A Comparative Analysis of Electronic Pedal Algorithms Using a Driver-in-the-Loop Simulator and System Identification of Driver Behavior

Ryan Boris, Chris Vermillion, and Ken Butts

Abstract—In modern automobiles, the driver’s accelerator pedal position is fed to an electronic control unit, which has traditionally interpreted this pedal input as desired throttle angle but can interpret the pedal position in other ways as well. In this paper, we consider three interpretations of pedal position, namely desired throttle angle, desired net engine torque, and desired wheel torque. We design separate controllers for each pedal interpretation. For each controller, we evaluate drivers’ abilities to simultaneously track a speed setpoint and keep high frequency vehicle acceleration to a minimum, relying on classical control theory to come up with preliminary hypotheses and a driver-in-the-loop simulator for determining which hypotheses hold. We also perform parametric system identification for each of the subjects used in this study, for each of the controllers, to assess any differences in driver behavior across the different controllers. We have concluded, for the vehicle platform studied here, the engine torque control provides comparable performance to direct throttle control, with improved drivability, whereas both throttle and engine torque control yield performance that is far superior to wheel torque control.

I. INTRODUCTION

The automobile represents an inherently modular, inner-loop/outer-loop control system, wherein an outer loop human driver interacts with inner loop ECUs through the manipulation of an accelerator pedal. Today, an electronic throttle is used in most new vehicles, where the pedal position is an input to an ECU. The pedal input need not be interpreted as desired throttle angle (as has traditionally been the case); it can be interpreted as desired engine torque, desired wheel torque, or even desired engine/wheel power. This provides design freedom but also brings up a debate regarding what the driver really wants when he/she depresses the pedal. Several papers have proposed control strategies where the pedal position is interpreted as desired engine torque [1]-[5], and the engine control inputs are manipulated in order to track this desired torque closely while obtaining good fuel economy and emissions. Other papers, particularly those within the hybrid vehicle literature, such as [6]-[7], interpret the driver’s pedal position as desired wheel torque. For split power hybrid vehicles, this interpretation arises from practical necessity, since the controller needs a torque demand as a basis for determining the torque split between the electric motor and the internal combustion engine. While alternative pedal interpretations are motivated (or even necessitated) in the literature, the performance tradeoffs associated with different pedal interpretations has not been studied in detail. This paper presents this detailed study for a particular engine and vehicle platform; a more general study is a topic for future work.

The performance of each controller will depend upon the properties of the plant and controller itself, but also on the ability for the human driver to interact with the controller and plant to produce an acceptable vehicle response. Results from human-in-the-loop modeling and control theory lead to varying hypotheses regarding what “should” occur when other pedal interpretations are considered. If driver maintains constant behavior regardless of the controller, a reasonable prediction of which pedal interpretation will work best could be made by examining the frequency response characteristics of the combined plant and controller. On the other hand, the famous “crossover model” of [8]-[10], which was supported by the results of many papers including [11], hypothesizes that the open loop transfer function of a driver cascaded with a plant has an invariant 20 dB/dec roll-off at crossover frequency, regardless of the plant. Additionally, preview/predictive models [12], which contain both a pursuit (feedforward) and compensatory (feedback) term, and do not necessarily produce results that coincide with those of the crossover model. The complex nature of the problem at hand, involving different controllers as well as a human in the loop, along with the diversity of results on how a driver interacts with the vehicle, motivates further investigation for our specific problem.

In this paper, we first present designs for separate controllers for three pedal interpretations, namely, throttle, engine torque, and wheel torque. For each controller, we analyze the properties of the dynamic relationship between the pedal position and the vehicle outputs, where this dynamic relationship is referred to here as the effective plant. This analysis, which is based on classical control theory, allows us to make some preliminary hypotheses regarding what we expect to happen when a driver is placed in the loop. In order to assess the merits of each potential pedal interpretation and validate (or invalidate) our hypotheses, we evaluate longitudinal driver/vehicle performance based on results we obtained with a driver-in-the-loop simulator. Finally, in order to gain a better understanding of how the
drivers compensate for variations in the effective plant (if at all), we perform system identification of driver behavior, based on a parametric driver model.

The paper is organized as follows. In Section II, we discuss the vehicle model and the problem formulation, including the key variables that will be used to quantify driver performance. In Section III, we describe the control design for each of the three pedal interpretations and analyze the effective plants resulting from each control strategy. In Section IV, we provide driver-in-the-loop simulation results. Finally, in Section V, we present driver identification results and an analysis of driver behavior.

II. VEHICLE MODEL ESSENTIALS AND PROBLEM FORMULATION

A. Essential Model Components

Our work considers the longitudinal dynamics of a full sized-vehicle with a 2.4 liter, 4 cylinder engine and 4-speed automatic transmission. The overall model description is given in Fig. 1, which shows the signal flow between the engine, transmission/driveline, and vehicle. The signals in Fig. 1 are given in Table I.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Units</th>
</tr>
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<tbody>
<tr>
<td>( \theta )</td>
<td>throttle angle</td>
<td>rad</td>
</tr>
<tr>
<td>( IPW )</td>
<td>fuel injection pulse width</td>
<td>( \mu ) s</td>
</tr>
<tr>
<td>( VVT^{cmd} )</td>
<td>commanded valve timing</td>
<td>degrees adv.</td>
</tr>
<tr>
<td>( \delta )</td>
<td>spark advance</td>
<td>degrees BTDC</td>
</tr>
<tr>
<td>( T_{net}^{eng} )</td>
<td>net engine torque</td>
<td>N-m</td>
</tr>
<tr>
<td>( \omega_{eng} )</td>
<td>engine speed</td>
<td>rad/s</td>
</tr>
<tr>
<td>( GR )</td>
<td>transmission gear ratio</td>
<td>-</td>
</tr>
<tr>
<td>( T_{pump} )</td>
<td>torque converter pump torque</td>
<td>N-m</td>
</tr>
<tr>
<td>( F_{tire} )</td>
<td>tire/road force</td>
<td>N</td>
</tr>
<tr>
<td>( v_{veh} )</td>
<td>vehicle speed</td>
<td>m/s</td>
</tr>
<tr>
<td>( \omega_{tire} )</td>
<td>tire rotational speed</td>
<td>rad/s</td>
</tr>
<tr>
<td>( \theta_{tire} )</td>
<td>tire rotational position</td>
<td>rad</td>
</tr>
<tr>
<td>( a_{HF} )</td>
<td>high frequency vehicle accel.</td>
<td>m/s^2</td>
</tr>
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</table>

Fig. 1. Main diagram of the model components and interconnections between the engine, driveline, and vehicle models.

B. Problem Description and Key Performance Variables

The interpretation of pedal input as either a throttle demand, engine torque demand, or wheel torque demand each result in different powertrain control strategies. This leads to different effective plants as seen by the driver, where the term effective plant refers to all the dynamics between the accelerator pedal and observed variables (such as vehicle speed and acceleration). This is exhibited in Figs. 2-4, which show the control strategies and resulting closed loop block diagrams for throttle control, engine torque control, and wheel torque control, respectively. Note that \( p \) represents the driver’s pedal input and ranges from 0 to 1.

For our research, two signals will used to characterize performance, namely tracking error and high frequency acceleration. Tracking error, \( e \), is simply characterized by the difference between the actual vehicle speed, \( v_{veh} \), and the speed setpoint, \( v_{des} \), as follows:

\[
e(t) = v_{veh}(t) - v_{des}(t).
\]

In order to capture this high-frequency acceleration, \( a_{HF} \), we pass the vehicle acceleration, \( a_{veh} \), through a lead filter as follows:

\[
a_{HF}(s) = \frac{s + a}{s + b} a_{veh}(s),
\]

\[
a = 5,
\]

\[
b = 50.
\]

Fig. 2. Main diagram for the throttle control strategy.

Fig. 3. Main diagram for the engine torque control strategy.
III. CONTROL DESIGN AND ANALYSIS

A. Controller Design for Throttle, Engine Torque, and Wheel Torque Interpretations

For each different pedal interpretation, throttle angle (θ) is adjusted in a different way, in order to achieve either a direct throttle input or the tracking of either desired engine torque or desired wheel torque. In this study, the remaining control inputs, namely fuel injection pulse width (IPW), spark advance (δ), variable valve timing command (VVTcmd), and transmission gear ratio (GR) are determined using standard control designs that are common across each pedal interpretation.

For throttle control, throttle angle is simply taken to vary linearly with the pedal input, namely:

\[ \theta = 90p. \]  

(5)

Because the throttle controller is a static function of pedal position, with no feedback, it results in no alteration of the original engine/driveline/vehicle dynamics.

Remark 3.1: While the pedal-to-throttle mapping of (5) is linear, many cars use a nonlinear mapping, which varies widely from car-to-car. The effect of ones choice of nonlinear mapping is outside the scope of this work; however, the invertible static mapping from pedal to throttle, whether linear or nonlinear, does not result in a fundamental alteration of the original engine/driveline/vehicle dynamics.

The other two interpretations (engine torque and wheel torque) involve the use of feedback and therefore alter the original system dynamics. In order to maintain practicality in an automotive environment, it is essential that this feedback structure remain simple and intuitive for an operator to tune, while providing enough degrees of freedom to realize the performance potential of the control strategy. It was found that a PI controller, with the augmentation of a lead-lag element, could be used to achieve this compromise.

For engine torque control, the control law is given by:

\[ \theta(s) = \frac{b_2s^2 + b_1s + b_0}{a_2s^2 + a_1s}(T_{\text{eng},\text{des}}(s) - T_{\text{eng}}^\text{net}(s)), \]  

(6)

where engine torque, \( T_{\text{eng}}^\text{net} \), is estimated with a nonlinear engine observer, as follows:

\[ \dot{x}_{\text{eng}} = f(x_{\text{eng}}, \text{IPW}, \delta, \text{VVT}_{\text{cmd}}, \omega_{\text{eng}}), \]  

(7)

\[ \dot{T}_{\text{eng}} = g(x_{\text{eng}}, \text{IPW}, \delta, \text{VVT}_{\text{cmd}}, \omega_{\text{eng}}), \]  

(8)

where \( x_{\text{eng}} \) represents the states of the engine model.

The same control law (but with different gains) is used for wheel torque control, namely:

\[ \theta(s) = \frac{b_2s^2 + b_1s + b_0}{a_2s^2 + a_1s}(T_{\text{des}}(s) - \hat{T}_{\text{wheel}}(s)), \]  

(9)

where the estimated (and filtered) wheel torque, \( T_{\text{wheel}}^\text{filt} \), is given according to:

\[ \hat{T}_{\text{wheel}}(s) = \frac{1}{\tau_{\text{filt}} + \frac{1}{m_{\text{veh}}R_{\text{tire}}^2}}a_{\text{veh}}(s) + F_{\text{drag}}(s), \]  

(10)

where \( F_{\text{drag}} \) contains a combination of aerodynamic and rolling resistance drag terms.

B. Analysis of the Effective Plants Under Different Pedal Interpretations

Figs. 5 and 6 show the frequency domain behavior for linearizations of each effective plant, from pedal (\( p \)) to speed (\( \omega_{\text{veh}} \)) and pedal (\( p \)) to high frequency acceleration (\( \alpha_{HF} \)), respectively. Fig. 5 shows that the two torque controllers contribute to a slower effective plant than throttle control, with the wheel torque controller contributing the greatest decrease in effective plant bandwidth. Fig. 6 shows that the throttle and wheel torque controllers suppress the effect of pedal variations on high-frequency vehicle acceleration.

If the driver’s behavior remains the same under each of the pedal interpretations, the examination of Figs. 5 and 6 leads to two hypotheses:

1) Direct throttle control will lead to the best results in terms of our setpoint tracking objective, with engine torque control yielding slower responsiveness, and wheel torque control yielding slower responsiveness yet.

2) The largest high-frequency vehicle accelerations will be experienced under direct throttle control.

However, the crossover model of [8] and [9], would suggest that the driver will compensate for variations in the effective plant. In order to assess which hypothesis holds in this particular case (or to what extent each hypothesis holds), we conducted driver-in-the-loop simulation studies, followed by system identification of drivers’ behavior, which are detailed in the following sections.

IV. DRIVER-IN-THE-LOOP SIMULATION RESULTS

A. Overall Experimental Setup

The experimental setup consists of a physical driver simulator, a target PC, and a host PC, as depicted in Fig.
Fig. 5. Bode plot of the effective plant from pedal \((p)\) to vehicle speed \((v_{veh})\).

Fig. 6. Bode plot of the effective plant from pedal \((p)\) to high frequency vehicle acceleration \((a_{HF})\).

Fig. 7. Overall experimental setup, including the driver, pedal assembly, host and target PCs, and Quanser Q4 card.

TABLE II

<table>
<thead>
<tr>
<th>Metric</th>
<th>Throttle</th>
<th>Engine Torque</th>
<th>Wheel Torque</th>
</tr>
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<tbody>
<tr>
<td>(J_t)</td>
<td>277.6</td>
<td>321.8</td>
<td>380.2</td>
</tr>
<tr>
<td>(J_a)</td>
<td>689.0</td>
<td>627.5</td>
<td>744.1</td>
</tr>
</tbody>
</table>

at any given time.

B. Tracking Performance

Average tracking performance (over the 10 drivers) was best for the throttle interpretation followed by engine torque then wheel torque, as exhibited in Table II, where:

\[
J_t = \int_{t_0}^{t_f} (v_{veh}(t) - v_{des}(t))^2 dt. \tag{11}
\]

For 70 percent of the participants the engine torque tracking performance was better than it was for wheel torque performance. To provide qualitative insight, Fig. 8 shows the resulting speed traces for a single test subject whose results were very similar to the average behavior seen.

C. High Frequency Acceleration (Drivability) Performance

Average acceleration performance was best for engine torque interpretation followed by throttle interpretation then wheel torque interpretation, as is exhibited in Table II, where:

\[
J_a = \int_{t_0}^{t_f} a_{HF}^2(t) dt, \tag{12}
\]

and shown for a typical driver in Fig. 9. A zoomed in version of Fig. 9 is given in Fig. 10 to give an indication of the performance of each controller during non-gear shift events (which create large acceleration spikes and somewhat mask the remainder of the plot). 70 percent of the participants obtained the best score for engine torque interpretation.
D. Summary of Results

With the exception of high frequency acceleration under wheel torque control, our experimental results rank the performances of the various controllers in the same order that they would be ranked under the hypothesis that the driver did not adjust his behavior at all. However, the fact that the rank order remains the same does not mean that the drivers made no adjustment whatsoever; rather, it indicates that the adjustment made by the drivers was insufficient to overcome the differences between the different controllers. The next section analyzes driver behavior in order to determine the extent to which drivers adjusted their behavior across the different controllers.

V. IDENTIFICATION OF DRIVER BEHAVIOR

A. System Identification for Characterizing Driver Behavior Under Various Pedal Interpretations

To understand the way that drivers adjust their behavior for each controller, we use standard system identification techniques to identify coefficients to a driver model transfer function from $v_{des}$ and $e$ to $\theta_{pedal}$, where the transfer function structure is given as follows:

$$
p(s) = \frac{N_1(s) q_{des}(s) + N_2(s) q(s)}{D(s)} c(s),
$$

$$
N_1(s) = b_{13} s^3 + b_{12} s^2 + b_{11} s,
$$

$$
N_2(s) = b_{23} s^3 + b_{22} s^2 + b_{21} s + b_{20}.
$$

This structure corresponds to a standard interpretation of driver behavior, which includes a pursuit (feedforward from $v_{des}$ to $p$) term and a correction (feedback from $e$ to $p$) term.

In order to perform the system identification, we multiply each side of (13) by $D(s)$, where $\Lambda(s)$ is a stable polynomial of the form:

$$
\Lambda(s) = \lambda_3 s^3 + \lambda_2 s^2 + \lambda_1 s + \lambda_0.
$$

This results in a causal expression in terms of $v_{veh}$, $v_{des}$, $\theta_{pedal}$ and their derivatives, from which transfer function coefficients can be identified using standard least squares.

B. Driver Identification Results

Figs. 11 and 12 show the resulting identification results for an example driver (the same driver that was used for Figs. 8 and 9). Most other drivers yielded qualitatively similar results, suggesting that no clear attempt is being made by drivers to compensate for different effective plants. Furthermore, Fig. 13 shows the transfer function from e to $v_{veh}$ (the effective plant) under each controller. These transfer functions do not exhibit a 20 dB/dec rolloff at crossover frequency, nor is the rolloff consistent between different controllers, demonstrating that the crossover model does not hold under this particular study.
VI. CONCLUSIONS AND FUTURE WORK

The results of this paper have demonstrated, for a particular vehicle, that the interpretation of pedal position as desired engine torque serves as an attractive alternative to the traditional interpretation as throttle, depending on one's relative interests in tracking a setpoint vs. managing drivability, whereas wheel torque control leads to deteriorated performance. Driver identification has demonstrated that, contrary to the crossover model, drivers did not compensate for the slow wheel torque control in this particular study. Because this study only applies to a particular vehicle platform, future work will aim to generalize the results of this study.

REFERENCES