The IEEE 5th International Conference on Intelligent Transportation Systems 3 – 6 September 2002, Singapore. Stop & Go Vehicle Longitudinal Model

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Abstract – In this paper a vehicle longitudinal model targeted towards low velocity adaptive cruise control systems is presented through a review of conventional models. The vehicle model obtained has two inputs, namely the engine input and the brake input. Also it is described by the composite of the power-train system model, the brake system model and the drive-train model. The response of the proposed model is examined through numerical simulation.

I. INTRODUCTION

The automotive industry has been encouraging the development of advanced concepts that may lead to more efficient traffic flow. Already various driver assistance systems including lane-keeping and distance-keeping have been successfully developed and are being commercialized. Conventional adaptive cruise control systems have been available for more than ten years, [1] and [2]. They operate at speeds above 40km/h by controlling the throttle input. However, they do not cope successfully with low speed travel due to deficiencies in the adaptive control algorithms used. Advanced adaptive control concepts are therefore required to facilitate the development of a slow motion adaptive cruise control system, i.e. Stop & Go Adaptive Cruise Controller (Stop & Go ACC) [3]. In this report, a longitudinal vehicle motion model used in Stop & Go ACC is discussed.

II. OUTLINE OF STOP & GO ACC

The goal of Stop & Go ACC is to assist human drivers whilst in traffic jams by reducing the need for them to repeatedly accelerate and/or stop their vehicles. The basic configuration of a Stop & Go ACC is shown in Fig.1. An outline of the differences between conventional ACC systems and Stop & Go ACC systems is summarized in Table 1 [2].

TABLE 1 - OUTLINE OF DIFFERENCE BETWEEN ACC AND STOP & GO SYSTEMS

	ACC	Stop & Go
Condition of operation	ACC performs driving assistance between 40 ± 10 km/h and 170 ± 10 km/h by action on accelerator and brakes	Stop & Go performs driving assistance between 0 km/h and 40 ± 10 km/h by action on accelerator and brakes
Preceding vehicle selection	Moving vehicle in the driving path of equipped vehicle	Moving and stopped vehicles in the driving path of equipped vehicle

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Fig 1. The basic configuration of Stop & Go System

The majority of published adaptive cruise control models are related to high speed traveling. Engine dynamics combined with automatic transmission and brake system characteristics under slow motion conditions are usually disregarded. The main purpose of this paper is a reconstruction of the simplified longitudinal vehicle model targeted towards the Stop & Go ACC.

III. LONGITUDINAL VEHICLE MODEL

Most vehicles currently on the road have an internal combustion engine as their source of power, an automatic transmission, a rear axle differential gear, a brake subsystem and wheels. Fig.2 shows the dynamic interactions between vehicle components. Mathematical models of these components have frequently been examined and include some non-linear input output relations. MacMahon and Hedrick [4] have described the vehicle model development for longitudinal analysis with the consideration of such non-linear characteristics in detail. The resulting model with nine state equations is often called a "Full Order Model (FOM)".

There are several types of the simplified model from the FOM that have been derived under conditions that suppose high speed travel.

They can be classified as single input models and 2-inputs models. The input of most of the single input models is a command signal of the throttle opening angle or the engine output torque. The 2-inputs models have additionally considered the command signal to the braking system as a target braking torque.

Fig 2. Dynamic interactions among vehicle components [4]

We propose the total longitudinal vehicle model for a Stop & Go ACC system such as shown in Fig.5. under the following assumptions:

- a) no torsion of the drive axle, and
- b) no slip at the wheels

the drive-train dynamics are given as:

$$\frac{d}{dt}v(t) = \frac{1}{\beta} [T_e(t) - T_b(t) - M_r - hF_a(t) - F_g(t)] \quad (1)$$

$$\beta = mh^2 + J_{wf} + J_{wr} \quad (2)$$

$$M_r = k_r mgh \quad (3)$$

where k_r denotes the rolling resistance coefficient, $F_a(t)$ is the aerodynamic drag defined by equation (4) and $F_g(t)$ is the road grade force defined by equation (5). We adopt the first order lag system with pure-time delay [6] as the brake system dynamics model as follows

$$F_a(t) = K_d v(t)^2 \quad (4)$$

$$F_g(t) = mgh \sin \theta(t) \quad (5)$$

$$\frac{d}{dt}P(t) = -aP(t) + K_b u_b(t - L_b) \quad (6)$$

$$T_b(t) = \mu R A P(t) \quad (7)$$

where a and K_b denote the time-constant and the gain constant of the proportional reducing valve, L_b denotes the pure-time delay, μ denotes the frictional coefficient, R denotes the effective radius of brake plate, and A denotes the piston area. It will be noted that a , L_b , and μ have large uncertainties with regard to the command pressure amplitude.

IV. SIMULATION OF VEHICLE DYNAMICS

The proposed model for longitudinal vehicle dynamics is tested by simulation using MATLAB/SIMULINK. The parameters used in this simulation are shown in Table 2 and Table 3. The gear shifting function is described in Fig.3.

TABLE 2
PARAMETERS OF THE TEST VEHICLE [4]

Notation	Definition	Value	Unit
m	Vehicle mass	1769	kg
h	Wheel radius	0.315	m
J_{wr}	Inertia of wheels	2.8	kgm ²
J_{wf}			
k_r	Coefficient of rolling resistance	0.009816	
C_a	Aerodynamic drag coefficient	0.4298	kg/m
R_1	First gear speed ratio reduction	0.3424	
R_2	Second gear speed ratio reduction	0.6379	
R_3	Third gear speed ratio reduction	1.0	
R_4	Forth gear speed ratio reduction	1.4194	
R_d	Final drive speed ratio reduction	0.3521	

TABLE 3
PARAMETERS OF THE TORQUE CONVERTER CHARACTERISTIC [25]

c_0	c_1	c_2	r_0	r_1
-1.695 * 10 ⁻³	0.2946 * 10 ⁻³	1.874 * 10 ⁻³	-	2.2
			1.412

Fig.3 Gear shifting characteristic

The simulation results are shown in the following. In each case, the minimum torque command is input to the engine for idling and the brake system keeps the vehicle from going into motion during the period of 0-10(s). Fig.4 is the step response for the large amplitude of the engine input $T_{net}(t)=120+T_{min}$ (Nm) where $T_{min}=8.2$ (Nm) (Case-1). The proposed model adequately represents the vehicle dynamics, because the response of the vehicle speed increases smoothly and the oscillatory response of engine velocity caused by the gear shifting indicates a fairly similar transient characteristic to the result shown in [8].

Fig.4 Simulation of step response (Case-1)

Fig.5 shows the step response for the small amplitude engine command $T_{net}(t)=3+T_{min}$ (Nm) where the engine command is generated as a rectangular wave whose period and duty cycle are 180(s) and 50%, respectively (Case-2). In this case, the vehicle speed is limited to a very slow speed range. The automatic transmission gear is shifted up from first gear to second gear at 70(s) and the vehicle speed is reduced with the second gear engaged. At 80(s), the automatic transmission gear shifts down to first gear again and the vehicle accelerates. While the command input amplitude changes to the minimum value from 100(s) to 180(s), the vehicle is decelerated to a steady-state speed (about 4.2 [km/h]).

Fig.6 shows the step response with the additional brake torque input from 75(s), generated by a rectangular wave command signal whose period and duty cycle are respectively given as 170(s) and 50%, and results in the same situation mentioned above (Case-3).

Fig.5 Simulation of step response (Case 2)

Fig.6 Simulation result of step response (Case-3)

We can note that vehicle open loop dynamics in the slow speed range will show a bizarre and complex response because of the frequent gear shift motion and the mix of brake torque together with engine brake torque and the brake system output torque.

V. CONCLUSIONS

A longitudinal model using the of Stop & Go ACC systems design has been discussed based on the review of conventional intelligent vehicle control systems. This model includes both the dynamics of the power-train system and of the brake system. The power-train system has been modeled using the first order lag system with input and output disturbances by taking the engine dynamics and the torque converter characteristic into account. The time-constant of the power-train model results depend on vehicle speed. It has been shown that, when the vehicle comes to a stop, the power-train model changes to the first order simple nonlinear model. Brake system dynamics has been described as a first order lag system with pure-time delay. Further, the responses of the proposed vehicle model have been examined by the use of numerical simulations. We intend to develop the speed and/or headway distance controller that is available in the slow speed range based on the proposed vehicle model as the next step to the realization of a Stop & Go ACC system.

VI. REFERENCES

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