Nonlinear Algorithms for Automotive Engine Control

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ABSTRACT: This article discusses nonlinear automotive engine control methods that would provide a more pleasing shift. In the course of a typical automotive transmission upshift, there is an extreme decrease in the transmission's output torque during the torque-phase of the shift; this is due to the torque being transferred from a high to a low gear ratio. In the speed phase, due to inertial effects and the negative slope of the friction coefficient, the output torque rises and peaks when the shift ends, changing engine speed. Modulating engine torque can smooth these output torque transients. The sliding-mode control method is used, and a solution to the multivariable control problem (i.e., coordinated throttle and spark control) is presented. A speed-control approach is used, and relationships between speeds and desired torque (or acceleration) trajectories are discussed.

Introduction

Modulation of engine torque during a shift can help smooth the shift transient. This article discusses the sliding-mode approach to this engine control problem. These control algorithms can also be used for traction control or other applications in which closed-loop torque or speed modulation is desired. To lay the groundwork for these algorithms, a short discussion of the parts of the shift is appropriate.

Consider the following simplified conceptual schematic of a two-speed transmission. In this illustration, \( \omega_e \) is the engine speed, \( \omega_w \) is the torque converter turbine speed, \( \omega_t \) is the transmission reaction carrier speed, \( \omega_m \) is the wheel speed, and \( R \)'s represent first, second, and differential gear ratios. In the first gear, all the torque is transferred through the first clutch, and the second clutch carries no torque. As the shift starts, the second clutch pressure increases so that the input torque is now split between both clutches. The first clutch is still locked, and the second clutch is slipping. This part of the shift is called the torque phase; it is characterized by a drop in the transmission output shaft torque.

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From this description, it is clear that one cannot avoid the torque-phase of the shift in an actual transmission (i.e., infinite bandwidth sensors and actuators would be necessary to instantly bring the first clutch torque to zero and instantly increase the second clutch torque for a smooth transient). Also, in order to produce a smooth shift, the engine torque must be increased during the torque phase to fill the torque hole caused by the different gear ratios, since inertia effects are negligible in this phase. During the speed phase, the engine torque can be decreased to offset the increasing transmission output torque, caused by inertial effects and increased engine torque of its mid-speed range. After the slip-speed across the second clutch goes to zero, the engine torque can be increased to maintain a smooth transmission output shaft torque. This required engine torque modulation (or speed control) is the subject of this article. Runde [1] has developed dynamic transmission models and calculated examples of the ideal engine output torque profiles for a smooth shift.

Several automobile manufacturers and suppliers are conducting research in the power train control area, with application to power train control [2]-[5] and traction control [6]. Most current engine control algorithms are based on steady-state maps; however, this article develops algorithms for dynamic compensation.

Engine Modeling for Control Development

A four-state, lumped parameter, dynamic engine model has been developed by the au-
to assess controller designs or system re-

ders. It can be used as a simulation model to study systems the throttle body, idle air control, E.G.R. control, intake manifold, fuel delivery, torque production, and rotational dynamics. A block diagram of the engine model, from controls to indicated torque, is shown in Fig. 3. Mass flow rates are given as \( m \), with subscripts for air (\( a \)) and exhaust gas recirculation (\( egr \)) and fuel (\( f \)) in (\( i \)) and out (\( o \)) of the intake manifold. Controls are throttle angle (\( \alpha \)), digital exhaust gas recirculation (E.G.R.) valve signal, idle air control (I.A.C.) signal, fuel injector commands (\( \eta_\text{f} \)), and spark advance (Spark). Other variables shown are stagnation pressures and temperatures of prethrottle air (\( T_a, P_a \)) and exhaust manifold (\( T_i, P_i \)), engine speed (\( \omega_e \)), intake manifold pressure (\( P \)), air/fuel ratio of combustion (\( A/F \)), and engine indicated torque (\( T_i \)). The algorithms in this article will provide control laws for throttle angle (\( \alpha \)) and spark advance (Spark), while fuel and E.G.R. commands will be given by the production controllers.

This engine model can be used in three ways. It can be used as a simulation model to assess controller designs or system responses as a validation model to compare model and actual outputs for parameter identification (given the inputs), and it can be used as a part of the control algorithm within a vehicle. These models vary by what inputs are available to the model (such as intake manifold pressure, engine speed, etc.), and what is the required output of the model (i.e., brake torque, engine/dynamometer shaft torque, etc.). Only the simulation and control models will be considered in this article. Details of the engine model are given in [7]-[9]. The equations for the gross indicated stoichiometric fuel conversion efficiency come from the work of Chang [10]. The torque converter inertia is combined with the engine inertia as a lumped parameter. The other inertias and models (i.e., torque converter, transmission, etc.) can then be added on this model, which effectively provides an additional torque on the engine/pump inertia, to vary the engine speed.

The control model is the simplest model, which utilizes all available measurements to estimate flows and torques. Airflow out of the intake manifold can utilize the mass flow measurement of airflow rate coming into the intake manifold through the throttle (\( m_{th} \)) and the idle air control valve (\( m_{iAC} \)), intake manifold pressure (\( P \)), estimates of the exhaust gas recirculation flows in and out of the manifold (\( m_{e_g} \) and \( m_{e_g0} \)), and the ideal gas law (\( R \) is the gas constant, \( T \) is the mean manifold temperature, and \( V \) is the manifold volume), to give:

\[
m_{th} + m_{iAC} = \text{mass airflow sensor output}
\]

\[
P = \text{measured intake manifold pressure}
\]

\[
dm_{e_g0}/dt = (m_{e_gi} - m_{e_g0})/\tau
\]

\[
m_{e_g} = m_{e_g} + m_{e_gi} - m_{e_g0} + (P \cdot V) \cdot R \cdot T^{-1} \cdot [(T/T_0) - (P/P_0)]
\]

\[
= m_{e_g} + m_{e_gi} - m_{e_g0} - (V \cdot \dot{P})(R \cdot T).
\]

Since the time constant of the E.G.R. is a mixing or flow-related phenomena, the time constant (\( \tau \)) should definitely be a function of engine speed (\( \omega_e \)) and volumetric efficiency (\( \eta_v \)). By comparing the response of this filter model with a partial pressure model (see [8] for details), the authors found that a close approximation can be obtained by choosing the time constant as:

\[
1/\tau = 0.095 \cdot \eta_v \cdot \omega_e
\]

Speeds are usually inputs into the model, and the brake torque is the model output (or turbine torque in the torque converter). These simple engine models can be inverted to give an estimate of the desired engine control (throttle or spark advance). The sliding-mode formulation then provides a closed-loop feedback to compensate for modeling errors.

**Speed-Error Control Formulation**

Before discussing the control algorithms, it is useful to look at the strengths and shortcomings of various approaches to the control problem. Also, preliminary groundwork must be established to understand why the authors use the speed-error formulation. In the sliding-mode nonlinear control formulation, the control engineer defines an error, which is the difference between a measured time-domain trajectory and the desired trajectory. Assume the nonlinear state-space equations are in the following form:

\[
dx/dt = f(x, t) + g(x, t) \cdot u^*
\]

The error (\( e \)) is then defined as the difference between the nth state (\( x_n \)) and the desired trajectory for that nth state (\( x_{nd} \)) or:

\[
e = x_n - x_{nd}
\]

The sliding surface is defined when a selected linear combination of the error and its derivatives go to zero, or:

\[
S = \alpha_1 \cdot e + \alpha_2 \cdot d e/dt + \alpha_3 \cdot d^2 e/dt^2 + \ldots
\]

A Lyapunov-like function (may not contain all states), where \( \eta \) is positive definite, then assures global stability (assuming there are no modeling errors) by making the sliding surface attractive when \( S \) is nonzero. If model errors occur and are bounded by some known function, then the formulation can be ad-

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*Throughout this article, the notation \( f(x, t) \) represents a function of parameters \( x \) and \( t \), \( \omega_e(t) + \Delta \omega_e \) represents \( \omega_e \) evaluated at \( t + \Delta \omega_e \), and \( e(t - t_o) \) represents the product of \( e \) and \( t(t - t_o) \). If the parameters are multiplied, there will be a dot; otherwise, the parentheses represent a functional relationship. Also, a "d" will be added to a subscript if it is a desired variable, such as a desired nth state trajectory (\( x_{nd} \)) as in Eq. (7).*
justed appropriately to still assure global sta-
tility [9], [11]. In the following equations, a boundary layer of thickness $\phi$ [12] is est-
established around the sliding surface in order to alleviate any chatter that results from a
digital or discrete-time implementation of this controller.

$$S = \eta \cdot \text{sat}(S/\phi)$$

sat$(S/\phi) = S/|S|,$ if $|S| \geq \phi$

= $S/\phi,$ if $|S| < \phi$ (10)

The following figure shows how chatter can be eliminated by incorporating boundary layers and illustrates the path of the error function [Eq. (8)] for a digital implementation with and without boundary layers. Because the implementation is not continuous, the error function does not slide along the surface but is always attracted to it at each time-step, causing chatter. Within the boundary layer, the error function dynamics respond as a stable (if $\eta/\phi > 0$) first-order lag filter driven by the model errors; so chatter is eliminated if boundary widths are appropriately chosen [see Eq. (33)]. In this article, the sliding surface will be defined only in terms of the error $[i.e., \alpha(t) = \alpha(t - 1),$ and all other $\alpha$’s are 0 in Eq. (8)]. This yields a sliding-mode equation of:

$$\dot{x}_e - \dot{x}_{ref} = \eta \cdot \text{sat} \left( \frac{x_e - x_{ref}}{\phi} \right)$$ (11)

From the introductory discussion, it seems intuitive to define the error in terms of torques, since the smoothness of the output torque is a measure of pleasing shift quality. There are several practical problems associated with this approach (used by Harrigan [13] and Allen [14] for engine control). If the sliding surface is defined in terms of engine

The resulting sliding mode equation is given as:

$$S = \dot{T}_b - \dot{T}_{ind} = -\eta \cdot \text{sat} \left( \frac{(\dot{T}_b - \dot{T}_{ind})}{\phi} \right)$$ (14)

This equation requires the brake torque $\dot{T}_b$ to be differentiated with respect to time; then, the rate of change of the mass airflow out of the intake manifold (acceleration of air) is needed. This is calculated from differ-
entiating the intake manifold Eq. (4). To implement this control algorithm with the control model, $m_{air}, m_{eng}, m_{eng},$ and $P$ would have to be measured or estimated, which does not appear feasible. The basic problem is that, although the model predicts the mean-torque well within its bandwidth, one must be extremely careful when using time deriv-
atives of the model to predict engine events (prediction results using the derivatives of a mean-
torque predictive model have extremely large errors because it is an inappropri-
ate use of a mean-predictive model). An-
other problem is that the engine brake torque would have to be measured or estimated since it is the feedback parameter. Because of the relatively high cost and low reliability of most torques sensors this is not a cost-effective solution. These problems are the driving force behind the development of the speed-
control algorithms discussed here and in Refs. [9], [16]. This engine model is also used by Allen [14] and Cho [15] for fuel control, and similar problems arise in their formulation. In their fuel control work, the sliding surface is defined as a function of the air-and-fuel flow rate out of the intake manifold to maintain a stoichiometric air/fuel ratio (Eq. 3.4.1.4 of Allen and Eq. 20 of Cho), or:

$$S = m_{air} - (A/F)_{hok} \cdot m_{air}$$ (15)

Upon differentiating $S$ to form the sliding-mode equation, the same implementation problems as stated above appear (i.e., the need to measure or estimate $m_{air}, m_{eng},$ and $P$). This article avoids these problems and addresses the multivariable engine control problem by defining the sliding surface so that derivatives of the mean-torque pre-
dictive model do not have to be used in control algorithms. Also, speed measurements are very easy to obtain, and the transducers are cost-effective and dependable.

The speed-error control approach requires that a desired shift be defined in terms of component speeds. Neglecting driveline windup and gear backlash, the reaction car-
dier speed is proportional to wheel speed. If the reaction carrier acceleration is held con-
tant throughout the shift at the value of the beginning of the shift, then the shift would not be detectable to the driver (except for engine and transmission sounds). The sliding surface for the carrier can be defined as a function of the carrier speed ($\omega_c$) and the desired carrier speed (maintaining the initial carrier acceleration $\varphi$ throughout the shift), where $t_1$ is the time at the beginning of the speed phase:

$$\varphi = \omega_c(t_1)$$ (16)

This sliding surface is given as:

$$S = \omega_c - \omega_c(t_1) \cdot \varphi \cdot (t - t_1)$$ (17)

This surface leads to the control of the second clutch, or $T_{C2}$. The turbine speed $\omega_t$ is given at the beginning and end of the speed phase because the first clutch slip-speed is zero at the start and the second clutch slip-speed is zero at the end of this phase, or:

$$R_1 = \omega_t(t_1)$$ (18)

$$R_2 = \omega_t(t_1 + t_{shift})$$ (19)

In the simplest approach to determining the desired turbine speed trajectory, these end points $[\omega_t(t_1)$ and $\omega_t(t_1 + t_{shift})]$ can be linearly interpolated where the initial turbine acceleration ($\xi$) is given (below $t_{shift}$ is the shift duration).

$$\omega_d = \omega_t(t_1) + \xi \cdot (t - t_1)$$ (20)

$$\xi = \omega_t(t_1 + t_{shift}) \cdot \left[ R_2^{-1} - R_1^{-1} \right] + \varphi R_2$$ (21)

The turbine speed sliding surface is then given as a function of speed-error, or:

$$S = \omega_t - \omega_d$$ (22)

More sophisticated desired turbine speed trajectories can be devised, but this discussion describes how the desirable shift can be de-

fined in terms of component speeds. J. H. Park’s (a Ph.D. Research Assistant at M.I.T.) transmission control work involves connecting the turbine speeds at the begin-
ning and end of the speed phase [Eqs. (18) and (19)] with a higher-order polynomial in order to try to match anticipated post-shift carrier accelerations.
Sliding-Mode Engine Control Algorithms

The engine controls can be used to modulate engine speed and turbine speed, and both of these control algorithms will be given in this article. In this article, throttle angle and spark advance will be the two control parameters. Fuel flow rate will be controlled by production fueling algorithms, where throttle angle will be modulated, and fuel will be added to match the resulting airflow. The carrier speed will typically be controlled by modulating clutch pressures.

To control the engine speed ($\omega_e$), the sliding surface can be defined in terms of engine speed-error, or:

$$S_1 = \omega_e - \omega_{sd}$$ (23)

This function can be differentiated with respect to time to give the sliding-mode equation, where $T_p$ is the engine indicated torque, $T_{fp}$ is the engine friction and pumping torque, $T_z$ is the torque converter pump torque, and $J_e$ is the engine and pump polar moment of inertia, or:

$$dS_1/dt = -\eta_1 \cdot \text{sat} [(\omega_e - \omega_{sd})/\phi_f] = (T_p/J_e) - (T_{fp}/J_e) - (T_z/J_e) - \omega_{sd}$$ (24)

By using the engine model equation for indicated torque [Eq. (13)] and substituting a simple torque converter model by Kotwici [17] for pump torque ($T_p$), this equation becomes:

$$-\eta_1 \cdot \text{sat} [(\omega_e - \omega_{sd})/\phi_f] = [K_e \cdot TF \cdot m_{in}(t - \Delta t)] 
\cdot AFI(t - \Delta t) \cdot SL(t - \Delta t)/
(J_e \cdot \omega_e(t - \Delta t)) \nonumber 
- (T_{fp}/J_e) - [(\alpha_2 \cdot \omega_e^2 + \beta_2) 
\cdot \omega_e \cdot \gamma_2 \cdot \omega_h(t)/J_e] - \omega_{sd}$$ (25)

In this equation, the engineer can substitute his/her favorite torque converter model to estimate pump torque.

Equation (25) can be used to calculate engine spark advance or throttle angle, given a desired engine speed trajectory ($\omega_{sd}$). The spark influence function ($SI$) is a normalized function of spark advance that represents how indicated torque is affected as a function of the spark advance from MBT. Spark control will be limited to being retarded from MBT. This will avoid knock problems associated with advanced spark timing, particularly common when the spark is advanced from MBT, and the spark influence function is monotonic and one-to-one (in spark advance) so that it can be inverted to give the control. Given the engine model, current spark advance is calculated. $\Delta t_s$ is the time delay from the intake valve closing to torque production, and $\Delta t_t$ is the delay from spark to torque production.

$$S_A = MBT - 51.289 \cdot \sqrt{1 - \Psi_1 \cdot \Gamma_1}$$ (26)

$$\Psi_1 = \omega_e(t - \Delta t)/[K \cdot TF \cdot AFI(t - \Delta t)] 
\cdot [m_{in}(t - \Delta t) - (T_f/P(t - \Delta t))/
R \cdot T(t - \Delta t)]$$ (27)

$$\Gamma_1 = T_{fp}(t + \Delta t) + T_z/(t + \Delta t) + J_e 
\cdot \omega_e(t + \Delta t) 
- \eta_1 \cdot J_e \cdot \text{sat} [(\omega_e(t + \Delta t) 
- \omega_{sd}(t + \Delta t))/\phi_f]$$ (28)

$$\Delta t = \Delta t_s - \Delta t_t$$ (29)

$\Psi_1$ is the inverse of the engine indicated torque if the spark were at MBT, and $\Gamma_1$ contains the ancillary torques on the engine inertia as well as the desired engine acceleration (from differentiating Eq. (23)) and the sliding-mode feedback term (for error compensation). Note that these are noncausal equations (because of $t + \Delta t$). However, they may be used as causal equations (assume $\Delta t_t = 0$) since the future prediction time-step $\Delta t_t$ is very small, and errors in using current values of engine speed and pump torque are negligible. For implementation, the throttle is assumed to move slow enough to maintain near stoichiometric air/fuel ratio, and $AFI$ is greater than 0.9 and is set to 0.95. Similarly, if the spark advance is known, Eq. (25) can be inverted to calculate the desired throttle angle ($\alpha$). This is possible since the throttle characteristic (TC) is monotonic and one-to-one, and it can be inverted to give throttle angle. An intake manifold model is used to estimate the desired airflow into the intake manifold, given the desired airflow out of the intake manifold. This throttle control equation is given below. MA, TC, and PRI represent the maximum possible flow through the throttle body (a function of stagnation pressure and temperature), throttle characteristic (normalized throttle flow as a function of throttle angle), and pressure ratio influence (normalized throttle flow as a function of pressure ratio across the throttle body), respectively.

$$\alpha = \frac{1}{\alpha_1} \cdot \left\{ [\Gamma_1 \cdot \omega_e/(K \cdot TF \cdot AFI 
\cdot SL(t + \Delta t)] + (V^p \cdot P) \right\}$$ (30)

$$\Omega_2 = J_e \cdot \omega_e(t + \Delta t) + T_z/(t + \Delta t) 
+ T_{fp}(t + \Delta t) 
- \eta_1 \cdot J_e \cdot \text{sat} [(\omega_e(t + \Delta t) 
- \omega_{sd}(t + \Delta t))/\phi_f]$$ (31)

Again, $\Omega_1$ represents the ancillary torques, desired engine acceleration, and sliding-mode error feedback terms. Note that these are also noncausal equations, but the causality problems are more severe for the throttle control than for the spark control. These problems are a result of $\Delta t_s$ (i.e., intake to torque delay) being larger than the spark to torque delay. If this delay is assumed to be negligible, there can be severe control problems, or even instability, for rapid throttle actuation. These problems are managed by slowing or rate-limiting the throttle response, since the delays have no effect in steady state. Another problem with the spark and throttle control equations given above is that they cannot be used simultaneously, since they are not independent equations.

An innovative solution (sliding-mode gain weighting) is proposed, which allows the spark and throttle to be coordinated in a desirable manner. It is desirable to have the spark slightly retarded from MBT so that good combustion can occur. If the spark becomes too retarded from MBT, incomplete combustion and misfires occur, which increase emissions significantly and seriously degrade the fuel economy of the engine. The control solution makes the spark advance tend toward MBT, with excursions only when large decreases in torque are desired during the shift.

The technique uses one sliding surface for both the throttle and spark control [Eqs. (26)-(31)] but has different sliding-mode gains for each control. Also, the throttle angle will be calculated, assuming the spark is slightly retarded from MBT; so the spark influence function is set to 0.95 in the throttle control, Eq. (30). Given that throttle setting, the spark is then calculated. The two sliding-mode gains now form a gain vector with a sliding-mode gain for each control:

$$\eta = [\eta_1, \eta_2]^T = [\text{throttle}, \text{spark}]^T$$ (32)

These gains can be weighted to provide the desired engine response, since they represent how much relative control effort will be given to each control to bring the error (or Eq. 23) to zero (analogous to loop gain). Note that
setting the spark influence to 0.95 can introduce large errors if the spark is severely retarded. The throttle gain is not increased to guarantee global stability with the throttle control equation. Instead, global stability is guaranteed with the spark control equation, since its gain can be increased to compensate for any modeling errors. (See [9] and [11], [12] for further discussions on how to guarantee global stability in the presence of large modeling errors.)

The control engineer can now weight the gain vector to get the desired engine response. Typically, the spark gain will be large in order to assure fast response. The spark control’s strength is it’s ability to change torque rapidly; however, it cannot make large engine torque changes without severe combustion and emission problems. Also, if the spark is at MBT, changing the spark alone cannot increase torque. Spark gain and boundary layer widths can be balanced by using the static-balancing condition for time-invariant boundary layers, where the maximum bandwidth (λ) from the spark control can be determined by varying spark at each occurrence. Boundary layers can be chosen using Eq. (33), where φ is the boundary layer thickness, η_{max} is the maximum sliding-mode gain, and λ is the desired break frequency.

\[ \phi \cong \eta_{\text{max}} / \lambda \]  \hspace{1cm} (33)

For a six-cylinder engine with a sliding-mode spark gain of 1000 sec^{-2}, the boundary layers should be larger than 6 rad/sec. The following figure shows how the throttle and the spark are coordinated to follow a desired engine speed trajectory, from 0.5 to 1.5 sec, for one set of gains. The weighting can be varied to change the speed of response of each control. Note that these simulations are demonstrations of how the algorithms work, rather than desired shift trajectories. A very difficult trajectory was intentionally chosen to challenge the controller.

Similar control laws can be developed to control the turbine speed along a desired trajectory, as defined in Eqs. (20)-(22). Recall that during the speed phase, the turbine inertia is dependent (i.e., both clutches have a nonzero slip-speed) and is forced by the two clutch torques and engine torque. The control laws for spark advance and throttle are given below. The new variables are first and second clutch torques (T_{c1}, T_{c2}), converter pump and turbine torques (T_p, T_t), and turbine inertia (J_t). \[ \Psi_t \] is given in Eq. (27).

\[ S_t = \text{MBT} - 51.289 \cdot \sqrt{1 - \Psi_t} \cdot \Gamma_t \] \hspace{1cm} (34)

\[ \Gamma_t = [T_p(t + \Delta t)/T_t(t + \Delta t)] \cdot \{T_{c1}(t + \Delta t) + T_{c2}(t + \Delta t) + J_t \cdot \dot{\omega}_t(t + \Delta t) - J_t \cdot \eta_t \cdot \text{sat} \{\omega_t(t + \Delta t)/\phi_t\} + J_t \cdot \omega_t(t + \Delta t) + J_t \cdot \omega(t + \Delta t) \} \]  \hspace{1cm} (35)

\[ \alpha = \text{TC}^{-1} \{[\omega_t/(K \cdot TF \cdot AFI] \cdot S_t(t + \Delta t)/T_t(t + \Delta t)/J_t \cdot \omega_t(t + \Delta t) - J_t \cdot \eta_t \cdot \text{sat} \{[\omega_t(t + \Delta t)/\phi_t] \} \} \]  \hspace{1cm} (36)

\[ \Gamma_t = T_{c1}(t + \Delta t) + T_{c2}(t + \Delta t) + J_t \cdot \dot{\omega}_t(t + \Delta t) - J_t \cdot \eta_t \cdot \text{sat} \{[\omega_t(t + \Delta t)/\phi_t] \} \]  \hspace{1cm} (37)

References [9], [16] detail the development of these turbine speed-control algorithms, along with simulation results. Turbine speed is more difficult to control because of the fluid coupling of the torque converter. Output torque of the torque converter can be modeled as a nonlinear function of the pump (engine) and turbine speeds; so turbine speed is related to the integral of engine speed. Also, torque is a nonlinear function of the engine controls; so engine speed is related to the integral of the controls. Therefore, control action of the throttle or spark is integrated twice in the model to get to turbine speed; thus, a large change in these controls is needed to affect turbine speed. This is a function of the drivetrain hardware and is independent of the control design. Turbine control can be enhanced by providing a stiffer coupling across the torque converter during shift, such as a variable geometry torque converter or selectively locking (or partial locking) of the converter. The control benefits must be weighed against the disadvantages of a stiff system (engine torque pulses transmitted to drivetrain) and additional system complexity.

**Conclusions**

This article discusses how the sliding mode nonlinear control design procedure can be used to enhance shift quality through engine control. The advantage this methodology has over other approaches is that global stability of the system can be guaranteed if the magnitude of modeling errors is known (i.e., stability robustness for nonlinear systems is included in the design). An innovative gain-weighting approach to the multivariable control problem is presented. The engine speed can be controlled quite well by using the gain-weighted sliding-mode controller. The turbine speed requires large control actions to follow desired trajectories. This is not the result of any control algorithm but is caused by drivetrain hardware (specifically by the fluid coupling of the torque converter). Turbine speed response from engine control can be improved considerably by increasing the damping of the torque converter (i.e., variable geometry, selective locking the converter, etc.). The advantages of increased response must be weighed against the disadvantage that engine torque pulses will be transmitted to the drivetrain through the stiff or locked torque converter. This disadvantage is more prevalent as the number of engine cylinders is decreased. The speed-control approach to sliding-mode engine control alleviates many of the implementation problems associated with torque control.
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