Performance identification and compensation of simulator motion cueing delays

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Abstract – *The lag existing between the command and the resulting cockpit motion in a motion-based simulator, commonly referred to as “transport delay”, is actually the sum of a fixed delay and a frequency-dependent phase delay. A measurement procedure for the identification of the overall transfer function of a motion system is first presented, then is used to design a PID compensator to reduce the apparent simulator lag in usual driving maneuvers. This procedure is carried out on RENAULT’s ULTIMATE high-performance driving simulator. For the reference driving task considered (slalom driving), this filter is shown to bring a 100-200 ms reduction of the phase delay, which is quite perceivable and preferred by test drivers.*

Résumé - *Le délai présent entre les commandes et le mouvement résultant du cockpit dans les simulateurs à base mobile, qui est généralement désigné comme "transport delay", est en réalité la somme d’un délai fixe et d’un retard de phase dépendant de la fréquence. Une procédure d’identification de la fonction de transfert d’un système de mouvement est présentée, puis appliquée à la conception d’un compensateur PID permettant de réduire le délai apparent du simulateur lors de manœuvres de conduites usuelles. Cette procédure est mise en œuvre sur le simulateur de conduite à hautes performances ULTIMATE de RENAULT. Dans la tâche de conduite de référence (conduite en slalom), ce filtre montre une réduction du retard de phase apparent de l’ordre de 100-200 ms, ce qui est tout-à-fait perceptible et préféré par les conducteurs.*

Introduction

Motion-based simulators rely on hydraulic or electro-mechanical actuators to render motion cues as accurately as possible in terms of amplitude, delay and frequency bandwidth. As for any computer-controlled mechanical system, a certain lag appears between the command and the resulting motion. In simulators, this lag is commonly referred to as “transport delay”, and is deemed as being a critical factor for the validity of driving simulators (Nordmark, 1994) and virtual reality systems (Bloche *et al*., 1997). Transport delay creates mismatches between sensory cues, which are considered as a main cause for the occurrence of motion sickness (Oman, 1990), although some level of driver adaptation is possible during active driving (Dagdelen *et al*., 2002). In any case, simulator designers should aim at reducing this apparent lag as much as possible.

The accurate measurement of this lag is often a practical problem, due to the technical complexity of motion cueing systems, which makes the identification of each individual controller and actuator often impossible for the end user. In this paper, we present a procedure based on external accelerometric measurements and numerical identification of the global transfer function. We then present a pseudo open-loop PID control system to enhance the global response of the motion cueing system and to reduce the apparent motion cueing lag. This algorithm takes advantage of the limited bandwidth of the driver commands, in particular the steering input. The approach chosen here is deliberately software-based, thus allowing a generalization to other motion-based simulators.

This development was carried out on the ULTIMATE high-performance driving simulator developed by RENAULT-Technical Center for Simulation, which is based on a X-Y rail actuator system combined with a hexapod (Bosch-Rexroth Hydraudyne B.V., The Netherlands). This motion system is driven in position mode by the SCANeR© software (www.scanersimulation.com) using a predictive motion cueing algorithm developed internally (Dagdelen *et al*., 2009). This simulator is being used for vehicle dynamics engineering applications, for which a typical validation scenario is the 1:1 scale simulation of a slalom at moderate speeds (Dagdelen *et al*., 2006). However, the apparent lag motion feedback was deemed by some expert drivers as being a disturbing factor for the subjective assessment of the transverse dynamics of the simulated vehicle, especially for faster maneuvers (e.g. ESC tests).

Identification of motion platform response

Apparent motion cueing delays

The apparent response lag of a computer-controlled actuator system is generally composed of two terms: a ‘pure’ delay and a phase delay.

The *pure delay* corresponds to the time taken by the computer system to transfer an input information into a command for the actuator system. The computation time, data buffering and numerical filters involved in the different algorithms of the simulator can create significant delays. In a typical multi-process architecture, the different cycle times and the communication protocols between processes can also participate in this delay. Depending on the underlying operating system, this delay may be variable (preemptive OS) or fixed (real-time OS). In most simulators, the data path followed by a driver input is complex and involves several sub-systems, which performance is often beyond the control of the simulator designer: data acquisition (digitization hardware, drivers), communication protocols (e.g. USB, reflective memory, TCP/IP, shared memory etc.), operating system scheduler, etc. This makes the identification of the resulting delay a difficult task without the help of external measurements.

The *phase delay* corresponds to the response time of the motion actuators system, and depends on the technology employed for the motion controllers (frequency and parameters of the control loop) and for the actuators themselves (load, power, damping). In electric motors, the drive electronics and motor coils generally behave as low-pass filters. In hydraulic actuators, the load and internal damping of the actuator also creates a low-pass behavior, not to mention the non-linearity of the valves and pressure supply system. These systems are generally designed to have a global linear response, and as for any linear system, the phase delay will depend on the input signal frequency and on the parameters of the system. For instance, a first-order low-pass filter $G(s)=\frac{1}{1+Ts}$ has a phase delay $t_\varphi = \arctan(T\omega)/\omega$ which varies with the input frequency *f* ($\omega = 2\pi f$) and time constant *T*, and a gain $G = \left(1+T^2\omega^2\right)^{-1/2}$. The performance of motion actuators is often expressed in terms of bandwidth f_c (cut-off frequency corresponding to -3 dB): for this low-pass filter *G*, this frequency is expressed as $f_c = \frac{1}{2\pi T}$ which gives another expression of the phase lag:

$$t_\varphi = \arctan\left(\frac{f}{f_c}\right)/\left(2\pi f\right)$$

The practical interpretation of the apparent delay is an ambiguous issue for the simulator designer: which is the delay that drivers are actually sensitive to?

As a comparison, the handling dynamics of a car (vehicle yaw or lateral acceleration response to steering inputs) also exhibit a certain phase delay, due primarily to the dynamics of the tires, suspension and chassis, and to the flexible structure of the steering system. For instance, the rise time (time to reach 90% of steady state value) for the lateral acceleration in response to a sudden step input on the steering wheel was measured for 169 vehicle models (Riede *et al.*, 1984) and shows a typical range of approx 300 to 600 ms. This rise time can be shown to be approximately 2.5-3.5 times greater than the phase delay for a range of sinusoidal steering inputs from 0.2 to 1 Hz. This estimation is the result of a Matlab simulation of a second-order transfer function between the steering angle and lateral acceleration, derived from a classical bicycle model of the vehicle dynamics (Peng *et al*. 1990). Drivers can make the difference between a relatively sluggish and a sporty reactive car, and are therefore sensitive to apparent phase delays between 100-200 ms (although modern cars would be mainly in the lower range). For a reference 0.2 Hz sinusoidal steering input (Norman, 1984), the additional phase delay introduced by a typical large-amplitude simulator motion system would be 32 ms for a relatively fast actuator (5 Hz bandwidth at -3 dB) and 157 ms for a slower one (1.0 Hz bandwidth). For advanced applications involving the assessment of transient responses of the simulated vehicle, a solution for reducing this additional delay is therefore critical.

Transfer function identification method

In theory, a simple step input is sufficient to estimate both the pure delay and the parameters of a linear transfer function. However, the accuracy of this method is very questionable in practice, due to the limited sampling resolution and the natural presence of noise in the measurements. Although not always perceptible, the position controllers of a motion platform generate a certain level of tremor when holding a set position under load, and accelerometers pick up this vibration quite well. Damping of this background noise would only result in introducing an artificial phase lag in the measurements. Increasing the signal-to-noise ratio is possible by using large platform movements, but this generally leads to reaching its limits in terms of displacement, velocity or acceleration.

Another approach is to design an input signal with sufficient frequency resolution given the expected response bandwidth of the system, while respecting its limits in terms of mechanical travel, speed and acceleration. A balanced frequency distribution is required to avoid biases in the identification procedure of the transfer function, which are generally based on a statistical fit of parameters. The duration of the input signal should also be minimized for practical reasons. The approach chosen here is to use a pseudo-white noise, passed through a low-pass filter to limit the signal to a bandwidth equal to 3~5 Hz for the rails system and 10 Hz for the hexapod system.

As the control algorithms of the motion platform are considered unknown *a priori*, the transfer function of the system G is being approximated by realistic standard models:

- P1D: first-order model, with pure delay $G(s) = \exp(-\tau_d.s).K / (1+T_{p1}.s)$
- P2D: second-order model with pure delay $G(s) = \exp(-\tau_d.s).K / (1+T_{p1}.s)(1+T_{p2}.s)$
- P2UZD: second-order model with pure delay and a zero pole:
- $G(s) = \exp(-\tau_d.s).K.(1+T_z.s) / (1+T_{p1}.s)(1+T_{p2}.s)$
- OE: general n-order model $G(s) = (a_n.s^m + \ldots + a_1.s + a_0) / (s^n + \ldots + b_1.s + b_0)$

For the OE identification, the pure delay term is approximated by a linear transfer function following the classical Padé approximation:

$$\exp(-\tau_d s) \cong (1 - 0{,}5\tau_d s) / (1+0{,}5\tau_d s)$$

For instance, the P1D model can be approximated by the second-order linear model:

$$G(s) = \exp(-\tau_d.s).K / (1+T_{p1}.s) \cong K.(1 - 0{,}5\tau_d s) / [(1+0{,}5\tau_d s).(1+T_{p1}.s)]$$

In this way, the pure delay system is linearized in a LTI system.

The parameters of the transfer functions are identified by means of an optimization method minimizing the errors between the actual measurements and the model outputs. The standard System Identification Toolbox of Matlab was used.

Data analysis and results

Pre-programmed command signals were injected either at the steering wheel input, or at the platform position input, and the resulting cabin motion was measured simultaneously with a set of accelerometers. Accelerometric measurements were fitted with a simple model composed of a pure delay and a n-order linear low-pass filter. The iterative function identification algorithm minimizes the prediction error of the linear, continuous-time model Y(s) = G(s)U(s) + E(s), where the plant model is G(s) = $e(-\tau_d.s)$.H(s). The results obtained with the different models P1D, P2D, P2UZD and OE yield very comparable results, therefore the simplest P1D approximation is sufficient. Non-linearities and higher-order behavior of the actuators are compensated for by the motion platform controllers, which reduce the apparent transfer function of the system to a equivalent filter composed of a low-pass and delay terms.

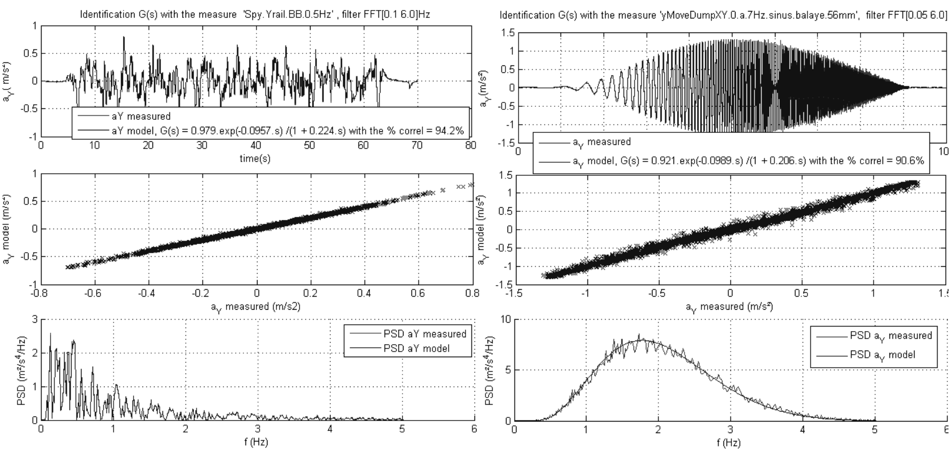

Figure 1. Example of identification output signal (top), comparison of model predictions vs. measurements (middle), and acceleration spectral distribution (bottom), for a pseudo-white noise (left) and swept sine (right)

The measured response of the ULTIMATE X-Y rail and hexapod systems is summarized in Table 1.

Table 1. Delay identification results for the Ultimate simulator

	Pure delay	Phase delay at 0.2Hz	Bandwidth (-6 dB)
Hexapod	30~35 ms	35 ms	7 Hz
Rails	15~20 ms	200 ms	1.25 Hz

The accurate identification of the pure delay term is critical for the stability of a phase delay correction algorithm presented in the following. As the pure delay term is of lesser amplitude for the rails system, its compensation is of lesser importance and is not considered here (moreover, the correction of pure delays involves specific algorithms with potential stability constraints).

Correction of the apparent motion cueing delay

The possibility to reduce the phase delay by modifications of the simulation software is analyzed here, as being a flexible and cost-effective alternative to extensive hardware modifications. Advanced methods are available to compensate different sorts of delays (e.g. Smith predictor, fuzzy logic or adaptive models for pure delays; PID for phase delay; others for non-linear and time-varying plant models), but they require a closed-loop control of the system. Their implementation in the simulator would require a substantial modification of the motion system. In the following, a pseudo open-loop solution, which relies on model predictions, is considered.

Implementation of a PID corrector

A PID corrector is a simple and robust way to shape the response of a given system, by using a negative feedback of its output. In our case, the measurement of the platform motion output entails additional hardware and measurement delays, and a numerical model of the motion system is used instead. The system model, comprising a pure delay and a first-order low pass filter, comes from the identification procedure presented above.

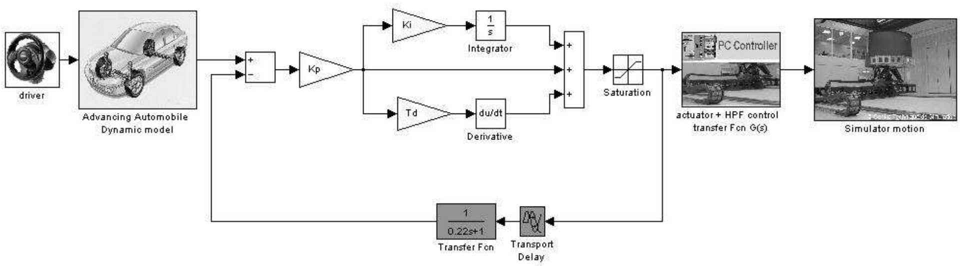

Figure 2. Structure of the proposed delay compensator

The performance of a PID corrector is generally assessed by analyzing the deviations from a step input command. Among the variety of available algorithms, the methods of Ziegler-Nichols and Cohen-Coon (De Laminart, 1993; Corriou, 2003) are considered here, as being particularly suitable for first-order systems with a pure delay.

Let us consider the general actuator model as identified for the motion platform:

$G(s) \cong K_G.\exp(-\tau_d.s)/(1 + T_{P1}.s)$

The Zielger-Nichols identification procedure yields the following PID parameters:

$Kp = 1{,}2.T_{P1}/ (K_G.\tau_d)$, with $Ti = 2\tau_d/Kp$ and $Td = 0{,}5\tau_d.Kp$

The Cohen-Coon identification procedure yields the following parameters, with a lesser sensitivity for the delay parameter:

$$Kp = (1/K_G)(T_{P1}/\tau_d)(4/3 + \tau_d/(4.T_{P1}))$$

$$Ti = \tau_d(32 + 6(\tau_d/T_{P1}))/(13 + 8(\tau_d/T_{P1}))/Kp$$

$$Td = 4\tau_d \, / \, (11 + 2(\tau_d/T_{P1})).Kp$$

The accuracy of the model parameters is crucial for the performance of the corrector. In particular, improper values for the pure delay τ_d will result in an unstable behavior of the corrector. According to De Laminart (1993), this PID controller gives an excellent result when T_{p1}/τ_d is important (i.e. over 5-10), which makes this technique applicable in our case (cf. Table 1).

Results

The simulation of the corrected system for a 0.1-3 Hz swept sine input (Figure 3) confirms the nulling of the phase delay and the correction of low-pass gain loss. Some over-amplification appears at higher frequencies, which is deemed acceptable at this stage, but which could be corrected by an adaptive PID parameter algorithm.

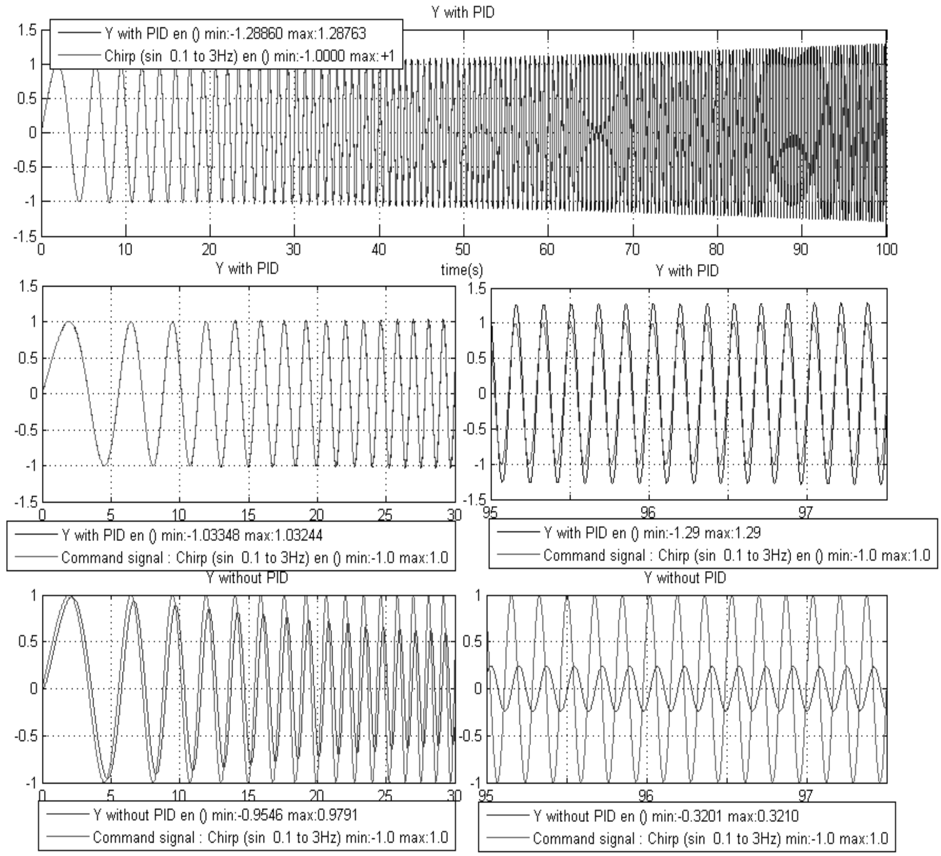

Figure 3. Simulation of PID compensator performance for a swept sine (top), focus on low frequencies (middle left) and high frequencies (middle right), and comparison without compensator (below)

This corrector was implemented in the ULTIMATE software by placing it after the vehicle dynamics model (Fig. 2), thus reducing the higher frequencies of the inputs of the filter which may create artifacts such as overshooting. Interactive driving tests were carried out with an accelerometer on the cabin. Despite the simple approximation of the actuator model, the performance of the corrector on the simulator performance is substantial. Figure 4 shows a result of off-line simulation evaluation for a slalom driving (delay reduction of 220 ms), and Figure 5 a real driving measurement with a delay reduction of 230 ms. Pilot subjective tests confirmed that drivers can identify properly (and prefer) the corrected configuration.

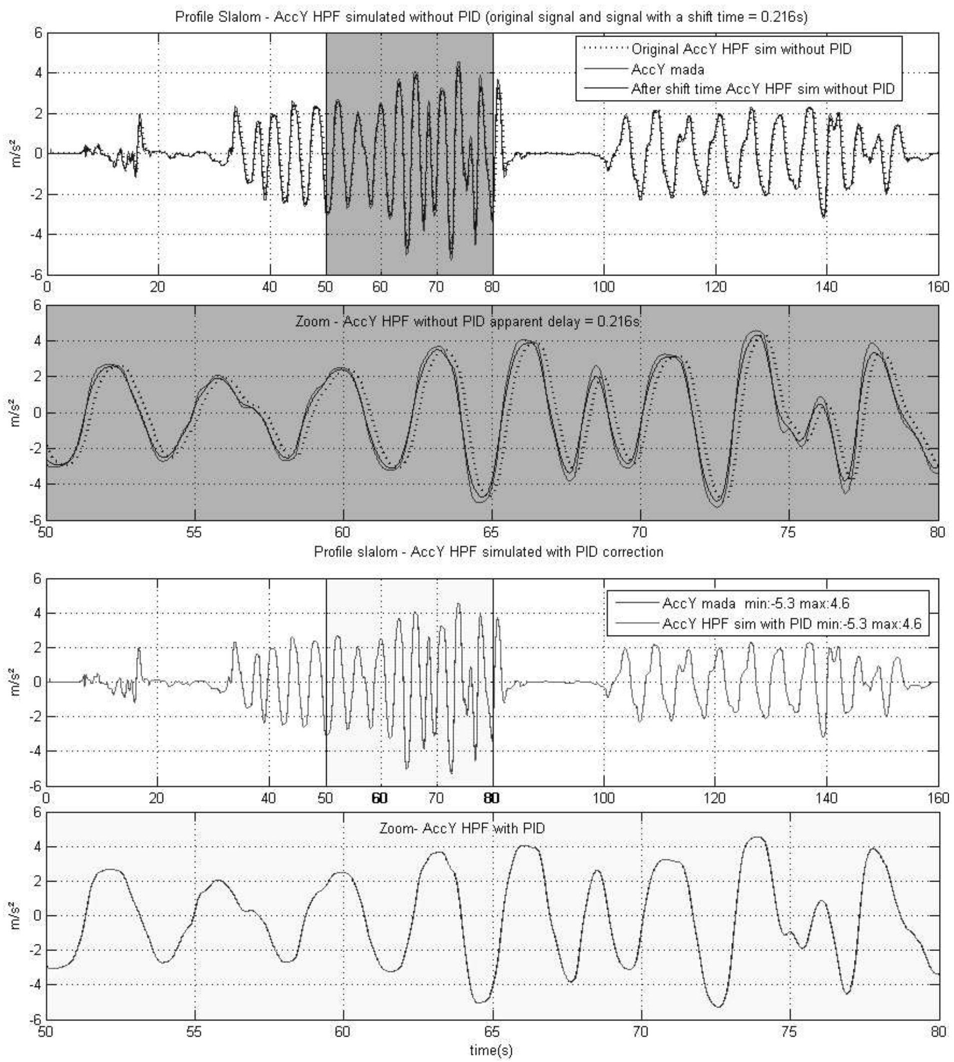

Figure 4. Comparison of off-line simulation of platform lateral motion with and without PID compensation

The validity of the parameter set chosen for the corrector actually depends on the driving scenario considered. Usual driving maneuvers correspond to steering inputs with a frequency bandwidth of [0-2] Hz, so the tuning described here will produce satisfactory corrections in most of the normal driving situations. However, faster maneuvers will call for a different tuning, with an improved PID parameter algorithm (adaptive, expert control etc.). This tuning may be necessary for instance during the simulation of lateral wind gusts, or for sudden braking maneuvers.

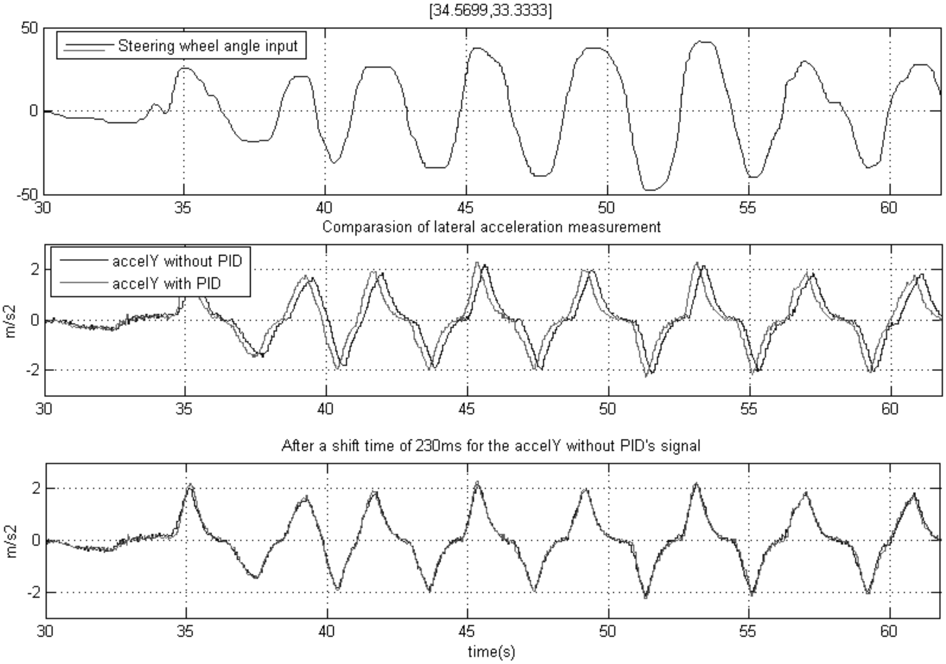

Figure 5. Measurements of a slalom driving session, with an estimation of the delay reduction

Conclusion

The present study shows that a substantial reduction of the apparent simulator delay is possible using a pseudo open-loop PID corrector tuned from a model of the motion system. This reduction depends on the phase delay of the actuators (which varies with frequency) and on the frequency range of the driver inputs, and typically varies between 100-200 ms. The resulting apparent delay of the simulator is therefore closer to the normal phase delay of the simulated vehicle, which was confirmed by comparative subjective driving tests.

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