Speed Control Experiments for Commercial Heavy Vehicles with Coordinated Friction and Engine Compression Brakes

M. Druzhinina and A. G. Stefanopoulou
University of Michigan, Ann Arbor

Abstract
In this paper we describe the development and experimental validation of a simple coordination scheme between friction and engine compression brakes for a Class 8 Freightliner truck used as a development platform in the California PATH program. The coordination scheme is tuned to minimize the use of friction brakes while maintaining the speed tracking performance of the closed-loop braking controller that utilizes the friction (service) brakes only. Through the coordination scheme, a PID controller command is interpreted as a braking torque demand and split into a friction brake command and compression brake command. The split is based on the models for the compression braking torque and friction braking torque developed and identified from “field” experiments performed in Crow’s Landing. The results of closed-loop experiments show that the integration of the compression brake into the speed control framework reduces the usage and wear of the conventional friction brakes.

1 Introduction
Braking capability is critical for the Heavy Duty Vehicles safe operation in today’s busy highways and for future applications in the automated highway systems. While the fuel efficiency and acceleration performance of heavy trucks have significantly improved over the recent years, at the same time the vehicle natural retarding capability has diminished. The main vehicle retarders, namely, the service/friction brakes (friction pads on the wheels) can provide a sufficient retarding power to brake the vehicle. They may not, however, be used continuously during long descends because of the potential damage/loss of performance due to overheating and increased wear [4]. Wear and overheating not only reduce the steady-state authority of the friction brakes but also are a cause of uncertainty in brake operation [6]. The presence of delays associated with pneumatic or hydraulic subsystems in the friction brake actuation lines impose additional difficulties in using friction brakes for the longitudinal control of HDVs [11].

One approach that indirectly addresses the HDV braking capability is the enforcement of low speeds for HDV traffic in commercial corridors and urban routes. This approach, however, creates large disparities in vehicle velocities and triggers a whole set of different traffic and policy problems. Other approaches that address the HDV braking capability directly are (i) sensing mechanisms that allow early warning of friction brakes malfunction thereby facilitating brake maintenance and contributing towards consistent retarding response, and (ii) actuating mechanisms that augment the traditional friction-based HDV braking capability and provide the consistent magnitude and unlimited duration of braking force thereby mitigating some of the limitations of speed control based solely on friction brakes. Our paper addresses the latter approach, with emphasis on the compression braking mechanism [1].

Compression braking is based on converting the turbocharged diesel engine, that powers the HDVs, into a compressor that absorbs kinetic energy from the crankshaft. During compression braking mode the engine dissipates the vehicle kinetic energy through the work done by the pistons to compress the air during the compression stroke. The compressed air is consequently released into the exhaust manifold through a secondary opening of the exhaust valve (brake valve opening) at the end of the compression stroke. In conventional compression braking mechanisms currently available in the market, the timing of brake valve opening is fixed relative to piston motion and only a finite number of possible braking torque values can be generated for a given engine speed. This number depends on the number of cylinders activated in the compression braking mode. We refer to this compression braking as discretely variable braking. The finest quantization of engine braking torque that can be achieved by this approach is defined by the number of cylinders of the engine [7].

Figure 1: Dash board of experimental Freightliner truck.

A Freightliner truck used as experimental platform in the California PATH program, has a Detroit Diesel engine equipped with a 3-stages Jake-brake mechanism. This mech-
anism can activate two, four, or six cylinders in the compression braking mode either by manually turning on a switch on the dash board (as shown in Figure 1) or by sending the voltage command from the on-board computer output port. Newly developed trucks with electronically controlled engines are equipped with a braking mechanism which allows activation of any number of cylinders [7].

Present day commercial systems use engine brake for cruise control, e.g., Eaton-Vorad collision-warning system EVT-300 with SmartCruise that activates compression braking automatically when an obstacle is detected. It is not clear, however, that “vehicle following” can be realized by using the discretely variable compression braking, unless the service brakes can smoothly compensate for the compression braking torque deficit at a given speed. Indeed, to satisfy stringent requirements of HDV following scenarios and other applications in Intelligent Transportation systems, a continuously variable braking torque is highly desired.

The continuously variable braking torque can be achieved through controlling a brake valve opening using a variable valve timing actuator [5]. We have previously studied the effects of continuously varying valve timing on vehicle response in [2, 3, 8, 9]. In [2], we have developed an adaptive controller that estimates the vehicle mass and road grade values and controls the continuously variable compression brake to ensure consistent and robust speed tracking performance in the face of unknown vehicle mass and significant road grade changes. However, the continuously variable compression braking represent an advanced technology which is not available on production vehicles at the present time. Hence, to enable experimental evaluation of the adaptive control scheme [2] with the existing hardware, we plan to pursue in our future work a modification of the adaptive controller to operate the discretely variable compression brake in coordination with the friction brakes.

In this paper we describe the development of a coordination scheme between friction and engine compression brakes to implement a non-adaptive longitudinal speed controller as a first step to conducting experiments for validation of our adaptive scheme. Specifically, we identify the characteristics of the existing 3-stage engine brake and friction brakes. Based on resulting models we develop a coordination scheme that maintains the speed tracking performance of the nominal PID controller which was originally designed for the case of friction brakes only [10]. Through the coordination scheme, the command of the nominal PID controller is interpreted as a braking torque demand and split into a friction brake command and compression brake command. The developments are validated on an experimental Freightliner truck with a Detroit Diesel DDEC III engine. The results of closed-loop experiments demonstrate that the integration of the compression brake into the speed control framework reduces the usage of the conventional friction brakes.

The paper is organized as follows. In Section 2 we describe the experimental hardware and software setup. The open-loop experiments that were conducted to identify the models for the compression brake are presented in Section 3. The development of the coordination scheme between the friction and compression brakes and the results of closed loop experiments validating its performance are reported in Section 4.

2 Hardware and software setup

In this section we summarize the relevant details of the hardware configuration used in our open-loop and closed-loop experiments (see [10] for additional details).

The experimental Freightliner truck is equipped with a Detroit Diesel DDEC III engine that uses an electronic fuel injection system. This system injects fuel based on the command communicated via a voltage signal. The air brake system with electronic actuation by ISE Research Corporation uses four proportional actuators and ten pressure transducers, one for each of the ten brakes. The brake signal is transmitted by wires from the on-board computer’s I/O ports to each proportional actuator. The engine brake (in two, four, or six cylinder mode) is turned on and off by two independent voltage commands (0-1). Note that the brake can only be turned on when there is no fuel command to the engine (i.e., when the input voltage to the fuel circuit is below 0.5 Volt).

Although the actuators can, in principle, respond to the full range of voltage command (between 0 and 10 Volt), they were limited within a smaller working range based on ride quality and safety requirements. An upper limit of 4.5 Volt for the voltage passed to the fueling actuator was imposed by the driveability requirements. There was also a lower limit of 1.2 Volt imposed on the voltage passed to the friction brake actuator since a voltage command lower than 1.2 Volt does not produce any friction braking torque (dead zone).

The on-board computer sends out all the signals to the actuators based on the signals from the sensors. The sensors provide measurements such as the wheel speed, engine speed, air brake pressure at the transducer location (attached to the brakes), Jake brake solenoid monitoring signals, etc. The transmission signals were not measured at the time when our experiments were conducted and no information about transmission control strategy was available to us. At the vehicle level we could only limit the highest gear selected by the transmission. This was done manually by setting the highest allowed gear (1-6) from the dash board (see Figure 1).

The control software is written in C and runs under QNX real-time operating system. The longitudinal control routine is a part of a larger program, that also includes lateral control. The longitudinal control routine is executed approximately every 20 ms.

3 Identification of the compression brake characteristics

As a first step to conducting full scale closed-loop speed control experiments, we performed open-loop longitudinal control experiments with the objective to determine the dynamic characteristics of the system, in particular, the dependence of the discretely variable compression braking torque on the engine speed and the number of activated cylinders (actuation
The longitudinal experiments were based on the following procedure. At the beginning of each run the vehicle is accelerated so that the engine speed is slightly higher than the engine speed at which we wish to engage the compression brake. Then we release both the brake and the acceleration pedal and wait for the speed to fall down to the desired engine speed value. At that moment we issue a compression braking command (to engage two, four or six cylinders) through the computer output port to the engine control module. Thus the vehicle starts to decelerate under the influence of the compression brake until it stops. Note that we enforced in our experiments a delay between fuel cutoff and compression brake engagement in order to avoid the transients caused by the transition from fueling mode to braking mode.

Figure 2 shows the engine speed response for an experiment where the vehicle is operated in first gear and where we first bring the engine speed to 2010 rpm and disable fueling. When the engine speed fall to 2000 rpm we sequentially engage the compression brake in six, four and two cylinder mode. Once the engine speed falls sufficiently low (approximately 700 rpm), the engine idle governor turns on and compression brake turns off automatically so that further decrease in the engine speed and engine stall are prevented.

3.1 Transmission logic

Since the compression braking torque affects the vehicle through the transmission, the transmission behavior needs to be taken into account when trying to coordinate the compression brake with the friction brake. Identifying transmission logic presented a number of challenges. In particular, transmission signals were not measured at the time when our experiments were conducted and we had no information about transmission control strategy. On the vehicle level we were only able to limit the highest gear selected by the transmission. In Figure 3, the gear ratio is estimated by dividing vehicle speed (estimated from wheel speed measurement under no slip assumption) by engine speed (measured). The transition from gear 1 to gear 2, as estimated on the basis of the speed ratio, is longer due to interactions with the torque converter and other dynamic effects. The transition between the second and third gears can be estimated very well on the basis of the speed ratio. Finally, for identification purposes we decided to use the data collected in the second and third gear when the torque converter is locked. This avoids the interactions with the torque converter dynamics. Our coordination scheme between the compression brake and the friction brakes incorporates a gear estimator, which is based on the speed ratio and additional logic.

Figure 3: The trajectories of vehicle speed, engine speed and estimated gear ratio.

3.2 Compression Braking Torque Identification

From the data collected in the deceleration experiments and known values for the vehicle parameters we identified the magnitude of the compression braking torque as a function of the engine speed and the number of activated “braking” cylinders using the Least Squares (LS) procedure as is described next.

The vehicle model that we use for identification purposes prescribes the dynamics of the engine speed from the balance between the engine torque on the driveshaft and the summation of aerodynamic resistance, road grade and rolling resistance torques. We assume that the vehicle is driven on a zero grade, the gear remains constant, the engine is not fueled and is operated in the compression braking mode. The engine speed, \( \omega \), is proportional to the vehicle speed assuming that the gear is fixed and the slip is negligible. Therefore, the system under consideration is of the form:

\[
J_t \dot{\omega} = -T_{Q,cb} - C_d q_A \omega^2 - r_g M \frac{g}{\rho} A f_r,
\]

where \( \omega \) is the engine crankshaft rotational speed in rad/sec, \( J_t = M_r^2 + J_r \) is the total vehicle inertia reflected to the engine shaft, \( J_r \) is the engine crankshaft inertia, \( M \) is the mass of the truck, \( r_g \) is the total gear ratio, \( C_d \) is the quadratic resistive coefficient, \( C_d \) is the aerodynamic drag coefficient, \( \rho \) is ambient air-density, \( A \) is the frontal area of the vehicle, \( f_r \) is the rolling resistance of the road, and \( g \) is the acceleration due to gravity. The engine torque applied to the crankshaft during compression braking by the engine is denoted by \( T_{Q,cb} \). For identification purposes we concatenated together trajectories from several experiments (also in different gears) that corresponded to the periods of time when the experimental operation matched assumptions of our model (1), i.e. deceleration in compression braking mode.
Approximations of various orders were considered for estimating $TQ_{cb}$ and the linear one in $\omega$ was found to be sufficient:

$$TQ_{cb}(\omega) = \theta_0 + \theta_1 \omega,$$

(2)

where $\theta_0$ and $\theta_1$ are unknown constant parameters to be identified. The braking torque at the engine shaft, $TQ_{cb}$, can be calculated from (1) as a function of known longitudinal vehicle parameters and an estimate of engine speed acceleration\(^1\). We found out, however, that the accuracy of the torque estimate obtained by this approach is not acceptable due to the signal noise and its implications when numerical differentiation is applied. Note that filtering the $\dot{\omega}$ estimate with a low pass filter does not improve much the accuracy of the torque estimates; moreover, it may result in biasing the torque estimate. Alternatively to differentiating and filtering the engine speed signals, we propose to apply the LS procedure to their integrated quantities that exhibit smooth behavior. For this purpose, we integrate both sides of the system equation (1) over the time intervals $[T^i, T^i+\Delta T]$, where $T^i = T^{i-1} + \Delta T$, $i = 1, \ldots, N$, and $\Delta T$ is the sample period of data acquisition system (20 msec). We have:

$$\omega(T^i) - \omega(T^0) = -\theta_0 \int_{T^0}^{T^i} \frac{1}{J_t} dt - \theta_1 \int_{T^0}^{T^i} \frac{1}{J_t} \omega(t) dt - \frac{C_q r_f}{J_t} \int_{T^0}^{T^i} \omega^2(t) dt - \frac{r_g f_r g M}{J_t} (T^i - T^0).$$

By defining for $i = 1, \ldots, N$

$$y_i = \omega(T^i) - \omega(T^0) - \frac{C_q r_f}{J_t} \int_{T^0}^{T^i} \omega^2(t) dt - \frac{r_g f_r g M}{J_t} (T^i - T^0),$$

$$\phi_i = \frac{1}{J_t} \int_{T^0}^{T^i} dt, \quad \frac{1}{J_t} \int_{T^0}^{T^i} \omega(t) dt$$

we obtain

$$y_i = \phi_i \cdot \theta, \quad i = 1, \ldots, N.$$ (3)

Let $\Phi = [\phi_1 \ldots \phi_N]^T$, $Y = [y_1 \ldots y_N]^T$, then $Y = \Phi \cdot \theta$, and the least squares estimate for $\theta$ is obtained as

$$\hat{\theta} = \Phi^T (\Phi \Phi^T)^{-1} Y.$$

Note that to calculate $y_i$ and $\phi_i$, it is necessary to calculate $\int_{T^0}^{T^i} \omega(t) dt$ and $\int_{T^0}^{T^i} \omega^2(t) dt$. These integrals have been calculated by passing the samples $\omega(T^i)$ and $\omega^2(T^i)$, $i = 1, \ldots, N$ through a Simulink integrator block.

The validation results of the resulting regression are shown in Figures 4-6. We compare the measured engine and vehicle speeds with the predicted engine and vehicle speed based on our model. Figure 4 shows the response in the second gear with 2 cylinders activated. Figure 6 shows the response in the third gear with 6 cylinders activated. Figure 5 shows the response in the 3rd gear with four cylinders activated. Note that the engine speed response is accurately predicted except at low engine speed, where the discrepancy is due to the idle governor turning on.

\(^1\)The acceleration signals were not measured at the time of our experiments.

4 Integration of compression brake into speed controller

In [10], a fixed-gain PID speed controller was implemented on an on-board computer to automatically determine the fuel and friction brake actuator commands so that desired vehicle speed can be tracked. The experimental results demonstrated that the fixed-gain PID controller is able to achieve good speed tracking performance for the nominal vehicle loading. Following these results, we conducted a set of closed-loop experiments with the objective to integrate the 3-stage Jake brake with the friction brakes and, therefore, investigate the benefits of using compression brake. In our approach we decided to preserve the existing longitudinal feedback speed controller [10] that worked quite well. However, we wanted to minimize the use of the friction brake. Thus, the integration strategy assigns high priority to the compression brake.

In this section we first review the nominal PID controller [10] for friction brakes only. We then describe our scheme to coordinate the compression brake with conventional friction brakes.
4.1 Nominal PID controller for friction brakes

Figure 7 shows a simplified diagram of the nominal PID control scheme (see [10] for details). In particular, the speed tracking error is passed through a PID controller to generate a control output $u$. Depending on the sign of the output, either fueling mode or braking mode are activated. The PID control output is then scaled to the physical voltage values corresponding to the actual fuel and brake commands, namely to the interval between 1.25 and 4.5 Volt in the fueling mode and to the interval between 1.2 and 3 Volt in the braking mode. After this conversion the signals are sent out to the computer I/O ports and finally reach the corresponding actuators. Since we are operating the longitudinal controller separately from the lateral controller, we send the same braking command to brake actuators at all wheels. In particular, we do not consider the effects of differential braking in this work that may be added on top of the nominal longitudinal friction braking torque.

We first evaluated the nominal PID controller with the compression brake disabled. The selected driving profile as shown in Figure 9 requires aggressive braking action to ensure that the vehicle speed is reduced from 15 m/sec to 5 m/sec within 5 sec. The measured vehicle speed shows good tracking of the desired vehicle speed. However, during this braking maneuver the friction brakes operate close to their limits. Indeed, as is shown in Figure 10, the voltage command to the friction brake actuator is as high as 2.8 Volt out of maximum of 3.0 Volt.

4.2 Coordination scheme between compression brakes and friction brakes

Recall that braking with the compression brake is preferable, because we want to minimize the use of conventional friction brake and, hence, the friction brake wear. Therefore, the braking strategy is to engage the friction brake only when it is necessary to supplement the compression brake. The approach pursued in this work relies on keeping the basic speed controller intact and then splitting the total braking demand of the nominal controller into compression torque demand and friction torque demand. Figure 8 explains how we extracted the compression braking command from the braking voltage command generated by the nominal PID controller.

4.3 Closed-Loop Experiments

While the basic idea of the coordination scheme is described in Figure 8, the actual implementation is more complex. In particular, we enforced the minimum residence time for the number of "braking" cylinders so that to prevent chattering between modes corresponding to different number of active cylinders. We also employed relays in switching between fueling mode and braking mode. Since the transmission signals, and in particular, the gear ratio signals were not available at the time when the experiments were conducted, a gear ratio estimator was implemented. Since the friction brake does not respond to voltage commands less than 1.2 Volt the scheme was designed to avoid this dead-zone. In particular, if the friction brake voltage demand is less than 1.2 Volt then we decreased the number of cylinders engaged in the compression braking mode to bring the friction braking torque outside the dead zone.

Based on the friction braking torque model, we first calculate the demanded friction braking torque at the vehicle level that corresponds to the voltage command of the nominal controller. Using the transmission model and the gear, we extract the demanded braking torque at the engine level. We then determine the maximum number of engine cylinders that can be operated in compression braking mode without exceeding the total demanded braking torque. The difference between total demanded braking torque and compression braking torque needs to be delivered by the friction brake. The difference is transformed through the gear ratio to determine the desired friction force and finally the actual command to be applied to the friction brake actuator.

Figure 7: Nominal PID controller diagram.

Figure 8: Coordination scheme diagram.

Figure 9: Nominal PID controller performance: trajectories of vehicle and engine speed.

\[ \text{2The dynamics for the fuel and friction brake actuators were identified during a series of open-loop experiments carried out by the UCLA team (see [10] for details of the HDV project with PATH).} \]
The next set of experiments involved activating the compression brake along with the friction brakes using our coordination scheme. To compare these experimental results with the previous set of experiments we used the same speed profile. Figures 11-12 summarize the closed-loop behavior with the compression brake active. First, the speed tracking performance is completely preserved as is shown in Figure 11. As can been seen from the upper plot in Figure 12, the total braking torque demand is split into compression torque demand and friction torque demand. The compression brake is engaged in the six cylinder mode during this braking maneuver to respond to the fairly aggressive braking demand. The friction brake is also active to complement the compression braking torque. The use of the friction brake is reduced by 35 percent during this braking maneuver as compared to the case when braking was done using friction brake only. This reduction is shown by the middle plot in Figure 12 which compares the friction brake command for the case when the compression brake is disabled and the friction brake command with enabled compression brake.

Acknowledgement

This research is supported in part by the California Department of Transportation through the California PATH Program under MOU 393 and TO 4202. We would like to thank PATH engineers B. Bougler, D. Empey, P. Kretz, D. Nelson, and C.-W. Tan for their help in arranging and conducting the experiments and driving the truck in the test site at Crow’s Landing.