Automatic Gear Shifting in Sport Motorcycles

Paolo Giani, Fabio Todeschini, Mara Tanelli, *Senior Member, IEEE*,
Sergio M. Savaresi, *Senior Member, IEEE*, and Mario Santucci

I. Introduction

GEAR shifting is undoubtedly a crucial operation for all vehicles, both for its impact on driveability and comfort and for its crucial link with engine and clutch performance and operation. In fact, in the scientific literature, several efforts have been done to study this maneuver and to propose active control systems to deal with automatic management of gear shifting, mainly tailored to four-wheeled vehicles (see, e.g., [1]–[5]).

Within this context, this paper first analyzes the characteristics of the motorcycle gear shift maneuver, and it proposes a way to estimate the maneuver quality from measured data (see also [6]). The final goal of this classification is to automatically and quantitatively describe the performed maneuver and label it with a quality attribute matching that assigned by the rider. Based on this initial step, an automatic control system will be designed, and its performance is evaluated due to the same cost functions that assess its suitability for the considered application. To realize the proposed controller, it is necessary to accurately regulate the clutch position to enable tracking a predefined reference profile for the engagement of the new gear.

In the motorcycle field, automatic clutches are not yet widely used; however, the increasing demand for transmission efficiency and performance, together with the availability of compact and reliable automatic drivetrains, calls for a more automated clutch system. In this context, as it is for racing cars, *wet* clutches are generally used, instead of the classical *dry* clutches, because of their better performance and heating efficiency. On the other hand, however, wet-clutch control is a more difficult task compared with the control of a dry-clutch system, mainly due to system complexity and stiffness, as pointed out in [5], [7], and [8].

In this paper, we will present the identification of the position dynamics of the electrohydraulic clutch installed on the test vehicle and design a position controller that fulfills the tracking specifications of the gear shift controller. It is worth noting that automatic gear shifting for two-wheeled vehicles is a new research area that is largely, if not completely, unexplored, and this work constitutes, to our best knowledge, the first scientific contribution to address the design of an automatic gear shift controller for this type of vehicle. An automatic gear shift might be of interest both for touring bikes, with the aim of reducing the burden of the riding activity, and for sport motorcycles, where a performance-oriented gear shift has the potential of improving vehicle driveability. The management of a motorcycle driveline system, in fact, is a complex task, due to many critical aspects related to the stability and safety of vehicle motion. Moreover, the needed operations have to be coordinated on a timely basis so as to achieve a good maneuver from the rider's point of view, while minimizing both the mechanical and the thermal stresses of the driveline components. In this context, automatic management allows the achievement of a prescribed and repeatable vehicle behavior independent of the rider's abilities. Further, although bike enthusiasts are sometimes reluctant to accept electronic aids that, in their view, may alter the riding experience, traffic conditions and oil prices have opened the way to the spread of two-wheeled vehicles for short-range travels. In this context, the vehicles are used by people that usually drive a car and, thus, welcome active control systems that make the two-wheeled vehicle dynamics easier to manage.

With respect to the existing literature previously presented, the strength of the proposed approach is its detailed validation in an experimental setting, which, of course, poses many more problems than a simulation-based setting, where the computational constraints, the sensor and actuator imperfections, and the related measurement errors, saturations, hysteresis, and the like are not present. Furthermore, as far as the control approach is concerned, our method is based on a standard control design method, with *ad hoc* refinements to deal with the aforementioned limitations of vehicle instrumentation. Further, the objective cost functions allow us to quantitatively measure the improvements enabled by the proposed approach. The proposed

Manuscript received June 14, 2012; revised December 20, 2012 and May 14, 2013; accepted October 15, 2013. Date of publication November 11, 2013; date of current version June 12, 2014. This work was supported in part by the MIUR project "New Methods for Identification and Adaptive Control for Industrial Systems" and in part by Piaggio & C. SpA. The review of this paper was coordinated by Prof. T. M. Guerra.

P. Giani, F. Todeschini, M. Tanelli, and S. M. Savaresi are with the Dipartimento di Elettronica, Informazione e Bioingegneria, Politecnico di Milano, Milano 20133, Italy (e-mail: giani@elet.polimi.it; todeschini@elet.polimi.it; tanelli@elet.polimi.it; savaresi@elet.polimi.it).

M. Santucci is with Piaggio & C. S.p.A., Pisa 56025, Italy (e-mail: Mario.Santucci@piaggio.com).

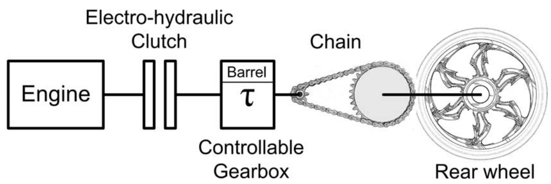

Fig. 1. Schematic view of the vehicle drivetrain.

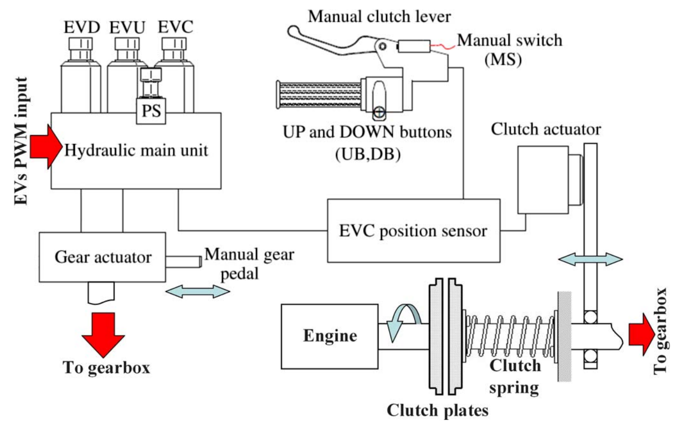

Fig. 2. Schematic view of the hydraulic layout of the vehicle transmission.

shifting control policy was tested on a sport vehicle, trying to capture all the relevant features of the considered maneuver, which are related both to performance and rider's comfort.

The structure of this paper is as follows. Section II provides a description of the test vehicle and of the considered transmission. Section III compares automatic and manual gear shifts. Further, Section IV presents the quality indexes designed to judge the gear shift quality. Section V focuses on the design of a novel gear shift controller, whereas Section VI discusses the experimental results obtained on an instrumented sport motorcycle.

II. Experimental Layout

The test vehicle considered in this work is an instrumented sport motorcycle, 750 cc and approximately 100 HP, equipped with a controllable gearbox. A schematic view of the transmission system is shown in Fig. 1.

The clutch mechanically connects the engine to the rear wheel, regulating the percentage of engine torque that flows through the drivetrain and reaches the rear wheel. More precisely, the transferred torque depends on the relative position of clutch plates, which is adjustable by regulating the oil pressure in the hydraulic clutch actuator chamber: If no force is applied on the clutch (*i.e.*, the oil pressure in its chamber is nearly zero), the clutch plates are firmly connected by the clutch springs, and all the available engine torque reaches the gearbox. On the contrary, if the oil pressure in the actuator chamber is large enough to counteract the spring force, the clutch is disengaged (*i.e.*, its plates are mechanically separated), and no traction torque reaches the rear wheel. The position interval within which the clutch plates effectively transmit torque represents a modest percentage of the whole clutch position excursion, generally approximately 25% $\div$ 30%. This interval starts at the so-called *kiss-point* position [5] and ideally ends when the clutch is completely closed. The *kiss-point* position is generally at approximately 70% of the overall clutch stroke, which makes the effective torque modulation interval to noticeably shrink, thus having a significant impact on the control design phase.

The main components that constitute the experimental layout are represented in Fig. 2.

The oil pressure in the clutch actuator chamber can be controlled both by the rider via the manual clutch lever and by the electronic control unit (ECU) via a proportional electrohydraulic valve (EVC). The EVC position sensor measures the displacement of the EVC valves and, assuming that the physical link between it and the clutch plates is rigid enough, this measurement can be considered proportional to the true displacement of the clutch plates.

The barrel can assume six different positions (corresponding to the six gears of the motorcycle) that can be sequentially selected both by the rider via the gear pedal (in manual gear shifts) and by the ECU via two three-way directional EVCs (EVU and EVD, respectively, which can assume open, close, and hold positions) that move the barrel from the current position to the upper or lower position. The barrel position can also be measured by means of a position sensor mounted on the gearbox crankcase.

All the hydraulic users share a common hydraulic circuit, which consists of an oil tank and an accumulator, the pressure of which is measured by means of the pressure sensor PS and is kept as constant as possible with an oil pump.

The experimental layout also allows the ECU to temporarily inhibit the spark plug ignition by activating the so-called *cut-off* signal.

Finally, the motorcycle is provided with a clutch switch sensor MS, which detects when the clutch lever is pulled by the rider, and two buttons, i.e., UB and DB, that the rider uses to request an automatic *up* or *down* gear shift, respectively. Other useful information (e.g., engine speed ω_e and throttle handle opening) can be accessed via the motorcycle CAN bus.

III. Analysis of the Gear Shift Maneuver

Here, a preliminary automatic gear shift logic is presented, and a comparison with manual shifts is considered.

A. *Quickest Automatic (QA) Gear Shifts*

The considered prototype is equipped with an automatic open-loop gear shift logic that comes from the racing-car practice and works as follows. When the rider requests the *up* gear shifting (i.e., when he/she presses the UB button), the *cut-off* and the directional valve EVU are activated. The spark plug ignition is inhibited, and the barrel is moved to the position corresponding to the incoming gear. As soon as the barrel reaches the target position, the *cut-off* is switched off, and the EVU is kept at the hold value for a fixed duration of time T_h. Once this time interval has elapsed, the gear shift is assumed to be completed.

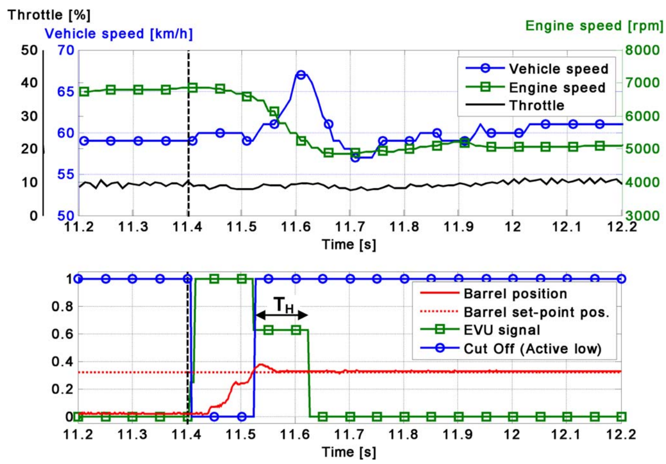

Fig. 3. Time histories of the signals recorded during a (I–II) gear shift with the QA logic.

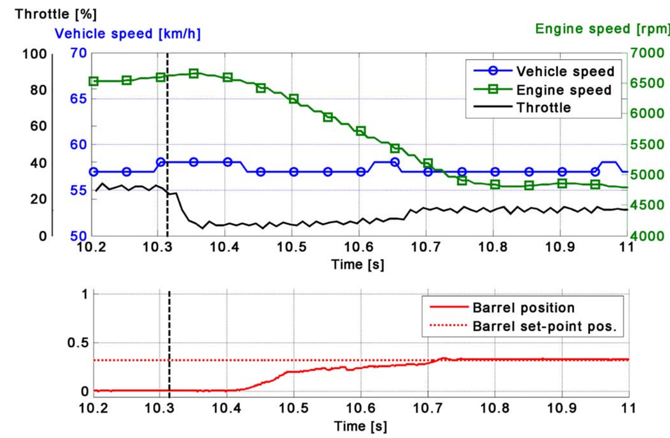

Fig. 4. Signals during an UP (I–II) manual gear shift.

The controller uses the *cut-off* signals and not the clutch to manage the gear shift, and the latter is not involved in the maneuver at all.

The time histories of the signals recorded during a QA gear shift maneuver are shown in Fig. 3. The *up* gear shift starts when the rider presses the UB button, i.e., $t \simeq 11.4$ s; immediately, the *cut-off* and EVU are activated (squared and circled lines in the bottom of Fig. 3). The engine is not fed, and the barrel (solid line) is moved with its maximum speed to the new position. As it reaches the target position (i.e., the position that corresponds to the next gear; dotted line in the bottom plot in Fig. 3), the *cut-off* is switched off, and the EVU is kept at the hold position.

The gear shift duration is the lowest possible (it only lasts within the time interval needed to move the barrel), but the maneuver is extremely uncomfortable for the rider, as shown in Fig. 3. When the *cut-off* is deactivated $(t \simeq 11.52$ s), the engine speed is almost equal to its value at the beginning of the maneuver (see the squared line in the top plot), but the transmission ratio between the engine and the wheel is now different (as can be seen from the barrel position in the bottom plot). This implies that huge variations in the vehicle speed occur (see the circled line in the top plot, for $11.5 < t < 11.7$ s) with negative consequences on both comfort and driveability.

B. Manual Gear Shifts

Manual gear shifts are performed in a completely different way. Specifically, the rider first opens the clutch, then moves the barrel to the position corresponding to the incoming gear while the clutch is open, and finally releases the clutch lever, thus closing the clutch.

In this case, all the operations are sequentially done, increasing the whole time duration of the maneuver (with respect to the previously discussed automatic logic) but keeping a good level of comfort. The sequence of operations is repeatable from rider to rider, but different driving styles (e.g., comfort-oriented or performance-oriented) sensibly modify the time duration of each operation, also impacting maneuver comfort.

The time histories of the signals recorded during a manual gear shift are shown in Fig. 4. The maneuver starts when the clutch lever is pulled ($t \simeq 10.3$ s, i.e., when the MS signal is activated (see the vertical dashed line in Fig. 4); when the clutch is open (note that the clutch position measure is not available in manual gear shifts), by acting on the gear pedal, the rider inserts the desired gear (see the barrel position signal; solid line in the bottom plot) and, finally, releases the clutch lever, to close the clutch. Different from what happened in the QA gear shifts, the vehicle speed is constant during the whole maneuver, whereas the throttle request significantly varies when the clutch is disengaged (solid line in the top plot), to avoid an engine speed overshoot when the clutch is opened.

IV. Objective Assessment of the Gear Shift Quality

In the industrial context, once the control design phase is accomplished and the control system is implemented in final products, an end-of-line tuning phase is usually scheduled to deal with constructive tolerances and production spreads (see e.g., [9] and [10]). This phase is usually carried out by human testers, who optimize the controller parameters based on personal driving preferences and experience. Thus, since no objective indexes to evaluate the gear shift performance and comfort exist, a gear shift can be qualified as comfortable by one operator but not by another. This means that final tuning can lead to very different gear shift behaviors on different vehicles of the same type.

To address this significant problem and to obtain indications that may guide also the design of the gear shift controller, here, we propose a way to estimate the maneuver quality from measured data. The final goal is to automatically classify the performed maneuver and label it with a quality attribute matching that assigned by the rider. This is to be seen as the initial step necessary to design and tune an automatic control system, where the controller parameter values can be adjusted according to predefined cost functions until a given quality of the gear shift is achieved.

For the considered maneuver, the most important characteristics that concur to define its quality are the gear shift duration (in principle, one would like to complete the maneuver in the shortest possible time), the rider's comfort during the disengagement of the outgoing gear and the engagement of the incoming gear, and the overall maneuver performance expressed in terms of the capability of continuously transmitting torque to the ground. These objectives, of course, give rise to a tradeoff, as in general, a quick gear shift tends to induce large accelerations and decelerations, which are the primary source of discomfort for the rider and of poor performance.

A. Duration Index $\mathcal{J}_T$

The duration index $\mathcal{J}_T$ is used to measure the time duration of the gear shift. According to this specification, $\mathcal{J}_T$ is defined as

$$\mathcal{J}_T = t_{fin} - t_{in} \tag{1}$$

where t_{in} is the time instant at which the gear shift commences, whereas t_{fin} is the final time instant. Since the gear shift starts with a rider's request, in automatic mode, t_{in} is uniquely determined as the time instant at which the UB or DB buttons are pressed; in manual mode, instead, the gear shift starts when the clutch-lever switch MS is activated.

To determine the gear shift end, otherwise, there are no measured signals that can be directly employed. Thus, we start from the observation that the gear shift has the objective of varying transmission ratio τ, which is defined as the ratio between vehicle speed v and engine speed ω_e, i.e.,

$$\tau = \frac{v}{\omega_e\, r} \tag{2}$$

where r is the average wheel radius. Since at steady state (i.e., once the gear shift is completed and the incoming gear is engaged) τ is constant, t_{fin} can be estimated as the time instant in which τ becomes equal to the value that corresponds to the incoming gear. More specifically, t_{fin} can be estimated as the first time instant t_1 at which transmission ratio τ assumes the value corresponding to the incoming gear.

In practice, however, this approach can lead to two problems. First, using t_1 as the final instant of the maneuver can cause sensible errors in the duration estimation, especially for particularly sharp maneuvers in which τ signal exhibits considerable oscillations around its nominal value also beyond t_1, which means that the maneuver cannot be actually considered completed at $t = t_1$. Second, also at steady state (i.e., when the gear shift is concluded and the motorcycle proceeds with the selected gear), oscillations of small yet nonnegligible amplitude of the transmission ratio around its nominal value τ^o may still be present. This is mainly due to the transmission elasticity and measurement noise and can cause errors in the correct determination of t_1.

As such, the scatter of the samples of τ measured on a very large number of gear shift maneuvers around the known nominal values τ^o was computed (see Fig. 5). This analysis showed that all sample values vary within a range of approximately $\pm 2.5\%$ around the nominal value. Therefore, the $\pm 3\%$ band around the nominal values τ^o is considered, which ensures a good approximation of the sample distribution, as confirmed in Fig. 5. Time instant t_1 is computed as the first time instant at which the computed transmission ratio τ enters in this band.

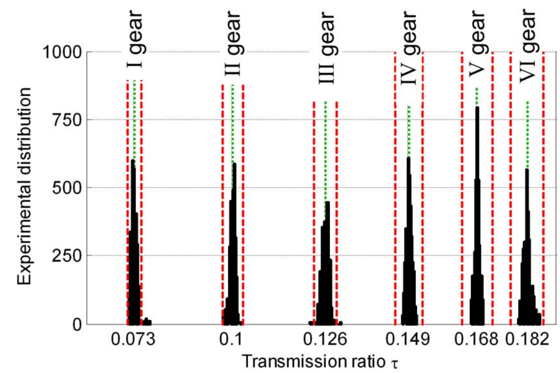

Fig. 5. Scatter plot of transmission ratio τ for six different gears: (black regions) experimental data distributions, (dotted lines) nominal values τ^o, and (dashed lines) $\pm 3\%$ band around τ^o.

Based on this rationale, time instant t_{fin} at which the gear shift ends is defined as

$$t_{fin} = t_1 : \{\tau(t_1) = \tau^o \pm 3\%\}\,. \tag{3}$$

Based on (3), the maneuver duration is computed as

$$T = t_1 - t_{in} \tag{4}$$

where t_1 is as in (3). The gear shift duration defined in (4) can be used to evaluate manual gear shifts only. In automatic gear shifts, in fact, the significant oscillations induced in the vehicle speed by the current controller also reflect on the transmission ratio signal.

To measure the gear shift duration in this case, one should be able to determine when the clutch is fully engaged and the new gear is selected. To do this, a statistical approach was used, and the duration index is computed as $\mathcal{J}_T = T\delta$, where T is as in (4), and δ is given by

$$\delta = \frac{(N + \frac{P}{a})}{N} \geq 1 \tag{5}$$

where N is the number of τ samples in a given time window, P is the number of such samples that lay outside the $\pm 3\%$ range, and a is a parameter that allows tuning the sensitivity of δ to the oscillation magnitude. More specifically, to compute δ, one must fix the duration of a time window over which to observe the transient of the transmission ratio, based on which parameter N is computed. In our work, experimental observations led to fixing the duration of such a time window to 300 ms, as it proved to be enough to account for the oscillations of the transmission. (Note, in fact, that the transient of the transmission ratio also comprises oscillations that are due to the wheel slip dynamics and that, in general, occur at higher frequency and reduced amplitude with respect to the transmission ones, but concur to yield a longer transient than one would have with the sole transmission elasticity.) As for parameter a, it is designed so as to penalize a shift in which there are oscillations in the transmission ratio with respect to another one in which they do not appear. Its value was tuned by looking at the *worst* shifts as far as the duration of the oscillations in the transient is concerned, so that these have a penalty of 100% (i.e., $\delta = 2$) with respect to the best shifts.

After comparative analysis, $a = 2$ was used, and the optimal time window length was set to $N = 300$ ms (with a sampling interval of 10 ms).

Remark 4.1: From (1) and (4), it is easy to show that the time instant at which the gear shift is completed is given by

$$t_{fin} = t_1\delta - t_{in}(\delta - 1). \tag{6}$$

Note that, in the absence of oscillations (i.e., in manual gear shifts), $\delta = 1$ (since $P = 0$), and then, $t_{fin} = t_1$. In other words, δ penalizes the standard duration only if there are oscillations on τ; otherwise, $\mathcal{J}_T = T$.

B. Discomfort Index $\mathcal{J}_D$

Several studies have been carried out in the automotive context, showing good results in evaluating comfort via acceleration measurements (see, e.g., [11]). More specifically, comfort is related to the absence of jerk, which represents the time derivative of the acceleration [12].

Thus, in principle, an ideal gear shift should take place with constant longitudinal acceleration, i.e., with jerk that is equal to zero. Using this assumption to build a jerk reference profile j^o, however, would result in a too demanding performance specification. Hence, an admissible jerk reference profile is defined starting from a speed reference profile v^o, which is defined as follows:

$$\begin{cases} v^o(t) = v(t), & t \in [t_{in} - \Delta t, t_{in}) \\ v^o(t) = \frac{v\left(t^R_{fin}\right) - v(t_{in})}{t^R_{fin} - t_{in}}(t - t_{in}) & \\ \qquad + v(t_{in}), & t \in \left[t_{in}, t^R_{fin}\right] \\ v^o(t) = v(t), & t > t^R_{fin} \end{cases} \tag{7}$$

where $v(t)$ is the measured vehicle speed, obtained as the average of the linear wheel speeds, and t^R_{fin} is computed as

$$t^R_{fin} = t^D_{fin} + \Delta t \tag{8}$$

where t^D_{fin} is the time instant after which transmission ratio τ stays within the $\pm 3\%$ band around the nominal value for all future times (up to the next gear shift), and $\Delta t = 300$ ms is a fixed time interval used to ensure the smoothness of the speed profile.

Remark 4.2: Note that, in manual gear shifts, $t^D_{fin} = t_{fin} = t_1$ [see also (3)]. This is correct, since a manual gear shift ends at $t = t_{fin}$ as no oscillations in the vehicle speed are present.

Once the speed reference profile v^o has been defined, the longitudinal jerk profile j^o can be easily obtained by differentiation, as

$$j^o(t) = \frac{d^2}{dt^2} v^o(t). \tag{9}$$

Based on the jerk desired profile, the discomfort index is computed as

$$\mathcal{J}_D = \frac{1}{\omega_e}\sqrt{\frac{\sum_{t=t_{in}}^{t^D_{fin}} \left(j^o(t) - j(t)\right)^2}{S}} \tag{10}$$

where S is the number of the samples in the time interval $t \in [t_{in}, t^D_{fin}]$, and the normalization with respect to the engine speed is necessary to remove the speed dependence in the discomfort evaluation.

Remark 4.3: To avoid jerk from noisy measured data in the case when only speed measurements are available (the test motorcycle is equipped with an inertial platform to measure longitudinal acceleration), it is possible to assess the gear shift discomfort by means of $\mathcal{J}_{D,v}$, which is defined based on the speed reference profile as

$$\mathcal{J}_{D,v} = f(\omega_e, v^2)\sqrt{\frac{\sum_{t=t_{in}}^{t^D_{fin}} \left(v^o(t) - v(t)\right)^2}{S}} \tag{11}$$

where S is as in (10), and $f(\omega_e, v^2)$ is a normalization function depending both on engine and vehicle speed. Suc correction has to be used to remove both the dependence from engine speed [as it was the case also for (10)] and to take into account the aerodynamic effects, which do not influence the jerk-based indicator. In what follows, the computation of the discomfort index is carried out using the jerk-based cost function (10).

C. Performance Index $\mathcal{J}_P$

An important characteristic of a gear shift is its performance, which is related to the capability of continuously transmitting the torque to the wheel during the maneuver, thus avoiding decelerations. Specifically, the performance index is defined as

$$\mathcal{J}_P = g(v)\left|\frac{a_B - a_D}{a_B}\right| \tag{12}$$

where a_B and a_D are computed as the average acceleration in a fixed time window before the gear shift and the average acceleration during the gear shift, respectively. Namely

$$\begin{aligned} a_B &= \frac{v^o(t_{in}) - v^o(t_{in} - \Delta t)}{\Delta t} \\ a_D &= \frac{v^o(t_{fin}) - v^o(t_{in})}{t_{fin} - t_{in}} \end{aligned} \tag{13}$$

and $g(v)$ is a speed-dependent weighting function that was tuned to remove the speed dependence exhibited by the accelerations. As a matter of fact, according to the specific gear shift, the acceleration involved can be significantly different. As a consequence of its definition, $\mathcal{J}_P$ assumes lower values for high-performance maneuvers and high values in poor shifts. From experimental analysis, a good value for the time-window length Δt_a used to compute a_B is $\Delta t_a = 500$ ms, while a more reliable estimation of a_D is obtained if t^R_{fin} is considered instead of t_{fin} in (12).

From experimental data, it was observed that the ratio $(a_B - a_D)/a_B$ linearly varies with the speed at which the gear shift commences, i.e., with $v(t_{in})$. As such, a speed-dependent weighting function $g(v)$ was tuned to remove such dependence. Note finally that, since $g(v) > 0$, the performance index is always nonnegative.

Remark 4.4: Expression (12) is always well defined except for $a_B = 0$. Although this seldom occurs in practice, since the rider in general requests a gear shift when he/she intends to obtain an acceleration variation and due to measurement noise, this event is properly managed. To this end, the rider was asked to perform several gear shifts, keeping the initial speed as constant as possible (so as to have $a_B \approx 0$). $\mathcal{J}_P$ was computed for these maneuvers, and a reasonable upper bound $\mathcal{J}_{P_M} = 2.5$ was found. Having determined $\mathcal{J}_{P_M}$, the final expression for the performance index is $\mathcal{J}_{Ps} = \min\{\mathcal{J}_P, \mathcal{J}_{P_M}\}$.

D. *Penalized Duration Index $\mathcal{J}_{PD}$*

The overall assessment of the gear shift quality expressed in 3-D space can be difficult. As such, one may notice that both the duration index and the performance index penalize the same aspect of the maneuver. Therefore, the two indexes can be merged to give the *penalized duration index* $\mathcal{J}_{PD}$, which is given by

$$\mathcal{J}_{PD} = \mathcal{J}_T(1 + \gamma \mathcal{J}_P) \tag{14}$$

where γ is a positive constant that can be used to weigh the two terms in $\mathcal{J}_{PD}$ as desired.

V. Proposed Gear Shift Controller

This section illustrates the proposed gear-shifting automatic control logic, which aims at maintaining the performance of the QA controller as far as duration is concerned, while improving comfort by minimizing the accelerations perceived by the rider during the maneuver.

A. *Gear Shift Controller*

To combine the positive aspects of both the existing gear shift policies, namely, the comfort of the automatic one and the duration of the QA, a *mixed* automatic control logic was designed, which makes use of the *cut-off* signal (as in the QA logic) but also adds the active modulation of the clutch position, to smooth the abrupt variations of the speed at the end of the maneuver, which occur when the *cut-off* is disabled and are a major source of discomfort. As in the QA approach, the maneuver commences when the rider presses the UB button. Immediately, *cut-off* signal is applied, and both clutch and barrel valves are activated. As no traction torque flows through the drivetrain in this time interval, the barrel can be rapidly and safely moved to the position corresponding to the incoming gear. Once this has been done, the *cut-off* is deactivated, and it is possible to smooth the engine torque transient by controlling the closing phase of the clutch. It is worth noticing that, according to this rationale, the *cut-off* signal has to be synchronized with the clutch position (and not only with the barrel), deactivating it only when the clutch plates reach the *kiss-point* position x_k. This operation is necessary to avoid overshoots in the engine speed, since in the experimental layout, the *cut-off* signal constitutes the only means to control both the torque in the drivetrain and the engine speed.

The given *mixed* control strategy admits two different implementations, namely, an *open-loop* (M-OL) and a *closed-loop* (M-CL) strategy. The first one consists in fully opening the clutch EVC valve for a fixed time interval T_{on}, without any direct control on its position, whereas the latter requires an

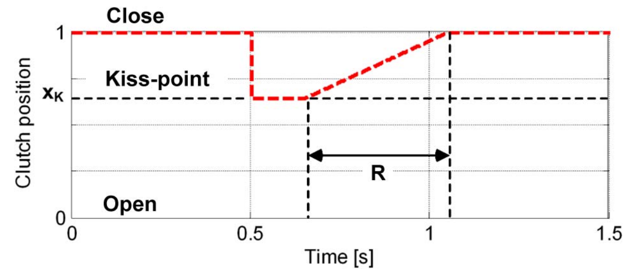

Fig. 6. Typical time history of the clutch set point $x^o(t)$ used in the M-CL control strategy.

accurate position controller, able to track a suitable position profile.

Remark 5.1: The M-OL strategy requires the definition of an optimal value for the duration of the time interval T_{on}^o, during which the clutch input current must be kept at its maximum value to ensure that when the barrel reaches its target position, the clutch is at its *kiss-point* position. In this way, the best tradeoff between gear shift time duration and comfort level is achieved. Note, in fact, that it is not possible to design a tracking loop that regulates the clutch position based on the barrel position feedback, as in the considered layout, the barrel position is a signal that takes on a finite number of values, which correspond to the positions of the different gears plus that of the neutral. When moving from one position to another, in fact, the output of the position sensor is not reliable, and thus, it is not possible to employ such a measurement for feedback. Thus, the clutch opening time T_{on} was tuned to be large enough to cope with the variability of the time that the barrel takes to move from one position to another under the different working conditions. Note, finally, that T_{on}^o slightly varies from one gear shift to another, and thus, it is difficult to fix a value off-line that provides good performance under all working conditions. This is the main motivation that led us to define the closed-loop mixed control approach (M-CL), which, at the price of an increased architectural complexity, allows us to better adapt the maneuver performance to the different situations via the clutch position feedback loop.

In the M-CL approach, the clutch position profile x^o is determined as follows. Its initial value is set to $x^o = 1$, i.e., it corresponds to the position of the closed clutch. When the UB button is pressed, the set point immediately changes stepwise to $x^o = x_k - \Delta x$, which allows us to open the clutch safely beyond its *kiss-point* position x_k (setting $\Delta x = 0.25 x_k$ was determined to be appropriate). The set point is kept constant until the barrel reaches its target position, ensuring that no torque flows through the drivetrain in this phase. Finally, the clutch is closed according to a ramp profile, the slope of which is a design parameter. A typical time history of the clutch set point $x^o(t)$ used in the M-CL control is shown in Fig. 6.

With respect to the M-OL approach, the tracking of such a reference position profile ensures a finer control of the transmitted torque at the end of the maneuver, not only giving the possibility of adapting the clutch closing phase (by varying the ramp slope) to different working conditions but also allowing synchronization between the barrel and the clutch. In fact, the closing ramp starts exactly when the barrel reaches its target

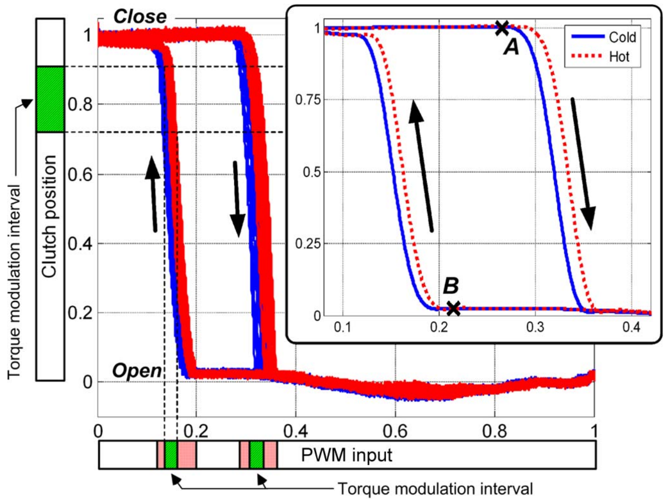

Fig. 7. Quasi-static system response in the input/output plane. (Left) Hysteresis diagram obtained in different tests. (Right) Difference in the hysteresis for (solid line) cold and (dashed line) hot clutches.

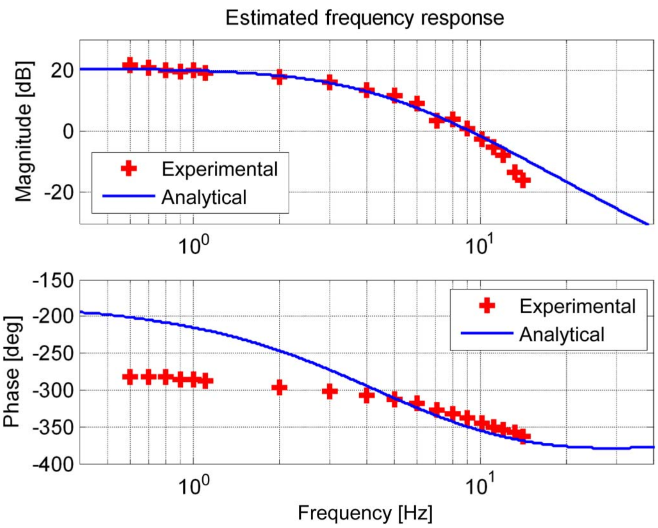

Fig. 8. Estimated frequency response $\hat{G}(j\omega)$.

position (which is a measured variable), thus inherently coping with its dynamic variations.

B. Identification and Control of the Clutch Position Dynamics

To implement the M-CL gear-shifting logic, a clutch position controller is needed. To design it, a control-oriented model of the actuator dynamics must be found. To this aim, identification experiments were carried out on the test vehicle. The first test aims at the characterization of the steady-state relationship between input variable u (i.e., the applied PWM on the EVC valve) and measured output x (i.e., the clutch position). To do this, the system was fed with a sequence of ascending and descending ramps of reduced slope. Several experiments were carried out considering two different temperature ranges $T \leq 70$ °C and $T > 70$ °C, and the obtained results are shown in Fig. 7 in the input/output plane x versus u.

By inspecting Fig. 7, one immediately notices that the system exhibits a significant hysteretic behavior. Moreover, the system is not repeatable, as the left plot shows a large spread among different experiments. In the right plot, a zoom of the input/output characteristic is shown, in which the average value obtained in all experiments carried out in the same temperature range is depicted. As can be seen, there is a considerable difference between the *cold* (solid line) and the *hot* (dashed line) behavior, mainly due to the viscousness of the clutch oil, which largely depends on its temperature. In the following, only the *hot* profile will be considered, since the normal operating temperature range of this component is above 70 °C.

Finally, consider the horizontal and vertical bars that refer to the left plot. The first represents the range of PWM input u, i.e., from 0 to 1, with the light- and dark- gray zones representing the (very limited) range of inputs that is of interest for the clutch movement (it is approximately 14% of the whole u range). Further, the dark-gray intervals, which are even smaller, are the range of values of u that manage the overall torque modulation. The corresponding values of the clutch position that are related to torque transmission are those in the dark-gray interval (accounting for approximately 20% of the entire clutch stroke) of the vertical line, which represents, again in a normalized description, the overall variation of the clutch position. This analysis reveals that the effective modulation range of the control variable is $u = [0.14, 0.18] \cup [0.3, 0.34]$, which is approximately 8% of the whole range. This would not be a problem *per se*, if the resolution of the PWM measurement were very high. Unfortunately, it is tuned according to the whole range, and it is $\Delta u = 0.001$. This means that to actuate the full torque transmission in the considered clutch, input u can take only 80 different values (over 1000), thus making the quantization effects significant. Further, due to the friction characteristics, the system exhibits a *two-state* behavior. Specifically, as soon as the excitation signal reaches an amplitude large enough to break the static friction, the clutch plates move to a fully open (or closed) configuration, although slowly. Due to the *two-state* behavior of the open-loop clutch, the identification experiments were carried out in closed-loop. To do this, a preliminary stabilizing position controller was set up to allow the tracking of a low-frequency position profile.

With the designed PI controller, further closed-loop experiments were carried out, using as input a single-tone sinusoidal signal and using a testbed where the engine speed is constant while the first gear is engaged and the rear wheel is free to roll. The controller was fed with a clutch position set point centered at $x^\circ = 0.5$. After steady-state conditions are reached, a single-tone sinusoidal signal is added to the set point x°. The tests spanned a frequency range from 0.5 to 14 Hz, with an amplitude equal to 25% of the overall position range, tuned to ensure that the linearity assumption holds.

For each experimental test, a nonparametric estimate of the system frequency response was obtained by windowed spectral analysis of the input/output cross-spectral densities [13]. The results are shown in Fig. 8.

The same test procedure was performed starting from different clutch positions (i.e., considering different values of the controller set point x°); however, no sensitive differences were found in the resulting frequency response estimates. Using

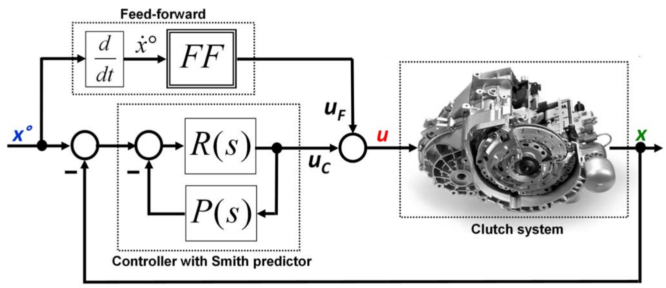

Fig. 9. Architecture of the control system.

an equation error method [14], a parametric estimated was obtained, i.e.,

$$G(s) = \frac{\mu(1+sT_z)}{(1+sT_{p1})(1+sT_{p2})(1+sT_{p3})}e^{-\tau_d s} = G'(s)e^{-\tau s}. \tag{15}$$

The obtained transfer function is shown in solid line in Fig. 8, and its parameter values are $\mu = -10.54$, $T_z = 0.005$, $T_{p1} = 0.0474$, $T_{p2} = 0.0325$, and $T_{p3} = 0.0246$. The delay $\tau_d = 0.019$ s, mainly due to the pipeline geometric size and the maximum admissible fluid flow rate, has been estimated via cross-correlation analysis of the input and output signals. As can be appreciated, the fitting between measured data and the analytical model can be regarded as quite satisfactory.

Remark 5.2: The phase diagram of the identified $G(s)$ given in the bottom plot in Fig. 8 does not explain the experimental points for $f \leq 1$ Hz, which appear to be all close to $\phi \approx -280°$. As will be clear later, at low frequency, the static friction effects are stronger, thus resulting in an extra delay that is not attributable to the linear dynamics of the system. For this reason, the control system must be augmented with an appropriate compensation for such an effect.

Based on the identified model (15), a linear controller was designed and complemented with a Smith predictor to deal with the system delay (see Fig. 9). The time delay τ_d was approximated with a second-order Padé polynomial expression $D(s)$ to achieve a good approximation up to approximately 15 Hz, yielding

$$D(s) = \frac{1 - 0.5\tau_d s + \tau_d^2 s^2/12}{1 + 0.5\tau_d s + \tau_d^2 s^2/12}$$

so that the predictor takes the form

$$P(s) = (1 - D(s))\,G'(s) = \left(\frac{\tau_d s}{1 + 0.5\tau_d s + \tau_d^2 s^2/12}\right) G'(s) \tag{16}$$

while the linear controller $R(s)$ is a second-order filter with transfer function

$$R(s) = g\frac{(1+sT_{z1})(1+sT_{z2})}{s(1+sT_p)} \tag{17}$$

where $T_{z1} = 0.0833$, $T_{z2} = 0.0325$, and $T_p = 0.0033$. The gain was set to $g = -5.91$, which yields a closed-loop cut frequency of $f_c \approx 10$ Hz.

Due to the significant hysteresis and static friction effects, the linear controller alone is not able to guarantee good system performance during transients, displaying a considerable delay in tracking the set point, particularly apparent when analyzing step responses. To overcome this limitation, a piecewise constant feedforward signal is added to the control action u_C, so as to keep the closed-loop system to operate within the region in which the linearity assumption holds. The feedforward signal u_F is computed based on both the first-time derivative of the set-point signal $\dot{x}°$ and the last value of the linear controller output $u_C(t - \Delta t)$. If the clutch is required to open (i.e., if the value of $x°$ decreases and $\dot{x}°(t) < 0$), the control action has to be brought near the opening branch of the hysteresis characteristic (point A in Fig. 7) by applying $u_F(t) = A - u_C(t - \Delta t)$. On the contrary, if $\dot{x}°(t) > 0$, the working condition must be near the closing branch (point B in Fig. 7), so that u_F is switched to $u_F(t) = B - u_C(t - \Delta t)$. Note that the signal u_F is kept constant, whereas $\dot{x}°$ does not change in sign. In particular, to avoid undesired commutations, a threshold with hysteresis was defined around $\dot{x}°$ zero value. Furthermore, in view of the spread of the system steady-state characteristic, values A and B were experimentally tuned (yielding $A = 0.245$ and $B = 0.215$), while $\Delta t = 1$ ms is the ECU sampling time.

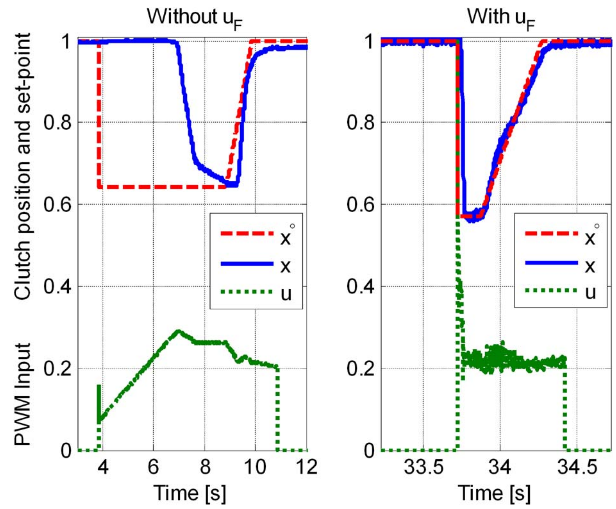

Fig. 10. Closed-loop system response (left) without and (right) with the feedforward action.

In Fig. 10, the closed-loop system response is shown with and without the feedforward component. In this experiment, the motorcycle was kept at constant engine speed, the first gear was engaged, and the rear wheel was free to roll. After reaching the steady state, the controller was fed with the clutch position set point represented with the dashed line in the figure. As can be seen, the feedforward action allows managing the static friction, which, in turn, makes it possible to track faster and larger position variations.

Remark 5.3: Based on the defined quality indexes, optimal control techniques could be, in principle, adopted for control design to achieve the best tradeoff of the cost functionals. To do this, however, one should have a very reliable model for the dynamics, which, as discussed, is hard to achieve with the needed accuracy and must be derived from blackbox identification experiments. Furthermore, the cost functions defined in this work would call for setting up a numerical optimization problem, which should be solved online with considerable computational effort, not compatible with the capabilities of the vehicle control

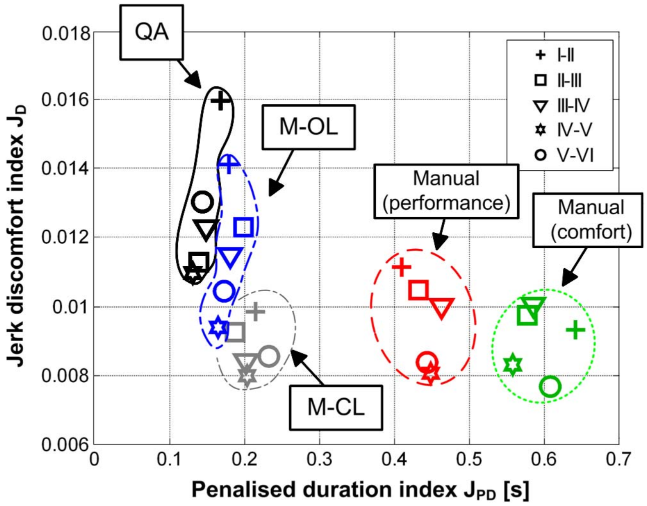

Fig. 11. Results of the quality assessment of *up* gear shifts in the (J_{PD}, J_D) plane: (solid line) QA, (dash-dotted line) M-OL, (line-dotted line) M-CL, (dashed line) manual performance-oriented, and (dotted line) manual comfort-oriented.

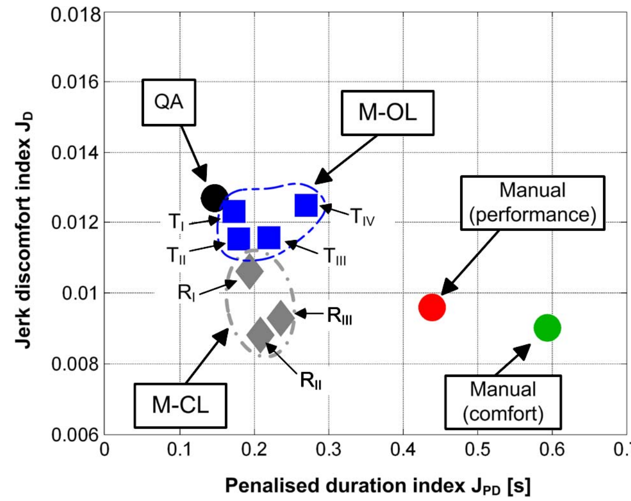

Fig. 12. Results of the quality assessment of *up* gear shifts in the (J_{PD}, J_D) plane obtained with the M-OL and M-CL controllers with different values of the tuning parameters.

unit. Moreover, the optimal solution of such an optimization problem is an open-loop sequence of control values. Although closed-loop optimization-oriented control approaches are available, the most significant of which is model predictive control (see, e.g., [15], [16] and references therein), its use would not allow employment of the selected cost functions, and its online implementation on vehicle electronic boards is hard to realize with the sampling time needed by the gear-shifting dynamics (1 ms). This is why in the considered application, it seems more reasonable to design the control system based on standard techniques and then use the cost functions to quantify the system performance and to objectively guide the tuning of the controller parameters.

VI. Experimental Results

To test the performance of the overall approach, experimental data were collected from tests carried out with a professional rider. Specifically, we will compare the performance of the manual and QA approaches with the M-OL and M-CL logics developed in this work. The rider was also asked to perform two different sets of manual shift: the first with a performance-oriented riding style and the second with a comfort-oriented riding style. The quality indexes presented in Section IV were computed for each type of gear shift, and the results are plotted in the $(\mathcal{J}_{PD}; \mathcal{J}_D)$ plane and shown in Fig. 11. To improve readability, a single point is represented for each gear shift, the coordinates of which are the average of all the maneuvers of the same type. Gear shifts of the same nature are then grouped to better appreciate their distribution in the plane.

As can be seen, M-OL gear shifts are quite disperse in the comfort direction and are scattered in the evaluation plane similarly as those obtained with the QA approach. On the contrary, the performance obtained with the M-CL algorithm in the different gear shifts is much more clustered, confirming the greater repeatability ensured by the closed-loop clutch controller. The M-CL approach also yields the best comfort level (superior to the comfort-oriented manual shifts) and a duration reduced by more than 60%, making it comparable to that of the QA shifts. Such good performance is mainly due to the closed-loop synchronization between the barrel position and the clutch position set point.

Finally, a sensitivity analysis of the performance obtained with the M-OL and M-CL control algorithms with respect to their design parameters was carried out. To this end, four different values for T_{on} have been used in the H-OL controller, namely, $T_{on} = [T_I \;\; T_{II} \;\; T_{III} \;\; T_{IV}] = [20 \;\; 30 \;\; 40 \;\; 50]$ ms, whereas three different ramp slopes are tested with the M-CL controller, namely, $R = [R_I \;\; R_{II} \;\; R_{III}] = [30 \;\; 50 \;\; 80]$ ms. The results are shown in Fig. 12. Again, a single point is depicted for each gear shift, representing the average of all the performed gear shifts of the same type. The results for the manual and QA shifts are also shown with a single point, which represents the centroid of the cluster obtained with the previous results shown in Fig. 11.

As already observed, the M-OL control logic is, on average, slightly faster than the M-CL control logic, although having a worse comfort level. The quality achieved with the M-CL approach is mainly influenced by the synchronization between the clutch and barrel position, and as already stated, the optimal performance is achieved when T_{on} is optimally tuned to $T_{on} = T_{on}^o$. In this case, in fact, the clutch correctly modulates the engine torque minimizing the value of the discomfort index. If $T_{on} < T_{on}^o$, the clutch overcomes its *kiss-point* position before the barrel reaches its target one; thus, when the *cut-off* is deactivated, the clutch is almost closed, resulting in abrupt vehicle oscillations. This happens, e.g., for $T_{on} = T_I$. On the contrary, if $T_{on} > T_{on}^o$, the resulting maneuver takes longer to complete. If the difference between T_{on} and T_{on}^o is small, no sensible impact on the comfort is recorded (see the case of $T_{on} = T_{III}$), whereas for larger values of T_{on}, the comfort level worsens (see the results for $T_{on} = T_{IV}$).

VII. Concluding Remarks and Outlook

This paper has addressed the problem of automatic gear shift control in two-wheeled vehicles, offering one of the first contributions in this area. Two main results were presented: an objective quality assessment of the gear-shifting performance and a new approach to the gear shift control problems. The results obtained on an instrumented vehicle confirm the suitability of the proposed gear shift strategy and offer interesting insights into the tuning of the controller parameters. Future work will be devoted to employing the proposed gear shift controller as a basis to design a full shifting policy that may free the rider from this task.

REFERENCES

[1] B. Jacobson, "Outline of a new control concept for power shifting of fixed step ratio automotive transmissions," *Proc. Inst. Mech. Eng., Part D, J. Autom. Eng.*, vol. 215, no. 5, pp. 613–624, May 2001.

[2] W. Han and S. Yi, "A study of shift control using the clutch pressure pattern in automatic transmission," *Proc. Inst. Mech. Eng., Part D, J. Autom. Eng.*, vol. 217, no. 4, pp. 289–298, Apr. 2003.

[3] M. Goetz, M. Levesley, and D. Crolla, "Dynamics and control of gearshifts on twin-clutch transmissions," *Proc. Inst. Mech. Eng., Part D, J. Autom. Eng.*, vol. 219, no. 8, pp. 951–963, 2005.

[4] S. Savaresi, M. Tanelli, F. Taroni, F. Previdi, and S. Bittanti, "Analysis and design of an automatic motion inverter," *IEEE/ASME Trans. Mechatronics*, vol. 11, no. 3, pp. 346–357, Jun. 2006.

[5] L. Glielmo, L. Iannelli, V. Vacca, and F. Vasca, "Gearshift control for automated manual transmissions," *IEEE/ASME Trans. Mechatronics*, vol. 11, no. 1, pp. 17–26, Feb. 2006.

[6] P. Giani, F. Todeschini, S. Corbetta, M. Tanelli, S. Savaresi, and L. Fabbri, "Control-oriented analysis and quality assessment of gear shifting in motorcycles," in *Proc. IEEE Int. CCA*, 2011, pp. 326–331.

[7] X. Song and Z. Sun, "Pressure-based clutch control for automotive transmissions using a sliding-mode controller," *IEEE/ASME Trans. Mechatronics*, vol. 17, no. 3, pp. 534–546, Jun. 2012.

[8] F. Vasca, L. Iannelli, A. Senatore, and G. Reale, "Torque transmissibility assessment for automotive dry-clutch engagement," *IEEE/ASME Trans. Mechatronics*, vol. 16, no. 3, pp. 564–573, Jun. 2011.

[9] M. Corno, S. M. Savaresi, M. Tanelli, and L. Fabbri, "On optimal motorcycle braking," *Control Eng. Practice*, vol. 16, no. 6, pp. 644–657, Jun. 2008.

[10] M. Tanelli, G. Panzani, S. M. Savaresi, and C. Pirola, "Transmission control for power-shift agricultural tractors: Design and end-of-line automatic tuning," *Mechatronics*, vol. 21, no. 1, pp. 285–297, Feb. 2011.

[11] *Mechanical Vibration and Shock—Evaluation of Human Exposure to Whole-Body Vibration—Part 1: General Requirements*, ISO, 1997, iSO 2631-1.

[12] D. Hrovat and M. Hubbard, "A comparison between jerk optimal and acceleration optimal vibration isolation," *J. Sound Vibration*, vol. 112, no. 2, pp. 201–210, Jan. 1987.

[13] R. Pintelon and J. Schoukens, *System Identification: A Frequency Domain Approach*. Hoboken, NJ, USA: Wiley, 2001.

[14] E. Levi, "Complex curve fitting," *IEEE Trans. Autom. Control*, vol. AC-4, no. 1, pp. 37–43, May 1959.

[15] E. F. Camacho and C. Bordons, *Model Predictive Control*. London, U.K.: Springer, 2004.

[16] M. Morari and J. Lee, "Model predictive control: Past, present and future," *Comput. Chem. Eng.*, vol. 23, no. 4, pp. 667–682, May 1999.