Adaptive Haptic Feedback Steering Wheel for Driving Simulators

Hakim Mohellebi, Abderrahmane Kheddar, Member, IEEE, and Stéphane Espié

Abstract—Controlling a virtual vehicle is a sensory-motor activity with a specific rendering methodology that depends on the hardware technology and the software in use. We propose a method that computes haptic feedback for the steering wheel. It is best suited for low-cost, fixed-base driving simulators but can be ported to any driving simulator platform. The goal of our method is twofold. 1) It provides an efficient yet simple algorithm to model the steering mechanism using a quadripolar representation. 2) This model is used to compute the haptic feedback on top of which a tunable haptic augmentation is adjusted to overcome the lack of presence and the unavoidable simulation loop latencies. This algorithm helps the driver to laterally control the virtual vehicle. We also discuss the experimental results that demonstrate the usefulness of our haptic feedback method.

Index Terms—Adaptive haptic augmentation, driving simulator, haptic feedback steering wheel, user-centered design.

I. INTRODUCTION

Driving simulators are virtual reality tools used by researchers and designers in the domain of vehicular technologies to study the drivers’ behavior in various road situations, e.g., [1] and [2]. This tool is even more interesting because it is difficult to replicate most actual road/traffic situations in real conditions, even with sophisticated driving simulators. Hence, designing driving simulators is mainly a problem of dealing with astutely combining sensory illusions and sensory substitutions with technological solutions. Moreover, a simulation allows a wide spectrum of parameter adjustments and settings [3]. The aim of the outcome, however, is to put the user in front of driving situations so that she/he behaves as closely as possible as if she/he was in a real driving situation.

Various experiments show that motion perception is basically induced and dominated by three sensory modalities, i.e., vision, haptics, and vestibular channels of the inner ear. The integration of these modalities (see [4]) allows the driver to have both proprioceptive and exteroceptive perceptions [5] while driving.

In a driving simulator, the lack of a display or a poor-quality display of one or several stimuli results in an inconsistent perception of movement. Such misperceptions can disturb the drivers’ performance with nausea, i.e., cybersickness [6].

Some tasks that are easily achieved in a real driving situation (lane shift or queuing for instance) become tedious when the driver has to accomplish them using a driving simulator, particularly when using nonmoving or low-clearance platforms [7], [8]. Indeed, the absence of sensory stimuli, the latency between the driver’s actions and the restitution of their effects, the lack of coherence between each rendered cue, and the relative quality of that multimodal rendering (lack of immersion presence) prevent the driver from adequately controlling the virtual vehicle.

The time needed for the acquisition, computation, and rendering of the virtual scene generates latencies. Consequently, we noticed that most drivers tend to amplify the steering wheel rotations, which leads to simulation instability and loss of vehicle control. To operate the driving simulator, the user needs an adaptation period to learn to anticipate the dynamics of the feedback (dominantly visual). This problem is recurrent in most interactive simulations based on virtual reality techniques. Solutions developed in haptic feedback, consisting of bilateral damping, can apply to this context. As shown in this paper, the provision of an additional cue (e.g., haptic), which is consistent with the visual feedback, improves performance when controlling the virtual vehicle.

This paper addresses the haptic feedback issue on steering wheels for driving simulators. A modeling approach, which is based on the quadripolar representation of each module of the overall steering system, and a haptic feedback steering wheel have been integrated into the Institut National de Recherche sur les Transports et leur Sécurité (INRETS) fixed-base simulator. Moreover, a new haptic augmentation algorithm is devised. This algorithm runs as a corrector with a forward phase. It anticipates the drifting of the vehicle from the middle of its lane and transmits that information to the driver through the haptic steering wheel. This assist corrects the orientation of the steering wheel, helping the driver to better control the virtual vehicle. Fundamentally, it can be seen as a damping technique through haptic augmentation [9] or a look-ahead haptic guidance [10] that maintains the operator’s natural interactivity by compensating the simulation latencies and the lack of other sensory information. Subsequently, it maintains a similar behavior of the driver in a simulation context (which is important when studying driving behavior). Experiments highlighting the effects of haptic feedback on driving performance are conducted and analyzed.
II. BACKGROUND

According to the results of previous research [11], [12], the effort produced through the tire/road interaction, as it is perceived by the driver via the steering wheel, is among the most useful information, after vision, in guiding the vehicle. For these reasons, all of the systems coupled to the steering system aim at improving driving comfort, performance, and safety. The most widespread implemented system is power-assisted steering. It reduces the efforts that go up to the steering wheel [13], according to the vehicle speed [14], [15] (for most of the cases). Actual steering efforts are strongly reduced for low speeds to provide the driver with better maneuverability, and they are relatively amplified at higher speeds to secure driving.

Techniques such as steering-by-wire (SbW), e.g., [16]–[19], fully mechanically uncouple the steering wheel from the wheels. In SbW, force-feedback control is needed to recreate on a steering wheel (or another equivalent driver–car interface) a drive feeling like the one produced by a traditional mechanical steering system. Using these systems, haptic feedback on the steering wheel is accomplished through an electric actuator that simulates, in an artificial manner, the force coming up from the wheels, e.g., [20]–[22]. That way, the haptic feedback can fully be controlled, whereas it is possible to integrate haptic guidance algorithms.

Basically, replicating force feedback in driving simulators or in SbW systems shares common problems and methods. In both cases, there is no mechanical link between the vehicle’s wheels and the steering wheel. The hardware is obviously different since for real use, it should comply with several requirements in terms of robustness, safety, etc. However, both need a real-time dynamic model of the vehicle, with more embedded sensors in the case of SbW [17], [18]. Therefore, the method developed in this paper would certainly partly apply to SbW.

III. BASIC MODEL

The driver applies light efforts/torque on the steering wheel to maintain a desired trajectory of the vehicle. These applied efforts counter those coming through the steering column and that are the aggregate of the tire/road interaction forces and the vehicle dynamics. To replicate the steering dynamic effects on a driving simulation, it is necessary to accomplish the following.

1) Instrument the steering wheel of the driving simulator hardware with an actuator (the obtained overall system is the haptic interface).
2) Establish a model of the steering dynamics, including all known physical parameters (provided for each car).

Since the haptic interface links the simulated model and driver, we chose to model the overall system in a way similar to master/slave telerobotics and electrical network theory [30]. That is, the driver and the tire/road interaction are considered as dipoles that exchange energy, i.e., flow (speeds) and effort (forces/torques), through different linkage modules of the vehicle steering mechanism. Each module (or stage) is a subsystem.
of the overall steering mechanism that also exchanges energy with the neighboring subsystems. When such an exchange is expressed in linear differential equations, it can be represented as transfer function quadripole.

When this is the case for each module of the overall steering system, it can be represented by interconnected chain-type matrices (see Fig. 1). This model allows good modularity of the implementation, allowing changing or tuning a specific module without rewriting the equations for the entire steering system.

A chain matrix representation of each module is motivated by the very fact that the overall transfer function is obtained by a simple ordered product of the connected set of transfer matrices. The mechanical model of the steering system includes the dynamics induced by the following (see also [31]):

1) inertia and stiffness of the steering column;
2) pinion/rack link;
3) mass of the rack;
4) inertia and dynamic friction of the front wheels.

When the driver interacts with the steering wheel, the applied forces induce a torque noted \( \tau_h \) on the steering wheel. Another torque noted \( \tau_{col} \) goes up from the steering column. Both torques induce a rotation of the steering wheel of which the angular speed is noted \( \omega_h \) and the turning of the steering column of which the angular speed is noted \( \omega_{col} \). Let \( J_{sw} \) be the inertia of the steering wheel, \( \beta_{sw} \) be its viscous friction, \( J_{col} \) be the steering column inertia, and \( \beta_{col} \) be the steering column friction. The dynamic of the set (steering wheel and the steering column) can be represented by the differentials

\[
(J_{sw} + J_{col})\dot{\omega}_h + (\beta_{sw} + \beta_{col})\omega_h = \tau_h - \tau_{col}
\]

\[
\omega_h = \omega_{col}
\]

and its chain matrix form in the Laplace frequency domain is

\[
\begin{bmatrix}
\tau_h \\
\omega_h
\end{bmatrix} = \begin{bmatrix} 1 & 0 \\ (J_{sw} + J_{col})s + (\beta_{sw} + \beta_{col}) & 1 \end{bmatrix} \begin{bmatrix}
\tau_{col} \\
\omega_{col}
\end{bmatrix}
\]

where \( s \) is the Laplace complex variable.

The same reasoning applies to the remaining components (Fig. 1). The steering column is used to exchange efforts between the steering wheel and the rack. Let the torsional stiffness be \( k_{col} \), the steering column damping be \( f_{col} \), the torque that goes up from the rack be \( \tau_{rk} \), and the angular speed of the rack seen by the steering column be \( \omega_{rk} \). This steering module can be represented by the differentials

\[
k_{col}\int(\omega_{col} - \omega_{rk})dt + f_{col}(\omega_{col} - \omega_{rk}) = \tau_{col}
\]

\[
\tau_{col} = \tau_{rk}
\]

and in its chain matrix form

\[
\begin{bmatrix}
\tau_{col} \\
\omega_{col}
\end{bmatrix} = \begin{bmatrix} 1 & 0 \\ \frac{k_{col} + f_{col}s}{k_{col} + f_{col}s} & 1 \end{bmatrix} \begin{bmatrix}
\tau_{rk} \\
\omega_{rk}
\end{bmatrix}
\]

Similarly, let \( Q_{rk} \) be the quadripole representing the dynamics of the rack, \( Q_{ab} \) be the quadripole representing the dynamics of the direction bars (also called steering bars), and \( Q_{vfw} \) be the resultant quadripole representing the dynamics of the remaining parts of the vehicle’s front wheels. This quadripole \( Q_{vfw} \) is in fact composed from three quadripoles. The first quadripole links the kingpin system to the output, which is the tire stiffness-induced torque \( \tau_{stf} \) and angular speed \( \omega_{stf} \), coming from the lateral force of the tire/road interaction, which produces rotation of a part of the wheel. The second quadripole links the previous output parameters (input) to the shear surface. Finally, the third quadripole links the output shear surface to a part of the wheel that is constantly in contact with the road.

Every formerly considered quadripole describes a flow and effort exchange. The interconnection of these sequentially cascading quadripoles represents the entire steering system. The modularity of the representation allows 1) easy implementation, 2) easy interchange of any component of the steering system, and 3) the independent study of the effect of each subsystem. When only linear effects are considered, the equivalent quadripole represents the steering system transfer that can be composed as a product

\[
\begin{bmatrix}
\tau_h \\
\omega_h
\end{bmatrix} = \begin{bmatrix} Q_{sw} & Q_{col} & Q_{ab} & Q_{vfw} \end{bmatrix} \begin{bmatrix}
\tau_{ext} \\
\omega_{ext}
\end{bmatrix}
\]

\[
Q_{stf} = \begin{bmatrix} k_{col} & f_{col} \end{bmatrix}
\]

IV. HAPTIC FEEDBACK

A. Wheel Steering System

In our driving simulator, an actual car steering wheel and only a part of the steering column are installed. Using a dc actuator, the motorized steering wheel is linked with the dynamics of the steering system thanks to the simulated stiffness and damping of the steering column (see Fig. 2). These elements will behave as a bilateral coupling in a force-reflecting system [29]. To give to the simulator steering wheel similar dynamics to any given actual vehicle steering wheel while altogether computing force feedback during virtual driving, we have to compute the actuator’s desired torque.
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Fig. 2. Haptic feedback on the INRETS-FAROS low-cost driving simulator.

First, we write the equations that describe the dynamic behavior of the motorized steering wheel installed in the simulator carbone. Let $J_r$ and $\beta_r$ be, respectively, the inertia and dynamic friction of the steering wheel + actuator + transmission mechanism of the driving simulator, $\omega_{\text{desired}}$ be the speed angle of the steering wheel shaft, and $\tau_{\text{desired}}$ be the desired torque on the steering wheel shaft. The behavior of this system is represented as

$$J_r \omega + \beta_r \omega = \tau_h - \tau_{\text{desired}}$$

$$\omega = \omega_{\text{desired}}$$

The chain matrix form of (8) and (9) is

$$\begin{bmatrix} \tau_h \\ \omega_h \end{bmatrix} = \begin{bmatrix} 1 & J_r s + \beta_r \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \tau_{\text{desired}} \\ \omega_{\text{desired}} \end{bmatrix}.$$  

To compute the desired torque, we simply equalize (10) with (7), i.e.,

$$\begin{bmatrix} \tau_h \\ \omega_h \end{bmatrix} = Q_{\text{real}} \begin{bmatrix} \tau_{\text{desired}} \\ \omega_{\text{desired}} \end{bmatrix} = Q_{\text{SteeringSystem}} \begin{bmatrix} \tau_{\text{ext}} \\ \omega_{\text{ext}} \end{bmatrix}.$$  

From (11), we can write

$$\begin{bmatrix} \tau_{\text{desired}} \\ \omega_{\text{desired}} \end{bmatrix} = Q_{\text{real}}^{-1} Q_{\text{SteeringSystem}} \begin{bmatrix} \tau_{\text{ext}} \\ \omega_{\text{ext}} \end{bmatrix}.$$  

which allows us to compute the desired torque with

$$Q_{\text{real}}^{-1} = \begin{bmatrix} 1 & -J_r s - \beta_r \\ 0 & 1 \end{bmatrix}.$$  

To simulate the dynamic of the steering wheel and the force felt by the driver, we first propagate the known steering velocity $\omega_{\text{desired}} = \omega_h$. We then compute the torque to be fed back according to

$$\tau_{\text{desired}} = q_{11} \tau_{\text{ext}} + q_{12} \omega_{\text{ext}}$$

$$\omega_{\text{desired}} = q_{21} \tau_{\text{ext}} + q_{22} \omega_{\text{ext}}.$$  

Let $q_{11}$, $q_{12}$, $q_{21}$, and $q_{22}$ be the transfers of $Q_{\text{SimulatedSystem}} = \begin{bmatrix} q_{11} & q_{12} \\ q_{21} & q_{22} \end{bmatrix}$.

Now, we have two equations with three unknowns, i.e., $\tau_{\text{desired}}$, $\tau_{\text{ext}}$, and $\omega_{\text{ext}}$. However, we have an additional complex and highly nonlinear relation linking $\tau_{\text{ext}}$ and $\omega_{\text{ext}}$, which are mutually dependent. Indeed, the tire/road interaction dipole can be written as $\tau_{\text{ext}} = f(\omega_{\text{ext}})$ (see Fig. 4). Letting $k$ be a given computation step of the simulation, we can numerically solve system equations (11) and (12) using the following discret form:

$$\tau_{\text{desired}}^k = q_{11} f(\omega_{\text{ext}}^k) + q_{12} \omega_{\text{ext}}^k$$

$$\omega_{\text{desired}}^k = q_{21} f(\omega_{\text{ext}}^k) + q_{22} \omega_{\text{ext}}^k.$$  

To solve this system, we first compute $\omega_{\text{ext}}^k$ using (17). To avoid the algebraic loop simulation (i.e., resolve the indeterminacy of solving $\omega_{\text{ext}}$ for step $k$ using $f$), we use and estimate $\dot{f}$ of $f$, which numerically predicts $f$ from previous computed values $\omega_{\text{ext}}^{k-1}$, $i \geq 1$ (numerical predictor–corrector scheme that may need a few iterations to converge). Then, the obtained $\omega_{\text{ext}}^k$ allows computation of the desired torque $\tau_{\text{desired}}^k$ from (16) using the value of the external torque computed from $f(\omega_{\text{ext}}^k)$. Another solution, which is very computationally demanding, would be to compute the interaction forces by explicitly enforcing the constraints at the tire/road contact space through relative motion projection and back-propagating these forces to have $f(\omega_{\text{ext}}^k)$.

Once obtained, $\tau_{\text{desired}}^k$ is tracked by the inner actuator’s torque controller. The torque of the haptic dc actuator is proportional to its armature current. To track the desired torque, we convert the desired torque into the desired current as

$$\tau_{\text{desired}} = \frac{1}{r_{\text{link}}} k_{\text{i}} i_{\text{desired}}$$

where $r_{\text{link}}$ is the ratio of the mechanical linkage between the steering wheel’s shaft of the simulator and the actuator’s shaft (reduction coefficient), $k_{\text{i}}$ is the actuator’s constant (provided with the actuator), $i_{\text{desired}}$ is the armature current, and $\tau_{\text{desired}}$ is the desired shaft actuator torque. The electric equation of the actuator is

$$u = R i + L \frac{d}{dt} i + e$$

where $u$ is the armature applied voltage (in volts), $e$ is the voltage generated by the back electromotive force (in volts), $R$ is the actuator resistance (in ohms), $L$ is the armature inductance (in henry), and $i$ is the armature current (in amperes). The current proportional integral (PI) controller implemented to track the desired current is

$$u_{\text{com}} = k_p \left( \frac{\tau_{\text{desired}} - \tau_{\text{link}}}{k_{\text{i}}} - i \right) + k_i \int \left( \frac{\tau_{\text{desired}} - \tau_{\text{link}}}{k_{\text{i}}} - i \right) dt$$

where $u_{\text{com}}$ is the voltage applied to the actuator to track the desired torque, and $k_p$ and $k_i$ are the controller gains.
The practical implementation of the previously described quadripoles is shown in Fig. 3. The driver and the tire/road interaction are represented by dipoles. The direction of the various arrows highlights the flux/effort exchanges between the various quadripoles.

**B. Road Dipole**

Here, we compute the dipole function $f$ linking $\omega_{\text{ext}}$ to $\tau_{\text{ext}}$, which requires taking into account the coupling between the vehicle dynamics and the tire/road interaction models. The force applied by the driver on the steering wheel produces a rotation on the sheared surface. Consequently, various normal and lateral forces caused by the vehicle dynamic induce a torque on the kingpin axis of the wheel. To provide the driver with high-fidelity haptic sensations, we modeled the most relevant phenomenon. The resultant torque is computed according to Fig. 4. We consider haptic feedback for moving drive, small maneuvers, and at stops.

During vehicle motion, normal and shear forces induce a torque on the kingpin axis of the wheel. This (front wheel) axis is inclined with an angle $\phi_p$ (in radians) according to the vehicle longitudinal plane and with a caster angle $\phi_c$ (in radians) according to the vehicle transversal plane. The distance, which is noted $d$ (in meters) and measured on the ground, between the middle contact of the tire and the pivoting axis of the wheel is called ground offset. The distance that is noted $c$ (in meters) between the intersection of the kingpin axis with the ground and the contact point of the wheel with the ground is called the mechanical trail. When the vehicle is moving, a pneumatic trail that is noted $a$ appears because of the tire shearing and is added to the mechanical trail. The wheels generally are not totally vertical but slightly tilted toward the outside with a wheel camber angle $\gamma$ (in radians) [32]–[34] (Fig. 5).

Several effects create a torque on the kingpin axis. However, in our case, the only torques to be taken into account are those that create the prevailing efforts, i.e., which are felt by the driver and those which are most relevant to guide the vehicle. These torques are due to three distinct types of forces: 1) the sliding force, 2) the wheel camber thrust, and 3) the vertical forces [35]. These forces cause a resulting torque $\tau_{\text{ext}}$ on the kingpin axis and are described hereafter.

1) **Sliding Force:** Without constraint, a wheel turns according to its longitudinal axis. As soon as a lateral force is applied to it, the direction of its movement diverts from its longitudinal axis, causing the deformation of the contact surface of the tire. This deformation creates a sliding angle $\alpha$ between the longitudinal axis of the wheel and the axis of its motion’s direction. The deformation results in a sliding lateral reaction force $F_\alpha$ on the tire. $F_\alpha$ is a function of $\alpha$ and of the static front load on the tire, which is denoted $N_f$ (in newtons). There are various models of $F_\alpha$. The best known model is proposed by Pacejka and Besselink [36]. It is an empirical model based on experimental measurements that allow these parameters to be set. In our case, we have chosen a simple empirical model that allows easier parameterization. The obtained curve is less precise but enough for a simulation intended to achieve haptic feedback on the steering wheel, i.e.,

$$F_\alpha = \mu(\alpha)N_f.$$ (21)
\( \mu(\alpha) \) is the adherence coefficient approximated by [37]

\[
\mu(\alpha) = 2C_p\alpha_p \frac{\alpha}{\alpha^2 + \alpha^2}.
\] (22)

\( C_p \) is a known coefficient that characterizes the adherence linked with the road type (for instance, for dry asphalt \( C_p = 0.8 \), for wet asphalt \( C_p = 0.2 \), etc.) [38]. When adherence enters a saturation area, \( \alpha_p \) (in radians) represents the sliding angle that corresponds to the sliding force maximum value. \( N_f \) (in newtons) represents the maximum reaction force provided by the tire before it reaches the adherence limits. It is defined as the normal force that is exerted on the nose gear of the vehicle depending on the vehicle's mass, center of gravity, and pitch and roll motions.

The sliding force results in a self-alignment torque that tends to bring the front wheels back to the longitudinal axis of the vehicle—when the vehicle speed is not nil. It can be expressed as

\[
\tau_{t_1} = F_c(c + a) \cos(\phi_c)
\] (23)

where \((c + a) \cos(\phi_c)\) represents the lever arm of the sliding force, and \(\phi_c\) (the caster angle), \(c\) (the geometric caster), and \(a\) (the tire caster) are known for a given vehicle (see Fig. 5), whereas the variables \(\alpha\) and \(N_f\) are computed using the simulated vehicle dynamic model.

2) Wheel Camber Thrust Induced by the Vehicle’s Roll: The wheel camber force (or thrust) is lateral to the tire force, whose approximation makes it proportional to the wheel camber angle. Letting \(k_c\) be the thrust coefficient (known for a given vehicle), we have

\[
F_c = k_c \gamma.
\] (24)

The wheel camber angle \(\gamma\) is generally small (\(\approx 1^\circ\)). Thus, the wheel camber thrust is relatively negligible to the sliding force. However, this force can be amplified by the suspension (e.g., on curves); in this case, the wheel camber angle induced by the suspension movement is approximately proportional to the vehicle’s roll angle \(\phi\), i.e.,

\[
\gamma = \lambda \phi
\] (25)

where \(\lambda\) is a known proportionality coefficient for a given vehicle. The roll-induced wheel camber thrust is

\[
F_c = k_c \gamma = k_c \lambda \phi.
\] (26)

This force induces a torque \(\tau_{t_2}\) on the front wheel kingpin, acting with a lever arm \((c + a) \cos(\phi_c)\) as

\[
\tau_{t_2} = F_c(c + a) \cos(\phi_c).
\] (27)

The roll angle \(\phi\) is given by the simulated vehicle's state.

3) Normal Forces: On road curves, the vehicle body rocks to the external side, producing a variation of the weight distribution called load transfer. The latter changes the effective static load \(N_f\) exerted on each front wheel. The load transfer is quantified by \(\Delta N_f\) that is subtracted from the interior front wheel’s \(N_f\) and added to the external front wheel’s \(N_f\). This load transfer induces a torque on the kingpin axis, which is approximated by

\[
\tau_x = 2\Delta N_f b_a \sin(\phi_c)
\] (28)

where \(\phi_c\) (in radians) is the caster angle, and \(b_a\) (in meters) is perpendicular to the axis offset on the level of the hub. The load transfer \(\Delta N_f\) is given by the simulated vehicle’s dynamic model.

The total expression of the torque produced by the sheared surface/road contact on the wheel’s kingpin axis is

\[
\tau_{ext} = \tau_x + \tau_{t_1} + \tau_{t_2}.
\] (29)

4) External Torque Applied on the Sheared Surface When the Vehicle Speed Is Null: When the vehicle is stopped, the equations of the lateral mode dynamics are no longer valid. The front tires of the vehicle are sheared only by the force provided by the driver and by the tire–carriageway contact friction. In this case, when the driver tries to turn the front wheels, a ground reaction force is applied on the sheared surface (Fig. 6), preventing the wheels from turning. The wheels start turning only when the lateral force on the sheared surface reaches their saturation. This force becomes insufficient to constrain the force of the driver.

This can be simulated thanks to the Coulomb friction model

\[
\tau_{ext} = F_s \text{sign}(\omega_{sh}) \cos(\phi_c) c
\] (30)

where \(c\) (in meters) is the geometric caster wheel (Fig. 5), \(F_s\) (in newtons) is the lateral ground reaction force that is represented by \(F_s = \mu N\), \(\mu\) is the dynamic friction coefficient that characterizes the road surface (bituminizes, asphalts, etc.), and \(N\) (in newtons) is the vertical load on the front tires of the vehicle.

When the sheared surface speed is null, friction cannot be described as a function of velocity. Instead, it has to be modeled using the torque applied on the sheared surface in the following manner [39]:

\[
\tau_{ext} = \tau_{stf}, \quad \text{if } |\tau_{stf}| < F_s \cos(\phi_c)c
\]

\[
\tau_{ext} = F_s \text{sign}(\tau_{stf}) \cos(\phi_c) c, \quad \text{if } |\tau_{stf}| \geq F_s \cos(\phi_c)c.
\] (31)
C. Simulation Example

The vehicle model comprises the dynamics and kinematics computation algorithms that take as an input 1) the driver operations given by the acquisition module and 2) the road characteristics. In our case study, the vehicle is considered as one body with five degrees of freedom (longitudinal, lateral, roll, pitch, and yaw). Its complexity relates more to the motorization part than the chassis dynamic. The engine part is modeled by mechanical and behavioral approaches [37] based on the vehicle general characteristics (engine torque curves, clutch pedal position, accelerating proportioning, etc.). After updating the vehicle’s state, the resulting data on the engine are sent to the road dipole and to the traffic model server.

We carried several preliminary simulations. In this case, we set the maximum lateral force before losing wheel ground adherence to 1140 N. The sliding angle that corresponds to this force is 0.22 rad. To highlight the behavior of the tire toward the sheared surface when the wheel abruptly undergoes a canceled force, we set the tire stiffness lower than the real value. In this simulation, we apply to the wheel a force profile having a saw-shaped tooth. Fig. 7 shows three different parts.

1) In the first part, we apply a force to the wheel of the vehicle. The tire is deformed without inducing motion of the sheared surface. This is because the force coming from the operator is lower than the static Coulomb friction force.

2) In the second part, the effort applied to the wheel is higher, and the sheared surface is pulled by the movement of the wheel.

3) In the third part, the force applied to the wheel is abruptly canceled; therefore, the wheel is brought back to the sheared surface position thanks to the force produced by tire deformation while the sheared surface remains in the same position.

D. Results of the Steering Wheel Experiments

1) When the Vehicle Speed Is Not Nil: The steering wheel torque increases when the operator draws aside the steering wheel from neutral position (zero) (Fig. 8). When the operator decreases the force exerted on the steering wheel, it returns to its initial position (zero). This is primarily caused by the self-aligning torque that tends to bring back the vehicle’s front wheels to the vehicle speed direction.
Fig. 9. Steering wheel’s torque according to the steering wheel’s angle at the stationary vehicle (i.e., vehicle speed = 0). When the operator decreases the exerted force on the steering wheel, this returns to its initial position if the front tires’ force reaction does not reach the adherence limit area (case 1) and returns to the higher position in the other case (case 2).

2) When the Vehicle Speed Is Null: The neutral position is different from zero (Fig. 9). Indeed, this depends on the front wheel balance position. The torque increases when the operator draws aside the steering wheel from its initial position.

V. HAPTIC AUGMENTATION ALGORITHM

The idea behind haptic augmentation [9] is to adaptively adjust the haptic feedback with a supplementary amount of artificial tunable force that does not confuse the driver and does not alter her natural behavior. The haptic augmentation algorithm will compensate for latencies, driving instabilities, lack of immersion presence, and light imperfections. It can also be used in guidance scenarios, as previously proposed in [10], or for driving-skill teaching purposes. We compute the angle $\Delta \psi$ (representing the relative heading error) between the current vehicle heading and the virtual point determined as the point the vehicle should take to reach a given point $p$ on the middle of its actual lane at a fore distance $d$ (in meters) (Fig. 10). This angle ideally tends to zero by the so-called predictor function through a light additional force feedback.

Introducing the predictor function in the equations of the haptic interface dynamics, we have

\[
\left[ \tau_h + \tau_{\text{com}} \right] = Q_{\text{sw}} \times \left[ \tau_{\text{eval}} \right]
\]

\[
\tau_{\text{com}} = k_p \Delta \psi + k_d \Delta \dot{\psi}
\]

where $\tau_{\text{com}}$ is the predictor function, $k_p$ is the proportional gain of the proportional derivator (PD) corrector, and $k_d$ is the derived gain of the proportional derivator (PD) corrector. Note the elegance of the quadripole formulation that allows simple integration of our proposed haptic augmentation.

Vehicle guidance is shared between the force of the driver and the force brought by the predictor function. The assistance (haptic augmentation) provided to the driver can be modulated through the gains of the corrector. Consequently, the haptic augmentation effect may be reduced or increased thanks to these gains’ adjustments. The increasing haptic augmentation tends toward a fully automatic driving. This system counters various forces coming up from the steering column, particularly the autoalignment force, which evidently tends to reduce the necessary driver forces, allowing keeping the vehicle on its “ideal” trajectory. Among the other benefits, the support system minimizes the effects of latencies and nondesirable simulation oscillatory dynamics.

To illustrate the effects of the haptic feedback steering wheel and the driver haptic augmentation algorithm on the simulator, we recorded the relative heading of the vehicle according to various driving situations (without haptic feedback, with haptic feedback, and with haptic feedback + haptic augmentation).

Without steering wheel haptic feedback and without haptic augmentation, the subjects (at curve exits) produce oscillations on the trajectory of the vehicle (Fig. 11). It should be noted that such a situation most of the time leads to a loss of the virtual vehicle control. In addition, the amplitudes of the oscillations seem to be decreased in the presence of the steering wheel haptic feedback and to be more attenuated when haptic augmentation is activated (Fig. 11). Generally, these last two situations considerably reduce the loss of vehicle control. Nevertheless, as stated in [40], which investigated the effects of extra visual signals on the driver’s distance using a driving simulator, the introduction of aids to drivers should be accompanied by an evaluation of the tradeoff between direct assistance to the
driver and distraction of the driver from primary driving tasks. Therefore, experiments have been made to evaluate the effects of the modalities presented in real driving scenarios.

VI. PERFORMANCE EXPERIMENTS

Experiments were carried out with the INRETS driving simulator (Fig. 12). It is a car cabin with a fixed base, the virtual scene of which is displayed through three projectors on three screens covering 156° of the driver’s visual field. This system is embedded within a French Architecture parallèle multi-acteurs pour la simulation microscopique du trafic (ARCHISM) traffic simulator. This allows us to put the driver into various (and sometimes complex) traffic situations. In that way, it is possible to observe how the driver achieves the various maneuvers to assess his driving performance.

A dedicated system controls the cabin and communicates with the PC, which operates the traffic simulation and the dynamic model of the vehicle. It is a microcontroller, and it is also in charge of computer haptic and its feedback, i.e., the control of the haptic dc actuator. Separating the computation of the haptic feedback from the rest of the simulation makes high-frequency rendering possible (up to 1 kHz is possible for haptics).

From now on, we will discuss the contribution of the different haptic feedback strategies.

Experiments were carried out on a secondary road with two lanes of opposite directions, each of them being 3.5 m in width. It is a very winding circuit, with ten bends of variable curvature. This variety of bend curvature aims to confront the driver to various levels of difficulty. That way, there is one bend with a 400-m radius, five bends with a 150-m radius, and four bends with a 75-m radius. To enforce the driver to stay on lane, traffic is generated on the lane of the opposite direction.

Before starting the experiment, each subject is made familiar with the driving simulator through a training phase on a highway. A highway has been chosen because we want the driver to be familiar with the driving simulator without having prior details on the task (i.e., dealing with bends on a more constraint road). Further to this adaptation phase, the following order is given to the subject. During the experiment, the task consists of driving on the road, staying as much as possible in the center of the lane and knowing that the driving speed remains constant and that the subject can neither brake nor accelerate (the brake and acceleration pedals are made nonoperational). The subject is then informed that she/he will have to drive six times, being submitted to two constant speeds (65 and 85 km/h) and the following three modalities:

1) without haptic feedback on the steering wheel (noted as without SWHF);
2) with haptic feedback on the steering wheel, rendering the only tire interaction forces under the effect of vehicle dynamics (noted as SWHF);
3) with haptic feedback blending previous effects and haptic augmentation (noted as SWHF + assistance).

We carried out a peer experiment to ascertain the driver’s reaction toward the haptic augmentation system. In that way, we could observe that, on a bend, the subjects were unpleasantly surprised by the behavior of the steering wheel. Indeed, they did not like not having full control of the vehicle. In fact, this problem comes from the very fact that on bends, the haptic augmentation algorithm may turn off forces at the point where the driver requires sensing them, depriving her/him from an essential piece of information about the road curvature. Indeed, when driving within bends, control is made around a position that requires finely adjusting the vehicle heading based on the felt autoalignment forces.

Further to the preliminary experiments, we lowered the haptic augmentation in bends so that it would weakly be perceived by the subjects. In contrast, we keep it more active in straight lanes. Indeed, the drivers appreciate the effect of the system since it reduces undesirable simulation effects and keep them from simulation-induced instability. This can simply be explained that, in a straight line, the autoalignment forces are nil. The haptic augmentation algorithm provides information that allows them to instantly detect small heading variations (which, in real driving situations, induces lateral acceleration). Moreover, the same system damped undesirable small oscillations of the steering wheel, which are filtered.

To do so, we slightly adapted the algorithm of the haptic augmentation. The gains of the controller were lowered in bends, making it possible for the driver to better feel the interaction efforts between the wheels and the road.

Twelve subjects, with full driving licenses, took part in the experiment. The course order was at random to remove the consequences of habit effect on the recording of performance indicators.

To measure the subjects’ driving performance, two indicators were recorded in real time, i.e., the relative heading angle and the lateral deviation (distance between the vehicle and the middle of the lane).

The difference between the variances of the steering wheel position for both modalities (without haptic feedback and with haptic feedback) is not significant in bends, but it is clearly stressed on the bend exit [11]. Indeed, considering the driver as corrector, who has a haptic and visual feedback, the presence of latency on one of the feedbacks could make driving unstable when the steering wheel movement is speed, as it is the case when exiting a bend. As the difference between the modalities in trajectory vehicle control is more significant at the first 10 m

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Fig. 12. INRETS driving simulator.
after the bend, we focus on the analysis of the recorded data on this part of the road after the bend [11].

As previously mentioned, two indicators of performance (relative heading, lateral deviation) are recorded in real time during the driving simulator. For each one, we analyzed the following.

1) The **simple vehicle speed effect**. The simple vehicle speed effect is analyzed to mainly show the vehicle control difficulties according to vehicle speed variations.
2) The **simple steering wheel modalities effect**. The simple steering wheel modalities effect is analyzed to show the impact of steering wheel modalities on driving simulator stability.
3) The **interaction of steering wheel modalities effect with vehicle speed effect**. The analysis of the interaction between the steering wheel modalities effect and the vehicle speed effect mainly shows the effect of the steering wheel modalities on driving simulator control according to the vehicle variation speed.

### A. Experiment Results of the Average of the Relative Heading Variances

1) **Simple Effect of Speed**: A simple effect of speed can be observed. The average of the variances is significantly higher at 85 km/h than at 65 km/h (Fig. 13). This result could be explained by the increase of the vehicle control difficulty according to the vehicle speed increase.
2) **Simple Effect of the Steering Wheel Modality**: The Newman–Keuls mean comparison test shows that there is a simple effect of the steering wheel modality. The average of the variances of the relative heading is significantly higher without the SWHF condition than for the SWHF or SWHF + assistance condition ($p < 0.006$ and $p < 0.001$, respectively) (Fig. 14). This result shows that both conditions (haptic feedback and haptic feedback with augmentation algorithm) decrease the difficulty of the virtual vehicle control.
3) **Interaction Between Speed × Steering Wheel Modality**: The planned comparisons demonstrate a significant effect of speed on the steering wheel modality. The difference between the average of the variances of the relative heading is higher in the SWHF condition than in the SWHF + assistance (planned comparison $F(1.11) = 5.08; p < 0.05$). In other words, the difficulty to drive, which increases according to the speed increase (85 km/h) observed with the SWHF condition, is less with a haptic augmentation (Fig. 15).

### B. Experiment Results According to the Average of the Variances of Lateral Deviation

1) **Simple Effect of the Speed**: A simple effect of speed can be observed. The average of the lateral deviation variances is significantly higher at 85 km/h than at 65 km/h (Fig. 16). Similar to the relative heading indicator, the result shows that the vehicle speed increase causes an increase in the control of the difficulty of the virtual vehicle.
performance indicators (lateral deviation) for the SWHF condition as compared with the without SWHF condition (Fig. 18). Speed has a significant effect on driving performance. Indeed, the level of driving difficulty increases as the vehicle speed increases (Figs. 13 and 16). However, there is a significant decrease in that difficulty with the presence of the haptic augmentation system. The result can be observed on the average of the variances of the relative heading. It is significantly lower for the SWHF + assistance condition than for the SWHF condition (Fig. 15).

VIII. CONCLUSION

The integration of a device with haptic feedback linked to the simulator steering wheel has been proposed. Haptic rendering modeling uses a modular quadripolar representation of each component to be connected through chain matrices. In addition, an algorithm of haptic augmentation coupled with the command of the haptic interface is proposed to assess drivers having a better control of the virtual vehicle and to compensate for specific simulation limitations.

To show the impact of haptic feedback and the contribution of the haptic augmentation on a driving simulator, two essential conclusions are drawn.

1) Integrating a haptic feedback on the driving simulator improves driving on the simulator.

2) The effect of the haptic augmentation method is significant under difficult driving conditions; in other words, haptic augmentation is effective when simulator driving is difficult.

Concerning the scalability and porting issues, although not mentioned, the same algorithms have been ported without any major modification to a moving platform simulator [8]. The driving performance on a simulator is generally different, depending on the drivers, and the adaptation time of the subject to the simulator is variable. That is why our future works aim to develop a haptic augmentation system that will be able to adapt to the driver’s difficulties by monitoring a set of driving performance indicators.

APPENDIX

NUMERICAL VALUE OF STEERING SYSTEM CHARACTERISTICS

The following data are those of a real commercial vehicle and have been provided by a well-known French car factory:

- steering (wheel + column) inertia: $J_{sw} + J_{col} = 0.023$ kg · m²;
- static friction of the steering column: 0.18 N · m;
- dynamic friction of steering (wheel + column): $\beta_{sw} + \beta_{col} = 0.004$ N · m/(rad · s⁻¹);
- rack mass: $M_{rk} = 2.5$ kg;
- steering column stiffness: $k_{col} = 135$ N · m · rad⁻¹;
- steering column damping: $f_{col} = 0.4375$ N · m/(rad · s⁻¹);
- pinion radius: $R = 0.00726$ m;
- rack static friction: 130–155 N;
- lever arm: $r = 0.1273$ m;
- wheel inertia: $J_w = 0.45$ kg · m²;
• dynamic friction of the (steering wheel + transmission mechanic) of the simulator; \(\beta = 0.19 \, \text{N} \cdot \text{m} \)
• inertia of the (steering wheel + mechanic transmission) of the simulator \(J_r = 0.0636 \, \text{kg} \cdot \text{m}^2 \)
• steering wheel angle \(\approx 17 \times \text{front-wheels angle}\).

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Author: Hakim Mohelbi
Title: BS in electrical and engineering science from the University of Tizi-Ouzou, Algeria, and the DEA (Master by Research) and Ph.D. degrees from the University of Evry, Evry, France.
Affiliation: Research Assistant with Technocentre Renault, Saint-Quentin-en-Yvelines, France.
Specialty: Current interests include driving simulators, haptics, motion feedback control platforms, and hardware in the loop.
Abderrahmane Kheddar (M’04) received the B.S. degree in computer science from Institut National d’Informatique (INI), Algiers, Algeria, and the DEA (Master by Research) and Ph.D. degrees in robotics from the University of Paris 6, Paris, France. From September 2003 to August 2008, he was a Professor with the University of Évry, Évry, France. Since 2003, he has been the Codirector of the AIST/CNRS Joint Robotics Laboratory, Tsukuba, Japan. Since 2008, he has also been the Director of research with Centre National de la Recherche Scientifique (CNRS), Montpellier, France. His research interests include haptics and humanoids. Dr. Kheddar is a Founding Member and a Senior Adviser of the IEEE/RAS Chapter on haptics and is with the editorial board and is a Founding Member of the IEEE TRANSACTIONS ON HAPTICS.

Stéphane Espié received the Habilitation à Diriger les Recherches (HdR) degree from the University of Paris 6, Paris, France, in December 2004. He is the Head of the Driving Simulation Department Modélisation, Simulation et Simulateur de conduite (MSIS), Institut National de Recherche sur les Transports et leur Sécurité (INRETS), Arcueil, France, where he first worked on new sensors for traffic and then on traffic simulation and driving simulators. His main topics deal with traffic system simulation through a multiactor approach, “traffic-centered” driven simulators, simulation and evaluation of intelligent transportation system concepts, and driver’s evaluation. His current research interests include the design of hardware and software architecture that deals with both traffic simulation and driving simulator and distributed artificial intelligence, artificial life, and cooperative systems.