PARAMETER OPTIMIZATION OF ELECTRIC POWER STEERING INTEGRATED WITH ACTIVE FRONT STEERING FUNCTION

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Abstract: The dynanaic model of a novel electric power steering(EPS) system integrated with active front steering function and the three-freedom steering model are built. Based on these models, the concepts and the quantitative expressions of road feel, sensitivity, and operation stability of the steering are introduced. Then, according to constrained optimization features of multi-variable function, a genetic algorithm is designed. Making the road feel of the steering as optimization objective, and operation stability and sensitivity of the steering as constraints, the system parameters are optimized by the genetic and the coordinate rotation algorithms. Simulation results show that the optimization of the novel EPS system by the genetic algorithm can effectively improve the road feel, thus providing a theoretical basis for the design and optimization of the novel EPS system.

Key words: vehicle engineering; electric power steering; active front steering; road feel; genetic algorithm

CLC number: U461.4 **Document code**: A **Article ID**: 1005-1120(2012)01-0096-07

Nomenclature

$g/(\mathrm{m\cdot s^{-2}})$ Aacceleration due to gravity

$u/(\mathrm{m\cdot s^{-1}})$ Velocity

$\omega_r/(\mathrm{rad\cdot s^{-1}})$ Yaw rate

ϕ/rad Roll angle of vehicle

β/rad Sideslip angle of vehicle centre of mass

δ/rad Steer angle of front wheels

α_1/rad Sideslip angle of front wheels

α_2/rad Sideslip angle of front wheels

$k_1/(\mathrm{N\cdot rad^{-1}})$ Cornering stiffness coefficient of rear wheels

$k_2/(\mathrm{N\cdot rad^{-1}})$ Cornering stiffness coefficient of rear wheels

a/m Distance between vehicle centre of mass and front axle

b/m Distance between vehicle centre of mass and rear axle

m/kg Mass of vehicle

m_s/kg Mass of sprung

$I_x/(\mathrm{kg\cdot m^2})$ Inertia moment of sprung mass about x-axis

$I_z/(\mathrm{kg\cdot m^2})$ Inertia moment of sprung mass about z-axis

$I_{xz}/(\mathrm{kg\cdot m^2})$ Inertia product of sprung mass about x, z-axis

E_1 Front roll steer coefficient

E_2 Rear roll steer coefficient

$C_{\phi 1}/(\mathrm{N\cdot m\cdot rad^{-1}})$ Stiffness coefficient of roll angle of front suspension

$C_{\phi 2}/(\mathrm{N\cdot m\cdot rad^{-1}})$ Stiffness coefficient of roll angle of rear suspension

$D_1/(\mathrm{N\cdot m\cdot s\cdot rad^{-1}})$ Damping of roll angle of front suspension

$D_2/(\mathrm{N\cdot m\cdot s\cdot rad^{-1}})$ Damping of roll angle of rear suspension

$J_s/(\mathrm{kg\cdot m^2})$ Inertia moment of input shaft

θ_s/rad Rotation angle of input shaft

$B_s/(\mathrm{N\cdot m\cdot s\cdot rad^{-1}})$ Damping coefficient of input shaft

$T_{sen1}/(\mathrm{N\cdot m})$ Anti-torque of input shaft

$T_h/(\mathrm{N\cdot m})$ Torque of input shaft

Foundation items: Supported by the National Natural Science Foundation of China(51005115); the Risiting Scholar Foundation of the State Key Lab of Mechanical Transmission in Chongqing University (SKLMT-KFKT-201105); the Science Fund of State Key Laboratory of Automotive Satefy and Energy in Tsinghua University(KF11202).

Received date: 2011-01-15; **revision received date**: 2011-10-12

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$K_s/(N \cdot m \cdot rad^{-1})$ Stiffness coefficient of input shaft
θ_p/rad Rotation angle of AFS motor stator
u_q/V Voltage of q-axis of AFS motor
i_q/A Current of q-axis of AFS motor
u_d/V Voltage of d-axis of AFS motor
i_d/A Current of d-axis of AFS motor
R_s/Ω Armature resistance of AFS motor
$k/(A \cdot N^{-1} \cdot m)$ Voltage gain coefficient of AFS motor
k_f Feedback current coefficient of AFS motor
L_s/H Armature inductance of AFS motor
$w_{rf}/(rad \cdot s^{-1})$ Angular velocity of AFS motor rotor relative to its rotor
e_f/V Rotation potential of AFS motor
p Pole pairs of AFS motor
$T_s/(N \cdot m)$ Electromagnetic torque of AFS motor
$k_e/(N \cdot m \cdot A^{-1})$ Torque coefficient of AFS motor
$J_{p1}/(kg \cdot m^2)$ Inertia moment of AFS motor stator
$B_p/(N \cdot m \cdot s \cdot rad^{-1})$ Damping coefficient of AFS motor stator
$J_{p2}/(kg \cdot m^2)$ Inertia moment of AFS motor rotor
θ_w/rad Rotation angle of AFS motor rotor
$T_{sen2}/(N \cdot m)$ Anti-torque of AFS motor rotor
θ_e/rad Rotation angle of output shaft
$K_a/(N \cdot m \cdot A^{-1})$ Moment coefficient of assist motor
$B_m/(N \cdot m \cdot s \cdot rad^{-1})$ Damping coefficient of assist motor
θ_m/rad Rotation angle of assist motor
$T_m/(N \cdot m)$ Electromagnetic torque of assist motor
$T_a/(N \cdot m)$ Output torque of assist motor
$J_m/(kg \cdot m^2)$ Inertia moment of assist motor
$K_m/(N \cdot m \cdot rad^{-1})$ Output stiffness coefficient of assist motor
G Power transmission ratio of assist motor
U/V Terminal voltage of assist motor
L/H Inductor of assist motor
I/A Current of assist motor
R/Ω Armature resistance of assist motor
$K_b/(N \cdot m \cdot rad^{-1})$ Back-EMF coefficient of assist motor
$J_e/(kg \cdot m^2)$ Inertia moment of output shaft
$B_e/(N \cdot m \cdot s \cdot rad^{-1})$ Damping coefficient of output shaft
$T_r/(N \cdot m)$ Anti-torque of output shaft
$b_r/(N \cdot m \cdot rad^{-1})$ Damping coefficient of rack
x_r/m Displacement of rack
m_r/kg Effective mass of rack and pinion
r_p/m Radius of pinion
F_{TR}/N Axial thrust exerted on rack
$k_r/(N \cdot m^{-1})$ Equivalent elasticity coefficient
F_δ/N Random signal of pavement
M_r/kg Effective mass of reduction gear, pinion and rack
$B_r/(N \cdot m^{-1} \cdot s)$ Effective damping coefficient of reduction gear, pinion and rack
$K_r/(N \cdot m^{-1})$ Equivalent elasticity coefficient of tire, pinion and rack
$T_n/(N \cdot m)$ Torque of torque sensor
$K/(A \cdot N^{-1} \cdot m)$ Power gain coefficient
n_1 Transmission ratio of AFS motor
n_2 Transmission ratio between steering screw and steering wheels
$\boldsymbol{X}$ Design variable vector
lb Lower bound of ideal energy range of steering sensibility
ub Upper bound of ideal energy range of steering sensibility

INTRODUCTION

The electric power steering (EPS) system, firstly used for mini car, has already experienced development for twenty years since 1988[1-2]. It solves the problems associated with the hydraulic power steering (HPS). The motor only operates when steering assistance is needed, and hydraulic pump and piping are eliminated. In EPS, the static torque boost curves can be adjusted by modifying software in the electronic controllers without changing the torsion bars, and alternated according to vehicle velocity to improve steering feel.

Active front steering (AFS) system was introduced to improve handling stability under adverse road conditions[3-4]. Compared with the conventional steering system, the mechanical linkage between the steering wheel and the front wheels of an AFS system is complemented by an extra angle augment motor. Therefore, a small auxiliary front wheel angle, in addition to the steering angle imposed by the driver, can be used to stabilize the vehicle besides improving vehicle steering responses and avoiding critical handling situations[5-6].

At present, the EPS system cannot achieve variable transmission ratio control and active steering control, also cannot improve the steering control stability of steering system[7-8]. The existing AFS system uses the commercial HPS system, and it is complex and inevitably has the hydraulic oil leakage[9]. Additionally, compared with the EPS system to achieve power, the AFS system is difficult to achieve the returnability and damping control[10].

Therefore, on the basis of the AFS technology integrated with EPS technology, the steering system which can provide the function of both EPS and AFS systems will be the main development direction of the future automotive power steering technology. In this paper, a novel EPS system integrated with AFS function is developed. Firstly, the performance indexes of road feel, sensitivity, and operation stability of the steering are proposed. Then, the influence of parameters on the novel EPS system is analyzed. Lastly, the genetic algorithm (GA) and the coordinate rotation algorithm (CRA) are used to optimize the structure parameters of the novel EPS system, thus providing a theoretical basis for design and selection of the novel EPS system.

1 SYSTEM MODELING

The model of the novel EPS system is shown in Fig. 1. It includes two motors : One motor is assist motor used to provide assistance torque for the EPS control, and the other is AFS motor used to provide additional steering angle for the AFS control.

1.1 Three-freedom vehicle model

The dynamic differential equation for the three-freedom vehicle can be expressed by

$$\begin{cases} I_z\dot{\omega}_r - I_{xz}\ddot{\phi} = 2ak_1\alpha_1 - 2bk_2\alpha_2 \\ mu(\omega_r + \dot{\beta}) - m_sh\ddot{\phi} = 2k_1\alpha_1 + 2k_2\alpha_2 \\ I_x\ddot{\phi} - m_su(\omega_r + \dot{\beta})h - I_{xz}\dot{\omega}_r = \\ -(D_1 + D_2)\dot{\phi} - (C_{\phi1} + C_{\phi2} - m_sgh)\phi \end{cases} \tag{1}$$

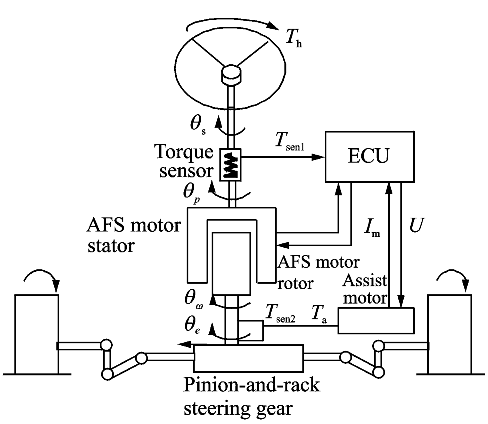

Fig. 1 Model of novel EPS system

where α_1, α_2 can be described as follows

$$\begin{cases} \alpha_1 = \beta + \dfrac{a\omega_r}{u} + E_1\varphi - \delta \\ \alpha_2 = \beta - \dfrac{b\omega_r}{u} + E_2\varphi \end{cases} \tag{2}$$

1.2 Input shaft model

Considering the viscous damping of the input shaft and steering wheel moment of inertial, the dynamic equation for the input shaft can be expressed as

$$J_s\ddot{\theta}_s + B_s\dot{\theta}_s = T_h - T_{sen1} \tag{3}$$

By analyzing the steering column and sensor torque, T_{sen1} can be expressed as

$$T_{sen1} = K_s(\theta_s - \theta_p) \tag{4}$$

1.3 AFS motor stator model

Considering the viscous damping of the AFS motor, the dynamic equation for the AFS motor stator can be expressed as

$$J_{p1}\ddot{\theta}_p + B_p\dot{\theta}_p = T_{sen1} - T_s \tag{5}$$

1.4 AFS motor rotor model

Considering the viscous damping of the AFS motor, the dynamic equation for the AFS motor rotor is derived as

$$J_{p2}\ddot{\theta}_w + B_p\dot{\theta}_w = T_s - T_{sen2} \tag{6}$$

For the output shaft, the anti-torque T_{sen2} can be written as

$$T_{sen2} = K_s(\theta_w - \theta_e) \tag{7}$$

1.5 Assist motor model

The equation for the electromagnetic torque can be given by

$$T_m = K_aI \tag{8}$$

Based on the dynamic analysis of the mechanical parts, the dynamic equation for the assist motor can be given by

$$J_m\ddot{\theta}_m + B_m\dot{\theta}_m = T_m - T_a \tag{9}$$

In the actual control, the equation for the assist torque can be written as

$$T_a = K_m(\theta_m - G\theta_e) \tag{10}$$

1.6 Output shaft model

Based on the dynamic analysis of the output shaft, the dynamic equation can be expressed as

$$J_e\ddot{\theta}_e + B_e\dot{\theta}_e = T_{sen2} + GT_a - T_r \tag{11}$$

1.7 Rack and pinion model

The dynamic equation for the rack and pinion can be given by

$$m_r \ddot{x}_r + b_r \dot{x}_r = \frac{T_r}{r_p} - F_{TR} \tag{12}$$

where F_{TR} is

$$F_{TR} = k_r x_r + F_\delta \tag{13}$$

The angle of rotation of the output shaft can be given by

$$\theta_e = \frac{x_r}{r_p} \tag{14}$$

According to Eqs. (1,5-7,9,11-12), the system dynamic equation can be expressed as follows

$$\begin{cases} J_s \ddot{\theta}_s + B_s \dot{\theta}_s + K_s \theta_s = T_h - K_s \theta_p \\ J_{p1} \ddot{\theta}_p + B_p \dot{\theta}_p + K_s \theta_p = K_s \theta_s - T_s \\ J_{p2} \ddot{\theta}_w + B_p \dot{\theta}_w + K_s \theta_w = K_s x_r \dfrac{x_r}{r_p} + T_s \\ J_m \ddot{\theta}_m + B_m \dot{\theta}_m + K_m \theta_m = T_m + G K_m \dfrac{x_r}{r_p} \\ M_r \ddot{x}_r + B_r \dot{x}_r + K_r x_r = G K_m \theta_m / r_p + \\ \qquad K_s \theta_w / r_p - F_\delta \end{cases} \tag{15}$$

where

$$\begin{cases} M_r = m_r + \dfrac{J_e}{{r_p}^2} \\ B_r = b_r + \dfrac{B_e}{{r_p}^2} \\ K_r = k_r + (K_s + G^2 k_m)/{r_p}^2 \end{cases} \tag{16}$$

2 STEERING PERFORMANCE

The novel EPS system should have following performance: good road feel, good steering sensitivity, and good steering stability. In this paper, the concepts and quantitative expressions of the novel EPS system are introduced.

2.1 Road feel

The road feel is analyzed by fastening the steering wheel. In this way, the interference information can be completely transferred to the driver. Additionally, it is easier to analyze the system, since one-freedom is reduced.

Assume that

$$\theta_s = 0 \tag{17}$$

When the assist motor is assumed as using current control strategy, and the torque sensor is reduced to the torsion bar spring, the measured value of the torque sensor can be expressed as

$$T_n = T_{sen1} = K_s(\theta_s - \theta_p) \tag{18}$$

Based on the current control strategy, there is

$$I = K T_n \tag{19}$$

Then T_m can be described according to Eqs. (11, 17-19), shown as

$$T_m = - K K_a K_s \theta_p \tag{20}$$

When the steering wheel and the steering screw of the steering actuator are equivalent, and the rotation angles of the AFS motor rotor and the output shaft are assumed to be equal, the steering system model can be simplified as follows

$$\theta_m = G\theta_e, \theta_p = n_1 \theta_w, \theta_w = \theta_e = n_2 \delta \tag{21}$$

With Eqs. (3-6, 9-11, 15-16, 20-21), the transfer function from T_r to T_h is obtained. This transfer function is just road feel and can be expressed as

$$E(s) = \frac{T_h(s)}{T_r(s)} = \frac{n_1 K_s}{X_1 s^2 + Y_1 s + Z_1} \tag{22}$$

where X_1, Y_1, Z_1 can be described as follows

$$\begin{cases} X_1 = J_e + n_1 J_{p1} + J_{p2} + G^2 J_m \\ Y_1 = B_e + n_1 B_p + B_p + G^2 B_m \\ Z_1 = n_1 K_s + G n_1 K K_a K_s \end{cases} \tag{23}$$

2.2 Steering sensitivity

The steering sensitivity is defined as the ratio between the vehicle yaw-rate and the angle of rotation of the steering wheel, its transfer function can be given by

$$\frac{\omega_r(s)}{\theta_s(s)} = \frac{\omega_r(s)}{\delta(s)} \frac{\delta(s)}{\theta_s(s)} \tag{24}$$

With Eqs. (4-6,8-11,18-19,21), we have

$$\frac{\delta(s)}{\theta_s(s)} = \frac{G K K_a K_s + K_s}{X_2 s^2 + Y_2 s + Z_2 + \dfrac{2dk_1}{n_2}\left(\dfrac{a}{u} \dfrac{\omega_r(s)}{\delta(s)} + \dfrac{\beta(s)}{\delta(s)} + E_1 \dfrac{\phi(s)}{\delta(s)} \right)} \tag{25}$$

where X_2, Y_2, Z_2 can be written as

$$\begin{cases} X_2 = n_2 G^2 J_m + n_1 n_2 J_{p1} + n_2 J_{p2} + n_2 J_e \\ Y_2 = n_2 G^2 B_m + n_1 n_2 B_p + n_2 B_p + n_2 B_e \\ Z_2 = n_1 n_2 K_s + n_1 n_2 G K K_a K_s - \dfrac{2dk_1}{n_2} \end{cases} \tag{26}$$

From Eqs. (1-2), they can be obtained as follows

$$\frac{\omega_r(s)}{\delta(s)}=\frac{\sum_{i=0}^{3}A_is^i}{\sum_{i=0}^{4}B_is^i},\frac{\beta(s)}{\delta(s)}=\frac{\sum_{i=0}^{3}F_is^i}{\sum_{i=0}^{4}B_is^i},\frac{\phi(s)}{\delta(s)}=\frac{\sum_{i=0}^{2}H_is^i}{\sum_{i=0}^{4}B_is^i} \tag{27}$$

2.3 Steering stability

Steering stability is stability of the novel EPS system and the vehicle system, and it should be guaranteed firstly. Therefore, it is necessary to study which circumstance can guarantee the stability of vehicle.

The denominator of the transfer function of the steering sensibility is chosen as the characteristic equation.

$$Q_6s^6+Q_5s^5+Q_4s^4+Q_3s^3+Q_2s^2+Q_1s^1+Q_0=0 \tag{28}$$

where Q_6, Q_5, Q_4, Q_3, Q_2, Q_1, Q_0 can be described as follows

$$\begin{cases} Q_6=X_2B_4 \\ Q_5=X_2B_3+Y_2B_4 \\ Q_4=X_2B_2+Y_2B_3+Z_2B_4 \\ Q_3=X_2B_1+Y_2B_2n_2+Z_2B_3+\dfrac{2dk_1}{n_2}\left(\dfrac{aA_3}{u}+F_3\right) \\ Q_2=X_2B_0+Y_2B_1+Z_2B_2+ \\ \qquad \dfrac{2dk_1}{n_2}\left(\dfrac{a}{u}A_2+F_2+E_1H_2\right) \\ Q_1=Y_2B_0+Z_2B_1+\dfrac{2dk_1}{n_2}\left(\dfrac{a}{u}A_1+F_1+E_1H_1\right) \\ Q_0=Z_2B_0+\dfrac{2dk_1}{n_2}\left(\dfrac{a}{u}A_0+F_0+E_1H_0\right) \end{cases} \tag{29}$$

In accordance with the Routh criterion, it requires that all formulas in the first column of the Routh table are positive.

3 PARAMETER OPTIMIZATION

3.1 Steering stability optimization

Some parameters of vehicle cannot be changed, such as the power gain coefficient of the assist motor K, which changes with the velocity of vehicle. Additionally the viscous friction coefficient of all parts cannot be adjusted, the power transmission ratio G always tends to be restricted in the optimal design, so the transmission ratio of the AFS motor n_1, the stiffness coefficient of the torque sensor K_s and the moment of inertia of the assist motor J_m are designed as the optimal variables.

In this paper, the optimal design variable is designed as $\boldsymbol{X}=(n_1,K_s)$ for two-parameter optimization of the novel EPS system. The bound of design variable is given as $\boldsymbol{X}_{\min}=(0.1,100)$, $\boldsymbol{X}_{\max}=(32,350)$ respectively, and the initial design variable is taken as $\boldsymbol{X}_0=(1,200)$. For three-parameter optimization, the optimal design variable is designed as $\boldsymbol{X}=(n_1, K_s, J_m)$, the bound of design variable is given as $\boldsymbol{X}_{\min}=(0.1,100,0.002)$, $\boldsymbol{X}_{\max}=(32,350,0.01)$ respectively, and the initial design variable is taken as $\boldsymbol{X}_0=(1,200,0.008)$.

3.2 Objective function optimization

In order to make the information from the pavement transfer to the driver′s hand as much as possible, it requires that the mean of the frequency domain energy of the road feel within a certain range of frequency domain can be as higher as possible. The mean of the frequency domain energy of the road feel in effective frequency range $(0,\omega_0)$ of the pavement information is defined as the objective function $f(\boldsymbol{X})$, and ω_0 is set to 40 Hz in the optimal design. The objective function $f(\boldsymbol{X})$ is described as

$$f(\boldsymbol{X})=\frac{1}{2\pi\omega_0}\int_0^{\omega_0}\left|\frac{T_h(s)}{T_r(s)}\right|^2 d\omega \tag{30}$$

3.3 Constraint conditions

The steering stability conditions of the novel EPS system must be satisfied, that is the transfer function of the steering sensibility must meet the Routh criterion, so the steering stability conditions could be written as

$$Q_6>0, Q_5>0, a_1>0, b_1>0, c_1>0$$
$$d_1>0, Q_0>0 \tag{31}$$

In the same way, in order to guarantee the driver get a good steering sensitivity, it requires that the mean of the frequency domain energy of the steering sensitivity within a certain range of frequency domain remains in a reasonable area. The mean energy of the steering sensitivity in the effective frequency range $(0,\omega_0)$ of the pavement information is defined as the objective function $g(\boldsymbol{X})$, and ω_0 is also set to 40 Hz in the optimal design. The objective function $g(\boldsymbol{X})$ is described as

$$\begin{cases} g(\boldsymbol{X}) = \dfrac{1}{2\pi\omega_0}\displaystyle\int_0^{\omega_0}\left|\dfrac{\omega_r(s)}{\theta_s(s)}\right|^2_{s=j\omega}\mathrm{d}\omega \\ g(\boldsymbol{X}) \in [lb, ub] \end{cases} \tag{32}$$

3.4 Optimization method

The flow chart of GA is shown in Fig. 2.

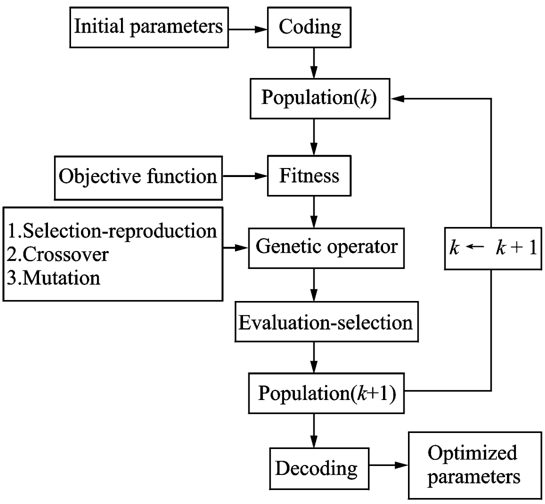

Fig. 2 Flow chart of GA

The optimization model of the road feel energy with GA can be described as

$$\begin{cases} \min(-f(\boldsymbol{X})) \\ \text{S. t.}\begin{cases} Q_6 > 0 \quad Q_5 > 0 \quad a_1 > 0 \quad b_1 > 0 \quad c_1 > 0 \\ d_1 > 0 \quad Q_0 > 0 \\ lb \leqslant g(\boldsymbol{X}) \leqslant up \end{cases} \end{cases} \tag{33}$$

3.5 Optimization results

With GA, the population fitness values with generation change are shown in Fig. 3. The optimization result is $\boldsymbol{X} = (6.257\ 29, 284.798\ 5, 0.009\ 82)$, that is $n_1 = 6.388\ 77$, $K_s = 278.334\ 27$ N·m/rad, $J_m = 0.01$ kg·m^2.

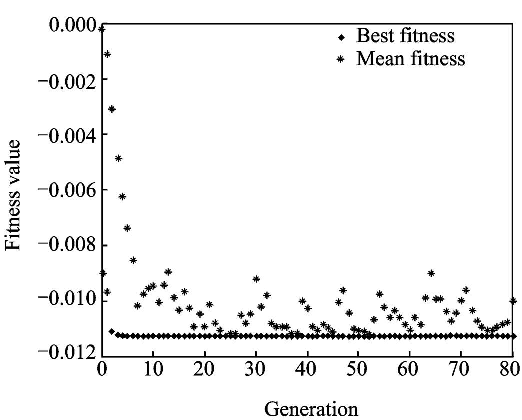

Fig. 3 Fitness population with GA (three-parameter)

With CRA, the optimization result is $\boldsymbol{X} = (3.644\ 4, 350.000\ 0, 0.003\ 8)$, that is $n_1 = 3.644\ 4$, $K_s = 350.000\ 0$ N·m/rad, $J_m = 0.003\ 8 \times 10^{-3}$ kg·m^2.

The road feel energy of three-parameter with different algorithms is shown in Fig. 4. From Figs. 3-4, with GA, the road feel energy after optimization is 0.011 287, increased 3.514 8 times than that without optimization (2.5×10^{-3}). With CRA, the road feel energy after optimization is 8.4×10^{-3}, increased 2.260 0 times than that without optimization (2.5×10^{-3}). The road feel energy optimized with GA is increased by 34.37% than that optimized with CRA. It shows that the novel EPS system with GA of three-parameter optimization can make the road feel attain optimal result.

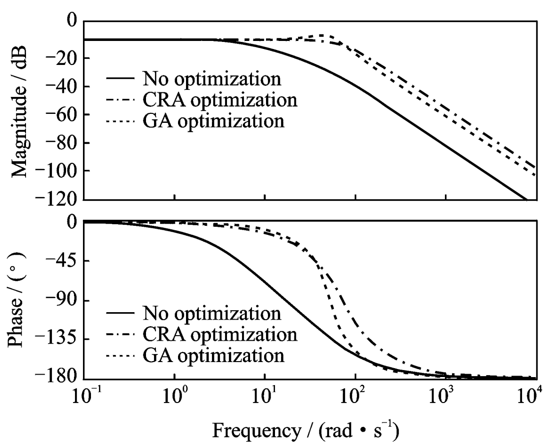

Fig. 4 Road feel energy with different algorithms (three-parameter)

4 CONCLUSION

In this paper, the novel EPS integrated with AFS function is proposed. The mathematical model for the novel EPS system and the three-freedom steeing model are built. Then, the concepts and the quantitative expressions of road feel, sensitivity, and operation stability of the steering are introduced. These parameters are optimized by GA and CRA, and results show that the road feel of the novel EPS system is effectively improved.

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集成主动转向功能的电动助力转向参数优化

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摘要:建立了融合主动转向功能的电动助力转向系统(新型EPS系统)动力学模型和整车三自由度转向模型;提出了新型EPS系统转向路感、转向灵敏度和转向稳定性的概念及量化公式;并根据多元函数有约束优化问题的特点设计了遗传算法。以转向路感为优化目标,以转向稳定性和转向灵敏度为约束条件,对EPS系统参数进行了多参数变量优化设计。仿真结果表明:应用多目标遗传算法进行EPS系统参数优化可在保证系统具有较好的转向稳定性和转向灵敏度基础上,有效提高了系统的转向路感,为新型EPS系统的设计和优化提供了理论基础。

关键词:车辆工程;电动助力转向;主动前向转向;路感;遗传算法

中图分类号:U461.4

(Executive editor: Zhang Huangqun)

基金项目:国家自然科学基金(51005115)资助项目;重庆大学机械传动国家重点实验室2011年度开放基金(SKLMT-KFKT-201105)资助项目;清华大学安全与节能国家重点实验室开放基金(KF11202)资助项目。