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Diagnostics for Automotive Electronic Throttle Body Systems

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Abstract— **If not properly diagnosed and corrected, temperature and position varying characteristics of friction observed in production automotive electronic throttle body (ETB) systems can lead to performance degradation, increasing warranty cost for automotive manufacturers. Consequently, friction diagnostics for ETBs has become an important issue for OEMs and ETB manufacturers. However, relatively little attention has been given to the issues of friction diagnostics for such systems. In this research, preliminary studies are presented for this problem via a mathematical formulation, thereby mitigating measurement and computation limitations in production hardware. A friction estimation scheme based on the principle of sliding modes is demonstrated in simulation for a typical production ETB. The contributions of this work lie in the friction modeling methodologies (incorporating experience in ETB operation) as applied to production ETB systems, and the novel control-theoretic approach for attaining useful diagnostic information.**

I. INTRODUCTION

Electronic throttle actuation has been widely recognized as an effective solution for the air loop control problems of automotive engines due to its numerous advantages over conventional mechanical linked throttle actuation systems [1]. Several engine control targets can be achieved simultaneously with a single Electronic Throttle Body (ETB) unit [1] [2] [9].

A typical configuration of an ETB is shown in Figure 1. Despite its simplicity in mechanical configuration, the actual system becomes complicated during production due to cost, safety, and packaging constraints. The nonlinearities, such as torque discontinuity from returning springs at the “limp home” position [12], backlash from gear driven mechanical linkage [9] and stick-slip friction [8] from brushed DC motor and position sensor, cause complicated open loop responses of ETBs. The signature of such complications is a hysteretic effect due to the interaction among those nonlinearities [8]. Among them, stick-slip friction is the dominant effect that usually affects ETB operation. In addition to the complex behavior of stick-slip friction, it is prone to vary with respect to temperatures and throttle positions. It is assumed here that the friction variation due to temperature changes may be one of the triggering events for the hunting issue in production ETBs. Some local oscillation phenomena are also observed due to the position varying nature of friction. Engine vibration, road surface condition and humidity effects all have potential contributions to friction variation. The friction variation becomes one of the most serious and costly quality problems for OEMs and ETBs manufacturers. In most cases, the stick-slip friction affects ETB operation at relatively low angular velocities and small angle manipulations when the throttle is operating below “limp home” position. Without a good handle on friction compensation, poor position tracking performances at such engine operating modes can lead to rough engine idle, bad driving perception, and inferior drive comfort. In the worst case, poor friction estimation, as well as corresponding compensation, can trigger the On Board Diagnosis (OBD) system to light up the ”Check Engine” light. Such control and detection problems definitely increase the warranty cost to automotive manufacturers. It is of great interest for OEMs and ETBs manufactures to obtain enough friction information for product quality management purposes.

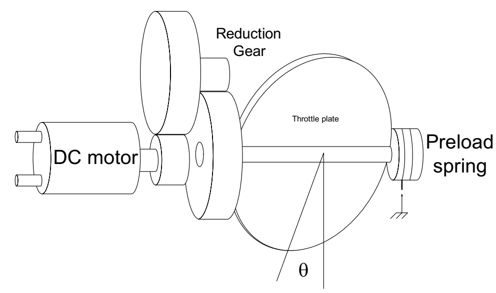

Fig. 1. Electronic Throttle Body Schematic

The topic of friction diagnostics is relatively new. However, friction estimation and compensation are not new problems. Several papers have addressed dynamic friction compensation of ETBs from different aspects [8] [10] [11] [12] [14]. Several types of dynamic friction models have been used in control design [17], including adaptation approaches [3] [5] [6] [18] [19]. However, little attention has been paid to the friction diagnosis problem [4] [13], especially for dynamic friction diagnostics. Among existing friction estimation and compensation schemes, adaptive approaches appear promising for friction diagnosis, especially because a strong tie exists between adaptive approaches [18] [19] and critical parameters of dynamic friction models which might give insight for friction in ETBs. However, the issue of parameter convergence is still an open question.

Relatively speaking, dynamics due to friction are of higher bandwidth than the throttle itself, and are a complex function of the velocity of the throttle shaft. However, the throttle shaft velocity is not directly measured on production ETBs. In this paper, we mitigate the measurement limitations on production ETBs and assume that a velocity measurement with sufficient accuracy is available in order to investigate possible solutions to the friction diagnostics problem. In addition, in order to capture the fast dynamics of friction, we lessen the restriction of production computa-

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tion power and assume we have adequate embedded computation power for this problem. The mathematical solution is emphasized here rather than a possible commercial solution. Because excessive wear or contamination can result in a higher peak stiction torque, the diagnostic task will mostly focus on the detection of faulty conditions of the peak stiction torque. The varying nature of the friction torque with respect to temperature and position are also considered in the diagnostic strategy.

This paper is organized as follows: the system model is developed in Section II; the friction estimator is developed in Section III. Simulation results are presented in Section IV, whereby conclusions and suggestions for future work appear in Section V.

II. MODEL

The generic ETB model as reported in [10] and [11] can be described by Equation 1; for simplicity, gear backlash nonlinearity is neglected:

$$\begin{aligned} \frac{di_a}{dt} &= -\frac{R_a}{L}i_a - \frac{K_eG_r}{L}\omega + \frac{u}{L} \\ \dot{\omega} &= \frac{K_tG_r}{J}i_a - \frac{T_f}{J} - \frac{T_{sp}}{J} - \frac{T_a}{J} \\ \dot{\theta} &= \omega \end{aligned} \tag{1}$$

Here, i_a is the rotor armature current, J is the total inertia measured at the throttle shaft, ω is the throttle shaft angular velocity and θ is the throttle angle. T_f, T_{sp} and T_a are friction torque, spring torque, and aerodynamic torque respectively. In following three subsections, more detailed model information about these three torques is provided.

A. Stick-Slip Friction Torque

A comprehensive stick-slip friction model, the Lugre model [6][17], is adopted in this research. Even though the distinct characteristics of friction observed in ETBs by [8] and [16] requires some modifications of the Lugre model to match open loop ramp responses of production ETBs, we believe that the Lugre model is adequate to describe stick, preslide, and slide, the three stages of friction torque which have been observed on ETBs. The Lugre model is given as

$$\begin{aligned} \dot{z} &= \omega - \frac{\delta_0|\omega|}{g(\omega)}z \\ T_f &= \sigma_0 z + \sigma_1\dot{z} + \sigma_2\omega \end{aligned} \tag{2}$$

where $g(\omega) = \alpha_0 + \alpha_1 e^{-(\frac{\omega}{\omega_s})^2} = T_c + (T_{st} - T_c)e^{-(\frac{\omega}{\omega_s})^2}$. z is the asperity deflection of the bearing contact, T_c is the Coulomb friction torque, T_{st} is the break away stiction torque, ω_s is the critical velocity, δ_0 is the stiffness coefficient, δ_1 is the damping coefficient and δ_2 is the viscosity coefficient.

Under abnormal conditions such as extra wear or contamination, difference between T_{st} and T_c are increased. Because of this increased difference, stick-slip effects can be even stronger at lower angular velocity. The approximated behaviors of the Lugre model under two different settings of stiction and slip friction torques are shown in Figure 2. As stated previously, friction torque varies with many factors such as temperature, position, vibration and time of usage, together with some degree of uncertainty.

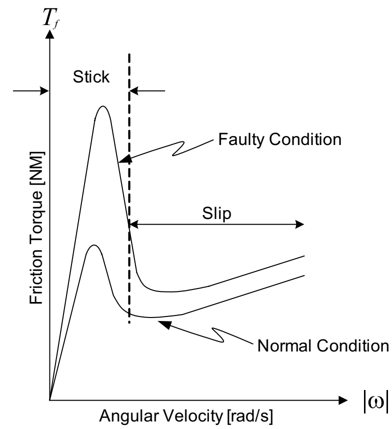

Fig. 2. Friction Torque under a Specific Control Input

B. Nonlinear Spring Torque

Production ETB systems are usually equipped with two springs. One spring is applied when the throttle angle is greater than the “limp-home” position, whereas the other is applied when the throttle angle is less than the “limp-home” position. The two springs may have different spring stiffness. Because of this configuration, a discontinuity of torque with a small dead zone is created at the “limp-home” position. Such a discontinuity creates some difficulties in control design if the control reference target position is near the “limp-home” position. Duer [12] has addressed this problem from a different aspect; as an illustration, consider an ideal realization of nonlinear spring torque, depicted in Figure 3. It was found that the spring stiffness (slop) is small in production ETBs. It is convenient to simplify the spring returning torques, T_{open} and T_{close}, as two constants.

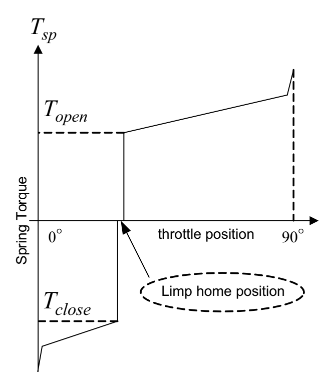

Fig. 3. Nonlinear Spring Torque

C. Aerodynamic Torque

When airflow passes the throttle plate (such as the butterfly valve), the aerodynamic torque is translated to the throttle shaft. A polynomial function was suggested by Sun [8] to approximate the aerodynamic torque, with a good degree of accuracy. Therefore, the aerodynamic torque is not considered here; for detailed information about aerodynamic torque, the interested reader is referred to [7] and [8].

D. Model Parameters

The range of parameter values which are commonly found in production ETBs are listed in Table I.

TABLE I
TABLE OF PARAMETER VALUES

Name	Definition	Range of Value	Unit
R_a	Rotor Resistance	1-5	*Ohm*
L	Rotor Inductance	$\leq 3 \times 10^{-3}$	*H*
J_m	Rotor Inertia	$\geq 2 \times 10^{-6}$	Kgm/s^2
J_{th}	Throttle Inertia	$\leq 50 \times 10^{-6}$	Kgm/s^2
K_e	Back *emf*	0.01-0.08	$Vsec/rad$
K_t	Motor Torque Constant	0.01-0.05	Nm/A
Gr	Gear Ratio	15-22	*Unitless*
T_{open}	Torque at opening	0.25-0.45	Nm
T_{close}	Torque at closing	0.3-0.4	Nm
T_{stick}	Stiction Torque	0.005-0.03	Nm
T_{slip}	Slip Friction Torque	0.0025-0.005	Nm
B	Viscosity coefficient	≥ 0.001	$Nmsec/rad$

III. FRICTION ESTIMATION

The Lugre model requires six parameters to characterize the nonlinear stick-slip friction. The parameters, T_{st} and T_c, are believed crucial for friction diagnosis. They are assumed unknown and are subject to estimation in this research. The remaining parameters are assumed to be invariable and can be identified off-line. From knowledge of the system, the model equation, and simulation studies, three properties of the Lugre model have surfaced which are useful for constructing estimation schemes for T_{st} and T_c.

Property 1:
The internal state, z, is bounded such that $max(\delta_0|z|) \leq T_{st}$. When friction is in a steady state, that is $\dot{z} = 0$ (or $z = z_{ss}$), the friction torque is $T_{fss} = \alpha_0 + \alpha_1 e^{-(\frac{\omega}{\omega_s})^2} + \delta_2 \omega$.

Property 2:
Given any initial condition z_0, there exists a set of input trajectories such that the state, z, can reach zero, while the friction torque passes peak stiction torque T_{st}, and stays at T_c.

Property 3:
Assuming initial conditions of z are positive, a velocity trajectory starting from a negative value and smoothly and monotonically increasing until $|\omega| \gg \omega_s$ and $|\dot{\omega}| < |f(\omega_s)|$, can guarantee that the friction torque passes from T_{st} and reaches T_c, where $f(\omega_s)$ is a smooth function of ω_s.

Given above three basis properties, a class of trajectories of input or the throttle shaft velocity can be constructed specifically for the purposes of friction diagnosis.

A. Friction Estimation using Current Measurement

Modern DC motor drives, such as the H-bridge drives which are widely used in typical ETB control modules, are already embedded with current sensing circuitry to supply motor current information. Together with an angular velocity measurement of the throttle shaft or the motor shaft, the angular velocity dynamics can be used to estimate friction. Again, the throttle shaft velocity dynamics are

$$\dot{\omega} = \frac{K_t Gr}{J} i_a - \frac{T_f}{J} - \frac{T_{sp}}{J} - \frac{T_a}{J} \quad (3)$$

Creating a sliding mode observer based on the above equation without considering the term $\frac{T_f}{J}$ yields

$$\dot{\hat{\omega}} = \frac{K_t Gr}{J} i_a - \frac{T_{sp}}{J} - \frac{T_a}{J} - \frac{M}{J} sgn(\hat{\omega} - \omega) \quad (4)$$

where $sgn(\cdot)$ is the signum function and M is a positive number. Letting $\tilde{\omega} = \hat{\omega} - \omega$, the error equation becomes

$$\dot{\tilde{\omega}} = \frac{T_f}{J} + d(\cdot) - \frac{M}{J} sgn(\tilde{\omega}) \quad (5)$$

where $d(\cdot)$ represents the unmodeled dynamics and disturbance. If $M \gg 0$ and control can be switched sufficiently fast, the sliding mode can be reached in finite time and $\tilde{\omega} \to 0$ ($\hat{\omega} \to \bar{\omega}$). When a sliding mode appears,

$$Msgn(\tilde{\omega}) \equiv T_f + d(\cdot) \quad (6)$$

Using the equivalent control concept and denoting T_{eq} as $Msgn(\tilde{\omega})$, the equivalent control T_{eq} is equivalent to $T_f(\theta, \omega) + d(\cdot)$, although it is in a form which can be difficult to interpret.

B. Obtaining Equivalent Control

In order to obtain the equivalent control, a low pass Filter (LPF) can be used to attenuate low frequency components of $T_f(\theta, \omega) + d(\cdot)$. The goal of creating a sliding mode observer is not to observe a variable which has already been measured; rather, it is to create a scheme to estimate friction torque. Usually it is assumed that the term $d(\cdot)$ is zero mean with a certain variance. Thus, the first moment statistics about T_{eq} are sufficient to estimate T_{st} and T_c. Because the sliding mode observer is implemented inside the controller, there is no limitation on the value of M. The only limitation in implementing the observer is switching frequency. If the control updating rate can be selected as fast as possible (some popular DSP chips can run floating point algorithms at 20 kHz effectively), estimation errors due to finite switching frequency are tolerable.

Using a LPF, the filtered information (estimated friction function) will have additional phase lag. However, because it is not used in either feedback or feedforward control in this research, the quality of the estimated friction torque will not affect the overall control response.

C. Diagnostic Strategy

Based on the properties of the stick-slip friction, the friction diagnostic strategy about T_{st} and T_c can be readily implemented under two situations:

1) When $|\omega| < |\omega_s|$, the friction torque is bounded by T_{st}. It is convenient to define a threshold to distinguish normal or abnormal stiction torque. Once the estimated stiction torque is beyond the threshold, the corresponding temperature and position information can be recorded and stored for offline analysis.
2) When $|\omega| >> |\omega_s|$, the friction torque is bounded by T_c. An approach similar to that for the stiction torque can be applied for slip torque diagnosis.

Based on the above diagnostic strategy, the position and temperature varying nature of the friction can be captured.

IV. Simulation Results

A proportional–derivative (PD) control, together with a simple static feedforward control term (capturing effects of static Coulomb friction and viscous friction), is adopted to investigate the potential of sliding mode concepts for friction estimation. The use of a simple PD control scheme may not be the most suitable choice, but is typically used in current production ETBs. Because this work is not intended to demonstrate, nor investigate the efficiency of (PD) control for ETBs, little effort was spent in tuning the PD parameters. With this in mind, the simulation setup is given in the following:

1) The friction estimator is updated at 20 kHz.
2) The PD controller is updated at 1 kHz.
3) Under normal operating conditions, $T_{st} = 0.0095 N.m$ and $T_c = 0.008 N.m$.
4) Under faulty conditions, $T_{st} = 0.00315$ N.m and the slip friction torque T_c is not changed.
5) The critical speed of the motor shaft, ω_s , is set at 140 rad/sec.
6) Two reference input signals are used to evaluate the performance of the friction estimator. The first reference input is a sinusoidal function with angular velocity at 30 rad/sec. The second reference input is a step input.

Simulation results under normal conditions are shown in Figure 4 and Figure 5; the simulation results under faulty conditions are shown in Figure 6 and Figure 7.

Under both sinusoidal and step reference inputs, the friction estimator in general can capture the trend of the friction torque with reasonably small errors. Under normal conditions, the differences between the estimated T_{st} and the estimated T_c are small. In contrast, under faulty conditions, the friction estimator clearly detects the difference between T_{st} and T_c, although it underestimates T_{st}. This minor discrepancy could be remedied by increasing the LPF attenuation frequencies.

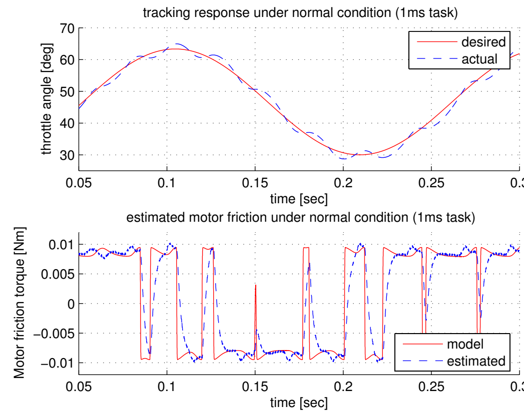

Fig. 4. Estimated friction torque under normal conditions with a sinusoidal input.

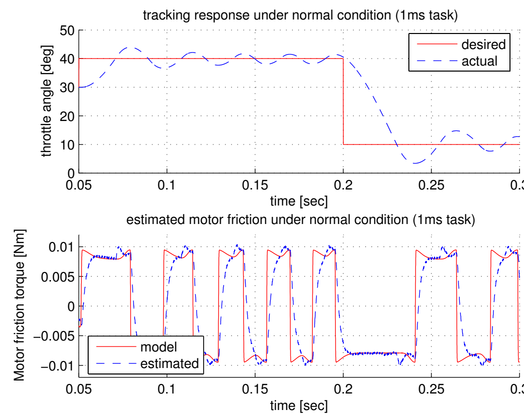

Fig. 5. Estimated friction torque under normal conditions with a step input.

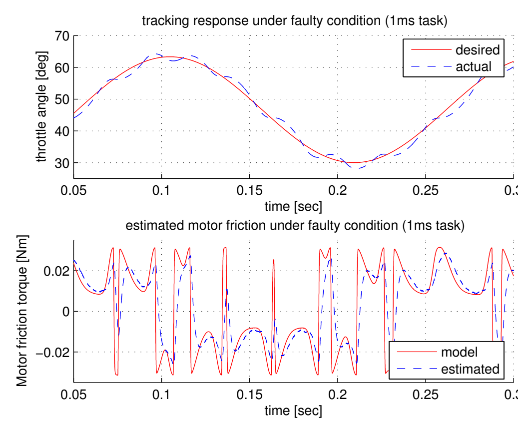

Fig. 6. Estimated friction torque under faulty conditions with a sinusoidal input.

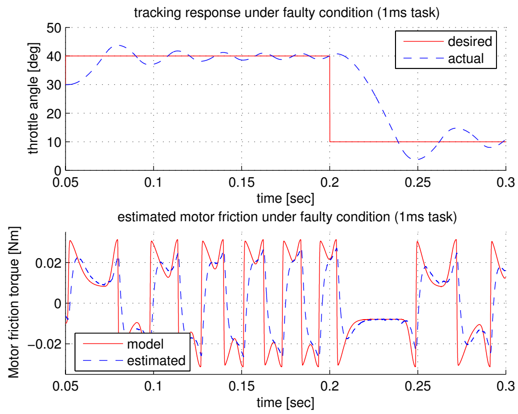

Fig. 7. Estimated friction torque under faulty conditions with a step input.

V. CONCLUSIONS AND FUTURE WORK

This research demonstrates a friction diagnostic algorithm based on the sliding mode concept in simulation. The sliding mode technique was used due to the nonlinear coupling in the velocity terms due to the friction term. The angular velocity dynamics of the throttle shaft are used to construct the friction estimator using the concept of sliding modes. The sliding mode friction estimator achieves reasonably small estimation errors and can be used for abnormal stick-slip friction torque diagnosis. The detected faulty conditions of the friction torque can be utilized to analyze the position and temperature varying nature of the friction torque observed in production ETBs. The Lugre friction model is utilized to construct the friction estimator. The approach can be extended to cover the variation of ω_s as well. In addition, it can be used to diagnose more complex friction behavior or used for purposes of friction identification.

This research demonstrates friction diagnosis only under PD control. Other forms of feedback control have yet to be tested. Moreover, the angular velocity measurement is assumed to be available in this research; current ETBs in production do not have this capability. In order to implement algorithms of the sort proposed herein, some modification will be needed. In experiment, the addition of a high resolution optical encoder on the motor shaft would be needed to validate the algorithm. It is also important to point out the limitation of the diagnostic theory as applied to this practical problem. Some design and manufacturing defects, such as the aluminum housing and the throttle shaft deflections due to uneven temperature distribution and partial engine load, can seize the throttle shaft movement. These phenomena are believed to be out of scope for this study, and beyond the capability of model-based diagnostics. Higher order dynamics excited by PWM control and aging effects from the worn geartrain also introduce nonlinear behavior. These issues will be addressed in an upcoming publication.

VI. ACKNOWLEDGMENTS

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