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Parameter Estimation and Supervisory Techniques for Robust Longitudinal Control of Heavy Vehicles

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CALIFORNIA PATH PROGRAM  
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# **Parameter Estimation and Supervisory Techniques for Robust Longitudinal Control of Heavy Vehicles**

**Maria Druzhinina, Lasse Moklegaard,  
Anna G. Stefanopoulou**

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# **Parameter Estimation and Supervisory Techniques for Robust Longitudinal Control for Heavy Vehicle**

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# Parameter Estimation and Supervisory Techniques for Robust Longitudinal Control for Heavy Vehicle

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## Abstract

This report describes the development and experimental validation of a coordination scheme between friction and discretely variable compression brakes for a Class 8 Freightliner truck used as a development platform in the California PATH program. The coordination scheme that we developed maintains the speed tracking performance of the nominal PID controller which was originally designed by the UCLA team for the case of friction brakes only. Through the coordination scheme, the command of the nominal PID controller is split into a friction brake command and compression brake command. The integration of the compression brake into the speed control framework reduces the usage and wear of the conventional friction brakes. This report describes the experimental hardware and software setup; open-loop experiments that were conducted to identify the models for the friction brake and the compression brake; the identification procedure of the friction brake and compression brake characteristics and the identification results; the development of the coordination scheme for the friction and compression brakes and the results of closed loop experiments that were used to validate the coordination scheme performance.

## Keywords

Advanced Vehicle Control Systems    Speed Control  
Brakes    Vehicle Dynamics  
Commercial Vehicle Operations

## Executive Summary

This report addresses the integration of vehicle retarders for safe automation of commercial heavy vehicles. Braking capability is critical for the Heavy Duty Vehicles (HDV) 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, thereby limiting the deceleration performance of HDVs. 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 to maintain the desired speed because of the potential damage/loss of performance due to overheating and increased wear. Wear and overheating not only reduce the steady-state authority of the friction brakes but also are a cause of uncertainty in brake operation. 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. The increasing need for retarding power with consistent magnitude and unlimited duration is addressed through truck manufacturers and fleet managers by the development of various retarding mechanisms in addition to the conventional friction brakes.

In this work we focus on one of the auxiliary retarders, that satisfies the low maintenance and weight-to-power ratio requirements, namely the compression brake. This retarding mechanism converts the turbocharged diesel engine, that powers the HDVs, into a compressor that absorbs kinetic energy from the crankshaft. The vehicle kinetic energy is dissipated 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 at the end of the compression stroke. Compression brake increases the overall decelerating capability of the vehicle and, therefore, allows to accommodate higher operational speeds. This retarder can potentially be used as a sole decelerating actuator during low deceleration requests and can be combined with the friction brakes during high deceleration requests. Therefore, the application and intensity of the friction brakes can be reduced resulting in a significant decrease in the vehicle maintenance costs.

Moreover, the use of the compression brake allows to mitigate some of the limitations of longitudinal speed control based solely on friction brakes. It is worth to note that compression brake is generally found in most well equipped privately-owned trucks and truck-fleets, hence, integrating this retarder with friction brakes can potentially have wide-spread application.

In this report we describe the development of a coordination scheme between compression and friction brakes for a heavy-duty Freightliner vehicle which is used as an experimental platform for the California PATH program. Specifically, we first identify the characteristics of the existing on-off engine brake and friction brakes from open-loop experiments on a vehicle conducted at Crows Landing in January, 2000. Based on resulting models we develop a coordination scheme that maintains the speed tracking performance of the nominal PID controller which was originally designed by the UCLA team for the case of friction brakes only. Through the coordination scheme, the command of the nominal PID controller is interpreted as an implied braking torque demand. Then, this braking torque demand is split into a part to be delivered by the compression brake and a part to be delivered by the friction brakes. The validation of the coordination scheme has been undertaken in August, 2001 and showed a sizable reduction in the friction brake usage during braking maneuvers. The reduction of the friction brake usage is beneficial in that the wear of the friction brake can be reduced and its service life can be prolonged.

While the performance of the fixed gain PID controller is quite good for the nominal vehicle parameters, to ensure consistent and robust speed tracking performance in the face of significant vehicle mass and road grade changes an adaptive controller is needed. In our prior work within MOU 393, 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 performance on a grade while avoiding the use of friction brakes altogether. The continuously variable compression brakes represent an advanced technology which is not largely available on production vehicles at the present time. Hence, in our future work we plan to pursue a modification of the adaptive controller to operate the discretely-variable compression brake in coordination with the friction brake as well as experimental validation of the resulting scheme on the test vehicle at Crows Landing.

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# 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, thereby limiting the deceleration performance of HDVs. 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* to maintain the desired speed because of the potential damage/loss of performance due to overheating and increased wear [7]. The current recommended practice to brake on a downgrade by intermittent application of the service brakes (or *snubbing*) rather than continuous application (or *dragging*) exemplifies these limitations [8]. Wear and overheating not only reduce the steady-state authority of the friction brakes but also are a cause of uncertainty in brake operation [9]. 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 [14].

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 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. Our report addresses the latter approach, with emphasis on the compression braking technology.

Compression brake is a powerful retarding mechanism which converts the turbocharged diesel engine, that powers the HDVs, into a compressor that absorbs kinetic energy from the crankshaft [2]. The consistent magnitude and unlimited duration of compression brake is increasingly important in order to integrate HDVs into the advanced transit and highway systems. The use of this retarder within automated operation of heavy vehicles allows to mitigate some of the limitations of longitudinal speed control based solely on friction brakes. Specifically,

- The compression brake increases the overall decelerating capability of the vehicle and, therefore, allows to accommodate higher operational speeds.
- This retarder can potentially be used as a sole decelerating actuator during low de-

celeration requests and can be combined with the friction brakes during high deceleration requests. Therefore, the application and intensity of the friction brakes can be reduced resulting in a significant decrease in the vehicle maintenance costs.

- Reducing the use of friction brake reduces the uncertainties in brake operation. Indeed, compression braking torque can be reliably estimated using steady-state engine torque maps (more accurately than combustion torque [11]).

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. The number of possible discrete torque values depends on the number of cylinders activated in the compression braking mode. We refer to this compression braking mechanism as discretely variable.<sup>1</sup> In order to satisfy stringent requirements of HDV following scenarios and of other applications in Intelligent Transportation systems a continuously variable in the braking torque is highly desired. This continuous variability can be achieved by integrating the compression brake with the friction brakes wherein friction brakes smoothly compensate for the compression braking torque deficit at a given speed. 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 a collision is imminent), however, none of them support integration with friction brakes. Of course, the integration with friction brakes raises critical liability issues.

It is worth noting that the continuously variable braking torque can be alternatively achieved through controlling a brake valve opening using a variable valve timing actuator. We have studied the effects of continuously varying valve timing on vehicle response in our previous work within MOU 372 [11]. We have shown that even with this advanced actuator the compression brake must be coordinated with friction brakes to handle large deceleration demands. Indeed, the compression brake capabilities are limited, therefore the friction brakes need to be engaged when compression brake saturates to provide sufficient

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<sup>1</sup>The acquired trucks for the Demo 2003 are equipped with a Cummins engine and with a communication protocol that allows the request of negative torque that is approximately matched by a combination of number of “braking” cylinders and gear ratio.

braking power [4],[10].

In this report we address the integration of the conventional discretely variable compression brake and of the friction brakes for vehicle speed control. The developments are validated on an experimental Freightliner truck with a Detroit Diesel DDEC III engine and 3-stage Jake brake. Specifically, we identify the characteristics of the existing on-off 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 by the UCLA team for the case of friction brakes only. 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. We demonstrate that the integration of the compression brake into the speed control framework reduces the usage and wear of the conventional friction brakes.

The report is organized as follows. The operating principles of the compression brake mechanism are reviewed in Section 2. In Section 3.1 we describe the experimental hardware and software setup. The open-loop experiments that were conducted to identify the models for the friction brake and the compression brake are presented in Section 3.2 followed by a discussion of the procedure that we used to identify friction brake and compression brake characteristics. 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. Finally, concluding remarks are made in Section 5.

## 2 Background

The increasing need for vehicle braking power is addressed through truck manufacturers and fleet managers by the development of various retarding mechanisms in addition to the service brakes (drum or disc brakes on the vehicle wheel). The main categories of these retarders are (A) the engine brakes and (B) the shaft brakes. In the first category, engine brakes enhance braking efficiency by modifying the conventional gas exchange process. In the second category, shaft retarders are devices attached to the transmission, driveline, or axle that use high turbulence or electro-magnetic forces to dissipate the rotational energy of the moving parts.

The engine compression brake is a retarder that enhances braking capability by altering the conventional gas exchange process in the cylinders of the engine and effectively converting the turbocharged diesel engine, that powers the HDV, into a compressor that absorbs kinetic energy from the crankshaft [2]. During the compression braking the fuel injection and combustion are inhibited. Through the work done by the pistons, using the crankshaft kinetic energy, the air in the cylinder is compressed in the compression stroke. At the end of the compression stroke, close to the time when fuel injection usually takes place, the exhaust valve opens dissipating the energy stored in the compressed air into the exhaust manifold. We call the secondary opening of the exhaust valve when the air is released into the exhaust as Brake Valve Opening (BVO) or braking event, and refer to the braking-event profile of the exhaust valve as “brake valve” (see Figure 1).

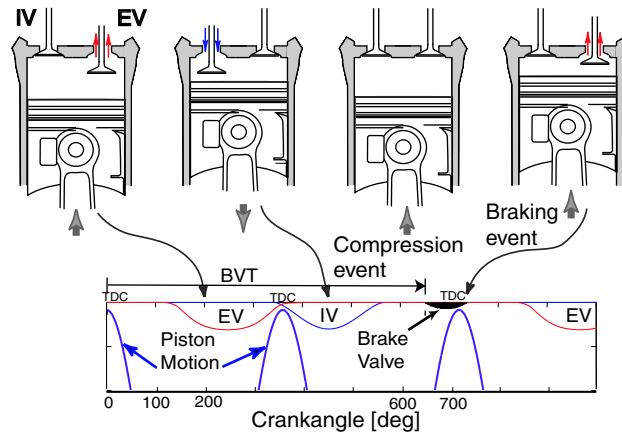


Figure 1: Lift profiles for exhaust, intake and brake events.

The activation of the brake valve is typically achieved through a master-slave hydraulic system. The exact profile and timing for the brake valve are designed to maximize the braking power, i.e., to generate the highest peak cylinder pressure. The brake valve profile

needs to satisfy constraints on component loading due to high in-cylinder pressure, and geometric clearance between the brake valve trajectory and the piston motion. Note that in the absence of BVO essentially all the potential energy stored in the compressed air will return to the wheels by the downward piston motion. With the secondary exhaust valve opening, however, the kinetic energy absorbed during the compression stroke can be dissipated into the exhaust manifold and, consequently, to the atmosphere through the engine cooling system. The existing engine cooling system typically manages the dissipation of the exhaust manifold heat and no additional cooling subsystems are introduced to the engine. It is important to note that the engine temperature during braking is not considerably lower than the one during combustion. As a matter of fact, the overheating of the injectors during braking at long descents is of concern to many engineers. The high temperature in the cylinders can damage the injectors that do not enjoy the beneficial cooling effect of fuel injection during the braking periods.

The concept of compression braking was introduced by Cummins in 1966, and typically, depends on an add-on device consisting of a cast iron housing with hydraulic circuitry added on top of the regular engine valve actuator system, that opens the exhaust valve. In conventional compression braking mechanisms, currently available in the market, the activation of the brake valve events is based on the mechanical link between the crankshaft and the camshaft, as shown in the left plot in Figure 2.

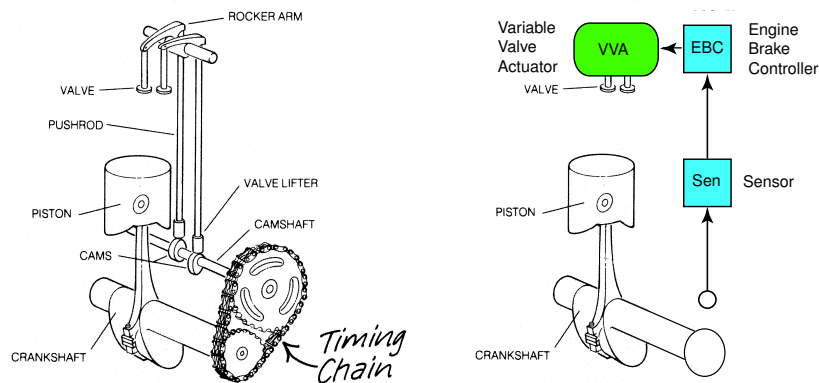


Figure 2: **Left:** Conventional valve lift system for fixed valve timing (Source: [12]). **Right:** Valve lift system that enables variable valve timing.

As a result, the valve opens at fixed degrees with respect to piston motion and only a finite number of possible braking torque values can be achieved for a given engine speed. The number of possible discrete torque values depends on the number of cylinders activated under compression mode. The finest quantization of engine braking torque that can be

achieved with this approach is defined by the number of cylinders of the engine. A further quantization of the braking torque delivered to the wheels can be achieved through the gear selection in the driveline. Thus, the request of negative torque can be approximately matched by combining a certain number of “braking” cylinders and a certain choice of gear ratio. This capability is achieved in the recently acquired Freightliner vehicle with an electronically controlled Cummins engine, a newly developed Jakobs 6-stage braking mechanism. It is not clear, however, that a vehicle following can be realized, unless the service brakes that are normally controlled by the driver can smoothly compensate for the compression braking torque deficit at a given speed.

The stringent requirements of HDV following scenarios and other application in Intelligent Transportation necessitate continuous variation in the compression braking torque. The continuous torque variation can be potentially achieved through controlling a brake valve opening using a variable valve timing actuator. Control of the brake valve timing allows us to continuously vary the retarding power of the compression brake mechanism. The right plot in Figure 2 shows a system where the valves are activated by electro-mechanical or electro-hydraulic actuators. The mechanical connections between the valve profile and the crankshaft are replaced by electronics, hence continuous variable valve timing is possible. Compression braking using a variable brake valve timing is very desirable because it allows smooth braking torque variations for a given speed, and thus, seamless integration with the service brakes. Although currently there are no commercial systems that achieve fully continuous variable compression braking, many engine manufacturers are striving for variable compression braking effort. Jacobs variable brake valve timing [6], Cummins’ discrete cylinder brake valve actuator [5], and Volvo’s variable compression braking with exhaust throttle actuator [1] are examples of advanced compression braking systems.

In this report we study an experimental Freightliner HDV with an old technology Detroit Diesel engine with a 3-stages Jake-brake mechanism which can activate two, four, or six cylinders in the compression braking mode. We, therefore, address a wider range of old generation vehicles that constitute a larger percentage of trucks rather than the new technology vehicles that are equipped with electronically controlled engines.

## 3 Identification of the compression brake characteristics

As a first step to conducting full scale closed-loop speed control experiments, in January, 2000 we performed open-loop longitudinal control experiments at Crows Landing. The objective was to determine the dynamic characteristics of the experimental Freightliner Class 8 vehicle with an old technology Detroit Diesel engine equipped with a 3-stages Jake-brake mechanism. This mechanism can activate two, four, or six cylinders in the compression braking mode. From these open-loop experiments we were able to determine the 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.

### 3.1 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 [13] 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 (one for the front tractor brakes, one for the rear tractor brakes, one for the left trailer brakes and one for the right trailer brakes) 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 <sup>2</sup>, 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.

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

## 3.2 Open-Loop Experiments

We now describe the open-loop experiments that we performed in more details.

The braking experiments were based on the following procedure. At the beginning of each run we accelerated the vehicle 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 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 make sure that the transients caused by transitions from fueling mode to braking mode have abated.

Figure 3 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.

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<sup>2</sup>There are two wheel speed sensors. The first sensor is used for measuring low wheel speed (corresponding vehicle speed is lower than 0.5283m/sec if slip is negligible) and the second is used to measure high wheel speed

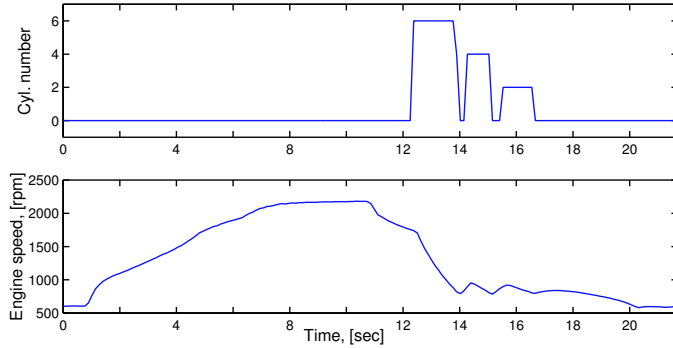


Figure 3: Engine speed response in a first gear to change in activated cylinders number. The upper plot shows the trajectories of activated cylinders.

### 3.3 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. Figures 4-6 illustrate some of the difficulties that we encountered when attempting to estimate the gear and transmission behavior. In Figure 4, the gear ratio is estimated by dividing vehicle speed (estimated from wheel speed measurement under no slip assumption) by engine speed. The transition from gear 1 to gear 2, as estimated on the basis of the speed ratio, takes a long time 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.

Another issue is when the downshifting occurs. Figures 5 and 6 show that downshifting occurs at lower rpm for friction brake (860 RPM) than for compression brake (980 RPM). This can be qualitatively explained by the fact that compression brake develops more torque at higher RPM, so it is beneficial to downshift early. The exact control strategy used to make these decisions was, however, not available to us. 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. <sup>3</sup>

<sup>3</sup>After we completed our experiments, the experimental setup was changed so that now the

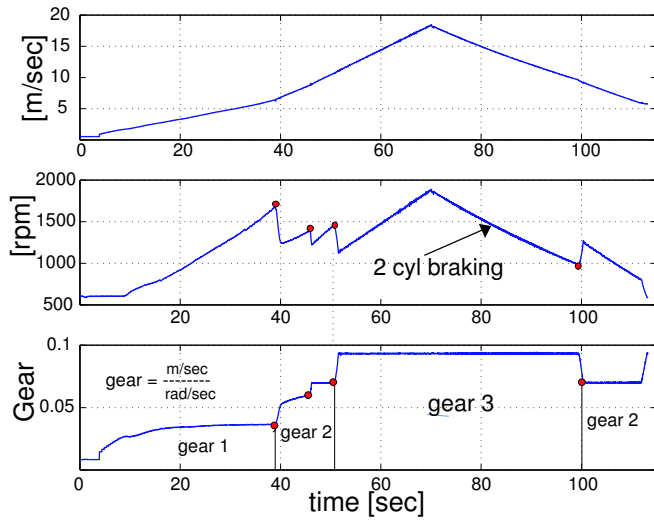


Figure 4: The trajectories of vehicle speed, engine speed and estimated gear ratio.

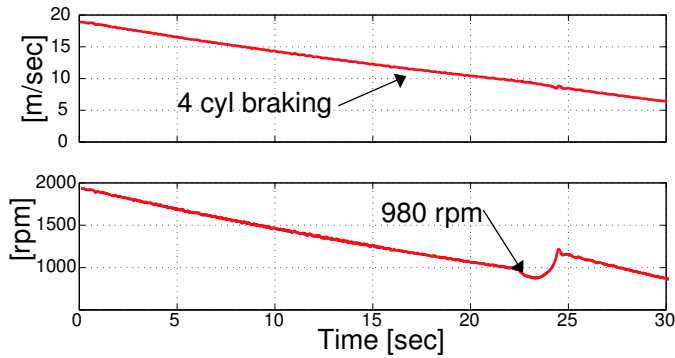


Figure 5: The trajectories of vehicle speed, engine speed with two active cylinders.

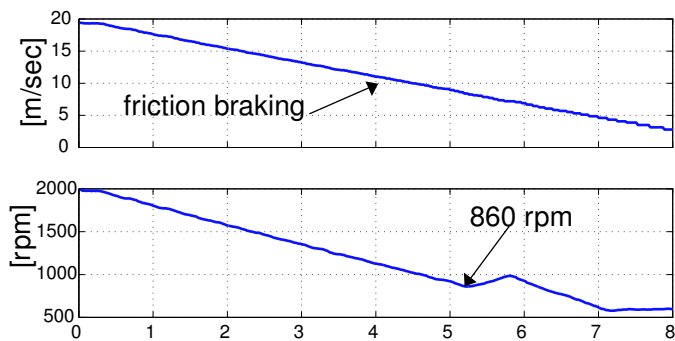


Figure 6: The trajectories of vehicle speed, engine speed with the active friction brake.

### 3.4 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 sum 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:

$$J_t \dot{\omega} = -TQ_{cb} - C_q r_g^3 \omega^2 - r_g f_r g M, \quad (1)$$

where  $\omega$  is the engine crankshaft rotational speed in rad/sec,  $J_t = Mr_g^2 + J_e$  is the total vehicle inertia reflected to the engine shaft,  $J_e$  is the engine crankshaft inertia,  $M$  is the mass of the truck,  $r_g$  is the total gear ratio,  $C_q = \frac{C_d A \rho}{2}$  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  $TQ_{cb}$ . Approximations of various orders were considered for estimating  $TQ_{cb}$  and the linear one in  $\omega$  was found to be sufficient:

$$TQ_{cb}(\omega) = \theta_0 + \theta_1 \omega, \quad (2)$$

where  $\theta_0$  and  $\theta_1$  are unknown constant parameters to be identified.

*Remark:* In our initial report presented at PATH conference in October 2000 we relied on the deceleration data in the first gear and used a quadratic regression for estimating the compression torque  $TQ_{cb}$ . Based on a more elaborate study here and deceleration data in the second and third gears, we found that a linear regression is adequate and a quadratic term does not add much accuracy. We, furthermore, believe that interactions with the torque converter dynamics significant in the first gear can obscure the identification results. In particular, we found that the quadratic regression obtained on the basis of the first gear data does not match well the response of the vehicle in the second and third gears.

The braking torque at the engine shaft,  $TQ_{cb}$ , can be apparently calculated from (1) as a function of known longitudinal vehicle parameters and an estimate of vehicle acceleration <sup>4</sup>:

$$TQ_{cb}(\omega) = -J_t\dot{\omega} - C_q r_g^3 \omega^2 - r_g f_r g M, \quad (3)$$

For identification, we used the following values: the mass of the truck is  $M = 19000 \text{ kg}$ , the aerodynamic drag coefficient is  $C_d = 0.55$ , the ambient air-density is  $\rho = 1.2$ , the frontal area of the vehicle is  $A = 10.03 \text{ m}^2$  and the rolling resistance of the road is  $f_r = 0.055$ .

An estimate of the braking torque was first obtained from (3) and numerical differentiation of  $\omega$  as shown in Figure 7. 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. The data shown in Figure 7 are for two different gears with the total gear ratio  $r_g$  equal to 0.07 and 0.0934 for the second and third gear, respectively.

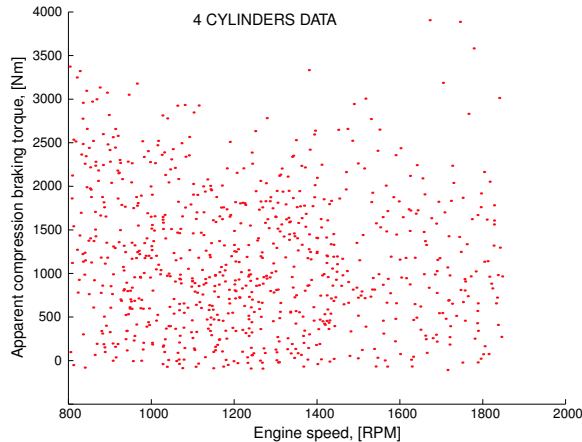


Figure 7: Apparent compression braking torque estimate based on numerical differentiation of  $\omega$  when four cylinders are active.

It is quite obvious that the wide spread in the torque estimates is due to the noise and numerical differentiation of the engine speed signal. Figure 8 shows that filtering of  $\dot{\omega}$  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).

Note also that the deviations of the torque estimates from the straight line, indicated by circles in Figure 8, coincide with the start of the individual trajectories that were concate-

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<sup>4</sup>The acceleration signals were not measured at the time of our experiments.

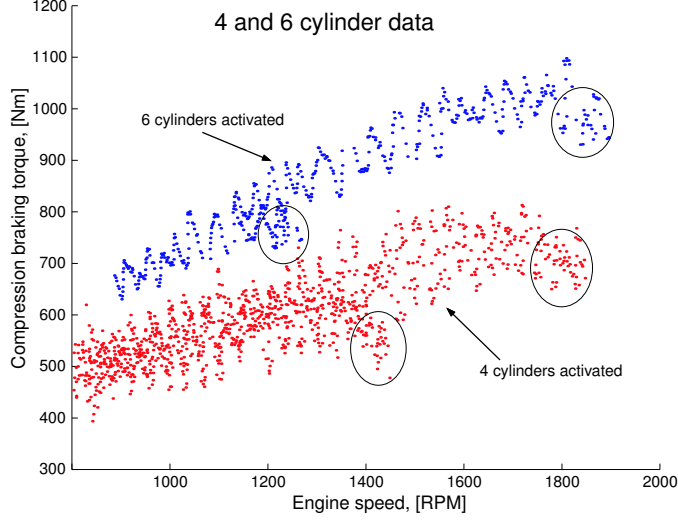


Figure 8: Compression braking torque obtained on the basis of filtered estimate of  $\dot{\omega}$ .

nated together for the purpose of identification. These deviations are caused by the initial transients in the filter that we used to estimate  $\dot{\omega}$ . Since these deviations can obscure the identification results, we can not use the data from the beginning of the experiments and, therefore, a lot of useful data may be missed.

Thus, to avoid the issues associated with 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  $[T^0, T^i]$ , where  $T^i = T^{i-1} + \Delta T$ ,  $i = 1, \dots, N$ , and  $\Delta T$  is the sample period of data acquisition system (20 ms):

$$\begin{aligned} \omega(T^i) - \omega(T^0) &= -\theta_0 \frac{1}{J_t} (T^i - T^0) - \theta_1 \int_{T^0}^{T^i} \frac{1}{J_t} \omega(t) dt \\ &\quad - \frac{C_q r_g^3}{J_t} \int_{T^0}^{T^i} \omega^2(t) dt - \frac{r_g f_r g M}{J_t} (T^i - T^0). \end{aligned} \quad (4)$$

By defining

$$\begin{aligned} y_i &= \omega(T^0) - \omega(T^i) - \frac{C_q r_g^3}{J_t} \int_{T^0}^{T^i} \omega^2(t) dt - \frac{r_g f_r g M}{J_t} (T^i - T^0), \\ \phi_i &= \left[ \frac{1}{J_t} (T^i - T^0), \frac{1}{J_t} \int_{T^0}^{T^i} \omega(t) dt \right], \end{aligned}$$

for  $i = 1, \dots, N$ , and  $\theta = [\theta_0, \theta_1]^T$ , we rewrite (4) as

$$y_i = \phi_i \cdot \theta, \quad i = 1, \dots, N. \quad (5)$$

Let denote

$$\Phi = \begin{bmatrix} \phi_1 \\ \vdots \\ \phi_N \end{bmatrix}, Y = \begin{bmatrix} y_1 \\ \vdots \\ y_N \end{bmatrix}.$$

Then, (4) has the following vector representation:

$$Y = \Phi \cdot \theta$$

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  $\phi_i$  in (4), 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  $\omega^2(T^i)$ ,  $i = 1, \dots, N$  through the Simulink integrator block.

The resulting steady-state dependence of the compression braking torque for the case when two, four and six cylinders are activated are shown in Figure 9 (see Appendix for the regression coefficients).

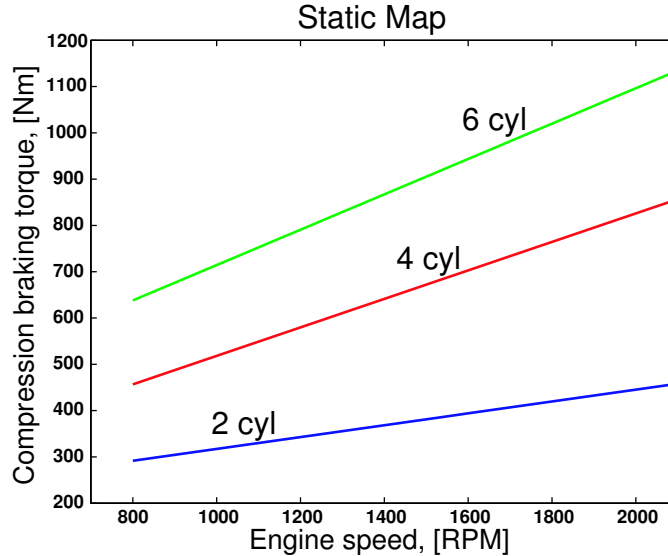


Figure 9: Steady-state engine braking map.

Figure 10 compares the actual values of  $y_i$ 's versus the regressed values  $\hat{y}_i = \phi_i \cdot \hat{\theta}$ . The plots show good accuracy on the fit.

As shown in the next section, the model also matches well the behavior of vehicle and engine speed thereby validating the fits as well.

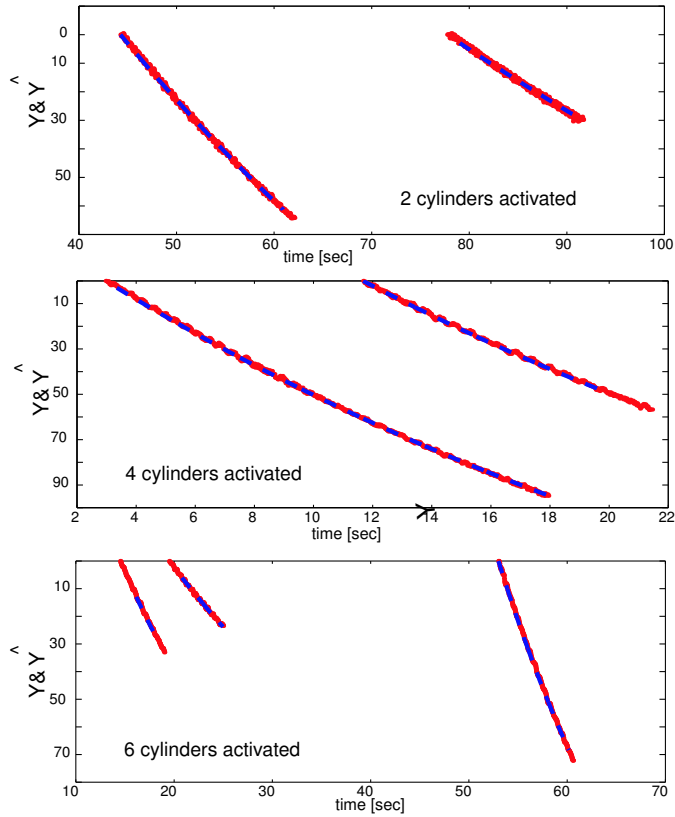


Figure 10: Comparison of the actual values of  $y_i$ 's with the regressed values  $\hat{y}_i$ .

### 3.5 Validation results

The validation results of the resulting regression are shown in Figures 11-13. We compare the measured engine and vehicle speeds with the predicted engine and vehicle speed based on our model. Figure 11 shows the response in the second gear with 2 cylinders activated. Figure 13 shows the response in the third gear with 6 cylinders activated. Figure 12 shows the response in the 3rd gear with four cylinders activated.

Moreover, we have also investigated the effect of the conventional friction brakes on the engine and vehicle speed.<sup>5</sup> The comparison between predicted and measured responses is shown on Figure 14 where the control signal of 2.46 Volt was applied to the friction brake actuator. 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.

<sup>5</sup>The dynamics for the fuel and friction brake actuators were identified during a series of open-loop experiments carried out by UCLA team in September 1998 (see [13] for details).



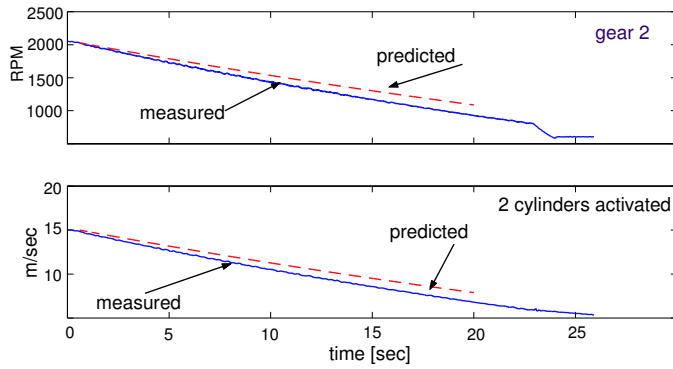


Figure 11: Validation simulation with two activated cylinders shows the trajectories of predicted and measured engine speed.

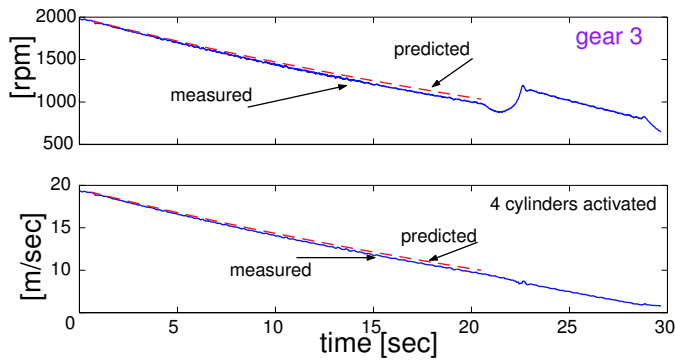


Figure 12: Validation simulations with four activated cylinders show the trajectories of predicted and measured engine speed.

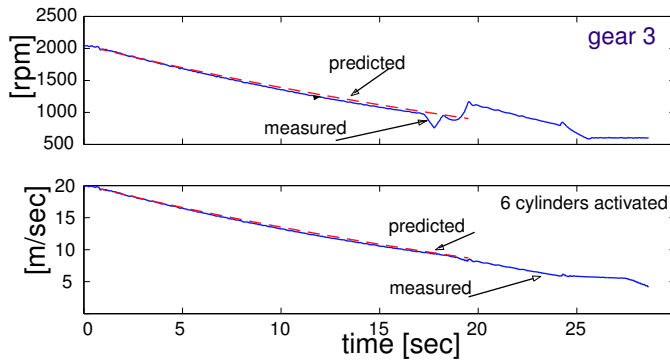


Figure 13: Validation simulations with six activated cylinders show the trajectories of predicted and measured engine speed.

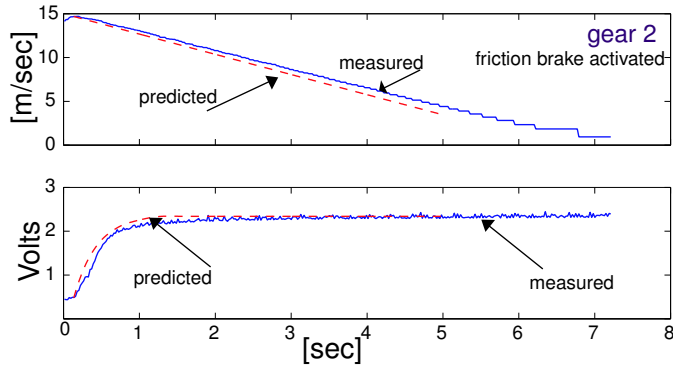


Figure 14: Validation simulations with activated friction signal (2.46V command) show the trajectories of predicted and measured engine speed (upper plot) and brake pressure in Volt at the front left wheel (lower plot).

## 4 Integration of compression brake into speed controller

In October 1998, the UCLA team experimentally tested a fixed-gain PID speed control algorithm that allowed the automated operation of the experimental truck [13]. The PID 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. Several speed tracking closed-loop experiments were performed for different desired speed profiles. The experimental results demonstrated that the fixed-gain PID controller is able to achieve good speed tracking performance for the nominal vehicle loading.

Following the closed-loop longitudinal control experiments conducted by UCLA, we conducted a set of closed-loop experiments in Crow's Landing in August 2001 with the objective to integrate the 3-stage Jake brake with the friction brakes and, therefore, investigate the benefits of using compression brake. The integration strategy assigns high priority to the compression brake because we want to minimize the use of friction brakes and the wear of their friction pads. In our approach we decided to preserve the existing longitudinal feedback speed controller designed by UCLA team that worked quite well. However, we wanted to enable it to operate with both the friction brake and the compression brake.

In this section we first review the nominal PID controller implemented by UCLA for friction brakes only. We then describe our scheme to coordinate the compression brake with conventional friction brakes. The idea of the coordination scheme is to split the total braking demand into two signals corresponding to discrete compression and continuous friction torques. Finally, we report the experimental evaluation of the control scheme with

the compression brake disabled and then for the case of integrated braking. To assess the improvements due to our coordination scheme, we use the same desired vehicle speed profile is used in both cases.

## 4.1 Nominal PID controller for friction brakes

In this section we consider the PID control scheme implemented by UCLA. Figure 15 shows a simplified diagram of this control scheme (see [13] 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 4 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.<sup>6</sup> An upper limit of 4.5 Volt for the voltage passed to the fueling actuator is dictated by the driveability requirements. There is also a lower limit of 1.2 Volt on the voltage passed to the friction brake actuator. It turns out that voltage command lower than 1.2 Volt does not result in any friction braking torque.

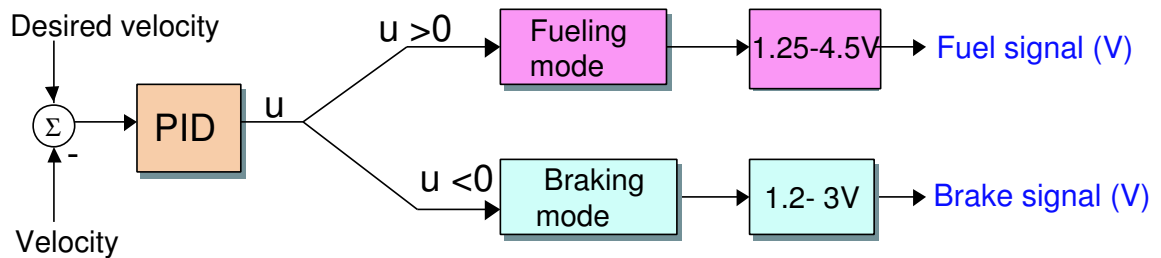


Figure 15: Nominal PID controller diagram.

<sup>6</sup>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 braking command.

## 4.2 Coordination scheme between compression brakes and friction brakes

In this section we describe a scheme for coordinating the compression brake with the friction brake. 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 16 explains how we extracted the compression braking command from the braking voltage command generated by the nominal PID controller in Section 4.1.

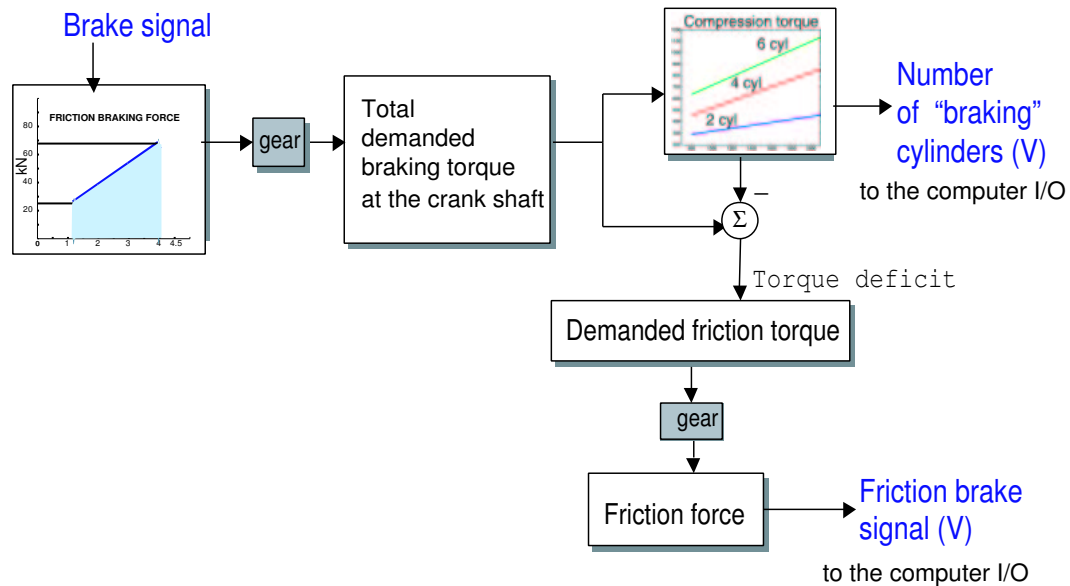


Figure 16: Coordination scheme diagram.

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.

While the basic idea of the coordination scheme is described in Figure 16, 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 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.<sup>7</sup> 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.

### 4.3 Closed-Loop Experiments

In our closed-loop experiments conducted in August 2001 we first evaluated the nominal PID controller with the compression brake disabled. The selected driving profile as shown in Figure 17 requires an 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 18, the voltage command to the friction brake actuator is as high as 2.8 Volt out of maximum of 3.0 Volt.

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 19-20 summarize the closed-loop behavior with the compression brake active. First, the speed tracking performance is completely preserved as is shown in Figure 19. As can be seen from the upper plot in Figure 20, 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

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<sup>7</sup>We already discussed some of the challenges that we encountered when dealing with the transmission and the need for the gear ratio estimator in Section 3.3.

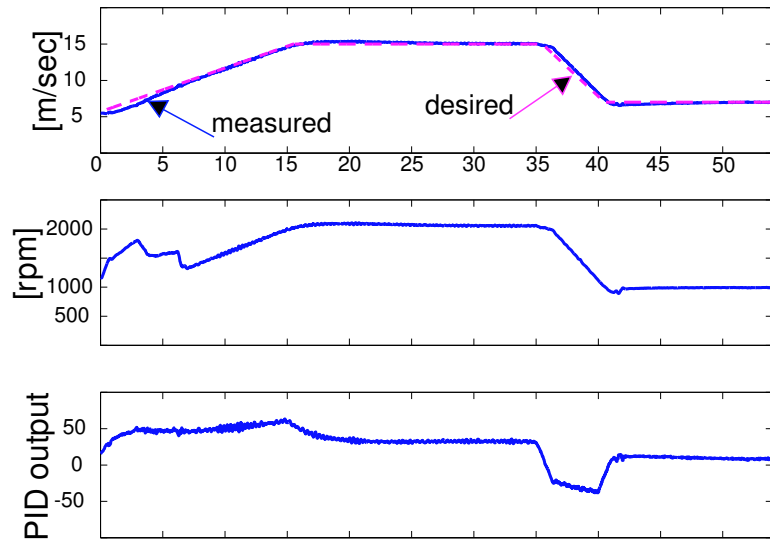


Figure 17: Nominal PID controller performance: trajectories of vehicle speed, engine speed and PID controller output.

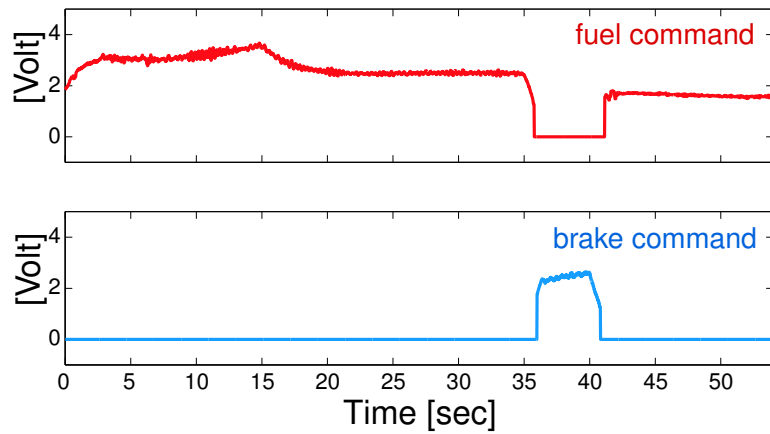


Figure 18: Nominal PID controller performance: trajectories of fuel and brake commands.

shown by the middle plot in Figure 20 which compares the friction brake command for the case when the compression brake is disabled and the friction brake command with enabled compression brake.

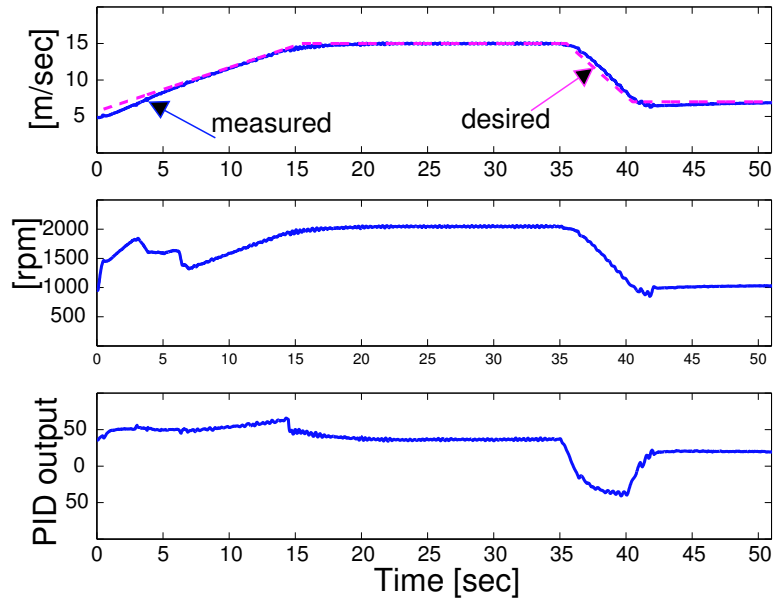


Figure 19: Coordinated PID controller performance: trajectories of vehicle speed, engine speed and PID controller output.

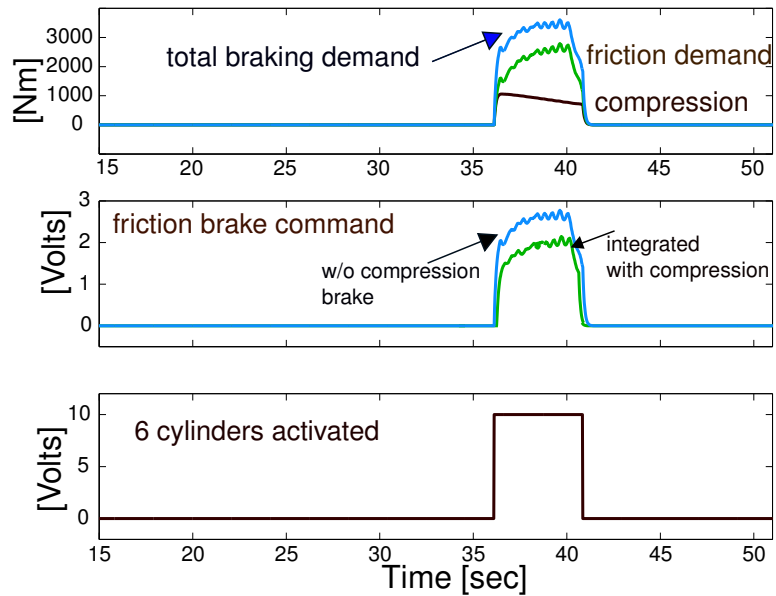


Figure 20: Comparison of the PID controller performance with and without compression brake: trajectories of the torque demands, friction brake command with the compression brake disabled/enabled, and the “braking” cylinders activation signal.

## 5 Conclusion

In this report we described the development of a coordination scheme between compression and friction brakes for a heavy-duty Freightliner vehicle which is used as an experimental platform for the California PATH program. Our approach was to keep the nominal PID speed tracking controller, designed by the UCLA team, intact, but to interpret its nominal friction brake command as an implied braking torque demand, and then split this braking torque demand into a part to be delivered by the compression brake and a part to be delivered by the friction brakes. The split is based on the models for the compression braking torque and friction braking torque that were developed and identified from open-loop experiments on a vehicle conducted at Crows Landing in January, 2000. The validation of the coordination scheme has been undertaken in August, 2001 and showed a sizable reduction in the friction brake usage during braking maneuvers. The reduction of the friction brake usage is beneficial in that the wear of the friction brake can be reduced and its service life can be prolonged.

While the performance of the fixed gain PID controller is quite good for the nominal vehicle parameters, to ensure consistent and robust speed tracking performance in the face of significant vehicle mass and road grade changes an adaptive controller is needed [13]. In our prior work within MOU 393 [3], 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 performance on a grade while avoiding the use of friction brakes altogether. The continuously variable compression brakes represent an advanced technology which is not largely available on production vehicles at the present time. Hence, in our future work we plan to pursue a modification of the adaptive controller to operate the discretely-variable compression brake in coordination with the friction brake as well as experimental validation of the resulting scheme on the test vehicle at Crows Landing.



## 6 Appendix

As described in Section 3.4, we approximate the engine torque applied to the crankshaft during compression braking as a function of the engine speed and the number of activated “braking” cylinders:

$$TQ_{cb} = \gamma_0 + \gamma_1 N_e$$

where  $N_e$  is the engine speed in RPM, and  $\gamma_0, \gamma_1$  have the following values:

**Case 1:** 2 cylinders activated:

$$\gamma_0 = 189.0566;$$

$$\gamma_1 = 0.1281;$$

**Case 2:** 4 cylinders activated:

$$\gamma_0 = 210.4114;$$

$$\gamma_1 = 0.3078;$$

**Case 3:** 6 cylinders activated:

$$\gamma_0 = 332.3492;$$

$$\gamma_1 = 0.3820.$$

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