International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 04 Issue: 12 | Dec-2017 www.irjet.net p-ISSN: 2395-0072

Vehicle Dynamics and Drive Control for Adaptive Cruise Vehicles

Dileep K[1], Sreepriya S[2], Sreedeep Krishnan[3]

[1,3] Assistant Professor, Dept. of AE&I, ASIET Kalady, Kerala, India
[2] Associate Professor, Dept. of AE&I, ASIET Kalady, Kerala, India

***

Abstract - *Adaptive cruise control (ACC) provides assistance to the driver in longitudinal control of their vehicle by controlling the accelerator, engine powertrain and vehicle brakes to maintain a desired time-gap to the vehicle ahead. Adaptive cruise control system can comprehensively address issues of tracking capability, fuel economy and driver desired response. Nonlinear vehicle dynamics is compensated by the lower level controller in the hierarchical control architecture and this enables tracking of desired acceleration. Proper switching between throttle and brake controller is needed for effective velocity and distance tracking by lower level controller. PID controller is used for throttle control and feedback control is used for brake control.*

***Key Words*: ACC, Hierarchical control architecture, PID controller, feedback control, Drive control.**

1. INTRODUCTION

Technological developments helps in the effort to build automated vehicles in order to travel safe and efficient using current highway system. Cruise control systems developed during the last decade and recently produced and made available to users by many automobile company is the first step toward that direction. The advanced cruise control (ACC) system offers three major advantages: maintained vehicle speed, improved fuel economy, and increased driver comfort on long distance trips.

A vehicle equipped with an ACC system (an ACC vehicle) uses on-board radar in front of the vehicle that measures the range and range rate between itself and the preceding vehicle to enable headway control. An on-board sensor measures the speed of the vehicle by measuring the wheel rotation [1]. An ACC system uses its two modes of longitudinal control as follows. If there is no preceding vehicle, then the ACC system enables its speed control mode and regulates the vehicle speed at a driver-defined setting. However, if the ACC vehicle is traveling using its speed-control mode and encounters a new, or target, vehicle traveling with a slower speed, then the ACC system must enable its headway-control mode to avoid a collision with the target vehicle. Headway control consists of two phases of operation; transitional and steady state. The procedure through which the ACC vehicle establishes the specified inter-vehicle distance behind the target vehicle is referred to as transitional operation. Steady-state has two operations; first, the steady-state spacing policy and the corresponding specified inter-vehicle distance must be selected. Second, the ACC system must implement a control algorithm that maintains the specified inter vehicle distance regardless of the movements performed by the target vehicle. This control algorithm must also provide the ACC vehicle with individual vehicle stability and string stability.

2. MODELING OF CAR FOLLOWING SYSTEM

The longitudinal dynamics of a vehicle are nonlinear. Its salient features include the static nonlinearity of engine torque maps, time varying gear position and aerodynamic drag force as a quadratic function of vehicle speed. A hierarchical controller consisting of a lower level controller and an upper level controller, as shown in Fig. 1 is used.

2.1 Hierarchical Control Architecture

Hierarchical control architecture consisting of a lower level controller and an upper level controller is utilized to compensate for nonlinear vehicle dynamics and enables tracking of desired acceleration [2].

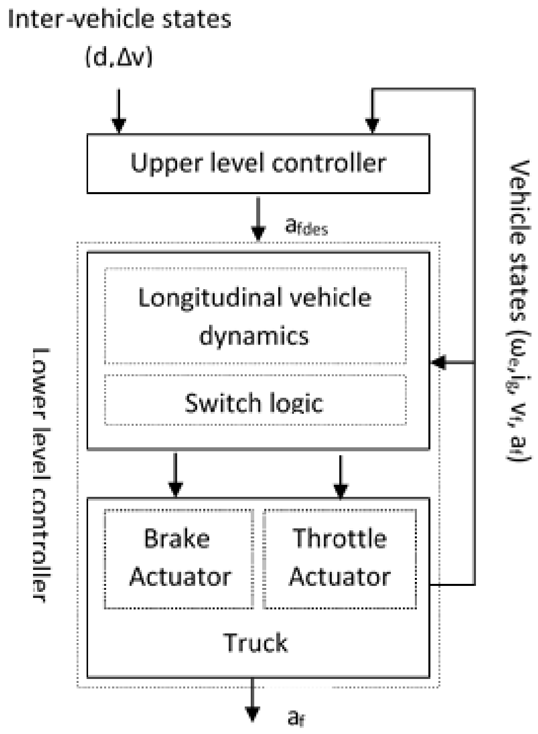

Fig-1 Hierarchical control architecture

The lower level controller determines the acceleration pedal position a_{accl} and brake pressure P_{brk}, so as to ensure that the desired acceleration a_{fdes} is tracked by the actual acceleration a_f of the vehicle. The upper level controller determines the desired longitudinal acceleration according to inter-vehicle states and vehicle states.

2.2 Longitudinal Vehicle Model

The model of a cruise control vehicle is shown in fig 2. Let v be the speed of the car and v_r the desired (reference) speed. The controller, receives the signals v and v_r and generates control signals u_1 and u_2 that is sent to the actuators that controls the throttle position and brake pressure respectively. Torque T delivered by the engine is in turn controlled by throttle, and is transmitted through the gears and the wheels, which generates a force F that moves the car. There are disturbance forces F_d due to variations in the slope of the road, the rolling resistance and aerodynamic forces.

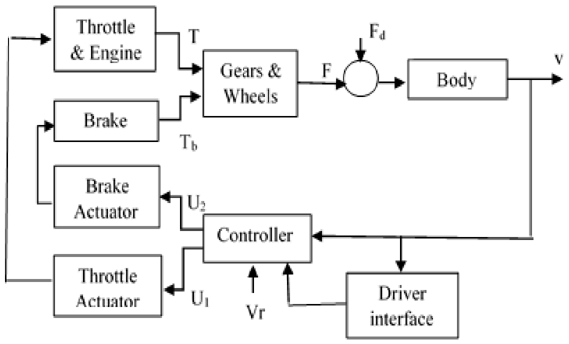

Fig. 2. Block diagram of Cruise control vehicle

Force balance for the car body is used to generate a mathematical model. Let speed of the car be represented by v, total mass by m (including passengers), Contact of the wheels with the road generates a force F, and all the disturbance force due to gravity, friction and aerodynamic drag is represented using F_d. Then motion of the car can represented using the equation

$$m\frac{dv}{dt} = F - F_d \qquad (1)$$

The control signal controls the throttle position which controls the fuel injection rate to the engine which in turn controls the force F generated by the engine, and its torque. The torque also depends on engine speed ω.

$$T(\omega) = T_m\left(1 - \beta\left(\frac{\omega}{\omega_m} - 1\right)^2\right) \qquad (2)$$

The maximum torque Tm is obtained at engine speed ω_m. β=0.4. Let n be the gear ratio and r the wheel radius. The engine speed is related to the velocity through the expression,

$$\omega = \frac{n}{r}v = \alpha_n v \qquad (3)$$

and the driving force can be written as

$$F = \frac{nu}{r}T(\omega) = \alpha_n u T(\alpha_n v) \qquad (4)$$

Typical values of αn for gears 1 through 5 are α1 = 40, α2 = 25, α3 = 16, α4 = 12 and α5 = 10. The inverse of αn has a physical interpretation as the effective wheel radius. The disturbance force Fd in equation (1) has three major components: Fg, the forces due to gravity; Fr, the forces due to rolling friction; and Fa, the aerodynamic drag. Letting the slope of the road be θ, gravity gives the force Fg = mgsinθ , where g = 9.8 m/s2 is the gravitational constant. A simple model of rolling friction is

$$F_r = mgC_r sgn(v) \qquad (5)$$

Where Cr is the coefficient of rolling friction and sgn(v) is the sign of v (±1) or zero if v = 0. A typical value for the coefficient of rolling friction is Cr = 0.01. Finally, the aerodynamic drag is proportional to the square of the speed:

$$F_a = \frac{1}{2}\rho C_d A v^2 \qquad (6)$$

Where r is the density of air, Cd is the shape-dependent aerodynamic drag coefficient and A is the frontal area of the car.

Newton's second law of motion is used to develop the model and is done on the assumption that the wheels has no unknown force acting on it. This assumption allows us to approximate the braking force as being proportional to the braking torque T_b. We need to control the brake line pressure defined as P_r in the brake system which is approximately proportional to the brake torque T_b. C_1P_r is the braking force.

Summarizing, we find that the car can be modelled by

$$m\frac{dv}{dt} = \alpha_n u_1 T(\alpha_n v) - C_4 P_r - mgC_r sgn(v) - \frac{1}{2}\rho C_d A v^2 - mgsin\theta \qquad (7)$$

Throttle angle command and braking command may be considered as the two control input variable available for longitudinal control, and one output that is vehicle speed. The other inputs like aerodynamic drag, road condition, vehicle mass changes may be treated as disturbances. Two inputs are separately considered, one output system into a throttle angle to speed and braking command to speed subsystems. In the design we are taking the assumption that the throttle and brake controllers are not allowed to act simultaneously. The throttle angle command is made zero whenever the braking command is sent to the brake actuator.

So the dynamic equations of vehicle following are:

$$\dot{v} = \begin{cases} \frac{1}{m}\left(\alpha_n u_1 T(\alpha_n v) - mgC_r sgn(v) - \frac{1}{2}\rho C_d A v^2 - mgsin\theta\right) & for\ throttle \\ \frac{1}{m}\left(-C_4 P_r - mgC_r sgn(v) - \frac{1}{2}\rho C_d A v^2 - mgsin\theta\right) & for\ brake \end{cases} \qquad (8)$$

3. LOWER LEVEL CONTROLLER FOR ACC

Brake and Throttle systems are separately controlled using a feedback and a PID controller respectively. A switching logic is designed to switch between the controllers effectively. Throttle control and brake control designed for lower level control are tested using computer simulations. Controllers are separately tested for tracking preceding vehicle speed and different acceleration. Finally lower level controller with proper switching between brake and throttle controller are tested.

3.1 Control Objective

Let v_l and v denote the vehice speed of leading and following vehicles respectively, dynamic equations of vehicle following are:

$$\dot{x}_l = v_l$$
$$\dot{x} = v$$
$$\dot{v} = \begin{cases} \frac{1}{m}\left(\alpha_n u_1 T(\alpha_n v) - mgC_r sgn(v) - \frac{1}{2}\rho C_d A v^2 - mgsin\theta\right) & for\ throttle \\ \frac{1}{m}\left(-C_4 P_r - mgC_r sgn(v) - \frac{1}{2}\rho C_d A v^2 - mgsin\theta\right) & for\ brake \end{cases} \qquad (9)$$

One control objectives is to make the speed v = v_l. . That is the v_l the desired speed trajectory is to be tracked by the following vehicle. Defining relative speed $v_r = v_l - v$ and relative distance $x_r = x_l - x$. Another control objective is to keep a desired inter-vehicle spacing S_d, measured from the front of the following vehicle to the rear of the leading vehicle. In our design, the desired inter vehicle spacing S_d is chosen as

$$S_d = hV_f - S_o \qquad (10)$$

Where h is known as the time headway, and $S_o > 0$ is a constant. This policy for the desired inter-vehicle spacing is called constant time headway policy. h can be chosen based on the performance characteristics of the leading and following vehicles in a worst stopping scenario, the accuracy of sensors, etc. If we use δ to denote the deviation of the relative distance xr from the desired spacing S_d, then we have

$$\delta = x_r - hv - S_o \qquad (11)$$

The overall control objective for vehicle following is to choose the throttle angle and braking torque so that δ→0 and v→ v_l [3].

3.2 Throttle Control Design

The dynamic equations of the throttle subsystem are

$$\dot{x}_r = v_l - v$$
$$\dot{v} = -a(v - v_e) + b(u_1 - u_e) - c\theta$$
$$\delta = x_r - hv - S_o$$
$$v_r = v_l - v \qquad (12)$$

Where (u1-ue) is chosen so that vr and δ→0 as t→α. A PID controller is used to generate the control signal u1.

$$u_1 = u_e + k_1 v_r + k_2 \delta + \int_0^t (k_3 v_r + k_4 \delta) d\tau \qquad (13)$$

This control equation includes the term ue which correspond to the equilibrium control value followed by a proportional term $k_2\delta$ a “derivative” term k_1V_r and an integral term. The gain constants k_1 to k_4 are to meet the control objective stated before. The use of $k_3V_r + k_4\delta$ in the integral action reduces the speed overshoot due to large position error.

3.3 Brake Control Design

When the leading vehicle is decelerating fast or during downhill vehicle following situations, engine torque alone may not be sufficient for vehicle following without using the brake. In this section we derive using the assume that the following vehicle has to be apply brake and we design a controller to control the brake line pressure.

$$\dot{x}_r = v_l - v$$
$$\dot{v} = \frac{1}{m}\left(-C_4 P_r - mgC_r sgn(v) - \frac{1}{2}\rho C_d A v^2 - mgsin\theta\right)$$
$$\delta = x_r - hv - S_o$$
$$v_r = v_l - v \qquad (14)$$

The equation can be rewritten as

$$\dot{v} = \frac{1}{m}(-c_4 P_r - c_1 - c_2 v - c_3 v^2) \qquad (15)$$

Where c1 to c3 corresponds to the coefficients associated with each term. We use feedback linearization method to transform the nonlinear system (15) into a linear system. By applying a control input Pr of the form (16) will linearize the nonlinear equation (15)

$$P_r = \frac{-1}{c_4}[mu_2 + c_1 + c_2 v + c_3 v^2] \qquad (16)$$

Equation (15) get linearized into the form $\dot{v} = u_2$. We can now use techniques from linear system theory to choose the desired input u_2 that stabilizes the system given by $\dot{v} = u_2$ and forces v_r to converge to zero and δ to the minimum allowed distance. From the desired input u_2 we calculate the desired line pressure P_r, by solving for P_r in (16). A feedback control is proposed, where k_5 and k_6 are design parameters.

$$u_2 = k_5 v_r + k_6 \delta$$
$$P_r = \frac{-1}{c_4}\left[m(k_5 v_r + k_6 \delta) + f_o + c_2 v + c_3 v^2\right]$$
$$\dot{v} = k_5 v_r + k_6 \delta \qquad (17)$$

3.4 Logic Switching For Throttle and Brake Controller

For the lower level controller brake and throttle controllers are not allowed to act together at any point of action, so we need an appropriate logic that does this switching procedure from the throttle controller to the brake controller and vice versa. The logic is developed on the assumption that that a good driver will not use the brake and accelerator pedals together, he will use brake only when he need a rapid and large deceleration. Under normal condition he will control the drive using the throttle pedal alone so as to avoid frequent switching from one pedal to another. The switching logic is based on the following situations:

- S_1 when $x_r < x_{min}$ and $v > v_l$

This situation arises when the following vehicle is at high speed (i.e. $v > v_1$) and relatively close to the leading vehicle (i.e. x_r, $< x_{min}$). The constants x_{min}, v_1 are design variables that are chosen according to the braking capabilities of the vehicle. In our design x_{min} = 6 meters and V_1 = 13.4 m/s. When this situation arises, the brake controller is switched on.

- S_2 when $x_r > x_{max}$

In this situation the relative distance is large enough and no braking is necessary. Therefore the brake controller is off independent of any other condition and the throttle control is made on. In our case the design constant x_{max} is chosen to be equal to 40 meters.

- S_3 when $x_{min} < x_r < x_{max}$ and $v < v_l$

Then no action is taken and the brake controller keeps its previous status

4. SIMULATION AND ANALYSIS

The dynamical equation (8) characterizing the behaviour of an adaptive cruise vehicle is validated by taking computer simulations. The physical parameters of the ACC were taken from [4] and are given in Table 1.

Table -1: Parameters of Adaptive cruise vehicle

Parameter	Meaning	Values
T_m	Maximum engine torque	190Nm
ω_m	Maximum engine speed	420rad/sec
M	Mass of vehicle	1000kg
C_r	Coefficient of rolling friction	0.01
P	Air density	1.3kg/m^3
Cd	Coefficient of aerodynamic drag	0.32
A	Frontal area	2.4m^2
Θ	Slope of the road	4^o
H	Time headway	3sec
S_o	Safety distance b/w vehicle	6m

4.1 Throttle Control

The dynamical equations (12) for the throttle subsystem are used to satisfy the control objectives. The leading vehicle is considered to be initially at rest and then accelerate to 10km\hr then to 20km\hr and then to 25km\hr at different acceleration. This is considered as the reference speed. The cruise vehicle is to move a speed such the relative speed between the vehicles is zero and the deviation between the relative distance and the desired spacing is zero. The control signal (13) keeps the vehicle at the desired speed. Fig 3 and Fig 4 gives the plot for vehicle speed, relative speed and deviation in spacing.

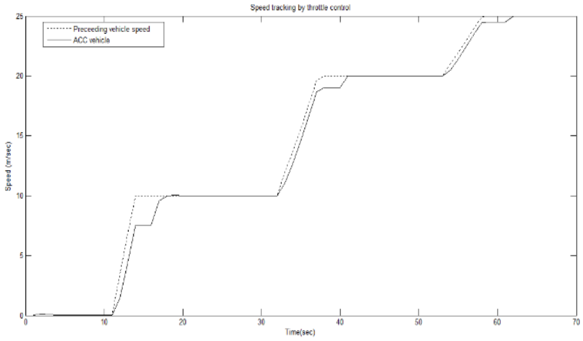

Fig 3 Vehicle speed plot using the throttle control

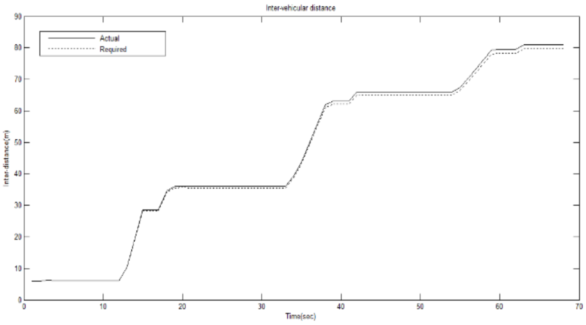

Fig 4 Inter- vehicular distance plot using throttle control

4.2 Brake Control

The dynamical equations (15) for the brake subsystem are used to satisfy the control objectives. The leading vehicle is initially at 20km\hr then decelerate to 15km/hr then suddenly made to stop so following cruise vehicle need to stop suddenly with the minimum required spacing in between.

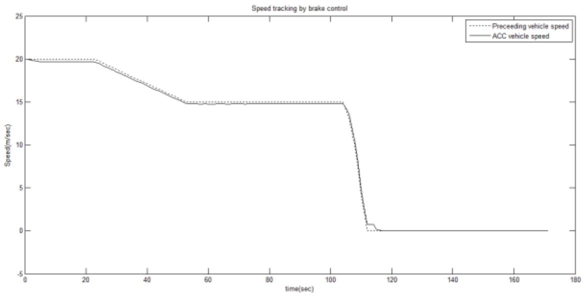

Fig 5 Vehicle speed plot using the brake control

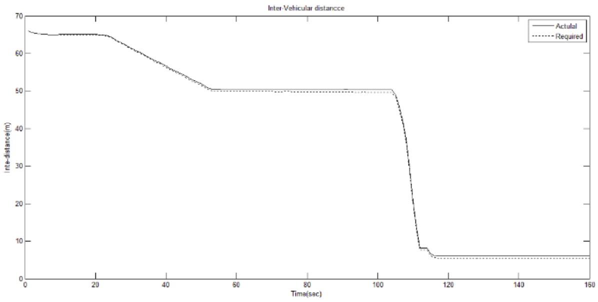

Fig 6 Inter- vehicular distance plot using brake control

The stopping time should be very small to avoid rear-end collision. The control signal (17) keeps the vehicle at the desired speed. The value of the gain parameters k5 and k6 are obtained by trial and error. Fig 5 and fig 6 gives the plot for vehicle speed and inter-vehicular distance.

4.3 Vehicle Speed Controls with Switching Between Throttle and Brake Control

The vehicle control need to switch between the throttle and brake depending up on the switching rules designed. The lead vehicle speed is given in fig 7 dotted line, the vehicle is initially at a speed 25km/hr which decelerates suddenly to zero and then again accelerates. This is taken as the reference the reference accelerates and decelerates at different rates, so proper switching needs to takes place for efficient control of the vehicle. Fig 7 shows the speed tracking and Fig 8 gives the distance tracking done by the cruise vehicle.

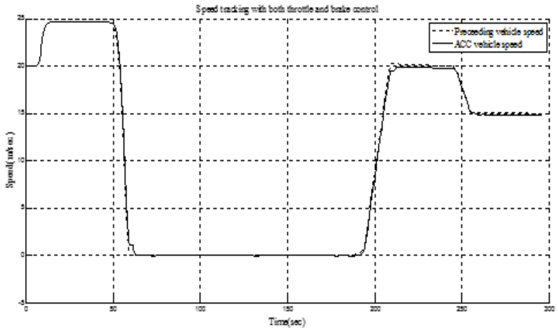

Fig 7 Vehicle speed tracking using both throttle and brake control

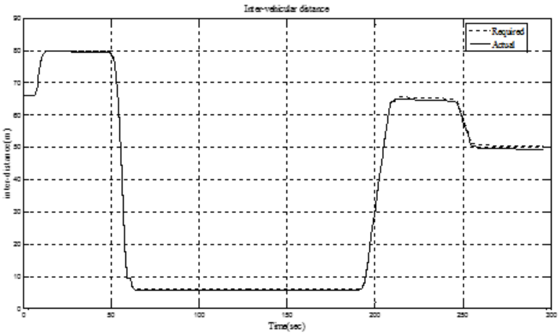

Fig 8 Inter vehicular distance with both brake and throttle control

5. CONCLUSIONS

A hierarchical control architecture was used in the paper for the vehicle design. Lower level controller is incorporating the basic vehicle dynamics separating the brake and throttle effect separately. Throttle is controlled using a PID controller and brake is controlled using a feedback controller. A switching logic is used to switch between the throttle and break control action. Matlab simulations showed good vehicle tracking results for different road scenarios using the controllers. By using a proper control algorithm for upper level controller adaptive cruise control can be easily achieved.

REFERENCES

[1] V. Bageshwar, W. Garrard, and R. Rajamani, "Model predictive control of transitional manoeuvres for adaptive cruise control vehicles," IEEE Trans. Veh. Technol., vol. 53, no. 5, pp. 365–374, Sep. 2004.

[2] S. Li, K. Li, R. Rajamani, and J. Wang, "Model predictive multi-objective vehicular adaptive cruise control" IEEE Trans. Control Syst. Technol., vol. 19, no. 3, may 2011.

[3] P. Ioannou and T. Xu, "Throttle and brake control systems for automatic vehicle following," IVHS J., vol. 1, no. 4, pp. 345–377, 1994.

[4] J. Jonsson, "Fuel optimized predictive following in lower speed conditions," M.S. thesis, Dept. Elect. Eng., Linköping Univ., Linköping, Sweden, 2003.

[5] J. Zhang and P. Ioannou, "Longitudinal Control of Heavy Trucks: Environmental and Fuel Economy Considerations" IEEE Intel Trans Syst Conf., Oct. 2004.

[6] P. Ioannou and M. Stefanovic, "Evaluation of the ACC vehicles in mixed traffic: Lane change effects and sensitivity analysis," IEEE Trans. Intel. Transport. Syst., vol. 6, no. 1, pp. 79–90, Jan. 2005.

[7] K. Yi, I. Moon and Young Do Kwon, "A Vehicle-to-Vehicle Distance Control Algorithm for Stop-and-Go Cruise Control," IEEE Conf. Intelligent Transportation Syst., Oakland (CA), USA - Aug 2001.