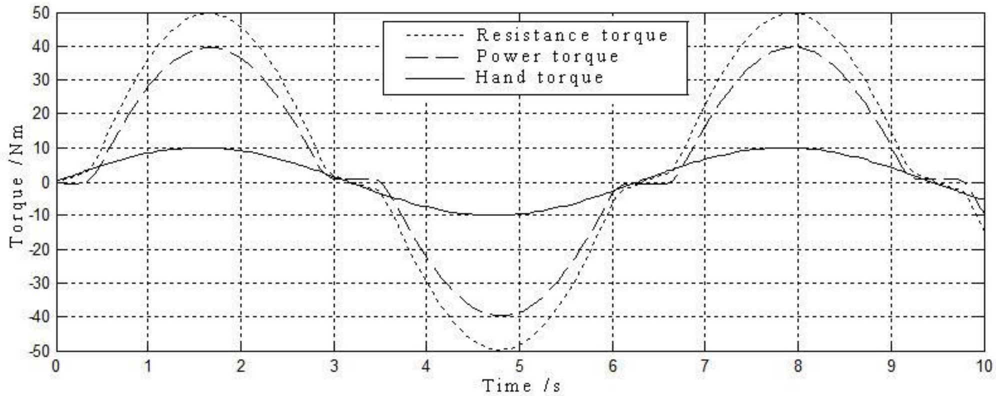
b. Simulation of steering portability and road sense

Improving steering portability in situ or at low speed is the basic function of power steering system. This paper analyzes it from the relationship between hand torque and power torque in different speed and load. Firstly, establish dynamical model and power characteristic model in MATLAB, and make a sine wave (its amplitude is 10Nm) for input to simulation continuous steering from left to right. The simulation in different speed and load is shown as following figures.

b.i When steering in situ

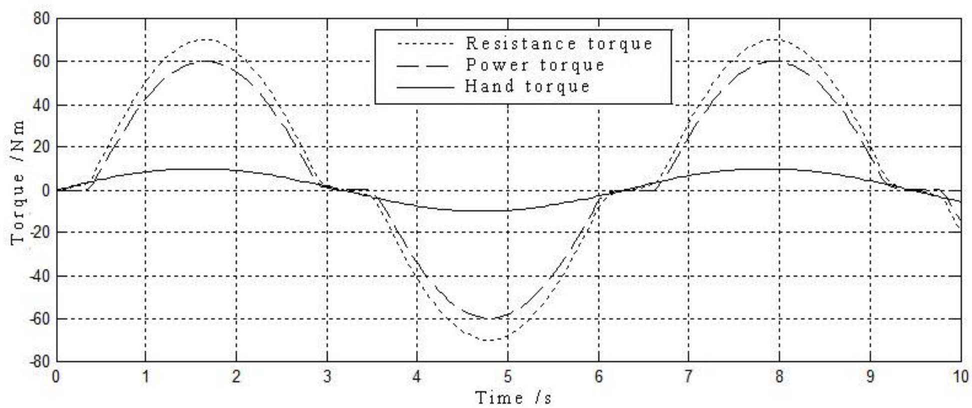


(a) steering in situ with non-load



(b) steering in situ with half load

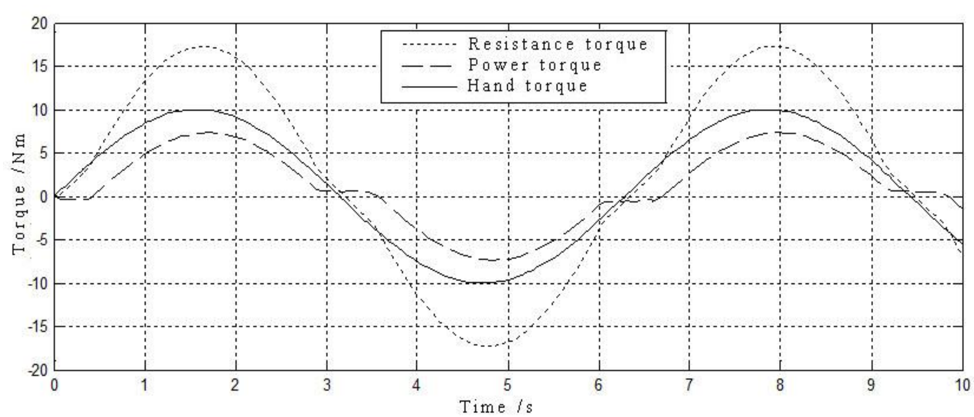
Resistance torque



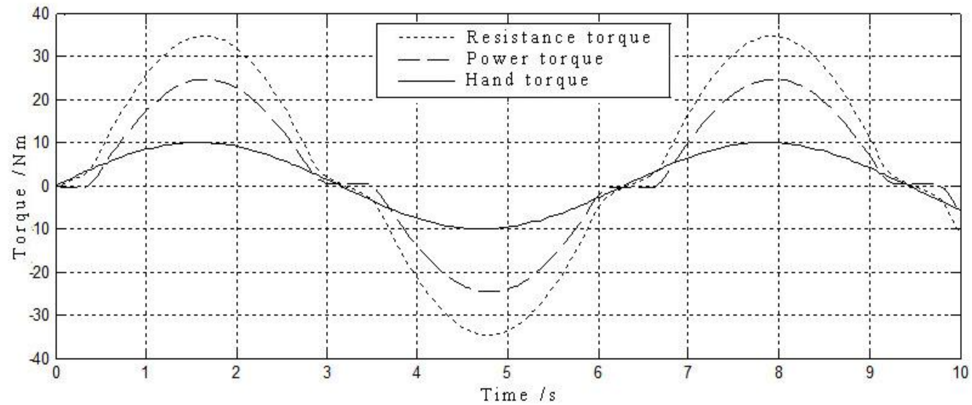
(c) steering in situ with full load

Figure 8. Simulation comparison when steering in situ

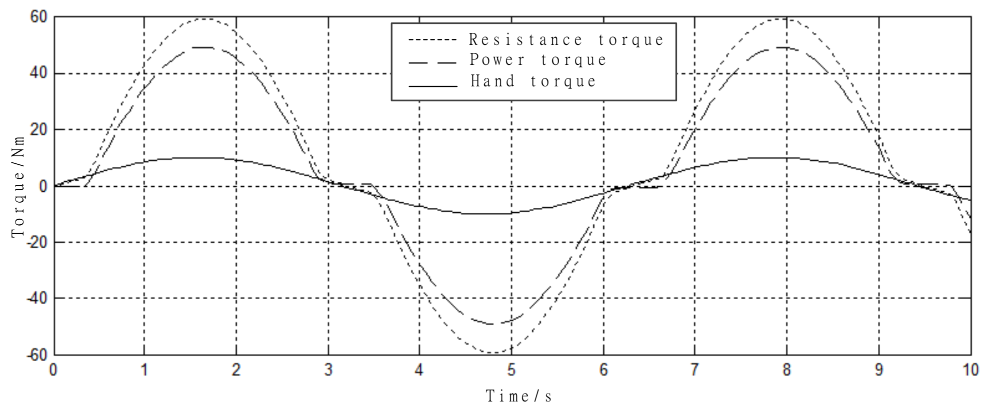
b.ii When steering at 10km/h



(a) when steering with non-load at 10km/h



(b) when steering with half load at 10km/h



(c) when steering with full load at 10km/h

Figure 9. Simulation comparison when steering at 10km/h

Comparing figure 8 and 9, the resistance torque when steering in situ is bigger than the torque of driving obviously, and motor provides bigger power to meet the demand of steering portability. When steering at 10km/h, resistance torque becomes smaller if remains unchanged hand torque, and motor power becomes smaller. It can provide better road sense to help drivers perceive road conditions. By the comparison of different load, resistance torque increases as load becomes heavier to save drivers' energy.

VI. CONCLUSIONS

As the development of EPS system becomes more mature, it is widely used in forklift and gets more and more attention. This paper takes a type of forklift as an example, and presents the structure, operating principle and force analysis of EPS system. Three variable power characteristics curve based on steering wheel torque, real-time speed, and load is established by using fuzzy control rule. The innovation of this paper is that load changes are being reflected on the target current calculation of power motor.

The dynamic model of EPS system is established. The simulation results show that three variable power characteristics curve can meet the demands of coordination between steering portability and road sense.

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