Dynamic modeling of primary commands for a car simulator

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Abstract— Simplified dynamic analytical models of primary commands of a car, i.e. steer wheel and gearshift, are identified and developed in the paper. The dynamic models are used to design the control law of force feedback devices, which will be integrated in a car simulator. The model simulation results match satisfactorily with the experimental available data. An experimental partial assessment of control law for the gearshift simulation has been performed with a commercially available force-feedback joystick. The gearshift simulation control is implemented with an hybrid model, based on a state machine. The results are presented and discussed.

I. INTRODUCTION

Vehicle simulators, and in particular aircraft simulators for use in training pilots, are generally equipped with various control devices such as acceleration pedal, brake pedal, clutch pedal, gear change lever and steering wheel, to enhance the realism of the simulation.

In most of such simulators [8], [2], [5], control commands are employed as passive sensorized systems, which measure the commands executed by the user. In such systems the force behavior is regulated by the mechanics of the control commands, designed to replicate the force response of real commands.

The adoption of Haptic Interfaces (HI) within simulator systems appears to be an added value which improves the functionalities of the simulator. The mechanical response of HI can be logically programmed and controlled. In such a way in tests within simulator users can evaluate the vehicle by changing either kinesthetics or Haptic parameters. This is particularly useful when ergonomics aspects are taken into consideration [1], [6].

In [9] the steer wheel was replaced with a force controlled Haptic Interface, whose behavior was based on an internal numerical model of the vehicle.

The ATARI corporations patented two inventions regarding the simulation of a gearshift [3], [4]. The devised systems are based on a mechanism which allows the gearshift lever to pivot around at least two axes. In the first invention [3] the mechanism is equipped with electrically operable clutches, that can cause only resistance to the movement of the lever. Positional sensors and strain gauges coupled to the gearshift lever are used to sense the lever movements and forces. In the second invention [4] a solenoid is coupled to the pivoting mechanism and controls the amount of force applied.

This paper concerns the dynamic modeling of the primary commands of a car, i.e. the steering wheel and the gear shift. Models have been set up to replicate the forces felt by the driver during a real drive in a car simulator, aimed at evaluating the ergonomics of internals of car in Virtual Environments [1]. The mechanical response (i.e. the exerted force related to the displacement imposed by the driver) of these primary controls will be controlled via software.

In this paper the control and dynamics models of the steering wheel and manual gearshift are presented and analyzed. The Haptic Interfaces, to be used for the simulation of the primary commands, and the simulator mock-up are currently under construction. The results of a preliminary evaluation of the gearshift control model are reported, as obtained with a commercially available force feedback joystick. The control law will be finally implemented on a 2 DOF Haptic Interface prototype, currently under construction at PERCRO.

II. THE STEERING WHEEL MODEL

The steering wheel torque estimation is based on a simplified model, that allows to perform the simulation in real time, by reducing the computing time required by a complete vehicle model. The solution is calculated through an algebraic formula, instead of a system of differential equations. The steering wheel simplified model calculates the torque $\tau$ at the steering wheel as a function of the vehicle longitudinal velocity $V_x$ and the steering wheel angular displacement $\delta_w$. The approach proposed in this section has been validated by experimental data collected for a FIAT PUNTO. The steering wheel torque can be split into different terms to clearly show the physical meaning of the equation:

$$
\tau = \tau_1(\delta_w) + \tau_0(A_y) + \tau_1(V_x, \delta_w) + \tau_2(\delta_w) + \tau_3(\delta_w) \quad (1)
$$

The term $\tau_1$ is the inertial torque. Knowing the steering wheel moment of inertia $J_w$, the value of $\tau_1$ is:

$$
\tau_1(\delta_w) = J_w \delta_w \quad (2)
$$

The term $\tau_0$ is the torque due to the vehicle lateral acceleration $A_y$, which has been derived from the linear time-invariant bicycle model. However, since it is not correct to
consider that the vehicle behaviour is linear in every condition, so the lateral acceleration of the bicycle model \( A_{y,b} \) has been adjusted to fit with experimental data. The relation between the \( A_y \) and \( A_{y,b} \) derived from the bicycle model is:

\[
A_y = K \cdot \max(A_y) \cdot \tan^{-1}\left(\frac{A_{y,b}}{K_0}\right)
\]  

(3)

On the basis of equation 3, the expression of \( \tau_0 \) determined by interpolation of track data acquisition is:

\[
\tau_0 = \frac{K_1 A_y}{1 + K_2 A_{y,b} K_3}
\]  

(4)

The term \( \tau_1 \) is the torque due to the tire spin. The parameter that most influences this torque is the vehicle longitudinal velocity \( V_z \). In order to have the torque sense and avoid discontinuity, the steering wheel angular velocity is also taken into account in the following equation:

\[
\tau_1 = \frac{1}{K_4 + K_5 V_z} \cdot \tan^{-1}\left(\frac{\delta_w}{K_6}\right)
\]  

(5)

The term \( \tau_2 \) is the torque due to the friction present in all the steering system:

\[
\tau_2 = K_7 \cdot \tan^{-1}\left(\frac{\delta_w}{K_8}\right)
\]  

(6)

The term \( \tau_3 \) represents the damping contribute:

\[
\tau_3 = K_9 \delta_w
\]  

(7)

To sum up, some comparisons between the values of the steering wheel torque, calculated with the proposed method, and the experimental data, collected on the track for an FIAT PUNTO, are represented below. Different kinds of manoeuvres have been used for the model validation: "steady state" manoeuvre (Fig. 1), "steering wheel angle imposed" manoeuvres (Fig. 2 and 4), "trajectory imposed" manoeuvre (Fig. 3).

III. THE GEARSHIFT MODEL

Differently from other primary controls, the gearshift force behavior is highly non-linear and unpredictable, since it is related to the instaneous collisions that occur in the gearbox. In the following it is presented an analytical approach for modelling the gearshift force response, based on the force contents displayed during the engagement as measured experimentally, and a preliminary assessment of results. Prior to introduce the model implementation, a brief description of the gear engagement is given, considering the involved mechanical aspects.

A. Mechanical description of a manual transmission gearshift

The gearshift allows to change the gear ratio between the primary shaft, connected by the clutch to the motor, and the secondary shaft, which conversely is permanently connected to the differential unit, and so to the wheels.
The lever from its equilibrium position.

After the lever has been released, another preset load must be applied to displace the coupling from its neutral position. In fact the coupling is hold in the neutral position by a second spring-ball mechanism (placed in the gear-box), whose function is to accomplish a softer mesh of the synchronizing ring and the coupling. This is the pre-synchronizing stage.

After the pre-synchronizing stage has been achieved, the synchronization can begin. The coupling engages the synchronizing ring and pushes it against the gear. The internal conical surface of the synchronizing ring is brought in contact with the external conical surface of the gear, which is rotating dragged by the primary shaft. The tangential friction forces, which the two bodies exchange in reason of their relative motion, are transformed in axial forces, through the taper of the synchronizing ring, and impede a further sliding motion of the coupling.

Until a relative motion between the synchronizing ring and the gear exists, the coupling (and so the gearshift lever) is blocked into a fixed position and can not go forward. Only when the relative velocity becomes null, the “synchro gate” can open and the coupling can continue its sliding motion.

At the end of synchronization, the block of the coupling is released and so the coupling teeth of the gear and synchronizing ring collide and then mesh. The main feature of the coupling tooth contact is that the generated impact forces are random, depending on the relative position of teeth at the moment of the engagement. After the full engagement, the lever reaches its mechanical stop, based on a spring-ball mechanism also.

The data displayed in figures 5 and 6 reveal typical force characteristics with respect to time and engagement position. Both plots have been collected by sensorizing the gearshift knob aboard an experimental car.

B. The gear engagement process

Three different main stages occur during the gear engagement, which characterize the particular force response of a gear-shift: the synchronizing, the engagement and the impact against the mechanical stop. Since the engagement is a multi-body dynamical process, each stage is associated to the interaction of different parts in the gear-box. In the following the forces felt during a gear engagement are explained according to the gear-shift dynamics.

A spring-ball system (placed in the housing of the lever mechanism) constrains the lever in the selected gear position. So an initial preset load must be applied to displace the lever from its equilibrium position.

Fig. 5. Force and position vs. time, as measured at the knob during neutral-first gear shift (courtesy of CRF)

The pre-synchronizing stage can be reasonably rid off in the simulation, since it generates a negligible force peak only.

During the synchronizing phase both the force and the
position are held constant, as shown in figure 5. The force reaches its maximum value, and the position is held constant for a definite period of time.

The engagement stage is characterized by an isolated peak force, that is lower than the synchronizing force peak. Moreover the magnitude of such pick force is variable, so that engagement peaks can vary remarkably. As shown in figure 6 the synchronizing and the engagement peaks occur at definite values of the x position. In particular the synchronizing stage reaches a peak value of about 8 Kgs, and has a duration of about 0.3 msec. The engagement peak is instantaneous instead, since it is due to the impact of the gears teeth.

The final stage gives raise to the stop impact peak. Since it is mediated by an elastic stop system, there is an overshoot and a following recovery to the equilibrium point, as in figure 5.

The forces that the driver exerts on the lever, when he changes into a gear, are so determined mainly by the stages of synchronizing, engagement and stop impact. A realistic simulation of a gearshift must replicate rigorously these three phases, and it can neglect the pre-synchronizing stage, because of both its low endurance and small forces.

C. The gearshift engagement model

An analytical model of the gearshift behavior was synthesized to replicate a correct force-feedback to the operator. The different phases of the gear shift have been modeled through a dynamic model with both continuous and discrete states (each discrete state was associated to a gearshift stage). The gearshift response has been developed as a MATLAB Simulink/Stateflow module, since it represents an hybrid model in itself.

The GEARSHIFT ENGAGEMENT model, shown in figure 7, takes as input the user forces exerted on the knob and provides as output the knob position and velocity. The model can be divided in two parts:

1. A time varying continuous dynamics which depends on the gear stage (GEARSHIFT DYNAMICS).

2. A discrete state machine (GEARSHIFT STATE-FLOW) which determines the gear stage on the basis of the knob position, the user’s force and the previous machine state.

The GEARSHIFT DYNAMICS has been implemented as a parametric mechanical system composed of a mass m, spring k, damper c with a stick-slip friction:

\[ F_{dr} = m \ddot{x} + c \dot{x} + k(x - x_0) + F_{fr} \]  

(8)

where \( F_{dr} \) is the driver force, \( x \) the position and the friction force \( F_{fr} \) is given by:

\[ F_{fr} = \begin{cases} F_{dr} & \text{if } F_{dr} - c \dot{x} - k(x - x_0) \leq F_{st} \\ \text{else} & F_{st} \end{cases} \]  

(9)

with \( F_{st} \) and \( F_{sl} \) respectively the static and dynamic friction coefficients. The parameters \( m, k, c, x_0, F_{st}, F_{sl} \) are set to different values for each stage by the GEARSHIFT STATE-FLOW module, according to the current stage.

The GEARSHIFT STATE-FLOW module receives as inputs the x position of the knob and the force exerted by the user. The discrete states of this module represent the different gearshift stages outlined in previous subsection, and so the synchronization, the engagement and the end impact. Moreover free motions states have been added to model the lever behavior out of these stages. Figure 8 shows a simplified scheme of the state machine, which simulates the engagement process.

![Engagement process](image-url)
The input information are used to manage the transitions among different states. For instance if the user is pushing forward during the synchronization state, a transition is activated to reach the engagement state. Conversely if the force is suddenly reverted, an in-transition to a free motion state is activated.

The conditions under which the transitions can occur are based not only on position and force, but on time also. So the duration of the synchronization stage is imposed by a time-dependent out-transition from this state.

Actions and events broadcasting are associated both to the transitions and to the states. In particular different assignments to logical variables are executed according to the active state. The logical variables, containing the information about the active state, are then output to the GEARSHIFT DYNAMICS module, to control the current values of the simulation parameters.

D. Simulation of the engagement model

To test the efficiency of such a model, a driver module was also implemented in the Simulink environment.

The driver has been modeled assuming that his force behavior is inversely linear related to the gearshift knob velocity:

\[ F = F_{\text{max}} \left( 1 - \frac{\dot{x}}{V_o} \right) \]  

(10)

where \( V_o \) is the target velocity and it is assumed equal to the maximum one measured during the engagement movement, while \( F_{\text{max}} \) corresponds to the maximum force that the user can apply. Such a force model with a velocity feedback of the driver can be adapted to fit the behavior of different drivers.

The driver module has so been interfaced to the GEARSHIFT ENGAGEMENT module in a velocity closed-loop (the position output of the GEARSHIFT ENGAGEMENT module has not been used in this simulation).

Figure 9 shows numerical results achieved during the simulation. As shown in the figure all main stages of engagement are replicated by the model. Model parameters have been regulated in order to match maximum forces, position and reference times to those of the experimental results.

E. Experimental validation of the gearshift model

A 2 DOF force feedback joystick has been employed for carrying out an experimental validation of the engagement model. For such purpose we adopted the "Wingman Force Feedback Joystick" by Logitech, a commercially available device with proper drivers and control board.

The Joystick is equipped with two DC motors and two analog potentiometers, which are connected to the handle through a 2 DOF parallel pivoting mechanism. Such a mechanism allows the lever to pivot around two co-planar axes. The motors transmit the torque through a capstan tendon transmission system, both to increase the motor torques and to reduce the mechanical plays between motors and potentiometers.

Since our aim was to test the efficiency of the algorithm, we excluded the built-in control processor, and connected the Joystick to a high performance control board (DSP DS1102 dSPACE). By a rough estimation of the maximum actuated force at the knob with the DSP arrangement, we found a value of about \( \sim 10N \), that is one tenth of the target force of 10 Kg we wished to replicate. Also the workspace dimensions were smaller than the required ones. So the presented results was scaled to the experimental ones of figures 5, just to test the validity of the control algorithm.

We interfaced the Joystick to the Matlab environments through the Real Time Workshop toolbox. The model was downloaded and run directly on the DSP board. A GUI interface panel was developed to control and capture the data of the simulation.

All simulations were performed at a frequency of 1 KHz to avoid problems related to simulation sampling time.

First we tested the force feedback during pure engagement operations. The force-feedback control scheme was based on a admittance force-display [7], which measures force and displays motion. The forces were estimated by the current inputs, sent to the motor drives, while the positions were read by two analogical potentiometers.

During simulation the y position of the joystick was constrained to zero to simulate a 1 DOF sliding constrain.

The following plots show the obtained results. The error on the force signal was determined by the electric noise produced by the analog potentiometers. Plot 10 shows the filtered (with a low pass Butterworth filter and a cut-off frequency of 20 Hz) and the non-filtered signal. The curve matches satisfactorily the experimental one, and also the feeling during the engagement was realistic and similar to a slow gear engagement during drive. The hardware restrictions did not allow to improve further the performance of the system (position signal resolution and noise, internal mechanical play, friction), but the devised control law seemed to work properly.

After the simulation of a single gear engagement, we stepped to the development of a complete gearshift. The
model was extended to consider the constraints imposed by the slides of each gear. The y axis was controlled by using a parametric module similar to that of the x axis, but with a lower number of states. In particular the workspace of the gear shift was divided, according to the x and y, into numerous areas, associated to a different state of the gear shift. A main state-flow module, called STATE-FLOW MANAGER, identifies the current area, and forwards this information to the X and Y controller modules. Inside these modules, the action executed, either selecting horizontally a gear or engaging vertically a gear, is recognized, and the corresponding state behavior is activated. For instance when the user is engaging a gear, the previously analyzed model of a single gear is activated. The slides constraints were replicated as mechanical impedances, which restrain the knob into rectilinear trajectories.

The final simulated workspace is shown in figure 12.

IV. FUTURE DEVELOPMENTS

The haptic interfaces dedicated to the primary commands simulation are currently under construction at PER-CRO and successively will be integrated in the mock-up of the system. The limitations encountered in the gearshift simulation control of a commercial joystick will be overcome by the new HI, specifically designed to satisfy the requirements of a gear shift simulator.

V. CONCLUSIONS

In the present paper an analytical model of the primary commands for a car simulator have been presented. The models are based on experimental data and simplified in order to accomplish a real-time simulation of the real force response in a Virtual Environment.

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