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Vehicle suspension and steering nonlinear integrated system coordinated control based on human-vehicle function allocation

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ABSTRACT

The coupled dynamics between the vehicle chassis suspension system and electrical power steering system(EPS) is analyzed, to establish the full-vehicle nonlinear model, EPS model, tire model and road input model. The tire's complex nonlinear model is approximated by utilizing the least square method, so as to obtain the integrated system model with the 22th-order. To simplify the nonlinear controller design, the nonlinear dynamics model is separated into two parts of nonlinear part and linear part. The state feedback optimal controller is designed for the linear part model, and the linear compensator based on the deviation separation is for the nonlinear part model, which can ensure the closed-loop control system is exponential asymptotic stability at the equilibrium point. The human-vehicle function allocation is adopted to adjust the two subsystems' output weights based on the fuzzy rules, to restrain the suspension roll motion and adaptively compensating driver's steering torque. The considerable simulations are carried out, and the results demonstrate that the suspension and EPS coordinated control system by applying bilinear control can obtain better performance than another nonlinear control method; the human-vehicle function allocation can further improve the vehicle whole-region control performance.
Keywords: suspension, steering, human-vehicle function allocation, bilinear control

1. INTRODUCTION

The vehicle control technology is an important way to improve the vehicle performance, which is paid special attention to by the researchers and engineers. The chassis is one of the most important parts in the vehicle, whose control system research and development is always the hot topic. Recently, it is very easy to find that the chassis dynamics is usually described as linear models[1]. Whereas, the linear chassis system model ignores much dynamic motion relation, which cannot fully reflect the vehicle dynamic characteristics. The nonlinear model can better describe the different motions and the coupled relations between the subsystems of the vehicle chassis[2,3].

The driver and vehicle are good at different functions. In order to improve the vehicle's whole performances, the relations between them need to be paid attention to, making them play their own respective advantages, so as to make the drivers have very good steering portability and road feeling, and the vehicle has better control performance. At the basis of considering their respective function and the vehicle current condition, the control function is allocated to different controlled objects to make one driver-vehicle integration control system with global best performance. The research about the driver-vehicle function allocation and active control is not found. In the literature [4], the driver's model, kinematic vehicle model and closed-loop control system are combined to give a mathematical driver model in the spatial equation form, which takes into account the previewed information of the path; Lyapunov-Krasovskii functional method is applied to analyze the exponential safety conditions of the human-vehicle-road control system based on this model. In [5], a driver-vehicle closed loop system simulation model including direction fuzzy PID optimized by genetic algorithms, speed fuzzy integrated control driver model and entire vehicle riding dynamics model is established, which simulates and analyzes typical modes, such as longitudinal speed one-way variation, lateral double lane and big curvature test road. Zhou et al[6] consider the yaw dynamics for the four-wheel drive vehicle, and the driver is applied to improve the fuzzy control system's robust performance. Cao et

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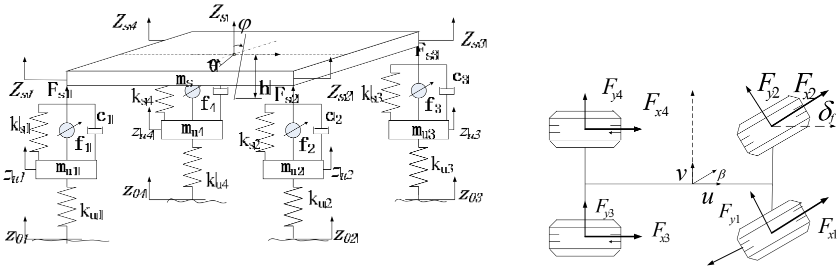
al[7] adopt the time-lag control method to analyze the driver and vehicle performance of the four-wheel steering nonlinear system. The related research has not referred to the function allocation between the vehicle and driver to realize the system function improvement.

Therefore, in order to analyze motion relationship and dynamics mutual association between the chassis suspension and steering system, In this paper, the movement coupling relations between the automobile chassis systems, steering and suspension are considered when non-linear model of the vehicle, the EPS model, tire model and filtered white noise road input model is established. On this basis, the complex nonlinear model is simplified to a polynomial nonlinear model, and then vehicle chassis coordinated control system is designed based on human-vehicle function allocation. By fuzzy rules setting the weight of the suspension subsystem's output force and steering subsystems power torque, the sub-control systems are coordinated to reach the global best performance. The bilinear control, H_∞ control are used to design nonlinear system controllers to build coordination control systems for simulation analyzing. Through simulation results' comparative analysis, it is to reflect the superiority of human-vehicle function allocation.

2. SYSTEM DYNAMIC MODELING

2.1 Vehicle nonlinear dynamic model

Considering the vehicle body's lateral motion, the vertical, yaw, pitch and roll motion and the vertical motion of the wheel, the 9-DOF vehicle nonlinear dynamic model is built[8]. Assuming that the left and right wheel longitudinal force is not differ largely, the vehicle is driving at a constant speed, and the disturbance from the lateral wind is not considered.



(a) Suspension part (b) Steering part

Figure 1–Framework of vehicle chassis system

Vehicle lateral motion model: $m(\ddot{y}_s + u\dot{\omega}) + m_s(-h_s\ddot{\varphi} + h_s\dot{\theta}\dot{\omega} - \dot{z}_s\dot{\varphi}) = \sum F_y$

Vehicle yaw motion model: $I_z\ddot{\omega} - I_{zx}\ddot{\varphi} + I_{zx}\dot{\theta}\dot{\omega} + (I_{yx} - I_{zx})\dot{\varphi}\dot{\theta} - m_s L_s \dot{z}_s \dot{\varphi} = \sum M_z$

Vehicle body vertical motion model: $m_s(\ddot{z}_s - u\dot{\theta} + \dot{y}_s\dot{\varphi}) - m_s h_s(\dot{\theta}^2 + \dot{\varphi}^2) + m_s L_s \dot{\varphi}\dot{\omega} = \sum F_{si}$

Vehicle body roll motion model: $I_x\ddot{\varphi} - I_{zxu}\ddot{\omega} - (I_{zs} - I_{ys} - m_s h h_s)\dot{\theta}\dot{\omega} - I_{zxs}\dot{\varphi}\dot{\theta} + m_s h \dot{z}_s\dot{\varphi}$

$$- m_s(\ddot{y}_s + u\dot{\omega})h = \sum M_{xs}$$

Vehicle pitch motion model: $I_{ys}\ddot{\theta} + (I_{xs} - I_{zs})\dot{\varphi}\dot{\omega} - I_{zxs}(\dot{\omega}^2 - \dot{\varphi}^2) = \sum M_{ys}$

Vehicle wheel vertical motion model: $m_{ui}\ddot{z}_{ui} + F_{si} - F_{0i} = 0$

$$F_{si} = k_{si}(z_{ui} - z_{si}) + c_i(\dot{z}_{ui} - \dot{z}_{si}) + f_i$$

$$F_{0i} = k_{ui}(z_{0i} - z_{ui})$$

$$\sum F_y = (F_{y1} + F_{y2})\cos\delta_f + F_{y3} + F_{y4} + (F_{x1} + F_{x2})\sin\delta_f$$

$$\sum M_{xs} = (F_{s1} - F_{s2})B_f + (F_{s3} - F_{s4})B_r + m_s g h \varphi$$

$$\sum M_z = (F_{y1} + F_{y2})\cos\delta_f L_f - (F_{y3} + F_{y4})L_r + (F_{x1} - F_{x2})B_f \cos\delta_f + (F_{x3} - F_{x4})B_r$$

$$\Sigma M_{ys} = (F_{s1} + F_{s2})(L_f - L_s) - (F_{s3} + F_{s4})(L_f + L_s) + (F_{x1} + F_{x2} + F_{x3} + F_{x4})(h + h_s)$$

2.2 EPS steering model

EPS system contains steering wheel, steering string, motor, decelerating mechanism and gear-rack. According the Newton's law of kinematics, the dynamics equations for each part are built. By the related constraint among the different part, the EPS system dynamic model is obtained.

By steering pinion dynamics relationship, $k_s(\delta_h-\delta_1)+T_a=T_r+J_p\ddot{\delta}_1+B_p\dot{\delta}_1$

The steering torque by the road to the pinion is $T_r=\frac{2dk_{cf}}{G_2}(-\frac{\dot{y}_s}{u}-\frac{L_f\dot{\omega}}{u}+\frac{\delta_1}{G_2})$

2.3 Tire model

The Pacejka nonlinear model is adopted as the tire model[9].

The front wheel cornering angle is $\alpha_f=\delta_f-\beta-\frac{L_f\dot{\omega}}{u}+G_f\varphi$, and the rear wheel cornering angle is $\alpha_r=\frac{L_r\dot{\omega}}{u}-\beta+G_r\varphi$ .The vehicle mass center cornering angle is $\beta=\ddot{y}_s/u$.

Applying the tier parameters to the tire model, the following equation can be obtained.

$$F_{yi}=5694.27\sin\{1.3\arctan[0.378083222\alpha_i-1.6217\arctan(0.144213\alpha_i)]\} \quad (1)$$

Analyzing the nonlinear joint working condition model with the least squares method[10], the complicated nonlinear model can be simplified to be the nonlinear polynomial model. The fitted cubic polynomial model of the tire is in equation (2). The errors between the fitted cubic polynomial model and the original model are shown in Table 1, which illustrates that the mean and variance between the fitted model and the original model are both small, and this fitting is of high accuracy. Consequently, the tire equationcan be substituted via the equation (2) with high approximation and simplification of system analyzing and controller design as both the models are nonlinear polynomial models.

$$F_{yi}=1362.1\alpha_i-101.1\alpha_i^2+2.4\alpha_i^3-116.3 \quad (2)$$

Table 1–Errors of tire fitted polynomial model

Indicator	Variance	Mean
Error, /N	1.5862	0.2477

2.4 Road model

Adopting the filtered white noise as the input model to the road model, the road model is $\dot{Z}_{0i}(t)=-2\pi f_0 Z_{0i}(t)+2\pi\sqrt{G_0 u}w_i(t)$.

2.5 Driver model

The driver is modeled via the optimal curvature preview method[11]as shown in Figure2.

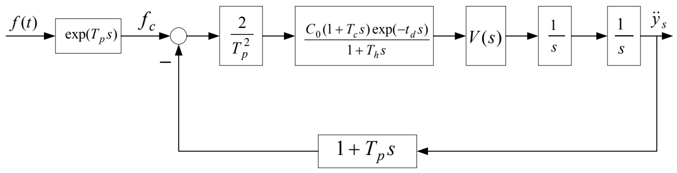


Figure 2–Diagram of the optimal preview curvature driver model

2.6 Nonlinear model of the vehicle

To reflect the dynamic relationship between the vehicle steering and suspension coupling system, the following 22 variables are taken as the state variables, including the four wheels' vertical displacements, the vertical displacement and velocity of the suspension，roll angle and its angular velocity, pitch angle and its angular velocity, yaw rate. The state vector $\boldsymbol{x}$ is:

$$\boldsymbol{x}=[z_{u1}\dot{z}_{u1}z_{u2}\dot{z}_{u2}z_{u3}\dot{z}_{u3}z_{u4}\ \dot{z}_{u4}z_s\dot{z}_s\ \varphi\ \dot{\varphi}\theta\dot{\theta}\ \dot{\omega}\dot{y}_s\delta_1\dot{\delta}_1\ z_{01}z_{02}\ z_{03}z_{04}]^T$$

The control input $\boldsymbol{u}$ is $\boldsymbol{u}=[\ f_1\ f_2\ f_3\ f_4\ T_a\ ]^T$, and the road white noise is the disturbing input $\boldsymbol{w}=[w_1w_2w_3w_4]^T$. The output vector is $\boldsymbol{y}=[\ \ddot{z}_s\ \dot{\theta}\ \dot{\varphi}\ \dot{\omega}\ \dot{y}_s\ ]^T$. Hence, the state model of the integrated system is as follows, which is a nonlinear model with the 22th-order.

$$\begin{cases} \dot{\boldsymbol{x}} = \boldsymbol{A}(\boldsymbol{x}) + \boldsymbol{Bu} + \boldsymbol{Fw} = f(\boldsymbol{x},\boldsymbol{u},\boldsymbol{w}) \\ \boldsymbol{y} = \boldsymbol{C(x)} + \boldsymbol{Du} \end{cases}$$

3. Coordinated control of the vehicle suspension and steering system based on the human and vehicle function allocation

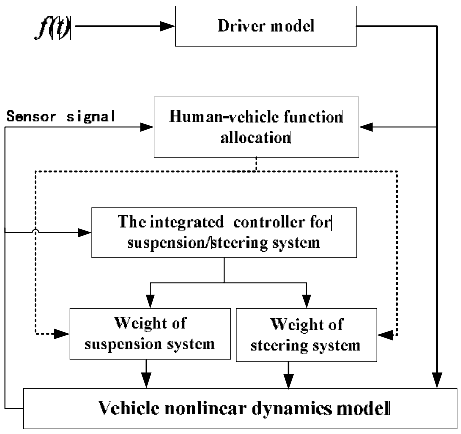


Figure 3–Vehicle coordinated control system based on function allocation

Under the steering situation, the driver outputs the lateral acceleration signal according to the optimal preview curvature model. The torque signal which outputs from the driver, the assistant torque from EPS and the resistant torque are acting on the turning system together. The human-vehicle function allocation module judges the real-time road status according to the sensors' signals, and adjusts the weights of the electric power torque according to the drivers' output torque, in order to obtain the best control performance for the steering system. The human-vehicle function allocation module coordinates the integrated controller to adjust the active suspension control forces weights, to ensure the vehicle body roll to reduce the roll moment as small as possible as shown in Fig. 3.

3.1 Nonlinear control for suspension and steering integrated system

3.1.1 Bi-linear control based on deviation separation(BCDS)

The whole vehicle model is a 22-oeder weak linear model, $\boldsymbol{x} = \boldsymbol{0}$ is checked to be the balance point of the system. Separating the linear part and nonlinear part in the nonlinear model, which can be obtained as $\dot{\boldsymbol{x}} = \boldsymbol{Ax} + \boldsymbol{Bu} + \boldsymbol{Fw} + \boldsymbol{D(x,u)}$ .Where the state equation of the linear part of the system is $\dot{\boldsymbol{x}} = \boldsymbol{Ax} + \boldsymbol{Bu} + \boldsymbol{Fw}$, and the nonlinear part is $\boldsymbol{D(x,u)}$ .Separating the control input $\boldsymbol{u}$ to be two parts, that is $\boldsymbol{u} = \boldsymbol{u_0} + \boldsymbol{u_e}$, where $\boldsymbol{u_0}$ is the linear control input, which optimizing the linear control, while, the $\boldsymbol{u_e}$ the impact from the compensation to $\boldsymbol{D(x,u)}$. The coefficient matrix $\boldsymbol{A}$ has positive eigenvalues, so the linear part of the system is not stable. To ensure the stability and the optimal control properties of the system, the optimal state feedback control $\boldsymbol{u_0} = -\boldsymbol{Kx}$ is adopted with the optimal control strategy design[12]. The control indicator of the control system is:

$$J = \lim_{T\to\infty}\frac{1}{T}\int_0^T [q_1(z_{u1}-z_{01})^2 + q_2(z_{s1}-z_{u1})^2 + q_3(z_{u2}-z_{02})^2 + q_4(z_{s2}-z_{u2})^2 + q_5(z_{u3}-z_{03})^2 + q_6(z_{s3}-z_{u3})^2 + q_7(z_{u4}-z_{04})^2 + q_8(z_{s4}-z_{u4})^2 + \rho_1\ddot{z}_s^2 + \rho_2\ddot{\varphi}^2 + \rho_3\ddot{\theta}^2 + q_9 {T_a}^2 + q_{10}\omega^2]dt$$

Which can be written as the matrix: $J = \lim_{T\to\infty}\frac{1}{T}\int_0^T (x^T Qx + u^T Ru + 2x^T Nu)dt$.

Applying the system variables and model parameters to the above equation, the matrices Q、R and N can be obtained. And programming in MATLAB software to solve the state feedback gain matrix $\boldsymbol{K}$ of the linear part, the closed-loop system state equation for the linear part can be gotten as $\dot{\boldsymbol{x}} = \boldsymbol{(A-BK)x} + \boldsymbol{Fw}$.

Table 2 – Control weighted coefficients

Performance index	Symbol	Value
Dynamic displacement of the front-right tire	q_1	40000
Dynamic displacement of the front-right suspension	q_2	50
Dynamic displacement of the front-left tire	q_3	40000
Dynamic displacement of the front-left suspension	q_4	50
Dynamic displacement of the rear-right tire	q_5	40000
Dynamic displacement of the rear-right suspension	q_6	50
Dynamic displacement of the rear-left tire	q_7	40000
Dynamic displacement of the rear-left suspension	q_8	50
Difference between the steering wheel angle and the pinion angle	q_9	10
Yaw rate	q_{10}	6
Longitudinal acceleration of the vehicle body	ρ_1	1
Roll angular acceleration of the vehicle body	ρ_2	1
Pitch angular acceleration of the vehicle body	ρ_3	1

$\dot{x} = Ax + B(u_0 + u_e) + Fw + D(x,u) = (A - BK)x + Bu_e + Fw + D(x,u)$. By constructing nominal function $\dot{\bar{x}} = (A - BK)\bar{x} + Bu_e$, $\dot{\xi} = \dot{x} - \dot{\bar{x}} = (A - BK)\xi + Fw + D(x,u)$.

By constructing $u_e = \gamma_1\xi + \gamma_2\dot{\xi}$, make

$$\|Bu_e + Fw + D(x,u)\| = \left\|B\gamma_1\xi + B\gamma_2\dot{\xi} + \dot{\xi} - (A - BK)\xi\right\| = \left\|(B\gamma_2 + I)\dot{\xi} + (B\gamma_1 - A + BK)\xi\right\|.$$

Furthermore, $\|B\gamma_1 - A + BK\|$ and $\|B\gamma_2 + I\|$ is assumed to be the least, then the nonlinear compensator u_e is acquired. Meanwhile make $L =$ 4.10, if $D(x,u)$ regarding to $\boldsymbol{x}$ satisfies $\|D(x_1,t) - D(x_2,t)\| \le 4.10\|x_1 - x_2\|$, $\|\xi\|$ is bounded. And if $\dot{D}(x) < 0$, $u_e = \gamma_1\xi + \gamma_2\dot{\xi}$ is established. The obtained bilinear control system is exponential stable at stability point $\boldsymbol{x} = \boldsymbol{0}$ [13].

3.1.2 Nonlinear H_∞ control

To compare with the bilinear controller, the nonlinear H_∞ controller for the suspension and steering integrated system is designed, which can restrain the compact to the output from the interference. Adopting the controlled output $\boldsymbol{z} = [\ \omega\ \ddot{z}_s\ \ddot{\varphi}\ \ddot{\theta}\ z_{s1} - z_{u1}\ z_{s2} - z_{u2}\ z_{s3} - z_{u3}\ z_{s4} - z_{u4}\ f_1\ f_2\ f_3\ f_4\ ]^{\mathrm{T}}$, the controlled output equation of the nonlinear system can be gotten as $\boldsymbol{z} = \boldsymbol{C_1(x)} + \boldsymbol{D_{11}w} + \boldsymbol{D_{12}u}$.

The nonlinear H_∞ control is designed to ensure the stability of the system and the minimal transfer function H_∞ norm. Based on the dissipation theory the control law $\boldsymbol{U(y)}$ is gained，making it satisfy gain inequality L_2 with closed-loop system which controlled system constructs: $\dfrac{\int_0^T \|y(t)\|^2 dt}{\int_0^T \|w(t)\|^2 dt} \le r^2$,Where $0 \le r \le 1$. As $D = D_{12}^T D_{12}$ is a positive definite matrix, the necessary and sufficient condition for the system to be a r-dissipation system is that smooth differentiable positive semi-definite storage function $V(x(t))$， $V(x_0) = 0$ does exist and the HJI inequality is met: $H[x, V_x(x), w, u] - r^2 w^T w \le 0$ .Where,

$$H[x, V_x(x), w, u] = V_x(x)^T\ [A(x) + B_{11}w + B_{12}u] + [C_1 x + D_{11}w + D_{12}u]^T\ [C_1 x + D_{11}w + D_{12}u]$$

$$V_x(x) = \frac{\partial V(x)}{\partial x}$$

In this way, the nonlinear H_∞ control issue is converted to the solution-getting issue of many control laws U and the corresponding storage function $V(x)$ of the inequality HJI. Solving the functions via Taylor series expansion method, the H_∞ controller can be obtained[14].

3.2 Human-vehicle function allocation module

The human-vehicle function allocation module adopts the different functions which the driver and vehicle are good at. During the steering process, as different drivers may have different control levels of the steering wheel, the electric power system exports adjustable power torques, so the drivers' steering torques can be compensated in real-time. That is, the human-vehicle function allocation module achieves the dynamic compensation process via adjusting the steering torques output weight coefficients of the integrated system.

The adaptive tuning of the steering system weight coefficients via fuzzy control rules is organized as Table 3 (supposing the anti-clockwise about *y*-axis of the body is the positive direction). Defining *e* is the error between the ideal lateral velocity and the real lateral velocity of the vehicle, $ec = \dot{e}$, the field $e, ec = \{-5,-4,-3,-2,-1,0, 1,2,3,4,5\}$, and the fuzzy subset is $e, ec = \{\text{NB}, \text{NM}, \text{NS}, \text{ZO}, \text{PS}, \text{PM}, \text{PB}\}$. The elements in the subset is negative big, negative middle, negative small, zero, positive small, positive middle and positive big. Supposing that both *e, ec* and q_i (i=1,2) are normal distribution, the membership grade of each fuzzy subset can be obtained. According to the assignment table of each fuzzy subset and the fuzzy control model of each parameter, the synthesis reasoning is utilized to design the fuzzy matrix tables of parameters. The modified parameters are looked up which are applied to $q_i = 1 + \{e, ec\}_i$, where 1 is the initial weighting value[16]. The membership grade table of q_i can be designed via the synthetic reasoning method. During the online operation, the self-tuning of q_i can be completed via the processing, table-look and calculation of the fuzzy logic rules' results. Setting the fuzzification factors k_e=4, k_{ec}=0.2, and the defuzzification factors $k_{q1} = k_{q2}$=0.15, the system can gain a good control performance. The fuzzy self-tuning rules table of steering system weight coefficient q_i is shown in Table 3.

Table 3–Steering system self-tuning rules table of weight coefficients

e \ *ec*	NB	NM	NS	ZO	PS	PM	PB
NB	PB	PB	PM	PM	PS	ZO	ZO
NM	PB	PB	PM	PS	PS	ZO	NS
NS	PM	PM	PM	PS	ZO	NS	NS
ZO	PM	PM	PS	ZO	NS	NM	NM
PS	PS	PS	ZO	NS	NS	NM	NM
PM	PS	ZO	NS	NM	NM	NM	NB
PB	ZO	ZO	NM	NM	NM	NB	NB

To minimize the vehicle roll degree as far as possible during vehicle steering process, the human-vehicle function allocation module is adopted to self-adaptively change the suspension active forces' left and right allocation rates by adjusting the weights of suspension active forces, to guarantee that the vehicle roll increasing trend is restrained.

Using fuzzy rules adaptive tuning the weights of the steering system, *e* is defined as the error between the ideal roll angle velocity and the real one, and $ec = \dot{e}$. The domain $e, ec = \{-5,-4,-3,-2,-1,0, 1,2,3,4,5\}$, and the fuzzy subset is $e, ec = \{\text{NB}, \text{NM}, \text{NS}, \text{ZO}, \text{PS}, \text{PM}, \text{PB}\}$. Supposing that both *e,ec* and q_i (i=1,2,3,4) are satisfying the normal distribution, the fuzzy self-tuning process is the same as the above steering one. Setting the fuzzification factors $k_{e1} = k_{e2} = 7$, $k_{e3} = k_{e4} = 3$, $k_{ec1} = k_{ec2} = 0.4$, $k_{ec3} = k_{ec4} = 0.23$ and the defuzzification factors $k_{q1} = k_{q2}$=0.12, $k_{q3} = k_{q4} = 0.21$, the fuzzy self-tuning rules of suspension system weight coefficients is shown in Table 4.

Table 4 –Suspension system self-tuning rules table of weight coefficients

e \ *ec*	NB	NM	NS	ZO	PS	PM	PB
NB	PB	PB	PM	PM	PS	PS	ZO
NM	PB	PB	PM	PS	PS	ZO	ZO
NS	PB	PM	PM	PS	ZO	ZO	NS

ZO	PM	PM	PS	ZO	NS	NS	NM
PS	PS	PS	ZO	NS	NS	NM	NM
PM	PS	ZO	NS	NM	NM	NM	NB
PB	ZO	ZO	NM	NM	NM	NB	NB

4. System simulation and analysis

Supposing the vehicle moves in a circumference route with a 30m diameter, 10m/s velocity and the steering wheel's angle step input with 1.57rad, the numerous simulations are carried out in this section for the human-vehicle function allocation control system by adopting the designed nonlinear controller and the non-function-allocation integrated system. Then the simulation results are compared and analysis.

Table 5 shows that performance indices have the smaller mean and variance by BCDS than by the nonlinear H_∞ control, where the vertical acceleration mean of vehicle body mass center is smaller than the nonlinear H_∞ control by 35.5%, while the variance decreases by 67.7%. The mean of vehicle body pitch angular acceleration is smaller than the nonlinear H_∞ control by 91.7%, while the variance of vehicle body roll angular acceleration decreases by 9.8% comparing with the nonlinear H_∞ control. It is also shown that the performance indices significantly decrease by the function allocation coordinated control system than the non-function-allocation integrated system.

Therefore, the suspension performance of the human-vehicle function allocation coordinated control system could been greatly improved by BCDS.

Table 5 –Comparison of the suspension subsystem performance indices

Index	Control method	Mean	Variance
The vertical acceleration of the mass center of the vehicle body, /m/s^2	No function allocation	0.052	1.27
	BCDS	0.031	0.31
	H_∞ control	0.042	0.52
Pitch angular acceleration of the vehicle body, /rad/s^2	No function allocation	0.047	0.51
	BCDS	0.012	0.15
	H_∞ control	0.023	0.21
Roll angular of the vehicle body, /rad	No function allocation	-0.079	0.98
	BCDS	-0.032	0.41
	H_∞ control	-0.040	0.45
The dynamic deflection of the front-left suspension, /m	No function allocation	-0.091	3.20
	BCDS	-0.040	1.81
	H_∞ control	-0.041	2.01
the dynamic deformation of the front-left tire, /m	No function allocation	0.021	0.47
	BCDS	0.002	0.21
	H_∞ control	0.006	0.32

Table 6 shows that under the BCDS, the peak value, the overshoot and the adjustment time of the yaw rate decrease by 0.5%, 13.6% and 8.2% than the nonlinear H_∞ control, and much more than the non-function-allocation. That is, the transient performance of the yaw rate is better with BCDS. The sideslip angle peak value and the overshoot of vehicle body mass center decrease by 3.1% and 266.7% with BCDS than the H_∞ control. The vehicle body roll angle peak value and adjustment time decrease by 50.0% and 25.0% with BCDS than the H_∞ control. Thus, the BCDS could improve the transient performance of yaw rate, sideslip angle of vehicle body mass center and vehicle body roll angle of human- vehicle function allocation coordinated control system.

Table 6 –Comparison of the performance indices of the steering subsystem

index	Control method	peak	overshoot/%	Adjustment time/s
Yaw rate, /rad/s	No function allocation	0.385	6.4	1.51

	BCDS	0.378	4.4	0.85
	H_∞ control	0.380	5.0	0.92
Sideslip angle of the mass center of the vehicle body, /rad	No function allocation	-0.0285	14.0	1.2
	BCDS	-0.0253	1.2	0.7
	H_∞ control	-0.0261	4.4	0.8
Roll angle of the vehicle body, /rad	No function allocation	0.25	733.3	10
	BCDS	0.04	33.3	2.8
	H_∞ control	0.06	100.0	3.5

5. CONCLUSIONS

(1) Considering the coupled relationship between vehicle chassis suspension and steering system, the whole vehicle nonlinear model, the EPS model, the tire model, the filtering white noise road input and the driver model are separately established. The nonlinear dynamic integrated system model with 22th order is obtained via fitting the Pacejka nonlinear tire model by the least square method.

(2) The bilinear controller is designed based on deviation separation method which separates the linear part and the nonlinear part of the integrated system. The controller design is easier to realize and the closed-loop control system is asymptotically steady at the balance point.

(3) Applying the human-vehicle function allocation module, the weights for the suspension output forces and the EPS power torque can be self-adaptively adjusted via the fuzzy self-tuning method. The vehicle roll angle can be reduced to the minimal degree.

(4) The simulation is carried out to the different coordinated control systems with BCDS and the nonlinear H_∞ control. The comparison of the simulation results verifies the easy implementation and optimal control performance of BCDS. Meanwhile, comparing with the non-function-allocation integrated control system, the human-vehicle function allocation system can reflect the differences between the driver and vehicle, and improve the vehicle ride comfort, handling and stability performance.

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