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Control-Oriented Models for vehicle longitudinal motion

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Abstract: The use of mathematical models is widespread both in simulating the dynamic behavior of vehicle longitudinal motion and in designing related controllers. This paper focuses on control-oriented models for longitudinal motion which better captured the plant dynamics for vehicles with internal combustion engines. Firstly, a review of some simplified models is presented and secondly, two more complex control-oriented models which take into account the powertrain dynamics are proposed.

Keywords:vehicle longitudinal motion, control-oriented model, mean value model, discrete-event model, internal combustion engine, powertrain.

1. Introduction

The dynamics vehicle models play a vital role in research and development of vehicles from bicycles to heavy road vehicles [1]. In the development phase, vehicle models serve as virtual prototypes, giving to developers a perspective on the influence of design parameters on vehicle behavior. Existing the possibility to simulate models, the developers can test and optimize their designs, control functions and several other functions. The vehicle models can be used in a series of applications as driving simulators, lap-time simulators, controller design for automated driving or product development. Depending on a certain application, these models can be represented as a point mass model or can be extended to multibody or even as inverse models with several hundred variables.
Control-oriented models for longitudinal motion of the vehicles are the foundation for advanced vehicle-control systems (AVCS) in the context of Intelligent Transportation Systems (ITS). AVCS is one of the four major areas of ITS [2] being dedicated to improve vehicle control by assisting and augmenting driver performance. One of the specific technological elements of AVCS is automatic longitudinal control which underlies the cruise control (CC), adaptive cruise control

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(ACC), cooperative adaptive cruise control (CACC) and thus, the vehicle platooning became possible. The first automatic longitudinal control was CC, which keeps a selected speed without the driver's involvement and then, by adding forward-looking sensors in order to detect and control headway to a lead vehicle, ACC system appeared [3]. The next step was CACC concept based on a combination of automated speed control with a cooperative element, such as Vehicle-to-Vehicle (V2V) and/or Infrastructure-to-Vehicle (I2V) communication [4], [5]. The vehicle platooning uses radar sensor to measure the distances between vehicles and V2V communications to form and maintain a close-headway formation between at least two in-lane vehicles, controlling both longitudinally and laterally vehicles speeds.

As vehicles have become more sophisticated, an intuitive development has become very expensive and also risky business. In an effort to predict and quantify the effects of the vehicle parameters changes, designers are increasingly turning to computer simulation techniques to evaluate design proposals [6]. The computer simulations can be useful if the software accurately reflects the behavior of the actual vehicle. If the previous statement is fulfilled, then considerable savings in time and costs can be obtained.

The longitudinal vehicle models can be also validated using experimental setups with real cars that consist in the following hardware: inertial measurement unit (IMU), CAN bus, Corrsys sensors, GPS, dynamometer and pedal robot as in [7]. This type of setup provides a very good opportunity for vehicle parameters identification. The proposed testbench can simulate real vehicle movements for a better modeling of the longitudinal vehicle model. Another real car-based setup that has as the main purpose the development of control systems for autonomous vehicles is proposed in [8]. This method implies a test vehicle that is freely moving on a test track controlled by a pneumatic actuator that controls the brakes and uses different on-board systems for the data acquisition. During the experiments, the influence of external environment factors is of special interest. These factors could be: wind loading, gradients, varying friction and other dynamic effects. In [9], a hardware-in-the-loop (HIL) setup for simulating a vehicle behavior using a scaled experimental testbed is described. This platform was used to validate decision and control algorithms for ITS applications. The built vehicles are equipped with wireless communication systems, GPS, and with on-board computers dedicated for solving decision, control and communication tasks. In order to create a driving simulator, the author of [10] developed a new vehicle dynamics model with ten degrees of freedom. The driving simulators have the role to help in research areas that concerns to human factors and the development of ADAS systems. The proposed model is intended to compute the motion of a passenger vehicle when driving in normal conditions, but it can also include a realistic and predictable behavior for some severe driving conditions such as collision avoidance maneuvers.

For all automatic vehicle longitudinal control applications, the accuracy of the longitudinal motion modeling is an important factor in obtaining the desired performances by the control systems. Usually, simplified models are used that lead to PID-type control algorithms. A review of some simplified models, often presented in the literature, e.g. [2], [11] or [12], is given in Section 2.

In this paper two control-oriented models for longitudinal motion control are also presented, which included the powertrain dynamics. Usually, these types of models are complex and are used for simulating the behavior of a vehicle in longitudinal motion and less frequently to design control laws [13]. The first proposed control-oriented model is a mean value one, which neglects the discrete cycles of the engine. The reciprocating behavior of the engine is captured by introducing in torque equation the *induction-to-power-stroke* (IPS) delay. The second model is a discrete-event model, more suitable to accurately capture the vehicle longitudinal motion dynamics. The two proposed control-oriented models were implemented in Matlab/Simulink and their dynamic behaviours have been comparatively tested in simulation.

2. Longitudinal motion simplified models – a short review

The system that models the vehicle longitudinal motion is represented in Fig.1 and is made up of:

- accelerator pedal which controls the throttle through the control u
- engine which produces the torque T_e related with the throttle position α
- vehicle driveline which transmits the engine torque T_e to the ground through traction force F_t generated by the driven wheels in contact with the road
- longitudinal force disturbance F_d due to the air/rolling resistance and the road slope θ
- car body of mass m having the speed v

The input of the longitudinal vehicle motion system is the control u generated by the accelerator pedal operation or a speed controller and the output is the vehicle speed and thus, it is possible to regulate the vehicle longitudinal speed by throttle adjustments.

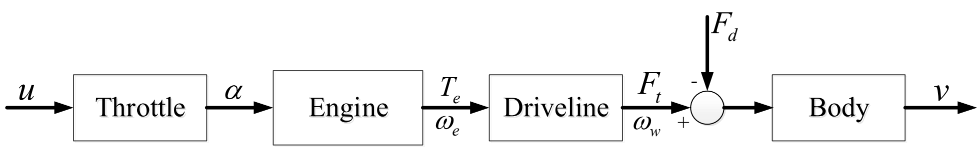

Fig.1 Vehicle longitudinal motion system

2.1. Simplified Models

The simplified model of the longitudinal dynamics of a vehicle is depicted in Fig. 2. The longitudinal movement of the vehicle is described using the force balance of the car body expressed by the Newton's second law for *x* direction:

$$m\frac{dv}{dt} = F_t - F_d \tag{1}$$

where m is the vehicle mass, v is the vehicle speed, F_t is the traction force generated by the contact of the wheels with the road and F_d is the disturbance force.

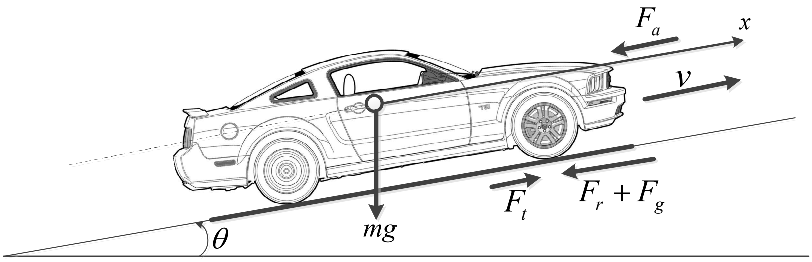

Fig. 2 Forces acting on the longitudinal axis of a vehicle

The force disturbance $F_d = F_g + F_r + F_a$ which influences the longitudinal motion of the vehicle has the components [7]:

- longitudinal component of the gravitational force: $F_g = mg\sin\theta$
- the rolling-resistance force: $F_r = C_r mg\,\mathrm{sgn}(v)$ (C_r – rolling resistance coefficient)
- aerodynamic-drag force: $F_a = 0.5\rho C_d A(v + v_{wd})^2$ (ρ – air density, A - vehicle cross-sectional area, C_d – drag coefficient, v_{wd} – wind speed)

The input of the model (1) is the traction force F_t and the output is the vehicle speed v. Despite being a simple model, equation (1) is used in [2] to determine changes in the vehicle-forward motion due to slopes, braking or acceleration. But the equation (1) is nonlinear and for this reason, it is more difficult to use conventional control algorithms.

To eliminate this drawback, the model (1) is linearized around an operating point characterized by the variables: $v = v_0$, $\theta = \theta_0$ and $v_w = v_{wo}$, resulting the simple one state model:

$$\dot{x} = -\frac{1}{\tau_v} x + \frac{K_v}{\tau_v} u + w \tag{2}$$

where $x = \Delta v,\ u = \Delta F_t$ and:

$\tau_v = m / \left(\rho C_d A\left(v_0 + v_{w0}\right)\right) \Rightarrow$ vehicle time constant

$K_v = 1 / \left(\rho C_d A\left(v_0 + v_{w0}\right)\right) \Rightarrow$ vehicle gain

$w = mg\left(f \sin\theta_0 - \cos\theta_0\right)\Delta\theta \Rightarrow$ disturbance

The linear model (2) does not consider the throttle, the engine and the driveline blocks from the vehicle longitudinal motion system (Fig. 1). In [14] and [2], the missing blocks are modeled taking into account a DC throttle actuator with a static gain that generates traction force output, resulting the transfer function:

$$G_a(s) = \frac{K_a}{s(\tau_a s + 1)} = \frac{F_t(s)}{U(s)} \tag{3}$$

where the input is the motor-drive duty cycle (percent). Computing the transfer function of the car using (2) and considering the actuator model (3), the simple model of the longitudinal motion is obtained:

$$G_l(s) = \frac{K_a K_v}{s(\tau_a s + 1)(\tau_v s + 1)} = \frac{V(s)}{U(s)} \tag{4}$$

Considering the longitudinal model (2) for CC, PI controllers can be designed with satisfactory performances and for the model (4), speed PID controllers are used in [2] with good behaviours of CC system.

2.2. Nonlinear model comprised the transmission system

The traction force is produced by the torque of the internal combustion engine, which is proportional with the rate of the fuel injection which in turn depends on the throttle position generated by the accelerator pedal or a speed controller through the control u. The engine torque also depends on the engine angular velocity ω_e and can be described considering the throttle fully opened, by the torque curve [7]:

$$T_e(\omega_e) = T_{em}\left(1 - \beta\left(\frac{\omega_e}{\omega_{em}} - 1\right)^2\right) \tag{5}$$

where T_{em} is the maximum torque obtained for ω_{em}. The torque curve is smoothed by the transmission system composed of the gearbox with the ratio i_t and the final drive with the ratio i_f. Thus, taking into account the ratio $i = i_t i_f$ of the transmission system of the driveline, the wheel angular velocity ω_w and the wheel torque T_w can be computed with:

$$\omega_w = \frac{1}{i}\omega_e; \quad T_w = iT_e \tag{6}$$

At the same time, if r is the wheel radius, and considering the linear speed of the wheel centre v_w is equal to the vehicle (body) speed v, for the wheel angular velocity results the expression:

$$\omega_w = v / r \tag{7}$$

and the traction force becomes:

$$F_t = \frac{i}{r} T_e(\omega_e)u \tag{8}$$

Taking into account the force balance of the car body (1), the following nonlinear longitudinal motion model results:

$$m\frac{dv}{dt} = \frac{i}{r} T_e(\omega_e)u - (mg\sin\theta + C_r mg\,\mathrm{sgn}(v) + 0.5\rho A C_d (v + v_{wd})^2) \tag{9}$$

The vehicle speed v is the state and the output, the input is the signal u that controls the throttle position and the disturbance is the longitudinal force disturbance F_d, which depends on the slope of the road θ. The model is nonlinear because of the torque curve, the aerodynamic-drag force, the rolling resistance force and the longitudinal component of the gravitational force. The simplified nonlinear model (9) takes into account the engine model through the engine torque curve (5) and the static model (6) for driveline that only considers the ratio i_t of the gearbox and the ratio i_f of the final drive.

3. Control-oriented model including powertrain dynamics

The wheel angular speed ω_w is highly influenced by the powertrain dynamics. For this reason, two realistic models of the vehicle longitudinal motion system which include the powertrain dynamics are proposed below. A vehicular powertrain consisting of engine and driveline is depicted in Fig. 3. The main parts of the driveline are the clutch, the transmission (gearbox and final drive), the shafts (propeller shaft and drive shafts) and the wheels.

3.1. Basic driveline model

The traction force on the driving wheel from (1) depends on the difference between the linear velocity of the wheel centre $v_w = r\omega_w$ and the vehicle (body) speed v which generates the longitudinal slip λ.
The behavior of the driveline dynamics is obtained for the acceleration/deceleration modes resulting a complex model, like the one in [13], but unsuitable for control. A control-oriented model is achieved neglecting the tire sliding ($\lambda = 0$).This implies a basic complete model of the driveline with the clutch engaged that can be developed considering the following assumptions [15]:

- $v_w = v$, the same supposition as for the model (9).
- the clutch and shafts are stiff
- the gearbox and the differential multiply the torque by the conversion ratio, without losses
- in the wheel model, the dynamical influence from the tire is neglected (the longitudinal tire slip is negligible)
- the inertia and the damping losses of the gearbox and the final drive are neglected
- the transmission is in the steady state (i.e. it is not undergoing a gear shift)

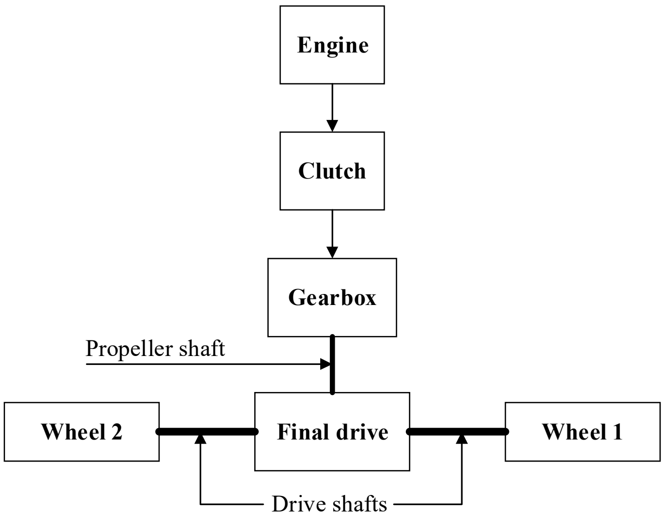

Fig.3 Powertrain consisting of engine and driveline

The flow of power and loads on the driveline components is given in Fig.3. It assumes that the angular speeds of the two wheels are equal and thus, the two drive shafts between the final drive and the two wheels are modeled as one shaft.
In order to obtain the driveline dynamics, first it is considered the engine dynamics given by the Newton's second law of motion:

$$J_e \frac{d\omega_e}{dt} = T_e - T_{load} \tag{10}$$

where J_e is the engine mass moment of inertia and T_{load} is the engine load torque that consists of the load applied by the clutch, and its load from the driven accessories (e.g., air-conditioning compressor or alternator). The load torque can be achieved by calculating the load on the engine from the wheels and the transmission. The load applied by the clutch is generated by the wheel torque T_w divided by the transmission ratio i. Taking into account the wheel dynamics given by the model:

$$J_w \frac{d\omega_w}{dt} = T_w - rF_t \tag{11}$$

where J_w is the wheel mass moment of inertia, the expression of the wheel torque required to produce the longitudinal speed v can be obtained using the longitudinal vehicle model (1), the wheel model (11) and considering $\omega_w = v / r$:

$$T_w = (J_w / r + rm)\frac{dv}{dt} + rF_d \tag{12}$$

From the torque flow on the driveline, the load on the engine results:

$$T_{load} = T_c = T_t = T_w / i = \frac{1}{i}(J_w / r + rm)\frac{dv}{dt} + \frac{r}{i}F_d \tag{13}$$

where T_c is the clutch torque and T_t is the transmission torque.
Including (13) in (10) and taking into account that:

$$\omega_e = i\omega_w = i\frac{v}{r} \tag{14}$$

together with the components of the longitudinal force disturbance, the following basic driveline model is achieved:

$$\begin{gathered}\left(J_w + mr^2 + i^2 J_e\right)\frac{dv}{dt} = riT_e - \\ r^2\left(0.5\rho A C_d (v + v_{wd})^2 + C_r mg\,\mathrm{sgn}(v) + mg\sin\theta\right)\end{gathered} \tag{15}$$

The driveline dynamics are described by the nonlinear model (15) that has one state, the longitudinal speed v, which is at the same time the output, and as input the engine torque T_e produced by combustion.

3.2. Torque generation for internal combustion engine

The engine torque T_e, which also contains the torque frictional losses depends on the engine dynamics and on the position of the accelerator pedal that generates the input signal u for the throttle. Control-oriented models used to describe the engine dynamics are divided into two main categories [16]: mean value models (continuous models which neglect the discrete cycles of the engine) and discrete event models (hybrid models that explicitly take into account the reciprocating behavior of the engine). The only physical link of the engine to the rest of the powertrain is the load torque.

A simple linearized model of engine dynamics considering a steady operating point is taken into account to obtain the engine torque. The symbol Δ was used to represent incremental variables that give changes about the steady values.

The mean shaft torque generated during the combustion process depends on the following variables of the engine subsystems: mass flow of air out of the manifold due to engine pumping $\dot{M}$, spark timing/advance δ and engine speed n, leading to the following nonlinear torque expression:

$$T_e = g(\dot{M}(t - \tau_{IPS}), \dot{m}_f(t - \tau_{IPS}), \delta, n) \tag{16}$$

The torque produced by the engine does not respond immediately to a throttle position change, but after the IPS delay [16]:

$$\tau_{IPS} \approx 2\pi / \omega_e \tag{17}$$

Using the assumption that the mass charge is a function of the manifold pressure p and the engine speed n, a linearized relationship of (16) is given in [2]:

$$\Delta T_e = K_p \Delta p(t - \tau_{IPS}) + K_f \Delta \dot{m}_f(t - \tau_{IPS}) + K_\delta \Delta \delta - F_n \Delta n \tag{18}$$

where K_p, K_f and K_δ are the influences of delayed pressure, delayed fuel and spark advance on torque and F_n is the engine friction. The first three terms from (18) define the so-called combustion torque with the components generated by the intake manifold, fuel system and spark command. The torque friction is considered in the sequel included in the engine torque.

Considering the linearized throttle model from [2]:

$$\Delta \dot{m}_a = K_\alpha \Delta \alpha \tag{19}$$

it results that the air mass inflow rate $\dot{m}_a$ is proportional to the change in the throttle angle. For manifold-filling dynamics, the Ideal Gas Law is used:

$$\dot{p} = \frac{RT}{V}(\dot{m}_a - \dot{M}) \tag{20}$$

where R is the ideal-gas constant, T is the air temperature and V is the manifold volume. After linearization of the equation (20), the following expression is obtained taking into account (19) and that the engine pumping mass flow rate $\dot{M}$ is a function of the manifold pressure p and the engine angular speed ω_e:

$$\Delta\dot{p} = \frac{-1}{\tau_m}\Delta p + K_m K_\alpha \Delta\alpha - K_m K_\omega \Delta\omega_e \tag{21}$$

where the expressions of the parameters K_m, τ_m and K_ω are given in [2].
The engine fuel-flow rate $\dot{m}_f$ is computed in [2] based on the linearization of an estimation of the mass flow rate, resulting:

$$\Delta\dot{m}_f = c(\omega_{e0}\Delta p + p_0\Delta\omega_e) \tag{22}$$

where c is a constant and ω_{e0} and p_0 are the engine angular speed and, respectively, the manifold pressure computed in the steady operating point where the linearization was done.
Equation (18), together with equations (21) and (22), lead to a control-oriented engine model having as input the change in the throttle angle $\Delta\alpha$ and as output the engine torque ΔT_e.

3.3. Models for the vehicle longitudinal motion

The engine is equipped with a throttle actuator, i.e., the control u from the driver or a speed controller does not directly command the angle change $\Delta\alpha$ of the throttle plate. Usually, the control u is the reference of a position system with a DC motor that has as controlled output the angle change $\Delta\alpha$. This position control system is frequently modeled by the following first order system [16]:

$$\Delta\dot{\alpha} = -\frac{1}{\tau_t}\Delta\dot{\alpha} + \frac{1}{\tau_t}u \tag{23}$$

where τ_t is the throttle time constant.
Taking into account (14), the equation (21) becomes:

$$\Delta\dot{p} = \frac{-1}{\tau_m}\Delta p + K_m K_\alpha \Delta\alpha - K_m K_\omega \frac{i}{r}\Delta v \tag{24}$$

For the driveline model (15), the effective inertia reflected on the engine side is noted with:

$$J = J_w + mr^2 + i^2 J_e \tag{25}$$

Further, combining the engine torque model (18), where the term generated by spark advance was replaced with $K_s \Delta\omega_e$ and the IPS delay was neglected as in [2], with the driveline model (15) and considering (14), it results:

$$\Delta\dot{v} = \frac{ri}{J}(K_p + K_f c\omega_{e0})\Delta p + \frac{i^2}{J}(K_f c p_0 + K_s)\Delta v - \frac{r^2}{J}F_d \tag{26}$$

Starting from the structure of the longitudinal motion system from Fig. 1, a new three state control-oriented model was created based on equations (23), (24) and (26). The mean value model developed is a continuous control-oriented model, which neglects the discrete cycles of the engine. The reciprocating behavior of the engine is captured by introducing in torque equation (18) the IPS delay. Taking into account the following states: the throttle angle change $\Delta\alpha$, the manifold pressure change Δp and the vehicle (body) speed change Δv, the mean value control-oriented model becomes:

$$\begin{cases} \dot{x} = Ax + bu + b_d F_d \\ y = \begin{bmatrix} 0 & 0 & 1 \end{bmatrix} x \end{cases}$$
$$x = \begin{bmatrix} \Delta\alpha \\ \Delta p \\ \Delta v \end{bmatrix};\ b = \begin{bmatrix} 1/\tau_t \\ 0 \\ 0 \end{bmatrix};\ b_d = \begin{bmatrix} 0 \\ 0 \\ -r^2/J \end{bmatrix} \tag{27}$$
$$A = \begin{bmatrix} -1/\tau_t & 0 & 0 \\ K_m K_\alpha & -1/\tau_m & -K_m K_{\omega_e} i/r \\ 0 & \frac{ri}{J}(K_p + K_f c\omega_{e0}) & \frac{i^2}{J}(K_f c p_0 + K_s) \end{bmatrix}$$

The real vehicle longitudinal motion model includes both continuous-time and discrete event subsystems. Among the discrete event subsystems is the torque production. The speed controller will be implemented using an Electronic Control Unit that will be described using a discrete-time approach. The torque production acts as a discrete system having the sampling period for a four-stroke engine:

$$T_s = \frac{4\pi}{\omega_e N_{cyl}} \tag{28}$$

where N_{cyl} is the number of cylinders of the engine. For $N_{cyl} = 4$, it immediately results that $\tau_{IPS} = 2T_s$.
Therefore, a discrete-event model is more suitable to capture more accurately the vehicle longitudinal motion dynamics.
Based on equation (24) and applying Laplace transform, the continuous model of the manifold is obtained:

$$\Delta P(s) = \frac{\tau_m K_m}{s\tau_m + 1}\left(K_\alpha \Delta\alpha(s) - \frac{i}{r} K_\omega \Delta V(s) \right) \tag{29}$$

The discrete event subsystem that generates the torque is based on the manifold, fuel and advance spark subsystem models. From equation (18), it results:

$$\begin{aligned} \Delta T_e(s) = K_p \Delta P(s) e^{-\tau_{IPS} s} \oplus^{T_s} K_f c(\omega_{e0} \Delta P(s) + \frac{p_0 i}{r} \Delta V(s)) e^{-\tau_{IPS} s} \\ \oplus^{T_s} \frac{K_s i}{r} \Delta V(s) \end{aligned} \tag{30}$$

where the sum $\oplus^{T_s}$ of the three terms is computed considering the sampling period T_s.
The output of the longitudinal motion model is obtained based on the equation (15) and the notation (25), resulting:

$$\Delta V(s) = \frac{1}{Js}\left(ri\Delta T_e - r^2 F_d \right) \tag{31}$$

Finally, the throttle actuator transfer function is computed from (23), yielding:

$$\Delta\alpha(s) = \frac{1}{\tau_t s + 1} U(s) \tag{32}$$

Thus, using the equations (29) - (32), the discrete event model of the vehicle longitudinal motion is achieved.

4. Simulation results

The two developed mean value and discrete event models for vehicle longitudinal motion were implemented in Matlab/Simulink and their dynamic behaviours were comparatively analyzed in simulation. The models can use an external disturbance force which influences the movement of the vehicle.The parameters of the two

models were taken from [2] and [15] and the parameters of the disturbance force from [7].
The dynamic behavior of the two models was tested considering the application of a step signal on the input *u* at *t* = 1s and a perturbation generated by the variation of the road slope at *t* = 8s. The responses of the two models are represented in Fig. 4.

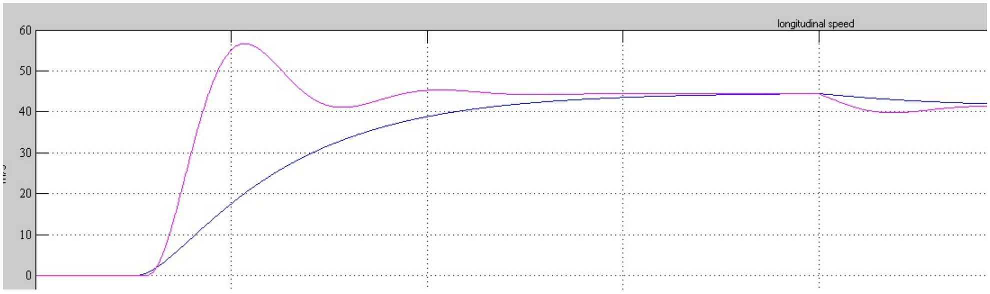

Fig.4 Longitudinal speeds for mean value and discrete event models

By neglecting IPS delay and ignoring the reciprocating behavior of the engine, the mean value model response is not oscillating both to the variation of the input signal and to the occurrence of the disturbance due to the slope of the road. However, by introducing the powertrain dynamics and by considering the IPS delay, the behavior of the two models should be an oscillating one. This oscillating behavior is faithfully respected by the response of the discrete event model. At the same time, it should be noted that the responses of the two models have the same speed steady values, both for the variation of the input signal and the occurrence of the perturbation. As Fig. 4 shows, the response of the discrete event model better reflects the plant dynamics by considering that the torque generated by the internal combustion engine is a discrete process and taking into account the IPS delay.

5. Conclusions

This paper was inspired by the need to enhance the control-oriented models for vehicle longitudinal motion. The proposed models take into account the powertrain dynamics so as to be suitable both for control and also to accurately reflect the dynamic behavior of the vehicle longitudinal motion. The first model was developed based on mean value modeling approach and the second on discrete event modeling. The two models were implemented and comparatively tested in simulation. The simulation results have proved the effectiveness of the proposed control-oriented models for vehicle longitudinal motion.

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