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A STUDY ON THE MODEL-MATCHING CONTROL IN THE LONGITUDINAL AUTONOMOUS DRIVING SYSTEM

S.-J. KWON[1)], T. FUJIOKA[2)], M. OMAE[3)], K.-Y. CHO[1)] and M.-W. SUH[1)*]

[1)]Sungkyunkwan University, Suwon 440-746, Korea
[2)]The University of Tokyo, 7-3-1, Hongo, Bunkyo, Tokyo 113-8656, Japan
[3)]Keio University, 5322, Endo, Fujisawa, Kanagawa 252-8520, Japan

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ABSTRACT–In this paper, the model-matching control in the longitudinal autonomous driving system is investigated by vehicle dynamics simulation, which contains nonlinear subcomponents and simplified subcomponents. The design of the robust model-matching controller is performed by the characteristics of the 2 degrees of freedom controller, which is composed of the feedforward compensator and the feedback compensator. It makes the characteristics of tractive and brake force to be equivalent to the specific transfer function, which is suggested as the reference model. Mathematical models of vehicle dynamic analysis including the model-matching control are constructed for computer simulation. Then, simple examples on open-loop simulation without any controller and closed loop simulation with the model-matching controller are applied to check the validity of the robust controller. As the practical example, the autonomous driving system in the longitudinal direction is adopted. It is proved that the model-matching control is effective and adequate to the disturbances and the perturbations, which are shown in the responses of the change of a vehicle mass and a road gradient.

KEY WORDS : Intelligent transport systems, Advanced vehicle control systems, Automated highways systems, Autonomous driving system, Model-matching control, Vehicle dynamics, Simulation

1. INTRODUCTION

By means of the rapid increase of automobiles, safety against traffic accidents, traffic congestion, environmental pollution, and economical inefficiency cause a public problem all over the world. ITS (Intelligent Transport Systems) is an activity to improve them with information, communication, and control strategy. It is embossed as the next generation technology of an automobile industry and a transportation organization (Hayashi, 1998; Kubozuka, 2002). The research and development has progressed actively on various divisions.

Above all, AVCS (Advanced Vehicle Control Systems) is applicable to intelligent and automated vehicles. In addition, AHS (Automated Highway Systems) aims to reduce congestion on highways by closer packing of automatically controlled vehicles into platoons (Rajamani *et al.*, 1998). The history of the research and development of intelligent vehicles started from GM "Futurama" exhibit at the 1939 Worlds Fair in New York which showed human's future dream. After that, the research for highway automation was undertaken in 1960's and 1970's (Furukawa, 2000). In recent years, AVCS and AHS that have been developed with an electronic control technique enable a vehicle to be controlled automatically instead of the human driver's operation.

The autonomous driving system controls throttle, brake, and steering functions by recognizing the road environment in order to maintain the traffic safety (Tamura and Furukawa, 1998). Nowadays, many researches have been carried out (Chee, 2000; Gortan *et al.*, 1995; Lee and Lee, 2001; Omae and Fujioka, 1998; Peng and Tomizuka, 1991; Prestl *et al.*, 2000; Won *et al.*, 2001) and some of the autonomous driving systems are realized. Especially, the researches on the design of longitudinal and lateral controllers for the autonomous driving system have been done on diverse areas. Seto *et al.* (1998) developed a headway distance control system using inter-vehicle communication to control vehicle longitudinal motion. Swaroop *et al.* (1994) analyzed two differential longitudinal control policies, which are spacing control strategy and the headway control strategy. Xu and Ioannou (1994) designed an adaptive throttle controller for speed tracking. Moreover, to design the longitudinal and lateral controllers, the researches on the vehicle dynamic analysis through computer simulation are treated as the useful method and the researches are also in progress widely with the

**Corresponding author.* e-mail: suhmw@yurim.skku.ac.kr

improvement of computing ability.

Although the researches on the longitudinal and lateral controllers are actively progressed, we still have stability problems that the control plant is affected by the disturbances and the perturbations. Generally, the vehicle is affected by the change of a vehicle mass, a road gradient, a wind, and so on. Also, it depends on the performance of each vehicle to follow the desired acceleration. Thus, the controller is demanded to be robust to those influences.

For the robust characteristics, we investigate the model-matching control in the longitudinal autonomous driving system. To design the model-matching controller, the 2 degrees of freedom controller is adequate to reconcile the model-matching characteristics and the robust-servo characteristics. The 2 degrees of freedom controller is composed of a feedforward compensator and a feedback compensator.

To check the validity of the model-matching controller, example problems on open-loop simulation without any controller and closed loop simulation with the model-matching controller are solved. Also, to analyze the robust characteristics of the model-matching controller in the autonomous driving system, the simulations including both the longitudinal controller and the model-matching controller are carried out.

2. VEHICLE DYNAMICS

This chapter explains the vehicle dynamic models used in this paper. The longitudinal vehicle dynamic models are consisted of the simplified vehicle model including non-linear and simplified subcomponents. Nonlinear subcomponents are engine, torque converter, automatic transmission, and wheels. Simplified subcomponents are the throttle actuator model, the brake actuator model, and the pure delay model of the automatic transmission.

2.1. Vehicle Model

In this study, the vehicle model including the power train model that was developed by Fujioka *et al.* (1994, 1995, 1996) is adopted. Vehicle parameters of a full-size sedan (4.5 liter V8) are used for vehicle dynamics simulation as detailed in Table 1. The longitudinal dynamics is calculated using a simplified vehicle model.

The equation of motion is represented as the Equation (1).

$$m\dot{V}=F_t+F_b-drag(V) \tag{1}$$

where m is the vehicle mass, V is the vehicle velocity, F_t is traction force, F_b is brake force, and $drag(V)$ is the rolling resistance and aerodynamic drag function of V. From the Equation (1), the vehicle acceleration $\dot{V}$ can be obtained.

The power train model is constructed including non-linear subcomponents of engine, torque converter, automatic transmission, and wheels. To reduce the calculation time, an engine torque and an engine speed are calculated by using independent variables, which are a throttle ratio

Table 1. Vehicle parameters.

Vehicle mass [kg]	2045
Wheelbase [m]	2.880
Tire rolling radius [m]	0.315
1st gear ratio (including final reduction gear)	9.850
2nd gear ratio (including final reduction gear)	5.463
3rd gear ratio (including final reduction gear)	3.538
4th gear ratio (including final reduction gear)	2.460
Torque transfer coefficient	0.93
Coefficient of the brake model	140.22

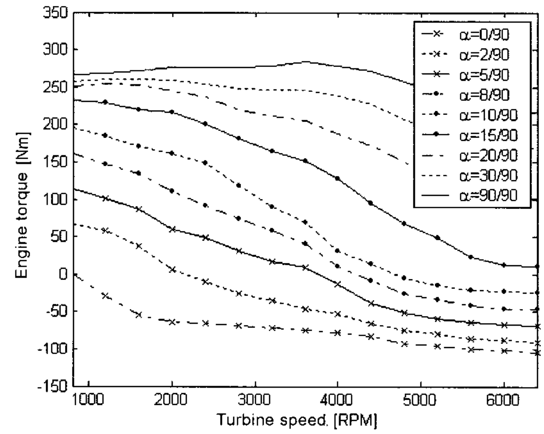

Figure 1. Engine torque model.

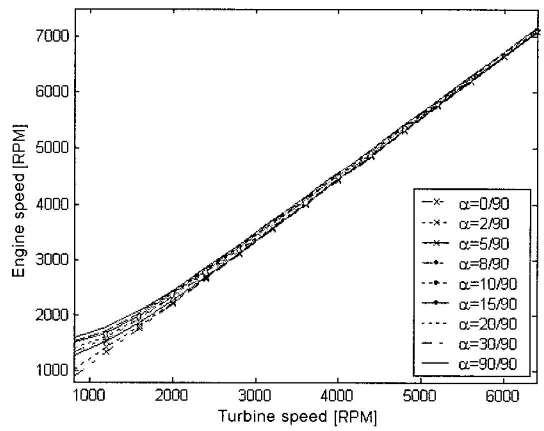

Figure 2. Engine speed model.

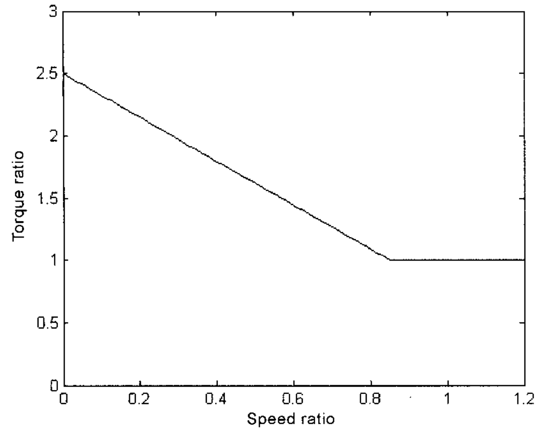

Figure 3. Torque ratio model.

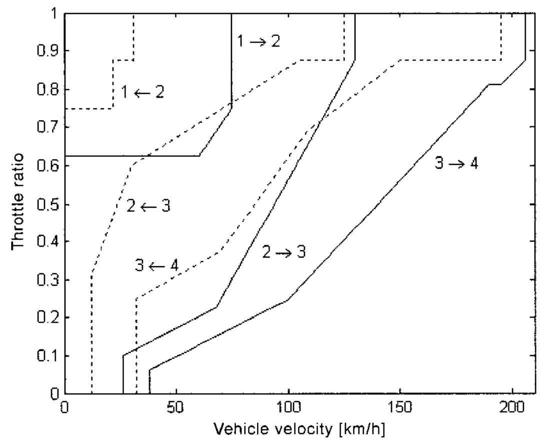

Figure 4. Automatic transmission model.

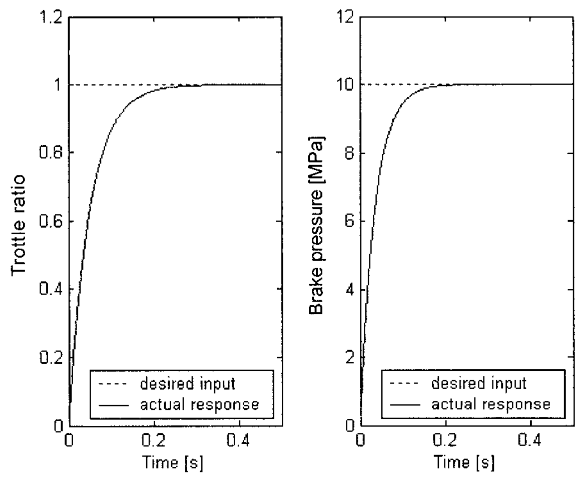

Figure 5. The step responses of the actuator models.

and a turbine speed of a torque converter. It can be represented as the following equations.

$$T_1=f_1(\alpha, N_2) \tag{2}$$

$$N_1=f_2(\alpha, N_2) \tag{3}$$

where T_1 is the engine torque, α is the throttle ratio, N_1 is the engine speed, and N_2 is the turbine speed of the torque converter. Also, f_1 and f_2 are the functions of the independent variables that are shown as the lookup tables of Figure 1 and Figure 2 (Fujioka *et al.*, 1995). A turbine torque and a turbine speed of a torque converter are calculated as the Equation (4) and the Equation (5).

$$T_2=T_1 \cdot f_3\left(\frac{N_2}{N_1}\right) \tag{4}$$

$$N_2=\frac{V}{2\pi r} \cdot f_4(V, \alpha) \tag{5}$$

where T_2 is the turbine torque of the torque converter and r is the tire radius. f_3 is the function of speed ratio to find the torque ratio that is shown in Figure 3. f_4 is the gear shift map of the automatic transmission as shown in Figure 4. Finally, traction force is calculated as the Equation (6).

$$F_t=\frac{\lambda \cdot T_2}{r} \cdot f_4(V, \alpha) \tag{6}$$

where λ is the torque transfer coefficient of the power train model. In addition, the relation between brake pressure and brake force is assumed linear based on the brake test. So, brake force can be calculated as the Equation (7)

$$F_b=-k_b \cdot P \tag{7}$$

where k_b is the coefficient of the brake model and P is brake pressure.

2.2. Actuator Model

Besides the referred vehicle model, we consider the actuator dynamics in this study. A throttle actuator (DC-motor system) and a brake actuator (electro-hydraulic system) are modeled based on the first order delay functions with the time constant of 0.05 and 0.035 separately. These constants are obtained by simplifying the responses of the actuators based on the experimental data. The step responses of the throttle and the brake actuator models are shown as Figure 5 respectively. The response of the brake actuator is modeled faster than that of the throttle actuator.

2.3. Pure Delay Model

Sometimes, vehicle dynamics simulation is very sensitive to the automatic transmission model. Due to the shift-busy characteristics of the automatic transmission model, vehicle dynamics simulation gets increased the nonlinear features of dynamic analysis so that it is hard to make out

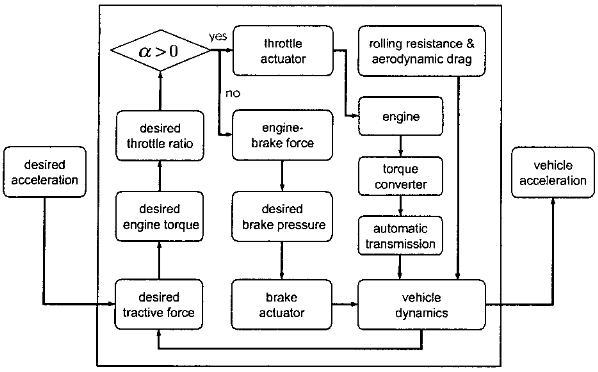

Figure 6. The flow chart of the control plant model.

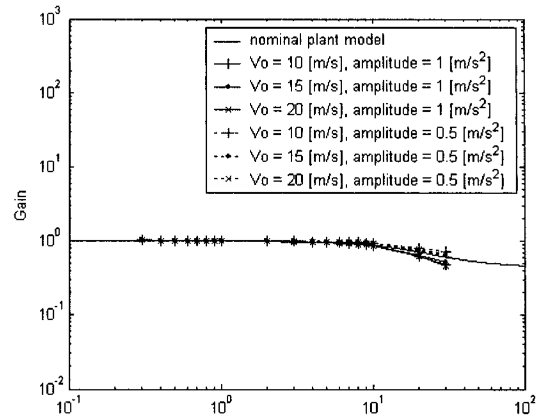

Figure 7. Gain characteristics of the control plant model and the nominal plant model by the frequency analysis.

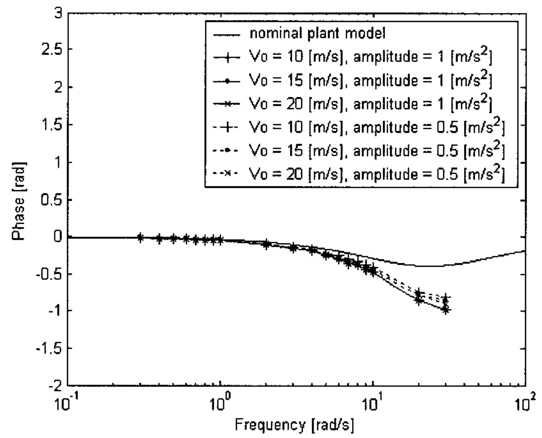

Figure 8. Phase characteristics of the control plant model and the nominal plant model by the frequency analysis.

the characteristics of responses. In this study, we suggest the pure delay function, which is installed in the automatic transmission model as the Equation (8).

$$L(u(t-a)f(t-a))=e^{-as}F(s) \tag{8}$$

where the Laplace transform of $f(t)$ delayed by time is equal to the Laplace transform $f(t)$ multiplied by e^{-as}. Also $u(t-a)$ means that the unit-step function that is delayed by time a. The pure delay function enables the shift of gear response to be smooth. The time constant used in this study is 0.05 to minimize the shift-busy characteristics of the automatic transmission model.

3. CONTROL PLANT MODEL

The control plant model is constructed to accomplish the autonomous driving system based on the desired acceleration calculated from the longitudinal controller. The main objective of the autonomous driving system is to control the throttle and the brake automatically without the human driver's operation.

3.1. Design of the Control Plant Model

In order to pursue the longitudinal autonomous driving system, the control plant model is composed as shown in Figure 6. The input variable of the control plant model is the desired acceleration, which is determined by a longitudinal controller.

Based on the desired acceleration, the desired tractive force, the desired engine torque, and the desired throttle ratio are calculated successively. According to the desired throttle ratio, the throttle actuator is operated when it is positive, and the brake actuator is operated when it is negative. After all, the output variable of the control plant model, which is the actual vehicle acceleration is calculated along with the vehicle dynamics.

3.2. Design of the Nominal Plant Model

The nominal plant model is necessary to design the model-matching controller. That is to make the response of the control plant model to be the simple transfer function. In this study, the nominal plant model is designed by the frequency analysis of the control plant model described in Figure 6.

Frequency responses (gain characteristics and phase characteristics) of the control plant model according to the change of an initial velocity and an amplitude are shown in Figure 7 and Figure 8. An initial velocity affects the variables of the control plant model like the gear status, the engine speed, the aerodynamic drag. An amplitude determines the magnitude of the sinusoidal input for the frequency analysis. From the frequency responses of the control plant model, the design of the nominal plant model $P_M(s)$ is mainly based on gain characteristics, which is presented by a specific transfer function as the Equation (9).

$$P_M(s)=\frac{0.45s+16}{s+16} \tag{9}$$

In Figure 7 and Figure 8, the nominal plant model is almost in accordance with gain characteristics of the control plant model. However, there are some differences of phase characteristics in high frequency. That is caused by the dead-time characteristics. It will be discussed on the next section of the robust stability.

4. MODEL-MATCHING CONTROL

Through the control plant model, the output variable varies according to the change of the parameters in the control plant model. Thus, the controller is demanded to be robust to those influences. Therefore, we make the reference model that has the normative characteristics of

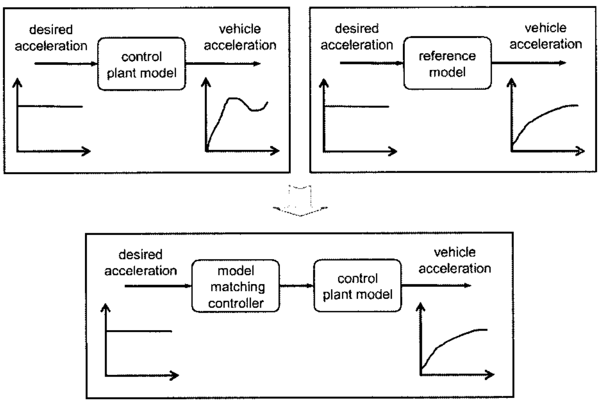

Figure 9. The concept of the model-matching control.

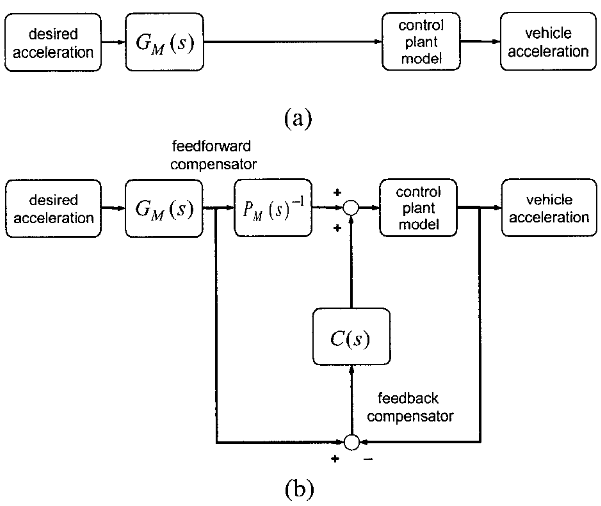

Figure 10. The flow chart of the open-loop system and the closed loop system. (a) Open-loop system without any controller. (b) Closed loop system with model-matching controller.

the responses. Then, the model-matching controller including the reference model enables the vehicle acceleration to follow the reference model even if the parameters of the control plant model are changed as shown in Figure 9.

4.1. Design of the Model-Matching Controller

Generally, the open-loop system without any controller for the autonomous driving is represented as Figure 10(a). In this system, the vehicle is affected by the change of a vehicle mass, a road gradient, a wind, and so on. Also, it depends on the performance of each vehicle to follow the desired acceleration.

The closed loop system with the model-matching controller is constructed as described in Figure 10(b). To design the model-matching controller, the 2 degrees of freedom controller is adequate to reconcile the model-matching characteristics and the robust-servo characteristics. The 2 degrees of freedom controller is composed of a feedforward compensator and a feedback compensator.

The model-matching controller can make the various characteristics of tractive and brake force to be equivalent to those of a specific transfer function, which is suggested as the reference model. On that account, it can be possible to make various vehicles to be controlled at the same performance.

4.2. Design of the Feedforward Compensator

The reference model has the normative characteristics of the model-matching control. In this study, the reference model $G_M(s)$ is decided to be the first order delay function as the Equation (10). Also, the time constant is assumed as 1. This reference model enables the desired acceleration from the longitudinal controller to be smoothly controlled by the time constant.

$$G_M(s)=\frac{1}{s+1} \tag{10}$$

The feedforward compensator has the model-matching characteristics as the Equation (11).

$$G_M(s)\cdot P_M^{-1}(s)=\frac{1}{s+1}\cdot\frac{s+16}{0.45s+16} \tag{11}$$

4.3. Design of the Feedback Compensator

The feedback compensator affects the robust stability. Also, it conduces to the reduction of the disturbance. To design the feedback compensator, the sensitivity of the closed loop $S(s)$ is used as the Equation (12).

$$S(s)=\frac{s}{s+w}=\frac{1}{1+P_M(s)\cdot C(s)} \tag{12}$$

where w is a parameter to adjust the robust stability of the closed loop. Using the Equation (12), the feedback compensator $C(s)$ is presented as the Equation (13).

$$C(s) = \frac{ws + 16w}{0.45s^2 + 16s} \tag{13}$$

4.4. Robust Stability

The multiplicative uncertainty $\Delta(s)$ of the dead-time characteristics is taken into consideration for the robust stability. It is presented as the Equation (14).

$$\Delta s = e^{-Ls} - 1 \tag{14}$$

where L means dead-time delay. The Equation (14) is satisfied under the multiplicative uncertainty.

$$\left\| \frac{\Delta(s)}{W(s)} \right\|_{\infty} \leq 1 \tag{15}$$

where the weighting function $W(s)$ can be obtained in the following equation.

$$W(s) = \frac{2.1Ls}{Ls + 1} \tag{16}$$

Figure 11 shows the relation between the multiplicative uncertainty and the weighting function when L is 0.2. Also, the feedback compensator is required to satisfy the Equation (17) to be a robust stabilizer for the dead-time characteristics.

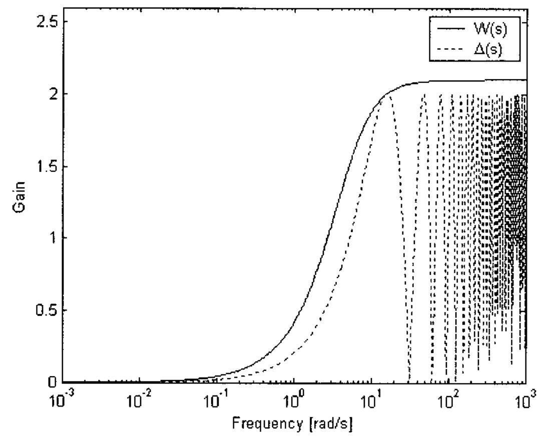

Figure 11. The weighting function.

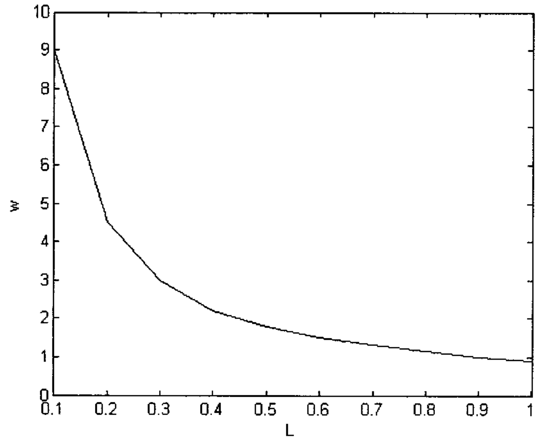

Figure 12. The relationship between w and L.

$$\left\| \frac{P_M(s) \cdot C(s) \cdot W(s)}{1 + P_M(s) \cdot C(s)} \right\|_{\infty} < 1 \tag{17}$$

Using the above equation, the relationship between w and L is obtained in Figure 12. This makes it feasible to design the feedback compensator.

5. VERIFICATION OF THE MODEL-MATCHING CONTROL

In order to confirm the robust characteristics of the model-matching control, simulation is carried out for two cases, which are in the fixed gear condition (the second gear) and in the free gear condition. Also, the simulation results for each case are compared between open-loop simulation and closed loop simulation that had the flow chart represented in Figure 10. For this purpose, the vehicle dynamic models are modularized in the computer program with the simulation step time of 0.001 [s]. The simulation compiler is Microsoft Visual C++ 6.0.

In the simulation, the desired acceleration is assumed as ±1 [m/s^2] to make a comparison between the acceleration and the deceleration. Also, the reference model is the first order delay function with the time constant of 1. The initial vehicle velocity of the control plant model is 10 [m/s]. To verify the robustness of the model-matching control to the disturbances and the perturbations, the vehicle mass is changed by ±50% and the road gradient is changed by ±5%.

5.1. Fixed Gear Condition

Figure 13 shows that the control plant model is simulated on open-loop simulation that is conducted without any feedforward and feedback compensator. It is shown that the vehicle acceleration depends on the vehicle mass and the road gradient.

To design the robust controller which is effective under the disturbance and the perturbation, Figure 14 is simulated on closed loop simulation with the model-matching controller, which is composed of the feedforward and feedback compensator. As a simulation result, it is confirmed that closed loop simulation with the model-matching controller has robustness to the variation of the vehicle mass and road gradient. Also, the model-matching control enables the vehicle acceleration to follow the reference one.

5.2. Free Gear Condition

Figure 15 shows that the control plant model is performed on open-loop simulation. The peaked point means the

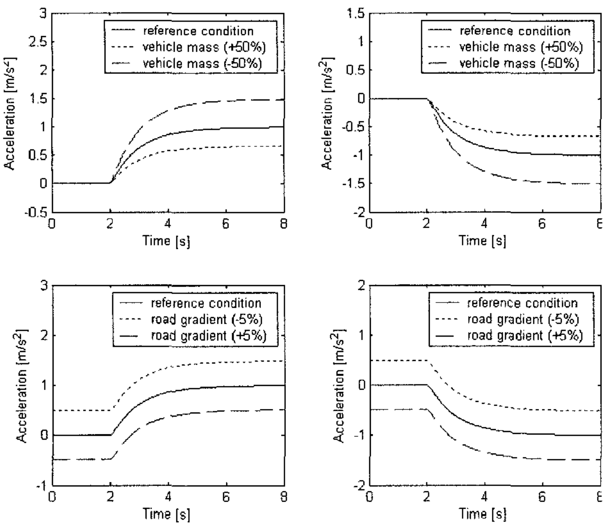

Figure 13. Open-loop simulation in the fixed gear condition.

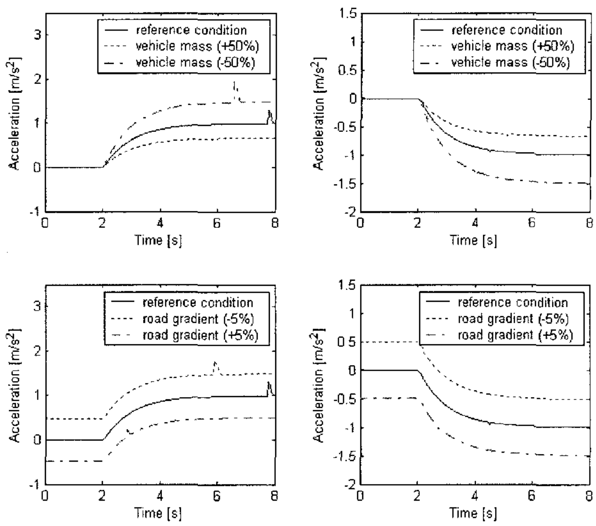

Figure 15. Open-loop simulation in the free gear condition.

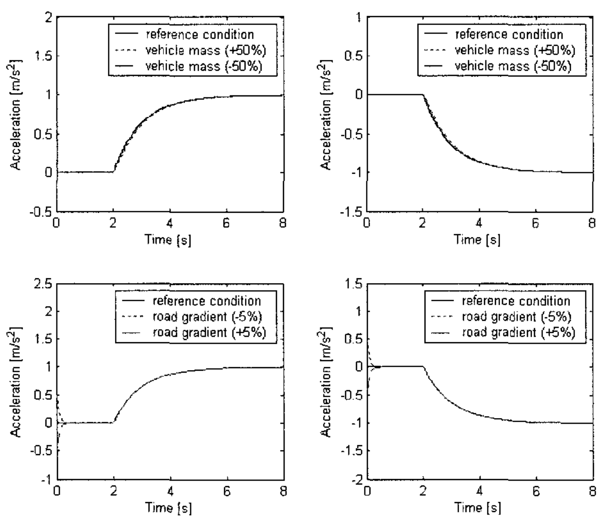

Figure 14. Closed loop simulation in the fixed gear condition.

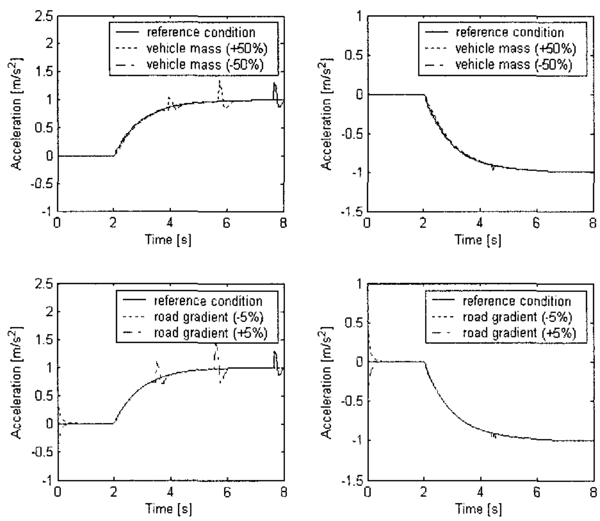

Figure 16. Closed loop simulation in the free gear condition.

gear shifting according to the variation of the throttle ratio and the vehicle velocity. If the gear ratio is changed, the throttle ratio must be changed depending on the current gear ratio. The vehicle acceleration has overshoot characteristics near the shift point of the gear since the throttle actuator model has the first order delay function. As a result of Figure 15 and Figure 16, the model-matching control is proved that it has robust characteristics even in the free gear condition.

6. APPLICATION EXAMPLE

The application example is applied to the autonomous driving system in the longitudinal direction. Figure 17 represents the vehicle following control that is one of the autonomous driving systems. According to the leading vehicle, the following vehicle equipped with the autonomous driving system tries to reduce the error of the relative distance between the leading vehicle and the following vehicle and maintains the desired distance. Therefore, to be controlled automatically, the longitudinal controller decides the desired acceleration depending on vehicle parameters, driving information from each sensor, and the control strategy.

For the simulation, the initial velocity of the leading vehicle is set up as 30 [m/s] and the leading vehicle has the acceleration of 0.5 [m/s^2] from 15 [s] to 30 [s]. Also, the initial velocity of the following vehicle is 30 [m/s],

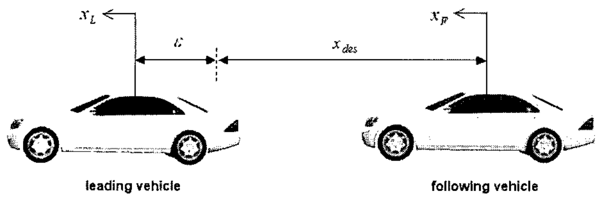

Figure 17. The autonomous driving system in the longitudinal direction.

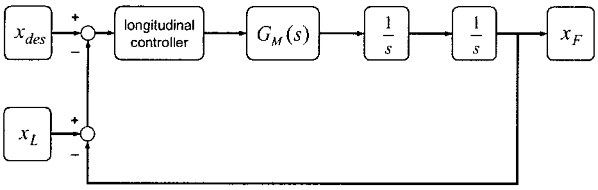

Figure 18. The flow chart to find out the weighting coefficients.

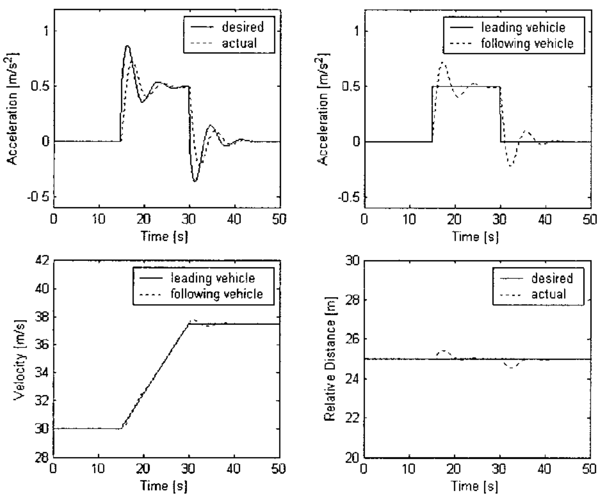

Figure 19. The autonomous driving simulation to find out the weighting coefficients.

the desired distance is fixed as 25 [m], and the initial error of the relative distance is 0 [m].

6.1. Autonomous Driving Simulation

In this study, the sliding controller for the autonomous driving system researched by Fujioka *et al.* (1995) is adopted to calculate the desired acceleration. To find out the weighting coefficients of the sliding controller easily, we use the characteristic of the model-matching controller, which has been proved in chapter 5 that the model-matching controller enables the vehicle acceleration to follow the reference model. So, the model-matching controller can be simplified as the reference model. It is

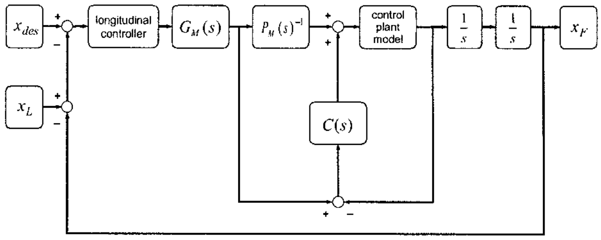

Figure 20. The flow chart to analyze the robust characteristics.

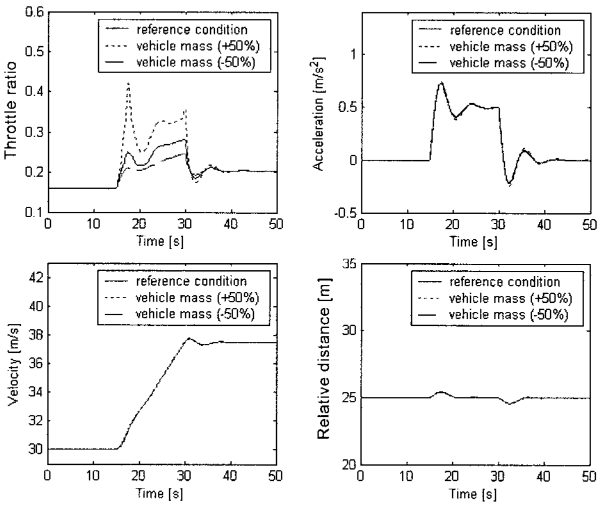

Figure 21. The simulation results according to the change of the vehicle mass.

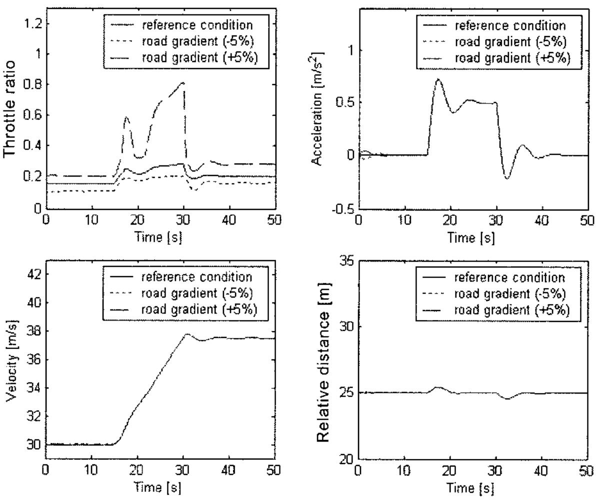

Figure 22. The simulation results according to the change of the road gradient.

shown as Figure 18.

Figure 19 shows the simulation result of the autonomous driving system using the flow chart in Figure 18 and the proper weighting coefficients. Since the reference model is the first order delay function with the time constant of 1, there are a little differences between the desired acceleration and the actual acceleration near 15 [s] and 30 [s]. However, it is shown that the following vehicle accelerates according to the leading vehicle and it maintains the desired distance.

6.2. Analysis of the Robust Characteristics

To analyze the robustness of the model-matching controller in the autonomous driving system, the simulations including both the longitudinal controller and the model-matching controller are carried out. It is shown in Figure 20.

The simulation results according to the change of the vehicle mass are shown in Figure 21. Also, the simulation results according to the change of the road gradient are shown in Figure 22. In these cases, the vehicle mass is changed by ±50%, and the road gradient is changed by ±5%. The reference condition means that it is simulated by the flow chart of Figure 20 without any change of parameters. Also, it has the same performance by the flow chart of Figure 18.

As a result of Figure 21 and Figure 22, in order that the model-matching controller performed the vehicle acceleration to chase the reference model, the model-matching controller adjusts the input acceleration of the control plant model. So, the throttle ratio of the control plant model is changed to generate the reference acceleration. Therefore, the simulation result under the change of the vehicle mass and the road gradient is almost identical with reference condition.

7. CONCLUSION

In this paper, the model-matching control in the longitudinal autonomous driving system is investigated by vehicle dynamics simulation. Above all, the vehicle dynamic models are consisted of the simplified vehicle model including nonlinear subcomponents and simplified subcomponents. Nonlinear subcomponents are engine, torque converter, automatic transmission, and wheels. Simplified subcomponents are the throttle actuator model, the brake actuator models, and the pure delay model of the automatic transmission. In addition, the control plant model is constructed to accomplish the autonomous driving system based on the desired acceleration calculated from the longitudinal controller. Also, the model-matching controller is proposed as a robust controller by using the 2 degrees of freedom controller, which is composed of a feedforward compensator and a feedback compensator.

To check the validity of the model-matching control, the comparison between open-loop simulation and closed loop simulation is conducted on various conditions. Also, fixed gear condition and free gear condition are compared. Moreover, to analyze the robustness of the model-matching controller in the autonomous driving system, the simulations including both the longitudinal controller and the model-matching controller are carried out.

It is confirmed that the model-matching control is effective and adequate to the disturbances and the perturbations, which are shown in the responses of the change of a vehicle mass and a road gradient. In addition, it is shown that the reference model of the model-matching controller has the normative characteristics and it enables the vehicle acceleration to follow the reference model even if the parameters of the control plant model are changed. Also, this robust characteristic of the model-matching control in the longitudinal direction can be useful and helpful to be expanded to the model-matching control in the lateral direction and the upper layer control such as the platoon-level control.

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