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Robust Active Suspension Control of Vehicles with Measurement Noises

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Abstract: In this paper, we propose a robust ride comfort control scheme for vehicles without using measurements of tire deflections. To realize good ride comfort without using measurements of the tire deflections, we propose using an estimator for the acceleration of the road disturbance, the derivative of the suspension strokes and the derivatives of the displacements from the road disturbance to the vehicle body. Using the estimates, we can design a combined ideal vehicle. Then, a tracking controller is designed so that the real vehicle can track the motion of the combined ideal vehicle. Moreover, by carrying out numerical simulations, the influence of measurement disturbances on control performance will be investigated. As a result, it is shown that the proposed ride comfort control scheme is effective even in the presence of measurement disturbances.

Keywords: Vehicle, Ride comfort, Active suspension, Estimator, Robust Tracking Control, Ideal Model

I. INTRODUCTION

Recently, in order to achieve good ride comfort and good handling qualities, a large number of control schemes using active suspensions have been proposed in [1]-[4]. In conventional schemes [1]-[4], the active suspensions are controlled so that the suspension strokes lie within an admitted range and the handling quality does not become worse. As a result, the ride comfort will be best at only one specified location on the vehicle body. In cases when the specified location has to be moved, too much time is needed to redesign an active suspension controller. To struggle with this problem, the authors have proposed some active suspension control schemes [5] [7]. In vehicle systems using controllers proposed in [5]-[7], the following good properties exist: (1) The ride comfort at a specified location will be best. (2) The best location can be easily moved by setting only one design parameter without redesigning the suspension controller. However, in the proposed schemes [5]-[7], it is assumed that the tire deflections can be measured. Since road surfaces are uneven, and using noncontact sensors such as laser position sensors, it is difficult to measure the tire deflections with high accuracy.

To overcome this problem, the authors have proposed a ride comfort control scheme without using the measurements of tire deflections [8],[9]. To realize good ride comfort without using measurements of the tire deflections, the signals are measured such as the vertical acceleration on the vehicle body, the displacement and the velocity of the suspension stroke, the vertical acceleration of the unsprung mass and the force added to the vehicle body. In addition, we propose using an estimator for the acceleration of the road disturbance and the derivatives of the displacements from the road disturbances to the vehicle body. In the papers [8],[9], however, the influence of measurement disturbances on control performance has not been considered. Moreover, it is assumed that the derivative of the suspension strokes can be rigorously measured.

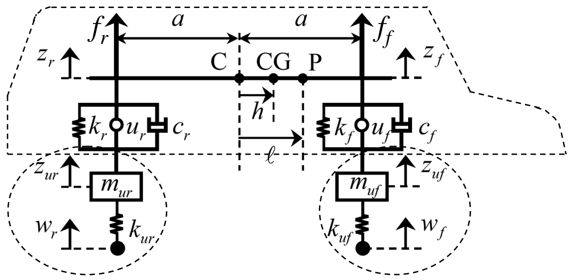

Fig. 1. Two wheels model.

In this paper, we propose an estimator for the acceleration of the road disturbance, the derivative of the suspension strokes and the derivatives of the displacements from the road disturbances to the vehicle body. A robust active suspension controller with the estimated signals is proposed. Moreover, carrying out numerical simulations, the influence of measurement disturbances on control performance is investigated. As a result, it is shown that the proposed ride comfort controller is very effective even in the presence of measurement disturbances.

II. VEHICLE MODEL

The two wheels model is shown in Fig. 1. The explanation for parameters is shown in Table 1. It is assumed that the pitching angle is small, and then, the dynamic

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equation of vehicles is given as follows.

$$\left.\begin{array}{l}\ddot{\boldsymbol{x}}_z(t) = \boldsymbol{d}(t) \quad H^{\ 1}\ddot{\boldsymbol{w}}(t)\\ \ddot{\boldsymbol{x}}_u(t) = \quad K_u\boldsymbol{x}_u(t) \quad M_u^{\ 1}\boldsymbol{f}(t) \quad \ddot{\boldsymbol{w}}(t)\\ \boldsymbol{x}_z(t) = H^{\ 1}[z_f(t) \quad w_f(t), z_r(t) \quad w_r(t)]^T\\ \boldsymbol{x}_u(t) = [z_{uf}(t) \quad w_f(t), z_{ur}(t) \quad w_r(t)]^T\\ \boldsymbol{d}(t) = M^{\ 1}H^T\boldsymbol{f}(t)\\ \boldsymbol{f}(t) = [f_f(t),\ f_r(t)]^T\\ \qquad = \quad C\dot{\boldsymbol{x}}_s(t) \quad K\boldsymbol{x}_s(t) + \boldsymbol{u}(t)\end{array}\right\}(1)$$

$$\left.\begin{array}{l}\boldsymbol{x}_s(t) = H\boldsymbol{x}_z(t) \quad \boldsymbol{x}_u(t)\\ \boldsymbol{u}(t) = [u_f(t), u_r(t)]^T,\ \boldsymbol{w}(t) = [w_f(t), w_r(t)]^T\\ M = (T_h^T)^{\ 1}\mathrm{diag}[m, i_c]T_h^{\ 1}\\ M_u = \mathrm{diag}[m_{uf}, m_{ur}], K = \mathrm{diag}[k_f, k_r]\\ C = \mathrm{diag}[c_f, c_r], K_u = M_u^{\ 1}\mathrm{diag}[k_{uf}, k_{ur}]\\ T_h = I_2 \quad Dh, H = \begin{bmatrix} 1 & \ a\\ 1 & \ a\end{bmatrix}, D = \begin{bmatrix} 0 & 1\\ 0 & 0\end{bmatrix}\end{array}\right\}(2)$$

The control objective is to develop an active suspension controller so that the ride comfort at a specified location becomes best. In general, humans feel uncomfortable in the vertical oscillation with the frequency about 1 Hz. Therefore, to achieve the control objective, we develop a controller so that the vertical acceleration can be reduced to a small value at any specified location ℓ on the vehicle body. To meet the objective, the following assumptions are made for actual vehicles.

A1 The accelerations $\ddot{z}_f(t)$, $\ddot{z}_r(t)$, $\ddot{z}_{uf}(t)$ and $\ddot{z}_{ur}(t)$ are measured.

A2 The force $\boldsymbol{f}(t) = [f_f(t), f_r(t)]^T$ added to the sprung mass are measured.

A3 Suspension displacement $\boldsymbol{x}_s(t)$ is measured.

A4 Vehicle parameters are known except for the length a, the front and the rear tire stiffness k_{uf}, k_{ur} and the tire mass m_{uf}, m_{ur}.

A5 The second and third derivation of the road disturbance $\boldsymbol{w}(t)$ are bounded.

III. ACTIVE SUSPENSION CONTROLLER

Fig. 2 shows the configuration of the vehicle active suspension system proposed in [7]. The combined ideal model shown in Fig. 2 has good properties. Namely, 1) The ride comfort at a specified location becomes best, 2) The best location can be easily moved by setting only one design parameter without redesigning the combined ideal vehicle. If the real vehicle can track the motion of the designed combined ideal vehicle, the control objective can be achieved. The active suspension control system in Fig. 2 is designed so that the real vehicle can track the motion of the combined ideal vehicle. To achieve the control objective, the signals of road disturbance $\ddot{\boldsymbol{w}}(t)$ and state variables $\boldsymbol{x}_z(t)$, $\dot{\boldsymbol{x}}_z(t)$, $\boldsymbol{x}_u(t)$, $\dot{\boldsymbol{x}}_u(t)$ $\boldsymbol{d}(t)$ are required in the controller proposed in [7].

According to the assumption A1, it can be seen that the following signals are available.

$$\left.\begin{array}{l}\ddot{\boldsymbol{x}}_s(t) = [\ddot{z}_f(t), \ddot{z}_r(t)]^T \quad [\ddot{z}_{uf}(t), \ddot{z}_{ur}(t)]^T\\ H\boldsymbol{d}(t) = [\ddot{z}_f(t), \ddot{z}_r(t)]^T\end{array}\right\}\ (3)$$

Table 1 Notation of vehicle model.

C, CG	center and center of gravity of vehicle body
z_{cg}, θ	vertical displacement at CG and pitching
z_f, z_r	vertical displacement of vehicle body at positions on front and rear wheel axle
z_{uf}, z_{ur}	vertical displacement of front and rear unsprung mass
w_f, w_r	vertical displacement of road disturbance added to front and rear wheel
v	longitudinal velocity of vehicle
m, i_c	sprung mass and moment of inertia of vehicle body
a	half of vehicle body length
h, ℓ	distances from C to CG and from C to P
m_{uf},m_{ur}	front and rear unsprung mass
k_f, k_r	front and rear suspension stiffness
c_f, c_r	front and rear suspension damping rate
k_{uf}, k_{ur}	front and rear tire spring stiffness
f_f, f_r	front and rear force added to sprung mass
u_f, u_r	front and rear active suspension control force

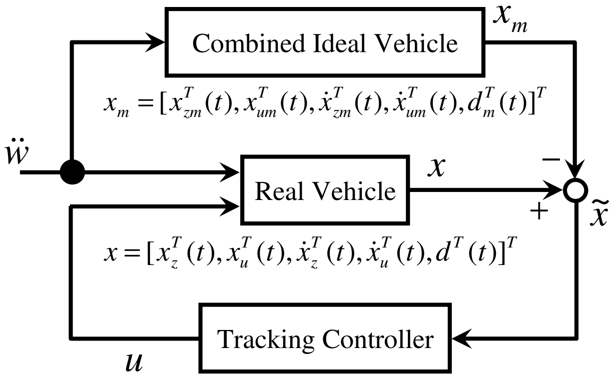

Fig. 2 Configuration of active suspension control system.

Moreover, it is easy from (1), (2) to ascertain that the following signals are also available.

$$\left.\begin{array}{l}\boldsymbol{p}_1(t) = K_u^{\ 1}\Big(\ddot{\boldsymbol{x}}_s(t) \quad (H\boldsymbol{d}(t) + M_u^{\ 1}\boldsymbol{f}(t))\Big)\\ \qquad = \boldsymbol{x}_u(t)\\ \boldsymbol{p}_2(t) = \boldsymbol{x}_s(t) + \boldsymbol{p}_1(t) = H\boldsymbol{x}_z(t)\end{array}\right\}(4)$$

If an estimator for $\dot{\boldsymbol{x}}_s(t)$ and $\dot{\boldsymbol{x}}_z(t)$ are developed, then, the signal $\dot{\boldsymbol{x}}_u(t)$ becomes also available. Therefore, we will develop an estimator for $\dot{\boldsymbol{x}}_s(t)$, $\dot{\boldsymbol{x}}_z(t)$ and $\ddot{\boldsymbol{w}}(t)$. Consider the new state $\boldsymbol{\eta}(t) = [\boldsymbol{\eta}_1(t)^T, \boldsymbol{\eta}_2(t)^T, \boldsymbol{\eta}_3(t)^T]^T = [\dot{\boldsymbol{x}}_s(t)^T, H\dot{\boldsymbol{x}}_z(t)^T, \ddot{\boldsymbol{w}}(t)^T]^T$. Then, the estimator for the state $\boldsymbol{\eta}(t)$ is proposed as

$$\left.\begin{array}{l}\widehat{\boldsymbol{\eta}}_1(t) = \alpha\boldsymbol{x}_s(t) + \boldsymbol{\zeta}_1(t),\\ \dot{\boldsymbol{\zeta}}_1(t) = \ddot{\boldsymbol{x}}_s(t) \quad \alpha\widehat{\boldsymbol{\eta}}_1(t),\\ \boldsymbol{\zeta}_1(0) = \quad \alpha(\boldsymbol{p}_2(0) \quad \boldsymbol{p}_1(0))\end{array}\right\} \qquad (5)$$

$$\left.\begin{array}{l}\widehat{\boldsymbol{\eta}}_2(t) = 2\alpha\boldsymbol{p}_2(t) + \boldsymbol{\zeta}_2(t),\\ \dot{\boldsymbol{\zeta}}_2(t) = \quad 2\alpha\widehat{\boldsymbol{\eta}}_2(t) + H\boldsymbol{d}(t) \quad \boldsymbol{\eta}_3(t),\\ \boldsymbol{\zeta}_2(0) = 2\alpha\boldsymbol{p}_2(0)\end{array}\right\} \qquad (6)$$

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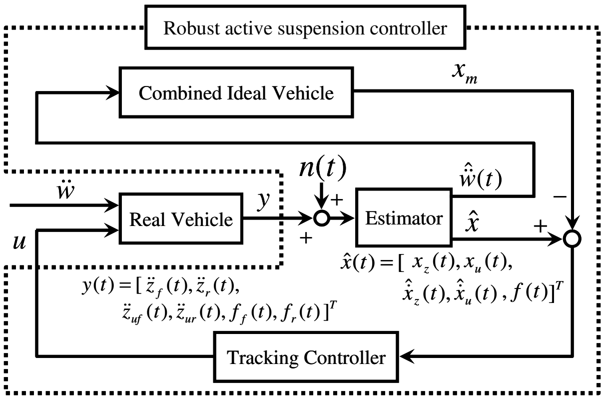

Fig. 3 Configuration of active suspension control system using proposed estimator.

$$\left.\begin{array}{l} \widehat{\boldsymbol{\eta}}_3(t) = \quad 3\alpha^2 \boldsymbol{p}_2(t) + \boldsymbol{\zeta}_3(t), \\ \dot{\boldsymbol{\zeta}}_3(t) = 3\alpha^2 \widehat{\boldsymbol{\eta}}_2(t), \\ \boldsymbol{\zeta}_3(0) = 3\alpha^2 \boldsymbol{p}_2(0) \end{array}\right\} \qquad (7)$$

where $\hat{\boldsymbol{\eta}}_i, i = 1, 2, 3$ are estimated signals for $\boldsymbol{\eta}_i, i = 1, 2, 3$, and α is a positive deign parameter introduced to improve performance of the proposed estimator. Differentiating the first equation in (5)(6)(7), we obtain the following equation.

$$\left.\begin{array}{l} \dot{\widehat{\boldsymbol{\eta}}}_1(t) = \alpha \widetilde{\boldsymbol{\eta}}_1(t) + \ddot{\boldsymbol{x}}_s(t) \\ \dot{\widehat{\boldsymbol{\eta}}}_2(t) = 2\alpha \widetilde{\boldsymbol{\eta}}_2(t) + H\boldsymbol{d}(t) \quad \widehat{\boldsymbol{\eta}}_3(t) \\ \dot{\widehat{\boldsymbol{\eta}}}_3(t) = \quad 3\alpha^2 \widetilde{\boldsymbol{\eta}}_2(t) \end{array}\right\} \qquad (8)$$

$$\dot{\widetilde{\boldsymbol{\eta}}}(t) = \begin{bmatrix} \alpha & O_2 & O_2 \\ O_2 & 2\alpha & I_2 \\ O_2 & 3\alpha^2 & O_2 \end{bmatrix} \widetilde{\boldsymbol{\eta}}(t) + \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} \ddot{\boldsymbol{w}}(t) \qquad (9)$$

Where $\tilde{\boldsymbol{\eta}} = \boldsymbol{\eta} \quad \hat{\boldsymbol{\eta}}$ and $\tilde{\boldsymbol{\eta}}_{\boldsymbol{i}} = \boldsymbol{\eta}_i \quad \hat{\boldsymbol{\eta}}_i, i = 1, 2, 3$. It is assumed that the proposed controller starts in the situation where the oscillation of the vehicle body dose not occur. According to the assumption A5, for the proposed estimator(5)(6)(7), the following theorem holds.

Theorem 1: For estimation $\widetilde{\boldsymbol{\eta}}_1(t)$, $\widetilde{\boldsymbol{\eta}}_2(t)$, $\widetilde{\boldsymbol{\eta}}_3(t)$, there exist bounded positive constants $\overline{\rho}_{Ei},\ i = 1, 2$ independent of the design parameter α such that

$$\left.\begin{array}{l} \|\widetilde{\boldsymbol{\eta}}_1(t)\|^2 = 0 \\ \|\widetilde{\boldsymbol{\eta}}_2(t)\|^2 \le \overline{\rho}_{E1}\alpha \ \ ^4 \\ \|\widetilde{\boldsymbol{\eta}}_3(t)\|^2 \le \overline{\rho}_{E2}\alpha \ \ ^2 \end{array}\right\}. \qquad (10)$$

It can be concluded from the theorem 1 that the estimated errors decrease as the design parameter α increases. In case when the design parameter α is set to be large enough, we can obtain the signals $H\dot{\boldsymbol{x}}_z(t)$ and $\dot{\boldsymbol{x}}_s(t)$ with enough accuracy. Then, the signal $\dot{\boldsymbol{x}}_u(t) = H\dot{\boldsymbol{x}}_z(t) \quad \dot{\boldsymbol{x}}_s(t)$ becomes also available. Namely, from (3), (4) and theorem 1, it can be concluded that the all required signals are available. Configuration of the active suspension control system using the proposed estimator is shown in Fig. 3.

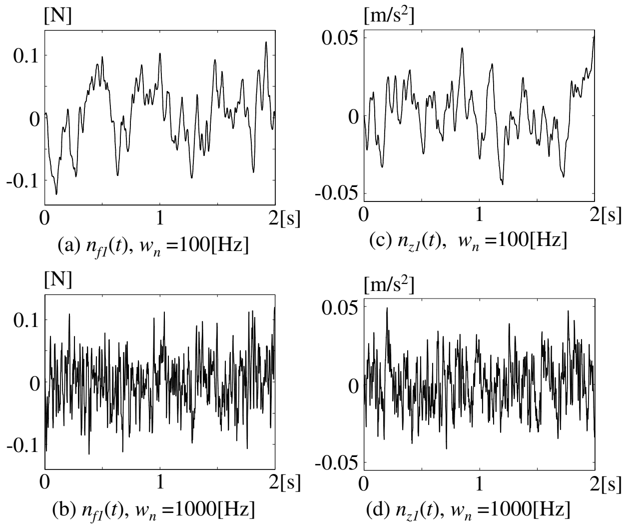

Fig. 4 Disturbance $n_{f1}(t), n_{z1}(t)$ added to measurements $f_f, \ddot{x}_z(t)$.

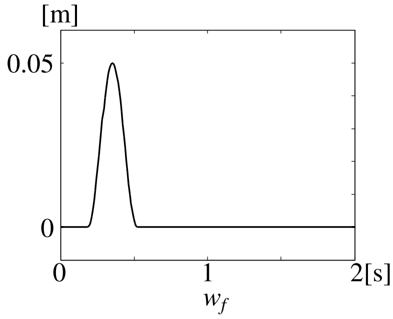

Fig. 5. Road disturbance $w_f(t)$.

IV. NUMERICAL SIMULATION RESULTS

To investigate the influences of measurement disturbances, numerical simulations are carrying out. The frequencies of measurement disturbances are set as 100[Hz] and 1000[Hz]. The measurement disturbance $n_{f1}(t)$ added to a signal of force sensor $f_f(t)$ are shown in Figs. 4 (a), (b), and the measurement disturbance $n_{z1}(t)$ added to a signal of acceleration sensor ($\boldsymbol{c}^T\ddot{\boldsymbol{x}}_z(t),\ \boldsymbol{c} = [1,\ 0]^T$) are shown in Figs. 4 (c), (d). The maximum values of measurement disturbances of force sensors and acceleration sensors are set as 0.12[N] and 0.05[$\mathrm{m/s^2}$], respectively. For the other sensors, similar measurement disturbances are added. The values of vehicle parameters are shown in Table 2. The design parameter α for the proposed estimator is set as $\alpha = 2000$. The design parameters for the tracking controller shown in Fig. 2 are set as a large value so that the tracking error between the actual vehicle and the ideal vehicle model can become small. And the vehicle velocity is set as $v = 100 \quad 1000/3600[\mathrm{m/s}]$. Road disturbance $w_f(t) = w_r(t \quad L),\ L = 2a/v$ is shown in Fig. 5.

Fig. 6 shows the responses of the vertical acceleration of the controlled vehicle shown in Fig. 3 at the specified location $l = \quad 1.0(l_p = \quad 0.7)$. The thin line shows the responses of the acceleration of the controlled vehicle without measurement disturbances. The thick lines show

Table 2 Nominal values of parameters.

m	781	kg	i_c	990	kgm^2
h	0.04	m	a	1.38	m
k_f	27160	N/m	k_r	29420	N/m
c_f	4000	Ns/m	c_r	2500	Ns/m
m_{uf}	69	kg	m_{ur}	96	kg
k_{uf}	229000	N/m	k_{ur}	255000	N/m

the responses of the controlled vehicle with measurement disturbances, and the dashed lines show the responses of the passive vehicle. It is seen from Fig. 6 that the acceleration responses do not vary even in the presence of the measurement disturbances.

V. CONCLUSION

We have proposed the active suspension control scheme in which tire deflections are not required. The proposed suspension controller has a good property that the location where the ride comfort becomes best can be easily moved by setting only one design parameter l_p. It has been shown by carrying out numerical simulations that in the closed loop system using the proposed ride comfort control scheme, the influence of measurement disturbances is analyzed by using numerical simulations. As a result, it is shown that the proposed ride comfort control scheme is very effective even in the presence of measurement disturbances.

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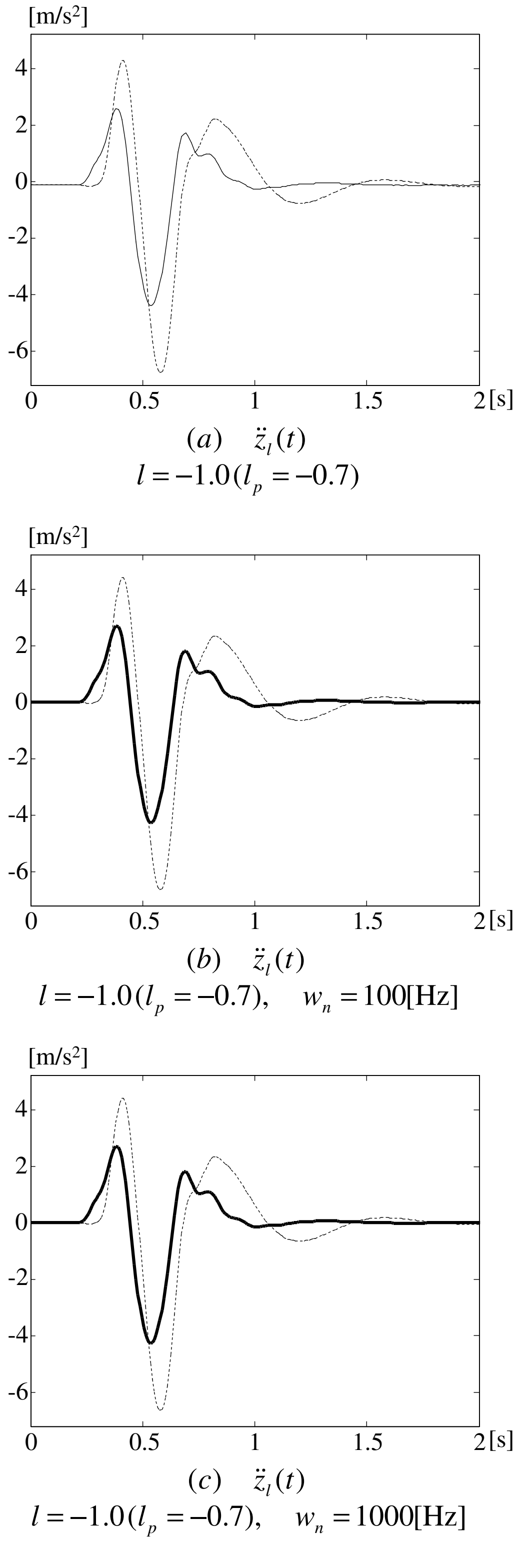

Fig. 6 Responses of the vertical acceleration of the vehicle at the specified location $l = \quad 1.0(l_p = \quad 0.7)$