Modeling of Heavy-Duty Trucks
for Longitudinal Platooning

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Abstract
This paper considers the modeling of longitudinal dynamics of Heavy-Duty-Vehicles (HDV) for automated vehicle short-inter-vehicle distance platooning. This is a challenging problem compared to passenger cars because a HDV is mass dominant, low power/mass ratio, large actuator delay and actuator saturation. The main purpose is to obtain a nonlinear model which is as simple as control design method can handle and as complicated as to capture the intrinsic vehicle dynamics. This includes extensive simplification and decoupling of sensor and actuator dynamics. After these procedures, the second order nonlinear vehicle body dynamics is feedback linearizable. Beside the vehicle body dynamics, other main dynamical components along the drive-train are turbocharged diesel engine, torque converter, transmission, transmission retarder, pneumatic brake and tire. A proper model for each component is either established or adopted from previous work.

1 INTRODUCTION

With successful demonstration of longitudinal and lateral control of automated passenger cars in automated highway system (AHS), development of similar techniques for Heavy-Duty Vehicles (HDV) arises. It is believed that automatic control of HDV can greatly reduce the drivers’ workload, relieve from fatigue and to avoid accidents. As reported in [16], eighty percent of the victims killed in crashes involving trucks are occupants of smaller vehicles. Besides, HDV platooning can reduce aerodynamic dragging force of follower vehicles by reducing inter-vehicle distance and thus can achieve better fuel economy [4]. However, to achieve this purpose, one has to face some challenging control problems, especially for longitudinal control, which are quite different from those for passenger cars. These differences mainly come from dynamic properties of the vehicles. This paper is to consider modeling of HDV equipped with turbocharged diesel engine for longitudinal control, particularly for short inter-vehicle distance platooning.

Contrasted to passenger cars, longitudinal control of a HDV has the following characteristics:

(a) Mass dominant: Any minor change (such as road grade or acceleration demand) of a mass related factor causes large variation in torque demand. This, in turn, causes high sensitivity to the model for real-time mass estimation;

(b) Low power/mass ratio: It is easily to cause actuator saturation which eventually leads to the loss of controllability. This ends up with the situation that there is no extra power for speed and distance control. In this case, both the closed-loop stability and string stability may be destroyed;

(c) Large actuator delays: These delays come from, but not restricted to, engine indicated torque production, torque converter, pneumatic brake and transmission retarder. These delays are the main obstacles to string stability;

(d) Disturbances during gear shifting are more prominent if vehicle is accelerating which is due to large vehicle inertia and zero engine torque passed to wheels.

These factors determine naturally that short-inter-vehicle distance platooning for HDVs is more difficult than that of passenger cars. To relieve this difficulty, it is necessary to analyze the main factors in modeling for control.

Vehicle modeling for longitudinal control would include the dynamics of the following components: vehicle body, engine, brake, transmission and tire. The overall dynamic system is intrinsically high dimensional and highly nonlinear. Previous research at PATH shows that control based on simplified linear model cannot achieve such a goal. This is understandable because, from real-time control design point of view, the following factors damage control performances: model mismatch, external disturbances, measurement noise and time delays.

To maximally achieve robust performance of a controller, one needs to reduce all those effects. External disturbances can be compensated for by robustness of properly designed controller. Measurement noise can be rejected by proper filtering and data fusion techniques which are to achieve the smallest estimation error, the strongest noise rejection property and the least time delay. However, The paramount issue for control design, among others, is to obtain a good model which should be as simple as the control design method can handle, but as complicated as to capture the intrinsic vehicle dynamics. This necessarily implies a nonlinear model. It is believed that the success of longitudinal control for platooning used for Demo-97 by PATH was mainly due to a reasonably good nonlinear vehicle
model. For such a purpose, this paper proposes the following approaches for vehicle modeling.

1. Turbocharger dynamics is separated from the engine dynamics under the assumption that the booster pressure (manifold pressure between turbocharger and the cylinders) is measured;

2. A static nonlinear engine mapping which gives a functional relation between engine speed, booster pressure, fuel rate and indicated torque is to be established;

3. For pneumatic brake system, the built-in Electronic Braking System (EBS) is used in control loop one the lower level actuator. For modeling, two delays are involved: pure time delays from actuation and release and pneumatic dynamic delay which is modeled as a time parameter. A variable structure second order brake model is established to count for the difference between activation and release delays;

4. Throttle control: The built-in engine speed and torque PI control is used in loop the other lower level actuator;

Tire slip directly reflect vehicle longitudinal dynamics and vehicle moving distance measurement. An moving average tire slip estimation proposed in [10] is adopted. With these separation and simplification, the second order vehicle dynamics is globally feedback linearizable. In vehicle dynamics, one of the major components is the total resistance which can be divided into three parts: aerodynamic dragging force term, the rolling resistance and engine breaking force. The dragging force term depends on drag coefficient which can be determined by wind tunnel experiment. Rolling resistance depends on vehicle mass and slightly on speed. It is the engine braking force, which is quite different from that of passenger cars and has not yet been reported in previous work.

Using Jake (compression) brake [2, 6] is a special feature of modern turbocharged diesel engine, which usually does not exist in passenger vehicles. For a 6 cylinder engine, one can switch on 2, 4 and 6 cylinders respectively to produce retarding torque to the vehicle. The maximum retarding power can be as high as the engine active horse power [3]. The advantages of using Jake brake is its fast response is faster response than pneumatic brakes, which is particularly desirable for longitudinal control. The disadvantage is that platooning control need continuous spectrum of braking torque at all vehicle speed. Obviously, Jake brake on current engine does not have such a feature. To compensate for this, a combined braking system, including Jake brake, pneumatic brake and transmission retarder will be used for future control design.

For reader's convenience, all the notations will be listed where the first time they are used.

2 VEHICLE DYNAMICS

Engine power transmitted through power-train includes two states: engine speed and net output torque. Engine speed corresponds to wheel speed while output torque to wheel acceleration. To establish a vehicle dynamics equation, one can choose either wheel speed, engine speed, or even a speed at any point in between, as the fundamental state variable. This paper choose wheel speed for the convenience of implementation. The power flow chart is depicted as in Fig. 1.

![Power transition flow chart](image)

**Figure 1: Power transition flow chart**

where

- \( M \) – vehicle mass
- \( \omega \) – engine speed
- \( \omega_{idle} \) – engine idle speed
- \( \omega_p \) – torque converter pump speed, \( \omega_p = \omega \)
- \( \omega_t \) – torque converter turbine speed
- \( \omega_{th} \) – turbocharger speed
- \( \omega_{tr} \) – transmission output speed
- \( \omega_{dr1} \) – propeller-shaft speed including front part of final gear
- \( \omega_{dr2} \) – drive-shaft speed including rear part of final gear, final drive end
- \( \omega_{dr} \) – drive-line speed (considered as lump sum), final drive end
- \( \omega_w \) – wheel angular speed
- \( v \) – vehicle wheel speed (longitudinal) is used for all control design
- \( a \) – acceleration
- \( \alpha \) – throttle angle
- \( \alpha_f \) – fueling rate
- \( I_e \) – engine inertia
- \( I_{tr} \) – transmission inertia
- \( I_{dr1} \) – drive line inertia (before final gear)
- \( I_{dr2} \) – drive line inertia (after final gear)
- \( I_{dr} \) – lump sum drive line inertia \( (I_{dr} = I_{dr1} + I_{dr2}) \)
- \( I_w \) – wheel inertia
- \( I_m \) – intake manifold pressure or turbocharger booster pressure
- \( T_d \) – drive-line torque loss
- \( T_{ind} \) – engine indicated torque
- \( T_{net} \) – engine net output torque
- \( T_p \) – torque converter pump torque, \( T_p = T_{net} \)
- \( T_t \) – torque converter turbine torque
- \( T_b \) – service brake torque
- \( T_{Jake} \) – engine brake torque
2.1 Engine Driving Mode

Engine net output torque equation:

\[ T_{net} = I_r \omega + T_p \]

(2.1)

where it is supposed that the clutch is locked up.

If torque converter is on, \( T_p \) should be calculated from torque converter model. A possible torque converter model is the popularly recognized static Kotwicki model [1].

Following the power flow in Fig. 1 and notice the following relationships

\[
\begin{align*}
\omega_w &= \omega_{dr2} = \frac{v}{h_r} \\
\omega_r &= \omega_{dr1} = \frac{v}{h_r r_d} \\
\omega &= \omega_t = \frac{v}{h_r} r_d r_g
\end{align*}
\]

Longitudinal vehicle dynamics is obtained as

\[
\ddot{v} = r_d r_g T_{net} - (r_d T_{i4} + T_b + F_a h_r + F_{eng-brk} + M g h_r \sin \theta)
\]

(2.2)

where

\[ F_a = 0.5 \ C_a \ \rho_{air} \ A \ V_a^2 \]

\( T_b \) and \( F_{total} \) will be modelled separately.

2.2 Other Modes

1. Engine Braking Mode: \( T_{net} = -T_{jake} \) (Jake = 0, 2, 4, 6) is used in (2.2);
2. Transmission Retarder Mode: \( T_{net} = 0 \) in (2.2);
3. Vehicle Mass Preliminary Estimation:

Vehicle mass can be preliminarily estimated by (2.2)

\[
M = r_d r_g T_{net} - \frac{1}{v} (r_d T_{i4} + T_b + F_a h_r + F_{eng-brk} + M g h_r \sin \theta)
\]

which is a singular expression since if

\[ \ddot{v} + g \sin \theta = 0 \]  

one cannot estimate the vehicle mass this way. Physically, this implies that to estimate the vehicle mass, it must have non-zero acceleration, which may be generated by engine torque and/or road grade.

It is noted that this formula only provide a rough mass estimation because all the measures contain noises. Besides, at low speed, torque converter plays an important part, which aggravates the situation. For good performance, an additional strategy for vehicle mass estimation is necessary. This will be addressed in future work.

3 ENGINE MODELING

Engine is the most important part of the powertrain. Powertrain modeling and control is referred to [13] while powertrain modeling for vehicle control is referred to [1]. Diesel and turbocharged diesel engine modeling and control have been conducted in [7, 11] and diesel engines with turbocharger and EGR (Exhaust Gas Recirculation) are considered in [9]. If booster pressure is measured and taken as input to the engine, this dynamic relation between engine and turbocharger is decoupled. This simplification is sufficient for vehicle control.
Strictly speaking, engine output torque is produced by discontinuous explosion strokes. However, for longitudinal control purpose, a continuous mean-torque-value output is assumed. A static engine mapping can be used to provide a dynamic relation between engine speed, net output torque, fuel rate and booster pressure \( (\omega, T_{out}, \sigma_f, P_{bo}) \) [1, 5]. The input to torque production delay, which depends on engine speed with other factors such as injection time, engine temperature, etc., is implicitly captured by the mapping. This mapping can be used to find (by interpolating) desired fuel rate from desired engine speed and torque. If engine fuel rate control is accessible, this approach implements the engine control [5, 14, 15]. However, access of fuel rate control on new trucks are prohibited due to internal engine control structure. Thus engine control part need to be reconsidered.

The Cummins N14 435 turbocharged diesel engine for Freightliner Century Truck have internal control (Fig. 2, Engine Control Module - ECM) which takes input from pedal deflection and interpret it as either speed control command or torque control command. These commands are passed through J-1939 Bus which is the main data Bus used for Century Freightliner. The output of the controller is fuel rate (as input) to the engine. The principle of the control strategy is based on desired torque at certain speed with some optimization with respect to, among others, emission, fueling time, output torque and fuel consumption, etc.

To practically use it, a simulator is necessary to interface between computer and the ECM. Because of this structure, one cannot use engine mapping anymore to generate fuel rate. Instead, the desired net output torque and desired engine speed are directly fed into the ECM. The internal controller here acts as an inverted engine mapping. Its performance will largely affect longitudinal controller.

### 3.1 Engine Braking Effect

**Engine braking effect** comes from the mismatch between engine speed and wheel speed while gear is engaged and either torque converter is on or clutch is locked-up. This is the case when throttle is released. It is an addition to the rolling resistance.

\[
F_{total} = \begin{cases} 
F_r, & \text{engine driving} \\
F_{eng-brk} + F_r, & \text{throttle released} 
\end{cases}
\]

Here \( F_{eng-brk} = C_{eng-brk}(\omega - \omega_{idle}) \)

\[
R_g = r_f r_d 
\]

This effect is taken into whenever throttle is released and a gear is engaged. Suppose vehicle acceleration \( a \) and road grade \( \theta \) are measured/estimated. These parameters can be obtained as follows:

1. When vehicle speed reach a specified value \( v_{max} \), release throttle and keep gear engaged to leave the vehicle for free rolling. From

\[
F_{total} = Ma + Mg \sin \theta 
\]

one obtains \( F_{total} \);

2. Repeating this procedure but setting gear to neutral, one obtains

\[
F_r = Ma + Mg \sin \theta 
\]

Then

\[
F_{eng-brk} = F_{total} - F_r
\]

A test result using Freightliner is shown in Fig. 3.

![Engine braking effect when throttle is released.](image)

**Figure 3:** Engine braking effect when throttle is released.

### 4 PNEUMATIC BRAKE

Freightliner Century trucks have built-in Electronic Braking System (EBS, Fig. 4). The main characteristics of this system are:

(a) Control on each individual wheel  
(b) Incompatibility in braking torque between tractor and trailer is compensated  
(c) Fault detect is available for brake components are they are managed in real-time  
(d) Simultaneity response and pressure buildup time  
(e) Optimized stability and tractive  
(f) Electro-pneumatic and pure pneumatic (parallel) redundancies for all brake circuits

Electro-pneumatic circuit is used for default and pneumatic circuit is used when the former fails.

To fully exploit the advantages of EBS, the internal control system will be used. To achieve this, a simulator circuit to generate modulated pulse-width signals to replace the Brake Signal Generator in EBS to interface with a computer.

From the brake structure, the functional relation from applied brake pressure to brake torque has the following relations:

\[
T_b = r_d \sigma_d P_t 
\]

\[
P_{\text{sup}} = \sigma \sqrt{\frac{P_b}{P_{\text{res}}}} 
\]

\[
P_{\text{sup}} \leq P_{\text{res}}
\]
road grade is measured by on-vehicle pitch sensor, unevenness of the road is considered as additive bounded disturbances to the vehicle dynamics, which is to be expelled by the robustness of the controller.

5.1 Accessories

The following components, when activated, produce large disturbances which needs compensation.

- Fan: $\max = 46.2 [\text{hp}]$
- Generator: $\max = 2.2 [\text{hp}]$
- Water pump: $\max = 2.6 [\text{hp}]$
- Compressor: $\max = 2.6 [\text{hp}]$
- Air conditioner compressor: $\max = 5.2 [\text{hp}]$

According to [3], they are approximately proportional to $(\omega - \omega_{\text{chr}})$.

5.2 Positive Feedback by Pitch Sensor

Pitch measurement is very important for longitudinal control of HDV because the term $Mg \sin \theta$ makes a substantial contribution to the desired torque due to large $M$. In practice, unless the road survey information is available and exactly corresponds to the current position of the vehicle, one has to estimate $\theta$ using a pitch sensor in real-time. However, problems arise in the following two aspects:

(a) There is a persistent measurement noise;
(b) Vehicle acceleration cause pitch up and deceleration cause pitch down, which gives positive feedback to the closed-loop, which destroys the closed-loop stability. To compensate for this, vehicle acceleration measurement is used to reduce this effect as

$$\theta_e = \mu (\theta_m) - \sigma (a_m)$$

$\theta_e$ - estimated pitch angle
$\theta_m$ - measured pitch angle
$\mu (\theta_m)$ - a filtered pitch angle
$\sigma (a_m)$ - filtered acceleration
Then $\theta_e$ is used to replace $\theta$.

5.3 Gear Shifting

The typical symptom gear shifting is large vehicle jerking which is a prominent disturbance to string stability. Although the string stability of the controller in ideal case guarantees that the uncertainties would be attenuated downstream along the platoon. In practice, there is always a time delay caused by distance measurement and filtering. For example, a Eaton Vo- rad Doppler radar has update rate of 65[ms]. Plus the time delay caused by filtering, the total practical delay would be about 100[ms]. Such delay will accumulate downstream along the platoon. It destroys the attenuation capability of the closed-loop controller and thus the string stability.

Vehicle gear shifting point depends on both vehicle speed and acceleration. For multiple vehicle platooning, it is very difficult to synchronize the shifting for all the vehicles because there are always differences between speed tracking errors as well as distance tracking errors between those vehicles.
6 CONCLUDING

This paper discusses the modeling of HDV for longitudinal control of short inter-vehicle distance platooning. This maneuver needs a good model to avoid throttle saturation caused by model mismatch. It requires that the model should be complicated enough to capture the intrinsic vehicle dynamics and simple enough for control design. The whole power-train is thus highly dimensional and highly nonlinear. To simplify the model, a static engine mapping is used to replace the turbocharged diesel engine dynamics. Engine braking effect is also taken into consideration for HDV, which is usually ignored for passenger cars. Pneumatic brake dynamics are modeled as a variable structure second order time delay system. Prominent disturbances to vehicle longitudinal dynamics are presented. A newly developed torque converter model which is more suitable for control design, Jake brake modeling for longitudinal control, and control design using these models will be addressed in future work.

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References


