Estimation of Vehicle Driving-Load with Application to Vehicle Intelligent Cruise Control

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This paper describes a vehicle driving-load estimation method for application to vehicle Intelligent Cruise Control (ICC). Vehicle driving-load consists of aerodynamic force, rolling resistance, and gravitational force due to road slope and is unknown disturbance in a vehicle dynamic model. The vehicle driving-load has been estimated from engine and wheel speed measurements using a vehicle dynamic model and a least square method. The estimated driving-load has been used in the adaptation of throttle/brake control law. The performance of the control law has been investigated via both simulation and vehicle tests. The simulation and test results show that the proposed control law can provide satisfactory vehicle-to-vehicle distance control performance for various driving situations.

Key Words : Intelligent Cruise Control, Vehicle Driving Load, Estimation, Wheel Speed, Brake

1. Introduction

Driver assistant systems and active safety systems have been active topics of research in the past decade. It has long been recognized that ordinary cruise control systems for passenger cars are becoming less and less meaningful because the increased traffic density rarely makes it possible to drive at a pre-selected speed. Intelligent Cruise Control (ICC) systems and Stop and Go (S&G) Cruise Systems control both speed and distance to preceding vehicles and can both improve the driving comfort and reduce the danger of rear-end-collision. Compared to the ordinary cruise control, the goal of ICC and S&G systems is to prevent the vehicle-to-vehicle spacing from dropping to an unsafe level in various driving situations, i.e., in high speed and low speed stop and go driving situations. The throttle and brakes should be gently controlled so that the driver is aware that the controller has taken over, but is not surprised by this action. In order to achieve high customer acceptance, a controller has to perform similarly to an experienced human driver.

There has been a lot of research conducted on the ICC and on the control of engine and brake torque for application to vehicle speed and distance control (Winner et al., 1996; Muller et al., 1992; Hoes et al., 1996; Chien et al., 1994; Yanakiev et al., 1996; Ioannou et al., 1993; Hedrick et al., 1991; Choi et al., 1995; Yi et al., 2000). Vehicle driving-load consists of aerodynamic force, rolling resistance, and gravitational force due to road slope and is unknown disturbance in a vehicle dynamic model. Although the vehicle driving-load has significant impact on the performance of the vehicle-to-vehicle distance control, relatively little research has

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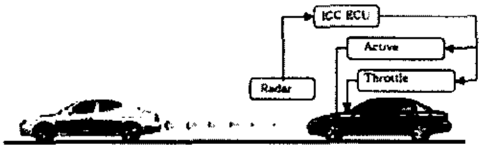


Fig. 1 Vehicle longitudinal control system

been done on the vehicle control with vehicle driving-load estimation. A robust cruise control law with a disturbance observer which estimates road gradient has been proposed (Inoue et al., 1993). A driving-load estimation algorithm for application to vehicle longitudinal motion control has been recently proposed (Kim et al., 2000). It has been revealed in the ICC vehicle tests that, in the case that the error between the nominal and actual vehicle driving-loads is not negligible, the vehicle-to-vehicle distance does not converge to the desired vehicle-to-vehicle distance. This paper presents a vehicle driving-load adaptive control law that was developed for vehicle-to-vehicle distance control. The vehicle driving-load has been estimated from engine and wheel speed measurements using a vehicle dynamic model and a least square method. The estimated driving-load has been used in the adaptation of throttle/brake control law. The performance of the control law has been investigated via both simulation and vehicle tests. The simulation and test results show that the proposed control law can provide satisfactory vehicle-to-vehicle distance control performance for various driving situations.

2. ICC Vehicle Model and Control Law

Figure 1 shows a vehicle longitudinal control system. The system consists of a radar sensor, a controller (ECU), a brake actuator (active booster) and a throttle actuator.

A block diagram of the vehicle and ICC algorithm is shown in Fig. 2. The distance to a preceding vehicle and the relative velocity are measured using a radar sensor. The distance and relative speed to the preceding vehicle and the pre-selected speed are fed to an ICC controller. Comparison of the headway distance and the distance

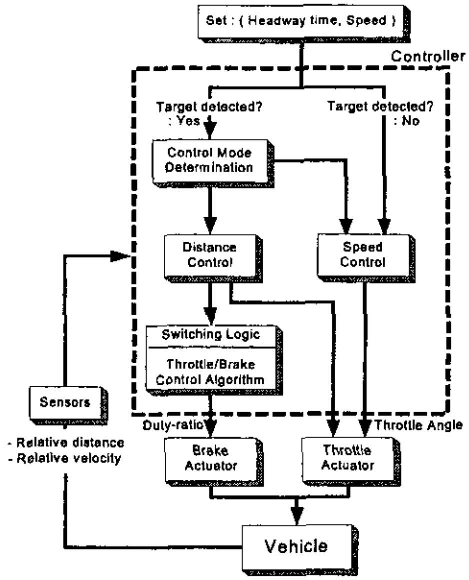


Fig. 2 ICC control algorithm

to the preceding vehicle is used to determine control mode between the speed control and the distance control. In the case of the speed control, the controller works like a conventional cruise control. The controller controls the throttle and brakes such that the vehicle acceleration tracks the desired acceleration, which is designed so that the vehicle-to-vehicle distance converges smoothly to the headway distance.

A vehicle longitudinal control system for an ICC can be represented as

$$M_v a = K_e T_e(\omega_e, \alpha) - K_b T_b(p_d) - F_L \quad (1)$$

where M_v is the vehicle mass, a the vehicle acceleration, K_e the lumped gain for the engine, T_e the engine net torque, ω_e the engine speed, α the throttle angle, K_b the lumped gain for the entire brake system, p_d the vacuum booster differential pressure, and F_L the vehicle driving-load. The vehicle driving-load consists of aerodynamic force, rolling resistance, and gravitational force due to road slope and is unknown disturbance in a vehicle dynamic model. The vehicle driving-load, F_L, is represented as

$$F_L = F_r + F_a + M_v g \sin\theta \quad (2)$$

where F_r is the rolling resistance force, F_a the aerodynamic drag force, g the gravitational constart, and $\theta$the road grade angle.

The desired acceleration, $u(=a_{des})$, is represented as follows (Yi et al., 2000):

$$u=-\kappa_1(d_h-d)-\kappa_2 v_{rel}\text{: distance control} \quad (3)$$

$$u=\kappa_3(v_{set}-v_{cc})\text{: speed control} \quad (4)$$

where d_h is the headway distance, d the vehicle-to-vehicle distance, v_{rel} the relative velocity between the preceding and the controlled vehicles, v_{set} the set speed of the controlled vehicle, v_{cc} the velocity of the controlled vehicle, and κ_1, κ_2, κ_3 are the gains. The headway distance, d_h, is defined as

$$d_h=v_{cc}t_h+d_o \quad (5)$$

where t_h is the headway time, and d_o a headway offset.

Depending on the desired acceleration that the ICC vehicle must follow, the ICC controller applies throttle or brake control. The engine net torque is represented as a function of engine speed, ω_e, and throttle angle, α, and typically the engine map is provided by the engine manufacturer as a look up table. The throttle angle, α, for the desired net engine torque for a given engine speed can be computed from the engine map that shows the throttle opening angles as a function of the engine speed and torque.

The throttle and brakes are controlled such that the vehicle acceleration tracks the desired acceleration. The throttle and brake control laws for the desired vehicle acceleration can be obtained from the vehicle model as follows:

$$\alpha=T_{en}^{-1}\left(\frac{M_{vn}u+\hat{F}_L}{K_{en}},\ \omega_e\right)\text{; throttle control} \quad (6)$$

$$P_d=T_{bn}^{-1}\left(\frac{K_{en}T_{en}(\omega,\ 0)-M_{vn}u+\hat{F}_L}{K_{bn}}\right)$$

; brake control (7)

where T_{en}, T_{bn} are nominal models of the engine map and the brake system, K_{en}, K_{bn} the nominal values of the lumped engine and brake gains, M_{vn} the nominal values of the vehicle mass, and $\hat{F}_L$ the estimated values of the vehicle driving-load.

3. Estimation of the Vehicle Driving-Load

A least square algorithm has been used for the estimation of the driving-load. It has been assumed that the vehicle acceleration, a, the engine speed, ω_e, the throttle angle, α, and the vacuum booster differential pressure, P_d, are available. A least square algorithm with forgetting factor for the vehicle driving-load, F_L, can be represented as follows:

$$\hat{\theta}(\kappa+1)=\hat{\theta}(\kappa)+\frac{P(\kappa)[y(\kappa+1)-\hat{\theta}(\kappa)]}{\lambda+P(\kappa)} \quad (8)$$

$$P(\kappa+1)=\frac{1}{\lambda}\left(P(\kappa)-\frac{P(\kappa)^2}{\lambda+P(\kappa)}\right)=\frac{P(\kappa)}{\lambda+P(\kappa)},\quad P(0)>0 \quad (9)$$

where $\theta(\kappa)=F_L(\kappa T)$,
$\hat{\theta}(\kappa)=\hat{F}_L(\kappa T)$,
$y(\kappa)=y(\kappa T)=K_{en}T_{en}(\omega_e(\kappa T),\ \alpha(\kappa T)) - K_{bn}T_{bn}(p_d(\kappa T))-M_{vn}a(\kappa T)$,

and λ is a forgetting factor. Sampling time T of 50 milliseconds and the forgetting factor of 0. 9 have been used in this study.

After many sampling, the estimation algorithm can be written as

$$\hat{\theta}(\kappa+1)=\hat{\theta}(\kappa)+(1-\lambda)[y(\kappa+1)-\hat{\theta}(\kappa)] \quad (10)$$

4. Stability Analysis of the Driving-Load Adaptive Control System

The driving-load estimation algorithm (10) can be rewritten as a continuous form as

$$\tau\frac{d\hat{\theta}(t)}{dt}+\hat{\theta}(t)=K_{en}T_{en}(\omega_e(t),\ \alpha(t)) - K_{bn}T_{bn}(p_d(t))-M_{vn}a(t) \quad (11)$$

where $\tau=\frac{T}{1-\lambda}$.

In the case of throttle control, plugging the throttle control law (6) into Eq. (1), Eq. (1) can be written as

$$M_v a=K_e\left(\frac{M_{vn}u+\hat{F}_L}{K_{en}}+\Delta T_e\right)-F_L \quad (12)$$

where ΔT_e is the error between the actual and nominal engine net torques for a given engine speed and throttle input, i. e.,$\Delta T_e=T_e(\omega_e,\ \alpha)-T_{en}(\omega_e,\ \alpha)$.

Differentiating Eq. (12) with assumptions that the vehicle driving-load be slowly varying, i. e., $\dot{F}_L\approx 0$ for a short time period and the engine torque error is time invariant, gives

$$M_v \dot{a} = \frac{K_e}{K_{en}}\left(M_{vn}\dot{u} + \dot{\hat{F}}_L\right) \tag{13}$$

Using Eq. (11) and defining the acceleration tracking error, e, as

$$e = u - a \tag{14}$$

the acceleration error dynamics can be written as

$$\tau \cdot \dot{e} + e = \tau\left(\frac{K_{en}M_v}{K_e M_{vn}} - 1\right)\dot{a} \tag{15}$$

In the case of the brake control, since the throttle input is kept in zero, the engine net torque is constant. Assuming that the brake model error is time invariant, the error dynamics in the case of the brake control can be written as

$$\tau \cdot \dot{e} + e = \tau\left(\frac{K_{bn}M_v}{K_b M_{vn}} - 1\right)\dot{a} \tag{16}$$

If the vehicle parameters are exactly known, i.e., $K_{en} = K_e$, $K_{bn} = K_b$ and $M_{vn} = M_v$, then the error dynamics becomes

$$\tau \cdot \dot{e} + e = 0 \tag{17}$$

Therefore, the acceleration tracking error tends to zero with first order dynamics in this case. In the case that there exist vehicle parametric errors, the error dynamics are excited by the vehicle jerk, $\dot{a}$. It should be noted that the effect of the jerk on the acceleration tracking error can be reduced by tuning the vehicle parameters as close to the true vehicle parameters as possible. The jerk excitation can be filtered out by the first order dynamics and the acceleration tracking error can be bounded even if there exist parametric errors.

5. Vehicle Tests and Simulations

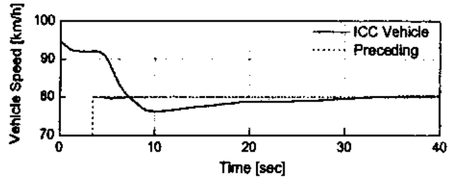
Computer simulations and vehicle tests have been done to evaluate the estimation algorithm and the performance of the proposed driving-load-adaptive control algorithm. Nonlinear powertrain model (Cho and Hedrick, 1989) has been used in the simulation study. Vehicle tests have been conducted using a test vehicle, a 2000cc passenger car equipped with a millimeter wave (MMW) radar distance sensor, a controller, an electronic vacuum booster (EVB) brake actuator and a step motor controlled throttle actuator. The differential pressure, p_d, of the vacuum booster was controlled by a PWM solenoid valve. The pressure was proportional to the duty ratio input to the solenoid valve. The already existing wheel speed sensors, engine speed sensor, and a Throttle Position Sensor (TPS) have used to estimate the vehicle driving-load and to implement the adaptive control laws.

5.1 The effect of the driving-load on vehicle-to-vehicle distance control

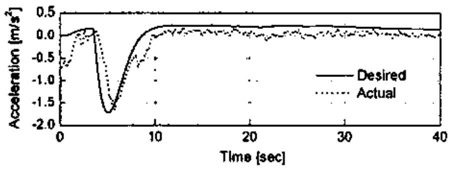
Vehicle distance control tests were done using two vehicles: a controlled vehicle and a cut-in-vehicle. Figure. 3 shows the test results: a comparison of the controlled and the cut-in vehicle speeds (a), a comparison of the desired acceleration profile and actual acceleration (b), the headway distance and the vehicle-to-vehicle relative distance (c), command and actual throttle angles (d), and a comparison of the desired and actual differential pressures of the EVB (e). The actual differential pressure of the EVB shows sensor offset when the brake command is not applied. The controlled vehicle's initial speed was 95 km/h and a vehicle of 80 km/h had appeared in the front of the controlled vehicle at 3 seconds. The initial relative distance was approximately 40 km/h and the headway time of 1.6 seconds has been used in this test. Since the speed of the cut-in vehicle is smaller than that of the controlled vehicle, the controller activates the brake control such that the relative distance converges to the headway time distance and the controlled vehicle speed converges to the preceding vehicle's speed. The throttle/brake controller forces the vehicle acceleration to converge to the desired acceleration. This test has been done on a flat test track and following nominal driving-load model has been used.

$$F_L = F_r + F_a = 260 + 0.36 \cdot v^2$$

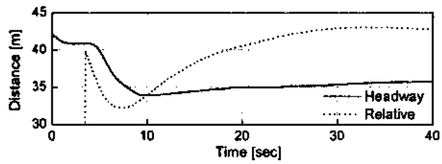
It is illustrated that although the controlled vehicle speed converges to the preceding vehicle speed, the relative distance does not tend to the headway distance and this is due to the acceleration tracking error.



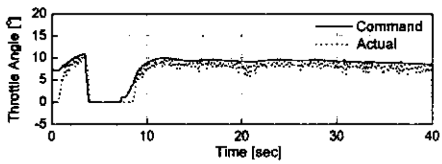
(a) ICC and preceding vehicle speeds



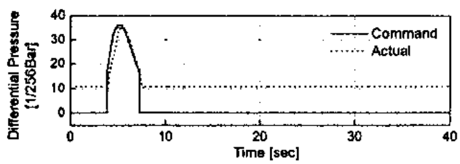
(b) Desired and actual accelerations



(c) Headway and relative distances



(d) Command and actual throttle angles



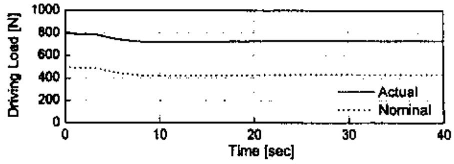
(e) Command and actual differential pressures

Fig. 3 Effect of the mismatched driving load on vehicle-to-vehicle distance control (Vehicle test results: Vehicle speed and distance control in case of a cut-in vehicle)

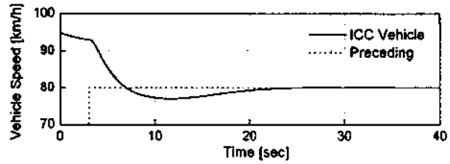
In order to illustrate that the distance and acceleration errors are due to the mismatched vehicle driving-load, computer simulation depicting the vehicle test has been done. Simulation results are shown in Fig. 4. In the simulation it has been assumed that the actual driving-load is represented as

$$F_L = 530 + 0.36 \cdot v^2$$

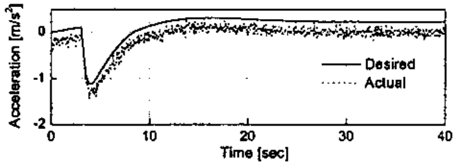
As illustrated in Fig. 4, the simulation results depict the test results very closely. The acceleration tracking error is due to the error between the actual driving-load and the nominal driving-load. The acceleration tracking error results in the distance error. It can be postulated from the simulation and test results that the distance error is due to the mismatched driving-load.



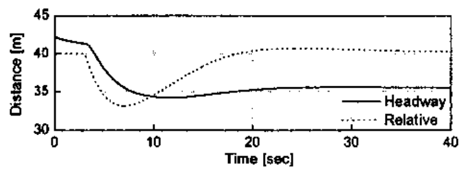
(a) Actual and nominal driving loads



(b) ICC vehicle and preceding vehicle speeds



(c) Desired and actual accelerations

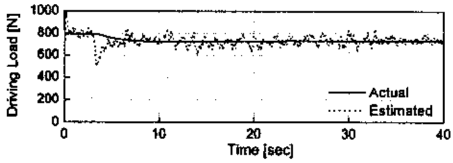


(d) Headway and relative distances

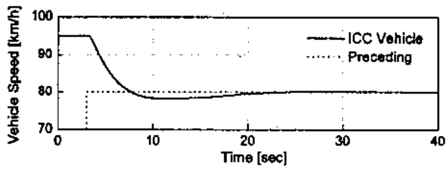
Fig. 4 Effect of the mismatched driving load on vehicle-to-vehicle distance control (Simulation results: Vehicle speed and distance control in case of a cut-in vehicle))

5.2 Driving-load adaptive control of the vehicle-to-vehicle distance

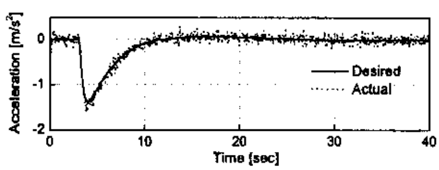
The driving-load has been estimated using the proposed algorithm and the estimated one has been used in the control. Figure 5 shows simulation results for the driving-load adaptive control of the vehicle-to-vehicle distance. Actual and estimated driving-loads in the simulation are compared in Fig. 5(a). It has been shown that the estimated driving-load is very close to the actual



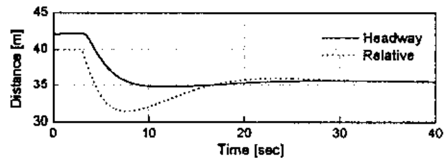
(a) Actual and estimated driving loads



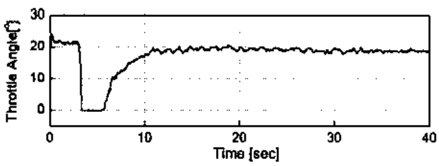
(b) ICC vehicle and preceding vehicle speeds



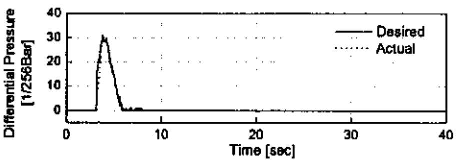
(c) Desired and actual accelerations



(d) Headway and relative distances

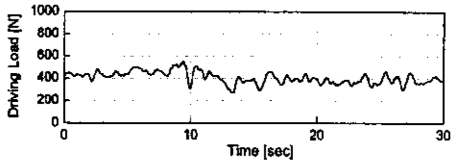


(e) Throttle angle

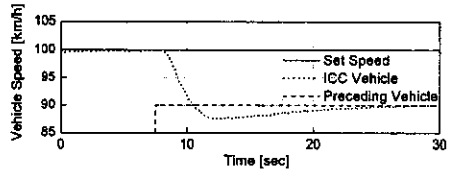


(f) Differential pressures

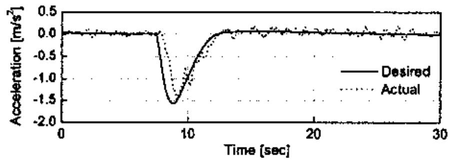
Fig. 5 Driving-load adaptive control of vehicle-to-vehicle distance (Simulation results)



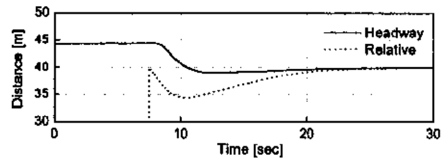
(a) Estimated driving load



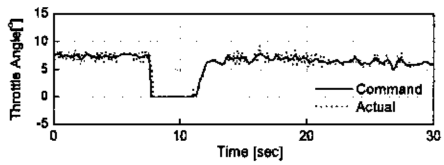
(b) ICC vehicle speed, set speed and preceding vehicle speed



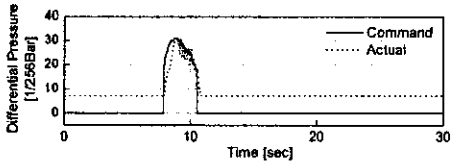
(c) Desired and actual accelerations



(d) Headway and relative distances



(e) Throttle angle



(f) Differential pressures

Fig. 6 Driving-load adaptive control of vehicle-to-vehicle distance (Vehicle test results)

one. The errors between the actual and estimated ones are due to the model difference between the nonlinear simulation model and the simplified vehicle model, i. e. Eq. (1), used in the estimation algorithm. Compared to the control without the driving-load estimation shown in Fig. 4, the vehicle speed, the acceleration, and the relative distance converge to the preceding vehicle speed, the desired acceleration, and the headway distance, respectively. As expected in the stability analysis of the driving-load adaptive control system, the acceleration tracking errors are bounded.

Vehicle test results for similar driving situation shown in Fig. 5 are illustrated in Fig. 6. Since the actual driving-load cannot be measured directly and is not available, only the estimated one is shown in Fig. 6(a). The mean values of the estimated driving-load are approximately 445N and 390N when the vehicle speeds are 100 km/h and 90 km/h, respectively. As expected in the simulation results, the vehicle acceleration tracks the desired acceleration and the relative distance converges to the headway distance.

6. Conclusions

A vehicle driving-load estimation method has been proposed. The vehicle driving-load has been estimated from engine and wheel speed measurements using a vehicle dynamic model and a least square method. The estimated driving-load has been used in the adaptation of throttle/brake control law for a vehicle Intelligent Cruise Control. The performance of the proposed estimation method and the driving load adaptive control law has been investigated via simulation and vehicle tests. The vehicle tests have proved that the proposed method can be effectively used for application to an ICC.

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