Research on Active Suspension Control Strategy Based on The Model With Parameters of Hydraulic System

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Abstract—**The disadvantage that ordinary active suspension model is in low accuracy and practicability which is caused by the neglect of actuator motion is brought forth through analyzing the assemble of active suspension. The model including hydraulic system parameters is obtained through building the active suspension hydraulic system model. The weighting coefficient matrix of car body acceleration, suspension deflection, and tyre displacement is determined through analyzing the principle of LQG control and the active suspension LQG control strategy is brought out. The conclusion that active suspension promote more control stability and riding comfort than passive suspension and PID control is made by simulation and comparing with passive suspension and active suspension PID control. The force output amplitude performance of the LQG control is better than the PID control which can be get from the force signals.**

Keywords- ***active suspension; hydraulic system model; state equation; LQG control; simulation***

I. INTRODUCTION

The active suspension was placed between sprung and unsprung mass based on the passive suspension. The real time vehicle state can be obtained through measuring the signal of acceleration sensor and displacement sensor. The output of actuator which is determined by the control strategy can ensure the vehicle own the favorable performance of steering, braking, acceleration [1]. Most research only takes the actuator into consideration and ignored the relationship between the actuator force and the actuator displacement. The model without the hydraulic system is not accuracy besides. The defects caused by these two make the results differ from the practical control. Ultimately, the practicability of simulation is decreased [2].

The nucleus of active suspension is control strategy while our country's researches are still in the stage of optimizing of control strategy and theory research. The main control mode in active suspension concludes PID control, fuzzy control, neuronic network control, sky-hook control. The PID control and H^{∞} control are representative control models. It needs substantial amount of time to accomplish PID parameter tuning, while the H^{∞} controller is always conservative and needs to be adjusted as a tradeoff between stability and robustness. The three evaluation index of the car body acceleration, suspension dynamic deflection, tyre dynamic displacement are took into consideration comprehensively in the LQG control, and guarantee of control stability and riding comfort in different working condition.

An accuracy model is built based on LQG control strategy which concludes hydraulic system. And the optimal result can be obtained through confirming the corresponding weighting matrix according to the optimum control target [3].

II. ACTIVE SUSPENSION MODEL

The active suspension consists of the ratio amplifier, ratio electric-magnet, slide valve, hydraulic actuator, and the hydraulic source. The 1/4 automobile suspension model is built for convenient research, in which the car body was supposed as a rigid body and the tyre was supposed as a spring of equivalent stiffness [4]. The physical model was shown in Fig. 1.

A Model of Road Input

The road input is very important to suspension model, the road White Noise Input in this paper is produced by a integrator through which a normal White Noise goes through. The model is shown as below [5]:

$$\dot{x}_g(t) = -2\pi f_0 x_g(t) + 2\pi\sqrt{G_0 u}w(t) \tag{1}$$

where $\dot{x}_g$, x_g stand for the velocity and displacement of the road stimulation. f_0 , G_0 , $w(t)$ denote the lower limiting frequency, road roughness coefficient and typical white noise separately.

B Model of Suspension

The dynamic equations of suspension are of the following forms:

$$m_b\ddot{x}_b = -k_s(x_b - x_w) - c_s(\dot{x}_b - \dot{x}_w) + u \tag{2}$$

$$m_w\ddot{x}_w = k_s(x_b - x_w) + c_s(\dot{x}_b - \dot{x}_w) - u - k_t(x_w - x_g) \tag{3}$$

where $\ddot{x}_b$, $\dot{x}_b$, x_b , $\ddot{x}_w$, $\dot{x}_w$, x_w denote the mass, velocity and displacement of the sprung mass and unsprung mass separately. Variables k_s , c_s , k_t are the stiffness and damping of the spring and tyre. f_0, G_0, $w(t)$ denote the lower limiting frequency, road roughness coefficient and typical white noise separately. m_b , m_w are the mass of sprung and unsprung element. u is the force of actuator.

978-1-4799-2886-6/13 $31.00 © 2013 IEEE
DOI 10.1109/GCIS.2013.33

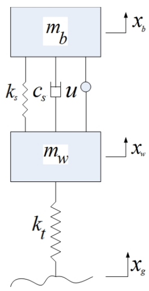

Figure 1. The active suspension model

C Model of Actuator

The dynamic equation set is shown as the following forms:

$$Q_L = A\frac{dy_p}{dt} \quad (4)$$

$$Q_L(s) = K_q y_v(s) - K_c P_L(s) \quad (5)$$

$$u = AP_L \quad (6)$$

where A , y_p , K_q , K_c and $P_L(s)$ denote the area of the piston, the absolute displacement of actuator, the gain of stream flow—displacement, the gain of stream flow—pressure and the pressure of piston. Q_L is the stream-flow of actuator.

D Model of Hydraulic System

The hydraulic system dynamic equations are of the following equations:

$$U_0(t) = k_e U_i(t) \quad (7)$$

$$U_0(t) = L_d\frac{di(t)}{dt} + i(t)R_s + k_v\frac{dy(t)}{dt} \quad (8)$$

$$F_d(t) = K_l i(t) - K_y y(t) \quad (9)$$

$$F_d(t) = m_j\frac{d^2y}{dt^2} + c\frac{dy}{dt} + k_s^{'}y(t) \quad (10)$$

$$F_d(t) = m_v\frac{d^2y_v(t)}{dt^2} + (c_v + c_f)\frac{dy_v(t)}{dt} + k_f y_v(t) \quad (11)$$

where k_e , k_v , k_l are the gains of the proportional amplifier, speed unti-electromotance and the electric current—force. k_y , k_f , $k_s^{'}$ are the spring stiff of ration electric magnet, pressure distribution equivalent stiff and the stiff of gag bit. m_j , m_v stand for the mass of gag bit and slide valve. c , c_v, c_f denote the gag bit damp, the friction coefficient between the spool and valve pocket and transient fluid power damping factor. y_v denotes the displacement of slide valve.

E The State Equation

The state variable can be set to (12) from the equations as below [6]:

$$X = \begin{pmatrix} x_b & \dot{x}_b & x_w & \dot{x}_w & x_g & i & y & \dot{y} & y_v & \dot{y}_v \end{pmatrix}^T \quad (12)$$

And the state equation is set as below:

$$\dot{X} = AX + BU_i + W\omega(t) \quad (13)$$

$$Y = CX + DU_i \quad (14)$$

In the equation, the coefficient matrixes are designed as below according to the equations from (1)—(14) [7]:

$$A = \begin{pmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -\frac{k_s}{m_b} & \frac{-c_s - \frac{A^2}{k_c}}{m_b} & \frac{k_s}{m_b} & \frac{c_s + \frac{A^2}{k_c}}{m_b} & 0 & 0 & 0 & 0 & \frac{k_q A}{m_b k_c} & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{k_s}{m_w} & \frac{c_s + \frac{A^2}{k_c}}{m_w} & \frac{-k_s - k_t}{m_w} & \frac{-c_s - \frac{A^2}{k_c}}{m_w} & \frac{k_t}{m_w} & 0 & 0 & 0 & -\frac{Ak_q}{m_w k_c} & 0 \\ 0 & 0 & 0 & 0 & -2\pi f_0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -\frac{R_s}{L_d} & 0 & -\frac{k_v}{L_d} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{k_l}{m_j} & -\frac{k_s^{'} + k_y}{m_j} & -\frac{c}{m_j} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & 0 & \frac{k_l}{m_v} & -\frac{k_y}{m_v} & 0 & -\frac{k_f}{m_v} & \frac{c_v + c_f}{-m_v} \end{pmatrix} \quad (15)$$

$$B = \begin{pmatrix} 0 & 0 & 0 & 0 & 0 & \frac{k_e}{L_d} & 0 & 0 & 0 & 0 \end{pmatrix}^T \quad (16)$$

$$W = \begin{pmatrix} 0 & 0 & 0 & 0 & 2\pi\sqrt{G_0 u} & 0 & 0 & 0 & 0 & 0 \end{pmatrix}^T \quad (17)$$

$$Y = \begin{pmatrix} \ddot{x}_b & x_b - x_w & x_w - x_g & \dot{i} \end{pmatrix}^T \quad (18)$$

$$C = \begin{pmatrix} -\frac{k_s}{m_b} & \frac{-c_s - \frac{A^2}{k_c}}{m_b} & \frac{k_s}{m_b} & \frac{c_s + \frac{A^2}{k_c}}{m_b} & 0 & 0 & 0 & 0 & \frac{k_q A}{m_b k_c} & 0 \\ 1 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & -1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{R_s}{L_d} & 0 & -\frac{k_v}{L_d} & 0 & 0 \end{pmatrix} \quad (19)$$

$$D = \begin{pmatrix} 0 & 0 & 0 & \frac{k_e}{L_d} \end{pmatrix} \quad (20)$$

III. LQG CONTROL

The vehicle body acceleration, suspension dynamic deflection, tyre dynamic displacement and the change rate of electromagnetic valve incoming current are supposed to be the performance index. The influence coefficient of the three performance indexes is determined through revising the weight matrix. And the goal function J can be structured as below[8]:

$$J = \lim_{T\to\infty}\frac{1}{T}\int_0^T [q_1\ddot{x}_b^2 + q_2(x_b - x_w)^2 + q_3(x_w - x_g)^2 + q_4\dot{i}^2]dt \quad (21)$$

where $Q_0 = \begin{bmatrix} q_1 & & & \\ & q_2 & & \\ & & q_3 & \\ & & & q_4 \end{bmatrix}$ stands for the weight matrix.

Equation (21) can be rewrite according to the theory of optimal control [9]:

$$J = \lim_{T\to\infty}\frac{1}{T}\int_0^T [X^TQX + U^TRU + 2X^TNU]dt \quad (22)$$

Substituting (14) into (21), provides:

$$J = \lim_{T\to\infty}\frac{1}{T}\int_0^T [X^TC^TQ_0CX + U^TD^TQ_0DU + X^TC^TQ_0DU + U^TD^TQ_0CX]dt \quad (23)$$

Substituting (15)—（20） into (23), provides:

$$
Q=\begin{pmatrix}
\frac{k_s^2q_1}{m_b^2}+q_2 & -\frac{k_sq_1}{m_b^2}(-c_s-\frac{A_1^2}{k_c}) & -\frac{k_s^2q_1}{m_b^2}-q_2 & -\frac{k_sq_1}{m_b^2}(c_s+\frac{A_1^2}{k_c}) & 0 & 0 & 0 & 0 & -\frac{k_sq_1k_qA_1}{m_b^2k_c} & 0 \\
-\frac{k_sq_1}{m_b^2}(-c_s-\frac{A_1^2}{k_c}) & \frac{q_1(-c_s-\frac{A_1^2}{k_c})^2}{m_b^2} & \frac{q_1k_s(-c_s-\frac{A_1^2}{k_c})}{m_b^2} & -\frac{q_1(c_s+\frac{A_1^2}{k_c})^2}{m_b^2} & 0 & 0 & 0 & 0 & \frac{(-c_s-\frac{A_1^2}{k_c})q_1k_qA_1}{m_b^2k_c} & 0 \\
-\frac{k_s^2q_1}{m_b^2}-q_2 & \frac{q_1k_s(-c_s-\frac{A_1^2}{k_c})}{m_b^2} & \frac{k_s^2q_1}{m_b^2}+q_2+q_3 & \frac{k_sq_1(c_s+\frac{A_1^2}{k_c})}{m_b^2} & -q_3 & 0 & 0 & 0 & \frac{k_sq_1k_qA_1}{m_b^2k_c} & 0 \\
-\frac{k_sq_1}{m_b^2}(c_s+\frac{A_1^2}{k_c}) & -\frac{q_1(c_s+\frac{A_1^2}{k_c})^2}{m_b^2} & \frac{k_sq_1(c_s+\frac{A_1^2}{k_c})}{m_b^2} & \frac{(c_s+\frac{A_1^2}{k_c})^2q_1}{m_b^2k_c} & 0 & 0 & 0 & 0 & \frac{(c_s+\frac{A_1^2}{k_c})q_1k_qA_1}{m_b^2k_c} & 0 \\
0 & 0 & -q_3 & 0 & q_3 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & (\frac{R_s}{L_d})^2q_4 & 0 & \frac{R_sk_vq_4}{L_d^2} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & \frac{k_vq_4R_s}{L_d^2} & 0 & (\frac{k_v}{L_d})^2q_4 & 0 & 0 \\
-\frac{k_sq_1k_qA_1}{m_b^2k_c} & \frac{(-c_s-\frac{A_1^2}{k_c})q_1k_qA_1}{m_b^2k_c} & \frac{k_sq_1k_qA_1}{m_b^2k_c} & \frac{(c_s+\frac{A_1^2}{k_c})q_1k_qA_1}{m_b^2k_c} & 0 & 0 & 0 & 0 & \frac{K_q^2A_1^2q_1}{m_b^2k_c^2} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{pmatrix} \tag{24}
$$

$$R=\frac{k_e^2}{L_d^2}q_4 \tag{25}$$

$$N=\begin{pmatrix}0 & 0 & 0 & 0 & 0 & -\frac{R_sq_4k_4}{L_d^2} & 0 & -\frac{k_vq_4k_e}{L_D^2} & 0 & 0\end{pmatrix}^T \tag{26}$$

The feedback gain matrix can be obtained under the help of Matlab function LQR [10]:

$$(K,S,E)=LQR(A,B,Q,R,N) \tag{27}$$

where S is the solution from Riccati equation, E is the system eigenvalue, K is the coefficient of force, the force output can be drawn below:

$$U=-Kx \tag{28}$$

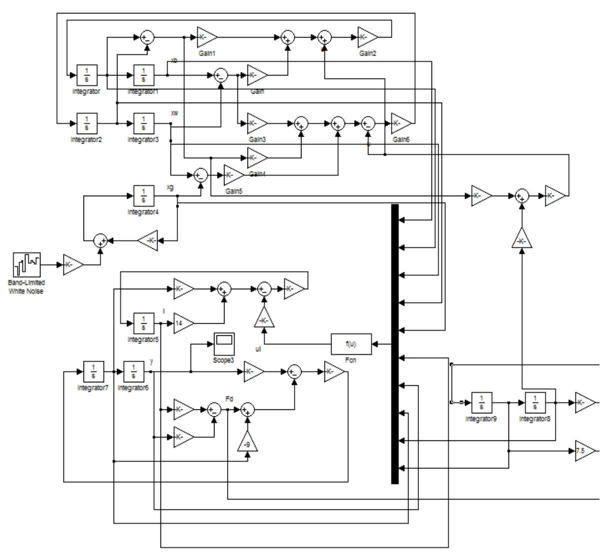

Figure 2. LQG simulation model

Table 1. LQG simulation parameter

The Parameter of The Simulation Vehicle	
Sprung m_b	320kg
Unsprung m_w	40kg
stiff of suspension spring k_s	20000 N/m
Damp of suspension damper c_s	1000 N/ms^{-1}
Tyre stiff k_t	200000 N/m

Table 2. Hydraulic system parameter

Hydraulic System Parameter									
parameter	$k_q(m3)$	$A_1(m^3)$	$k_c(m^5/Nm)$	$f_0(Hz)$	$L_d(H)$	$R_s(\Omega)$	$k_v(vs/m)$	$k_I(N/A)$	$K_y(N/m)$
value	0.88	0.00785	2.03e-10	0.01	0.12	4	32.4	60.64	16000
parameter	k_e	$m_v(kg)$	$c_v(Ns/m)$	$C_f(Ns/m)$	$m_j(kg)$	$c(Ns/m)$	$k_f(N/m)$	$k_s^{'}(N/m)$	
value	2.24	0.04	6.5	1	0.04	9	13000	13000	

IV. STIMULATION RESULT

The simulation model is shown in Fig. 2 and the parameter of the model is shown in Table 1 and Table 2.

The road input according to (1) can be emulated as Fig. 3.

The input from the road can be seen obviously From Fig. 3, and the largest input is 12*mm.*

The output of vehicle body acceleration, suspension dynamic deflection, tyre dynamic displacement are shown in Fig. 4—Fig. 6.

The force output can be seen from Fig. 7. Fig. 8 is the supplementary specification which shows the energy relationships between the LQG and PID control.

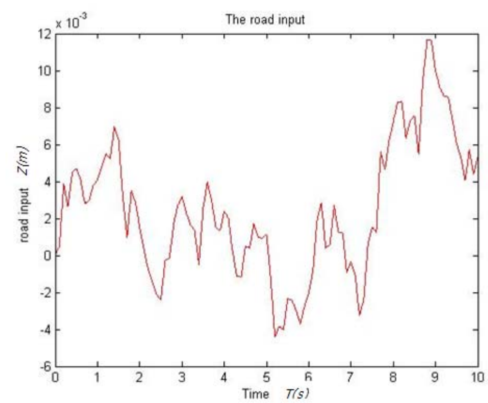

Figure 3. The road outline input

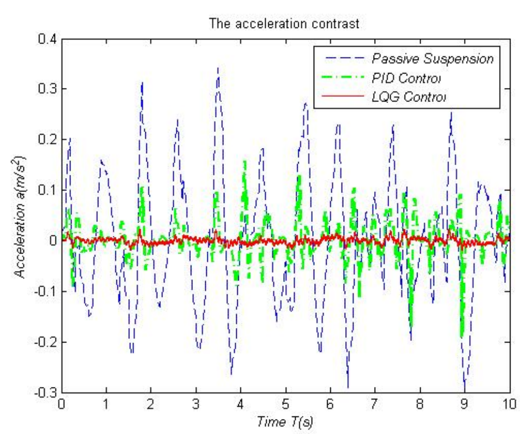

Figure 4. The acceleration contrast

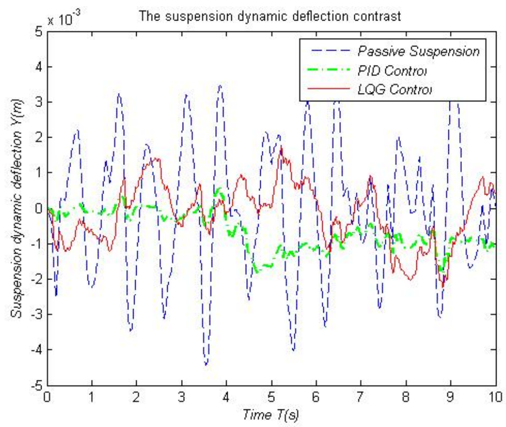

Figure 5. The suspension dynamic deflection contrast

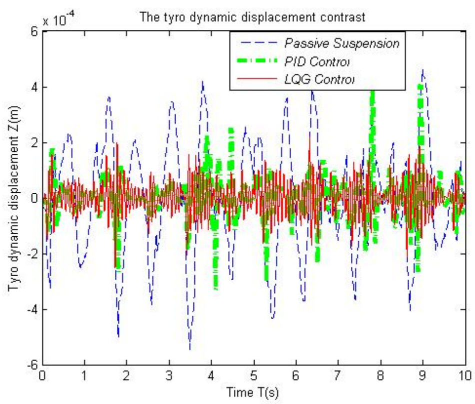

Figure 6. The tyre dynamic contrast

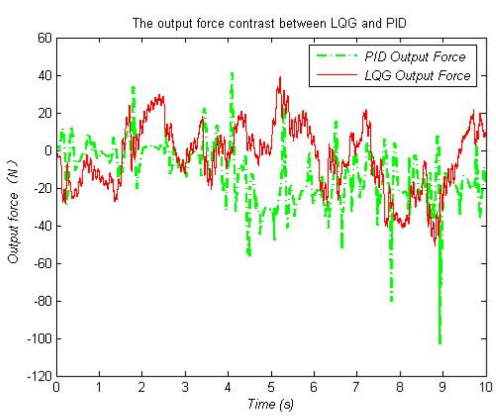

Figure 7. The output force contrast

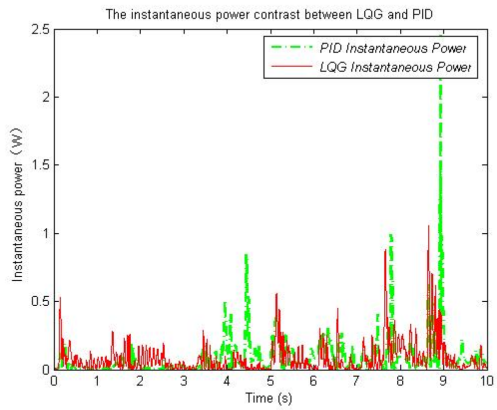

Figure 8. The instantaneous power contrast

Different colors in Fig. 4—Fig. 6 stand for passive suspension, active suspension PID control and active suspension LQG control separately.

Different colors in Fig. 7 — Fig.8 denote active suspension PID control and LQG control.

It can be concluded from the simulation results and the road input model that the PID control and LQG control both can curb the road stimulation. Compared to the passive suspension, PID control can decrease the acceleration amplitude more than 30%, while the LQG control can decrease more than 70%. To the dynamic suspension deflation, it implied that PID control and LQG control both can decrease the amplitude compared to passive suspension. The dynamic tyre displacement of LQG control is much better than PID control too. To the aspect of energy, the amplitude of force output in LQG control is lower than PID control. This conclusion can be get from the instantaneous diagram too.

V. CONCLUSION

In this paper, a complicated active suspension model which is more accurate than the common model is built. In this model, the hydraulic system parameters are included. Compared to the modern model, this active suspension model are more practicable. The active suspension LQG model was built through combining the suspension model with LQG control strategy. The result of simulation shows that the active suspension LQG control is more effective than PID control apparently.

ACKNOWLEDGMENT

We thank all the tutors who gave the genuine suggestions to us. Professor Zhao Xuan helped us modify the active suspension model. Shi Peilong thanks his classmate Chen Jing who corrected the equations for him and helped him revise this paper especially in the field of

English. We are appreciative of many our students who gave us their constructive criticisms of certain sections in this paper.

REFERENCES

[1] Lin, J.S. and Kanellakopoulos, “Nonlinear Design of ActiveSuspensions,” IEEE Control System Magzine, Vol. 17, No.3, pp. 45-49.

[2] Crosby, M. and D.C.Karnopp, “The active damper a new concept for shock and vibration control,” Shock Vibration. Bull, Part H, Washington D.C, pp. 35-42.

[3] R. Rajamani and J.K. Hedrick, “Performance of Active Automobile Suspensions With Hydraulic Actuators: Theory and Experiment” Proceedings of the American control conference June 1994.

[4] Esmailzadeh, E. and Taghirad, H.D, “Active Vehicle Suspensions With Optimal State-Feedback control,” Proceedings of CSME, Vol, 2, pp, 822-834.

[5] Lin, J.S and Kanellakopoulos, “Modular Adaptive Design for Active Suspensions,” Proc. 36th IEEE Conference on Decision and Control, 1997, San Diego, CA, pp.3626-3631.

[6] Jecqueline Wilkie, Michael Johnson and Raza Katebi, “Control Engineering: an Introductory Course, ” Pal grave Publishers 2003.

[7] Ogata. K, “Mordern Control Engineering,” Prentice-Hall, 3rd edition 1996.

[8] Wong J.Y, “Theory of Ground Vehicles,” John Wily and Sons Inc., 1988 , NewYork.

[9] Prof. Dr. George Rill “Vehicle Dynamics”, University of applied science, October 2006.

[10] Karnopp, D. , “Passive and Active Control of Road Vehicle Heave and Pitch Motion ”, Proc. IFAC 10th Triennal World Congress, 1991, pp. 181-183.