ACTUAL ENGAGED GEAR IDENTIFICATION: A HYBRID OBSERVER APPROACH

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Abstract: The knowledge of the actual engaged gear is essential to achieve high-quality control of an automotive power train. In cars equipped with manual gear shift, this information is not directly available. A hybrid observer for on–line identification of the actual engaged gear is proposed. Two hybrid models of an automotive driveline, a detailed one for verification and a reduced-order one for synthesis, have been developed. The identification algorithm is based on a novel approach to observer design for hybrid systems. The algorithm was tested on experimental data obtained at Magneti Marelli Powertrain on an Opel Astra equipped with a Diesel engine and a robotized gearbox and yielded excellent results.

Keywords: automotive control, discrete event dynamic systems, observers, system state estimation, traction control.

1. INTRODUCTION

Engine control strategies achieving high performance and efficient emissions control depend critically on on–line identification of the actual engaged gear. The knowledge of the actual engaged gear is necessary in engine torque control to compensate the equivalent inertia of the vehicle on the crankshaft. For Diesel engines, actual gear identification is very important to improve emissions control. In fact, particulate emissions (which consist of soot with some additional absorbed hydrocarbon materials) are particularly difficult to control with first gear engaged. At present, in commercial cars the problem of actual engaged gear identification is solved by comparing the revolution speed of the wheels with the revolution speed of the crankshaft. However, since both of them are affected by oscillations due to the elasticity of the transmission shafts and the tires, then this approach implies large time delays in the identification and may produce significant identification errors.

Applications of hybrid systems techniques to automotive control have been presented in (Baotic *et al.*, 2003; Balluchi *et al.*, 1999; Balluchi *et al.*, 2000; Balluchi *et al.*, 2001; Bemporad *et* *al.*, 2002; Bemporad *et al.*, 2003). A hybrid system approach to engine control allows to achieve better control accuracy with respect to standard mean–value models design, since the internal combustion engine of a car is intrinsically a hybrid system due to (i) the discrete nature of the four stroke engine cycle; (ii) the transitions between strokes that are determined by the continuous motion of the driveline, which in turn depends on the torque produced by each piston, the actual engaged gear and the connection clutch state.

In this paper, the design of a hybrid algorithm, for on–line identification of the actual engaged gear, based on the theory of hybrid observers, is proposed. The behavior of the driveline is described by a hybrid model, where the engaged gear and connection clutch state are represented as discrete states. Then, the design problem is formulated as the identification of the discrete state of the driveline hybrid model. For this purpose, the revolution speeds of the crankshaft and the wheels are assumed to be measurable and the engine torque to be available by estimation. The identification algorithm was derived by applying an approach to observer design for hybrid systems recently proposed by our group (see (Balluchi *et al.*, 2001; Balluchi *et al.*, 2002)). The algorithm was validated by extensive simulations on a hybrid model of the driveline that is much more detailed than the one used for synthesis and was successfully applied on a set of experimental data.

The paper is organized as follows. In Section 2, the theoretical background for the hybrid observer design is summarized. In Section 3, a detailed hybrid model (used for validation) and a reduced-order hybrid model (used for synthesis) of the driveline are proposed. The identification algorithm is developed in Section 4. In Section 5, simulations and experimental results are reported.

2. THEORETICAL BACKGROUND

In this section, the methodology for the design of hybrid observers proposed in (Balluchi *et al.*, 2001; Balluchi *et al.*, 2002) is briefly illustrated. The hybrid observer consists of two parts: a *location observer*, which identifies the current discrete state of the hybrid plant, and a *continuous observer*, which produces an estimate of the evolution of the continuous state of the hybrid plant. Since actual engaged gear identification corresponds to the identification of the discrete state of the driveline hybrid model, a location observer will be used to this purpose. The reader is referred to (Balluchi *et al.*, 2001; Balluchi *et al.*, 2002) for a description of the continuous observer.

Let H denote the hybrid model of a given hybrid plant (see (Henzinger, 1996; Lygeros *et al.*, 2003))

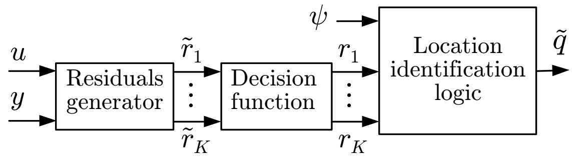

Fig. 1. Location observer structure.

with N discrete states $\{q_1, q_2, \ldots, q_N\}$ (referred to as locations). Assume that in each location q_i the evolution of the continuous state $x(t)$ of H is subject to a linear time–invariant dynamics, i.e.

$$\begin{array}{l} \dot{x}(t) = A_i x(t) + B_i u(t) \\ y(t) = C_i x(t) \end{array} . \tag{1}$$

The location observer receives as inputs the plant continuous inputs $u(t)$ and outputs $y(t)$, and the plant discrete outputs ψ (if any) and provides an estimate $\tilde{q}$ for the current location q of H. Assume that the location observer has properly recognized that the hybrid plant H is in a location q_i, i.e. $\tilde{q} = \{q_i\}$. When H makes a transition to some location $q_j \neq q_i$, then the location observer may identify such transition from the discrete output ψ. If either ψ is not available or it does not carry enough information, then the evolution of $u(t)$ and $y(t)$ are processed to identify the new location q_j, by detecting the change in the continuous time dynamics. The general scheme for the location observer is composed of three cascade blocks, see Figure 1:

Residuals generator. This block produces K residual signals $\tilde{r}_i$ that are used to supply additional information for location identification by processing $u(t)$ and $y(t)$, when needed. A simple and reliable approach for this purpose is to use K Luenberger observers, tuned on the K continuous time dynamics to be detected:

$$\begin{array}{l} \dot{z}_i(t) = (A_i - L_i C_i) z_i(t) + B_i u(t) + L_i y(t) \\ \tilde{r}_i(t) = C_i z_i(t) - y(t) \end{array} \tag{2}$$

where L_i are design parameters. If the hybrid plant is in location q_i, then the residual $\tilde{r}_i(t)$ converges to zero with rate adjustable by L_i.

Decision function. This block outputs K binary signals r_i, called signatures. A simple implementation is:

$$r_i = \begin{cases} \textit{true} & \text{if } \|\tilde{r}_i(t)\| \leq \varepsilon \\ \textit{false} & \text{if } \|\tilde{r}_i(t)\| > \varepsilon \end{cases} \tag{3}$$

where the threshold ε is a design parameter. In (Balluchi *et al.*, 2002) a sufficient condition ensuring $r_i = \textit{true}$ within a given time after the location q_i is entered, is presented.

Location identification logic. This block is a DES that identifies the current location on the basis of the K signatures r_i and the plant discrete outputs ψ (if any). Its design is obtained by the current–observation tree approach (see (Balluchi *et al.*, 2002)).

3. DRIVELINE HYBRID MODEL

In this section two hybrid models of an automotive driveline are presented. The first model, referred to as H_M, is very detailed and has been used for analysis and validation. The second model, denoted by H_m, is obtained by abstraction and reduction of the first one and has been used for the synthesis of the identification algorithm. The model H_M, which has 6048 locations and 12 continuous state variables, is able to represent precisely driveline discontinuities (see (Lemma, 2004) for details). As an example, in Figure 2 the elastic torsional characteristic of the driveline modeled in H_M is compared with experimental data. As it is clear from the figure, the model is able to represent the hysteresis of the characteristic, due to the tires, and the discontinuity of the elastic coefficient due to the engine suspension and backlashes.

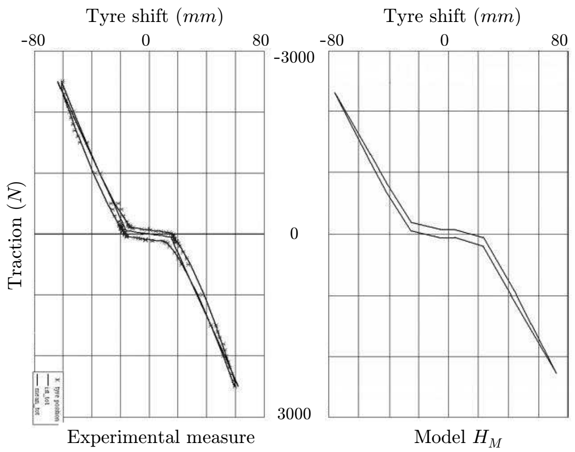

Fig. 2. Torsional elastic characteristic in second gear.

Since the hybrid model H_M has a very large number of discrete states, then it cannot be used for the synthesis of the location observer. As a matter of fact, for the identification of the gear and clutch actual states, a simpler driveline hybrid model can be used. The simplified hybrid model H_m has been obtained from the detailed model H_M by abstracting discrete states and reducing the continuous state space, after a deep analysis and extensive simulations.

The discrete state of the hybrid model H_m is denoted by q^m. The hybrid model H_m has 7 locations, i.e.

$$q^m \in \{q_1^m, q_2^m, q_3^m, q_4^m, q_5^m, q_{RG}^m, q_N^m\} \tag{4}$$

where: locations q_i^m, for $i = 1, \ldots, 5$, correspond to i–th gear engaged and clutch closed; location q_{RG}^m models reverse gear engaged; location q_N^m represents either driveline open (idle gear and/or clutch open) or clutch slipping. The DES D_m

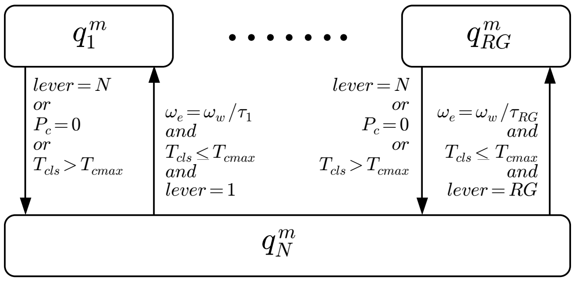

Fig. 3. The DES D_m of the hybrid model H_m

describing the discrete dynamics of H_m is depicted in Figure 3.

The inputs of the hybrid model H_m are: the position of the gear lever $lever \in \{1, 2, 3, 4, 5, RG, N\}$, the connection pressure of the clutch plates $P_c(t)$, the torque generated by the engine $T_e(t)$ and the wheel torque $T_w(t)$. The continuous state variables are: the driveline torsion angle $\alpha(t)$, the crankshaft revolution speed $\omega_e(t)$, and the wheel revolution speed $\omega_w(t)$.

When the clutch is slipping it transmits a torque $T_c(t) = \mu_d P_c(t)$, where μ_d is the kinetic friction coefficient. When clutch is locked and the i–th gear is engaged, it transmits the torque

$$T_{cls}(t) = k_i \alpha(t) + b_i \left(\omega_e(t) - \frac{\omega_w(t)}{\tau_i} \right)$$

where τ_i is the transmission ratio, and k_i and b_i are, respectively, the driveline equivalent elasticity and damping coefficients for i–th engaged gear. The clutch remains locked until the transmitted torque T_{cls} exceeds the static friction capacity, $T_{cmax}(t) = \mu_s P_c(t)$, where μ_s is the static friction coefficient.
The continuous dynamics in location q_N^m is

$$\dot{\omega}_e(t) = -\frac{b_e}{j_e}\omega_e(t) + \frac{1}{j_e}T_e(t) - \frac{1}{j_e}T_c(t) \ , \tag{5}$$

with: $T_c = 0$ when the clutch is open, and $T_c = \mu_d P_c$ when it slips. For $q^m = q_i^m$, with $i = 1, \ldots, 5$, and $q^m = q_{RG}^m$, the continuous dynamics is as in (1) with:

$$x = \begin{pmatrix} \alpha \\ \omega_e \\ \omega_w \end{pmatrix}, \quad u = \begin{pmatrix} T_e \\ T_w \end{pmatrix}, \quad y = \begin{pmatrix} \omega_e \\ \omega_w \end{pmatrix}$$

and

$$\mathbf{A}_i = \begin{bmatrix} 0 & 1 & -\frac{1}{\tau_i} \\ -\frac{k_i}{j_e} & -\frac{b_e + b_i}{j_e} & \frac{b_i}{\tau_i j_e} \\ \frac{k_i}{\tau_i j_w} & \frac{b_i}{\tau_i j_w} & -\frac{b_w + \frac{b_i}{\tau_i}}{j_w} \end{bmatrix} \tag{6}$$

$$\mathbf{B}_i = \begin{bmatrix} 0 & 0 \\ \frac{1}{j_e} & 0 \\ 0 & -\frac{1}{j_w} \end{bmatrix}; \quad \mathbf{C}_i = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

where j_e is the crankshaft inertia, j_w is the vehicle equivalent inertia, b_w is the driveline equivalent viscous coefficient and b_e is the crankshaft viscous coefficient.

The guards associated to the transitions of the driveline hybrid model H_m are as follows:

- from q_i^m to q_N^m

$$P_c = 0 \;\; or \;\; T_{cls} > T_{cmax} \;\; or \;\; lever = N \; ;$$

- from q_N^m to q_i^m

$$\omega_e = \frac{\omega_w}{\tau_i} \;\; and \;\; T_{cls} \leq T_{cmax} \;\; and \;\; lever = i \; .$$

4. LOCATION OBSERVER DESIGN

The on–line identification of the actual engaged gear corresponds to the identification of the current location of the driveline hybrid model H_m described in Section 3. In this section, a location observer O for the hybrid model H_m, obtained according to methodology reported in Section 2, is described. Notice that the hybrid model H_m does not provides any discrete output signal, since events causing discrete transitions are not measurable.

The proposed location observer O, depicted in Figure 4, receives as inputs: the crankshaft revolution speed $\omega_e(t)$, the wheel revolution speed $\omega_w(t)$ and an estimate $\bar{T}_e(t)$ of the mean–value of the engine torque. Six residual signals $\tilde{r}_i$, with $i = 1, \ldots, 5, RG$, are produced by implementing residuals generators as in (2), tuned on the continuous dynamics of the hybrid model H_m in locations $q_1^m, \ldots, q_5^m, q_{RG}^m$. Such residual are affected by several unknown disturbances:

- the quantization error on the measurements of $\omega_e(t)$ and $\omega_w(t)$;
- the mismatch between the actual continuous–time and pulsating engine torque $T_e(t)$ and the event–based estimate $\bar{T}_e(t)$, synchronized with engine strokes;

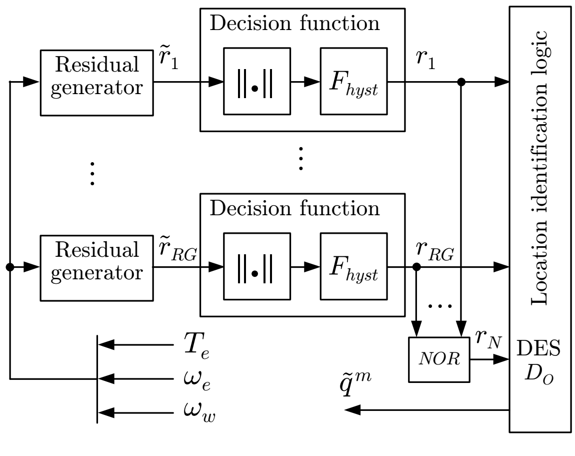

Fig. 4. Location observer O.

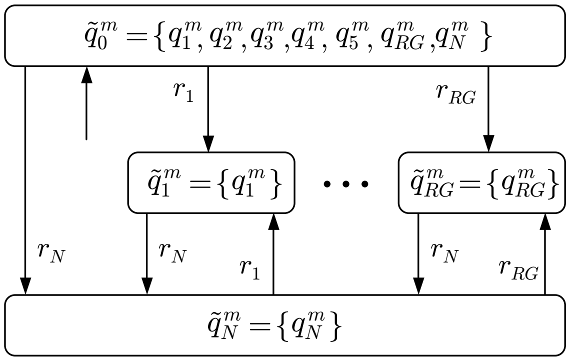

Fig. 5. DES D_O of location observer O.

- the not measurable wheel torque $T_w(t)$.

The effects of the disturbances on the residual signals have been minimized by appropriately tuning the residuals generators. Moreover, in order to reduce chattering of the signatures r_i obtained from the residuals $\tilde{r}_i(t)$, a passive–hysteresis relay and a debouncing algorithm have been used in the decision function in place of the relay (3) (blocks F_{hyst} in Figure 4).

Location q_N^m, representing either driveline open or clutch slipping, cannot be detected using the residual approach. This is because: (i) the continuous dynamics (5) is more sensitive to torque disturbances (especially during clutch slipping when the clutch plate torque T_c is nonzero); (ii) such disturbances cannot be satisfactorily compensated by the residual, being only one output (the engine speed) available for feedback. Hence, the signature r_N detecting location $\tilde{q}_N$ is obtained by negation of the other, i.e.

$$r_N = NOR(r_1, r_2, r_3, r_4, r_5, r_{RG}) \; .$$

The DES D_O that identifies the current location on the basis of the signatures r_i is depicted in Figure 5. The output of D_O, which identifies the actual engaged gear, corresponds to its current state.

5. SIMULATION AND EXPERIMENTAL RESULTS

The performance of the proposed algorithm for actual engaged gear identification was tested with both simulations and experimental data. The specification given by Magneti Marelli Powertrain was to achieve correct identification on a set of maneuvers within a delay of 250 *msec*, with an implementation of the algorithm in discrete–time with a sampling period of 12 *msec*.

Simulation results. Figure 6 reports simulation results obtained by applying the location observer O to the detailed hybrid model H_M (locations are encoded as follow: $q_N^m = 0$ and $q_i^m = i$).

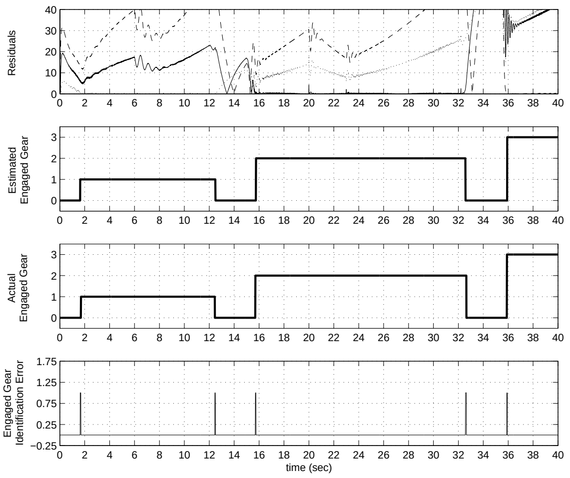

Fig. 6. Gear identification in a simulation result.

The maneuver starts with car at rest, clutch open and first gear engaged ($q = q_N^m$). After a clutch slipping phase ($q = q_N^m$), the clutch is locked ($q = q_1^m$). Later, second gear ($q = q_2^m$) and then third gear ($q = q_3^m$) are engaged, passing through idle and slipping ($q = q_N^m$). Maneuvers starting with car at rest are among the most critical ones for actual engaged gear identification. The simulation shows that the location observer identifies correctly the actual engaged gear with a worst case delay of 70 $msec$.

Experimental results. Experimental results were obtained in Magneti Marelli Powertrain using an Opel Astra equipped with a Diesel engine and a robotized gearbox SeleSpeed. The experimental data collected the measurements of the estimated engine torque $\bar{T}_e$, the crankshaft speed ω_e, and the wheel speed ω_w obtained by the engine control unit installed on the vehicle. The measurements of ω_e and ω_w are affected by delays, but the algorithm proved to be robust with respect to this non ideal situation. For the validation of the identification algorithm, the estimated engaged gear is compared to the signal on actual engaged gear provided by the control unit of the robotized gearbox.

The algorithm was tested on several maneuvers for a total of 250 gear engagements. The actual engaged gear was successfully identified within a delay of 250 $msec$ in 90% of cases. The unsuccessful cases have been obtained in very critical maneuvers such as gear engagements during sharp braking or clutch abrupt releases. In these cases, the residuals exhibit large oscillations that cause a delay up to 500 $msec$ in the identification. We believe that this delay can be reduced by using a more sophisticated decision function.

Figure 7 reports the results on actual engaged gear identification for a maneuver similar to that presented in the simulation results. The residual

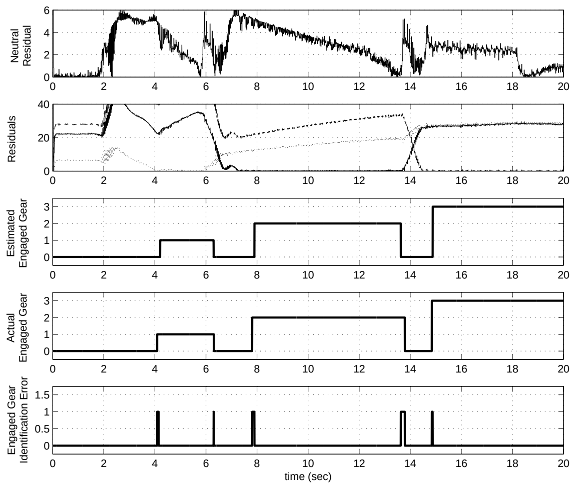

Fig. 7. Gear identification with experimental data.

$\tilde{r}_N$ of the idle gear obtained by using a dynamic residual as in (2) is shown in the first plot of Figure 7 and the residuals $\tilde{r}_i$ associated to the gear in engaged state in the second plot. As the figure shows, the residual $\tilde{r}_N$ is more affected by noise than the residuals $\tilde{r}_i$ and this is due to a larger degree of uncertainty for the friction during idle. For this reason, the signature r_N detecting location $\tilde{q}_N^m = \{q_N^m\}$ is obtained by negation of the other signatures.

Figure 8 shows more in detail the transition from second to third gear. The figure clearly shows that there is a delay in detecting the idle and slipping location ($q = q_N^m$). In fact, since the signature r_N is obtained by negation of the others, then the location $q = q_N^m$ is identified only when the residual $\tilde{r}_2(t)$ increases over the high value of the relay threshold.

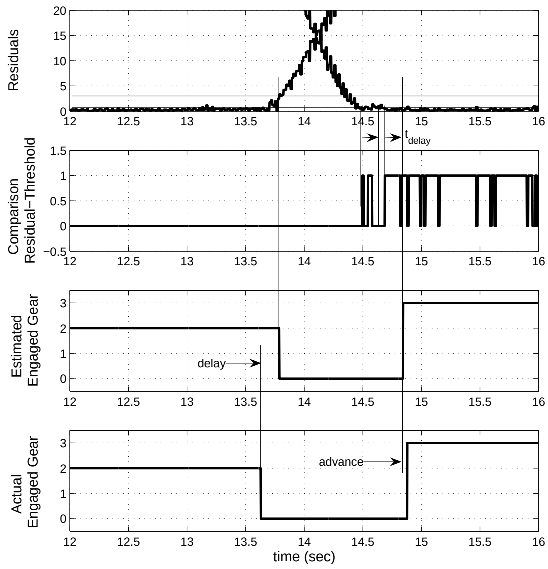

Fig. 8. Transition from second to third gear.

The figure also shows that the location $q = q_3^m$ is identified in advance with respect to the actual transition. To understand this situation, notice that during the slipping phase that precedes each clutch engagement, since the new gear has been already engaged, the continuous dynamics (5) of location q_N^m tends to that of the entering location q_i^m as the slipping decreases. Due to this behavior, transitions from q_N^m to q_i^m never cause abrupt changes in the continuous dynamics. Consequently, the residual norm $\|\tilde{r}_i\|$ gradually decreases during slipping and approaches the low value of the relay threshold before the plant model enters location q_i^m (see Figure 8). Hence, occasionally the observer is able to identify the new location q_i^m in advance with respect to the actual transition in the plant model.

Finally, the second plot of Figure 8 shows the signal to the input of the debouncing algorithm produced by the passive-hysteresis relay. Notice that, when the residual $\tilde{r}_3(t)$ decreases below the low threshold of the relay, the debouncing algorithm waits a time t_{delay} and then sets the signature r_3 to *true* only if the residual $\tilde{r}_3(t)$ is still below the threshold.

6. CONCLUSIONS

The design of an algorithm for on–line identification of the actual engaged gear has been presented. The proposed algorithm was derived leveraging recent results on observer design for hybrid systems. The robustness of the algorithm with respect to parameter uncertainties (e.g. vehicle inertia) and time-varying unknown disturbances (e.g. wheel torque and road slope) was validated by means of extensive simulation and actual experimental results obtained on an Opel Astra at Magneti Marelli Powertrain. The algorithm was able to identify correctly the actual engaged gear within 250 *msec*, as requested by specification.

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