Designing Steering Feel for Steer-by-Wire Vehicles Using Objective Measures

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***Abstract*—Drivers use torque feedback at the handwheel of a vehicle, or steering feel, to obtain information about the road and tire dynamics. This aids them in driving tasks like curve negotiation. Steer-by-wire vehicles, due to the mechanical decoupling of the front tires and handwheel, do not have any inherent steering feedback and require an artificial steering feel. One way to implement an artificial steering feel is to synthesize steering feedback with a model running on board the vehicle. Identifying the appropriate level of model fidelity required involves understanding what elements of steering feel are important. This paper proposes using objective steering feel measures used in industry as a means of understanding which elements of steering feel are most important. It also introduces a steering feel model of appropriate fidelity to capture these important elements of steering feel while remaining intuitive to tune objectively with a small set of parameters. Experimental data show that the desired performance measures obtained via dynamic vehicle simulation can be replicated on an actual steer-by-wire vehicle, validating the design technique.**

***Index Terms*—Force feedback (FFB), haptic, objective measures, steer-by-wire, steering feel.**

I. Introduction

DRIVERS make extensive use of the torque feedback at the handwheel of a conventional vehicle (known as steering feel) to obtain useful information about the road and tire dynamics. Using a driving simulator, Liu and Chang demonstrated that having steering torque feedback results in better curve negotiation and skid recovery by a driver [1]. Forsyth and MacLean used a driving simulator and observed that without steering feedback, drivers became disoriented in tight turns [2]. Similar results have been found for driver assistance systems, for instance, Switkes *et al.* showed that steering feedback must be carefully designed to ensure the stability of a lane-keeping controller [3]. Clearly, designing a good steering feel is important for driver comfort and safety.

Steer-by-wire systems replace the mechanical steering linkages between the handwheel and front tires with electronic sensors and actuators. This severing of the mechanical connection disrupts the steering feel, removing a useful source of driving feedback. As steer-by-wire technology moves into the first production vehicles in 2013, artificially creating this feel becomes increasingly important [4].

One way of creating steering feel on steer-by-wire vehicles is to feedback the steering motor torque directly to the driver. Shengbing *et al.* demonstrated using hardware-in-the-loop simulation that steering motor current can generate steering feedback [5]. Similarly, Asai *et al.* used a test bench setup to show that steering motor current coupled with a torque map can generate steering feel [6]. Though this technique generates a realistic steering feel, this feel is tied to the physical properties of the vehicle and cannot be varied easily in software.

Another way of creating steering feel is to use a steering model with some abstraction of the physical system. Low fidelity models, typically involving a spring model based on steering angle, have been used on simulators by Segawa *et al.* [7] and Oh *et al.* [8]. Williams implemented a speed-varying spring model for steering feel on heavy trucks [9]. Low fidelity models, though simple to implement, cannot capture all the important elements of steering feel like power assist and "heaviness" of steering. In order to capture more elements of steering feel than low fidelity models, higher fidelity models have been used in simulators by Salaani *et al.* [10] and Mandhata *et al.* [11]. Though higher fidelity models can capture more elements of steering feel, the increase in the number of model parameters makes tuning challenging. As a result, validation procedures such as the repeated subjective human evaluations used by Mandhata *et al.* [12] are required.

Currently, conventional vehicles have wide variation of steering feel. Steer-by-wire vehicles should not be different. Therefore, higher fidelity steering feel models, which can create a wide variation of steering feel, are necessary. However, as model fidelity increases, the tuning of these models to obtain desired feel becomes more challenging. Therefore, finding the appropriate level of steering model fidelity is critical. The model must be complex enough to capture all the elements of steering feel that modern drivers care about while remaining simple enough to be tuned intuitively. This paper analyzes objective steering feel performance measures used by industry as a means of understanding what elements of steering feel modern drivers care about. This paper then proposes using a steering feel model with the appropriate fidelity to capture these important elements while remaining intuitive to tune objectively with a small set of parameters.

This paper begins with a simple steering feel model which captures the effect of tire torques, power assist and the modification of inherent inertia, and damping of a steer-by-wire steering system. Objective steering feel performance measures used by industry are then introduced as a means of understanding and

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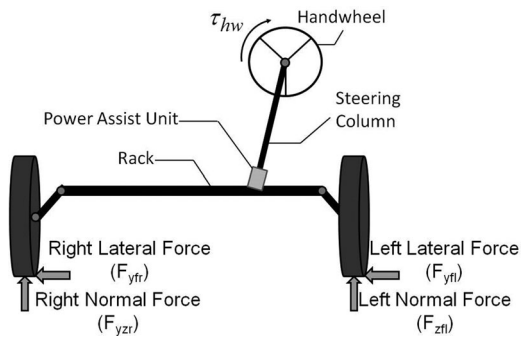

Fig. 1. Conventional steering system.

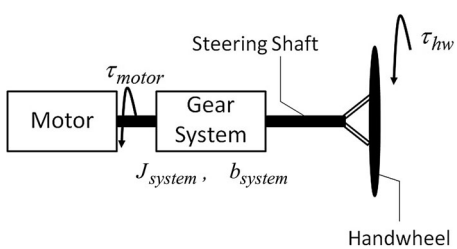

Fig. 2. FFB steering system.

characterizing steering feel. These performance measures do not depend only on the steering model but also on the vehicle dynamics. Using a simple dynamic vehicle simulation, a critical set of steering model parameters that tune these measures is identified and a desired feel is designed. Experimental data then show that the desired performance measures obtained via dynamic vehicle simulation can be replicated on an actual steer-by-wire vehicle.

II. Artificial Steering Feel

In conventional steering systems, the handwheel is mechanically connected to the roadwheels as illustrated in Fig. 1. In steer-by-wire vehicles, the handwheel is mechanically decoupled from the roadwheels. The driver's steering commands are transferred electronically to motors that in turn actuate the roadwheels. This mechanical decoupling removes much of the steering feel that would normally be present in a conventional vehicle.

Steering feel can be artificially created in a steer-by-wire vehicle using a force feedback (FFB) steering system, illustrated in Fig. 2. For this system, the handwheel torque felt by a driver (τ_{hw}) has the following dynamics:

$$\tau_{\mathrm{hw}} = \tau_{\mathrm{motor}} + J_{\mathrm{system}}\ddot{\delta}_{\mathrm{hw}} + b_{\mathrm{system}}\dot{\delta}_{\mathrm{hw}} \tag{1}$$

where the handwheel FFB motor torque (τ_{motor}) is commanded, the inherent system inertia (J_{system}) and system damping (b_{system}) are known from an identification of the mechanical system, and the handwheel angle (δ_{hw}) is a driver input.

A. Conventional Steering Feel

Developing an artificial steering feel for steer-by-wire vehicle involves having the FFB steering system feel like a conventional steering system. In conventional steering vehicles, there are three major torques that make up the steering feel. They are as follows:

1) tire moments;
2) torques due to inertia and damping;
3) torque due to power assist.

The weighted sum of these torques result in the steering feel felt by the driver.

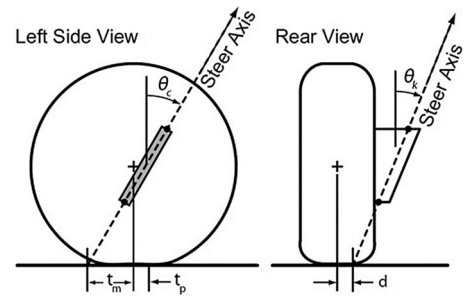

Fig. 3. Typical suspension geometry on conventional vehicle.

1) Tire Moments: Fig. 1 illustrates a conventional mechanical steering system with the right and left front lateral tire forces (F_{yfr}, F_{yfl}) and the right and left front normal tire forces (F_{zfr}, F_{zfl}) displayed. Fig. 3 illustrates a typical suspension geometry of a conventional vehicle with the steer axis, mechanical trail (t_m), pneumatic trail (t_p), caster angle (θ_c), kingpin angle (θ_k), and scrub radius (d) indicated. The moments generated by the lateral and normal tire forces about the steer axis form a major component of steer feel.

The moment about the steer axis generated by the lateral tire forces is known as the aligning moment and is given by (see [13])

$$\tau_{\mathrm{alignMoment}} = -\left(F_{yfl} + F_{yfr}\right)\left(t_m + t_p\right)\cos\sqrt{\theta_c^2 + \theta_k^2}. \tag{2}$$

The aligning moment results in a strong self-centering torque at higher speeds.

The moment about the steer axis generated by the normal tire forces is known as the jacking torque and is given by (see [13])

$$\tau_{\mathrm{jackingTorque}} = -\left(F_{zfl} + F_{zfr}\right) d\sin\theta_k \sin\delta_{rw} + \left(F_{zfl} - F_{zfr}\right) d\sin\theta_c \cos\delta_{rw} \tag{3}$$

where the δ_{rw} is the roadwheel angle. The jacking torque accounts for most of the self-centering effect of the steering feel at low speed when the effect of the aligning moment is small.

Transmission of the aligning moment and jacking torque via the steering linkages to the handwheel results in these two tire moments having a significant impact on the resultant steering feel.

2) Torques Due to Inertia and Damping: Fig. 1 illustrates that between the handwheel and the roadwheels, there exists a set of mechanical connections including the steering column, power assist unit, rack, and suspension, which results in the steering system having some inherent damping and inertia. This

inherent damping and inertia give rise to additional torques that are felt by the driver and form a part of the steering feel.

3) Torque Due to Power Assist: In modern power steering systems, drivers receive an assistive torque counteracting the total front tire moment to aid them in turning the roadwheels. The amount of this assistive torque depends on the difference in torque due to the total front tire moment and the driver's input torque as measured at the steering rack using a flexible torque-measuring element like a torsion bar. A greater difference in this torque results in a larger assistive torque. This assistive torque also features prominently in the steering feel.

B. Creating Artificial Steering Feel

Building a steering feel model with the appropriate fidelity allows the creation of artificial steering feel on steer-by-wire vehicles. The model presented in this section simplifies the steering feel model developed by Mandhata *et al.* [11]. Mandhata's model incorporated tire moments, torques due to the steering system properties, static friction, and a power assist model, which assumed the use of a torsion bar. The model presented in this section incorporates tire moments and torques due to the steering system properties while simplifying the power assist model to be more general. This more general model does not require a torsion bar windup as an input as steer-by-wire vehicles, like the one shown in this paper, need not have a torsion bar.

Based on this model, the motor torque (τ_{motor}) is given by

$$\tau_{\text{motor}} = \tau_{\text{damp}} + \tau_{\text{inertia}} + K\tau_{\text{assisted tire moment}} \tag{4}$$

where the damping torque (τ_{damp}) and inertia torque (τ_{inertia}) modify the inherent damping and inertia of the steering system, and the assisted tire moment ($\tau_{\text{assisted tire moment}}$) represents the combined effect of the aligning moment, jacking torque, and power assist. These are discussed in detail below. The tire moment gain (K) accounts for the transmission of moments from the tire to the handwheel and varies for different steering systems, suspension geometries, and steering ratios. A larger K results in a great portion of the assisted tire moment being incorporated into the steering feel.

1) Damping Torque (τ_{damp}): Damping torque modifies the effect of the inherent system damping (b_{system}) on steering feel. Conventional vehicles have more inherent damping than steer-by-wire vehicles as they have a more mechanically complex steering system. Therefore, adding damping torque in a steer-by-wire system is usually necessary. In a conventional vehicle, modification of inherent damping is challenging as it involves actual hardware modification. Having this modification be a design parameter that can be tuned in software makes changing the inherent damping much simpler. The change in inherent damping is created using the following model:

$$\tau_{\text{damp}} = -\Delta b\dot{\delta}_{rw} \tag{5}$$

where the change in damping (Δb) to the system is a design parameter.

2) Inertia Torque (τ_{inertia}): Inertia torque modifies the effect of the inherent system inertia (J_{system}) on steering feel. For the same reason as inherent damping, conventional vehicles have more inherent inertia than steer-by-wire vehicles. Having this modification be a design parameter that can be tuned in software makes changing the inherent inertia match desired values straightforward. The change in inherent inertia is created using the following model:

$$\tau_{\text{inertia}} = -\Delta J\ddot{\delta}_{rw} \tag{6}$$

where the change in inertia (ΔJ) to the system is a design parameter.

3) Aligning Moment (τ_{align}): Front tire forces and their resulting moments have a large effect on steering feel. The effect that the front *lateral* tire forces (F_{yf}) have on steering feel is through the aligning moment. By assuming that the caster angle (θ_c) and kingpin angle (θ_k) are small in (2), the aligning moment can be approximated as the product of the front lateral tire forces and the total trail of the vehicle as shown by

$$\tau_{\text{align}} = -F_{yf}(t_m + t_p) \tag{7}$$

where t_m is the mechanical trail and t_p is the pneumatic trail of the vehicle. Taking the small angle assumption reduces the number of parameters needed for the model.

The brush tire model proposed by Fiala [14], and presented in the following form by Pacejka [15], gives a useful approximation of the nonlinear relationship between tire force and slip angle:

$$F_y = -C_\alpha \tan\alpha + \frac{C_\alpha^2}{3\mu F_z}\tan\alpha\,|\tan\alpha| - \frac{C_\alpha^3}{27\,(\mu F_z)^2}\tan^3\alpha \tag{8}$$

where the surface coefficient of friction (μ) and tire cornering stiffness (C_α) are design parameters. The normal load (F_z) can be obtained from the mass of the vehicle.

For simplicity, the mechanical trail (t_m) is assumed to be constant and the pneumatic trail (t_p) is modeled as a decreasing linear function of slip angle as introduced by Hsu *et al.* [16]

$$t_p = t_{p0} - \text{sgn}(\alpha_f)\frac{t_{p0}C_{\alpha f}}{3\,\mu F_{zf}}\tan(\alpha_f) \tag{9}$$

where the pneumatic trail at zero front slip angle (t_{p0}), surface coefficient of friction (μ), front tire cornering stiffness ($C_{\alpha f}$) are design parameters. This simplification reduces the number of tuning parameters.

This aligning moment term reflects the current road condition to the driver via the steering feel. Real-time friction estimation approaches, like that proposed by Hsu and Gerdes [17], can be used to obtain the surface coefficient of friction. This results in the appropriate front lateral tire force and pneumatic trail being used to generate an aligning moment term that corresponds to the current road condition.

4) Jacking Torque (τ_{jack}): The front *normal* tire forces (F_{zf}) affect the steering feel via the jacking torque. Jacking torque creates the centering feel for a driver at low speeds.

By assuming that the sum of the normal tire forces is much greater than their difference, i.e., $((F_{zfl} + F_{zfr}) >> (F_{zfl} - F_{zfr}))$, and that the roadwheel

angle (δ_{rw}) is small, we can approximate (3) as a spring model. This spring model gives a useful approximation of jacking torque and captures the important centering effect in a simple model. In a production vehicle, the steering system typically has a deadband in which the stiffness is lower. The two spring constant model given below allows the deadband characteristics to be incorporated into the model:

$$\tau_{jack} = \begin{cases} -k_{db}\delta_{rw}, & \text{if } |\delta_{rw}| \leq \delta_{db} \\ -k_{\text{jack}}\delta_{rw} - k_{db}\left(\text{sgn}(\delta_{rw})\delta_{db}\right), & \text{if } |\delta_{rw}| > \delta_{db} \end{cases} \tag{10}$$

where the steering deadband (δ_{db}), deadband spring constant (k_{db}), and jacking torque spring constant (k_{jack}) are design parameters. The resultant τ_{jack} given by this function is continuous. The two spring constants represent the different stiffnesses due to the steering deadband region.

5) Power Assist: In modern power steering systems, drivers receive an assistive torque counteracting the total front tire moment to aid them in turning the roadwheels. This torque factors into the steering feel. Ryu and Kim demonstrated that the effect of this assistive torque on the resultant steering feel is that a smaller fraction of the total front tire moment is included in the steering feel [18]. Using the expressions for aligning moment and jacking torque obtained earlier, the total front tire moment can be approximated by

$$\tau_{\text{tire moment}} \approx \tau_{\text{jack}} + \tau_{\text{align}}. \tag{11}$$

The fraction of the total front tire moment felt by the driver is then modeled as a weighting function (W_f) that depends on front slip angle. The resultant assisted total front tire moment is given as

$$\tau_{\text{assisted tire moment}} = \tau_{\text{tire moment}}\, W_f. \tag{12}$$

Building on the work of Ryu and Kim, a good weighting function must fulfill some critical properties [18]. This weighting function must have a unity value at zero front slip angle, which decreases as the magnitude of front slip angle increases. This indicates that there is no assist at zero front tire force and the level of assist increases with the increasing magnitude of front tire force. The weighting function must also have a lower limit which translates to a saturation of the power assist effect when the magnitude of the front tire force exceeds a given threshold which is set based on the desired power assist characteristics. Modeling the weighting function (W_f) as a Gaussian function fulfills these critical requirements. The weighting function (W_f) is given by

$$W_f = e^{\frac{-\alpha_f^2}{2\sigma_{ps}^2}}(1-\gamma) + \gamma \tag{13}$$

where the front slip angle (α_f) is known, while the standard deviation (σ_{ps}) and lower limit (γ) for the weighting function are selected to create the desired power steering effect. A typical weighting function is illustrated in Fig. 4.

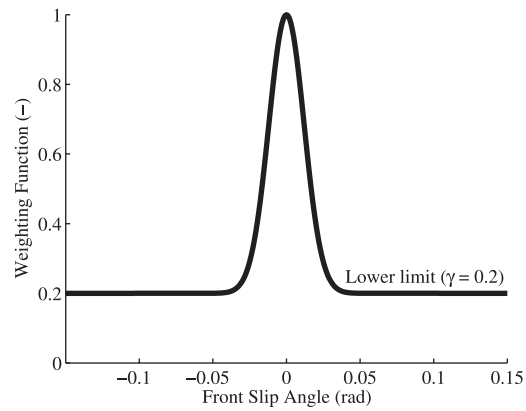

Fig. 4. Power assist weighting function.

III. Objective Steering Feel Performance Measures

Having a model to generate steering feel, as presented in the section above, is only one part of the solution to designing an artificial steering feel. In order to design a good steering feel, an effective way to characterize the steering feel must also be employed. One way to do this is to use surveys as a means of steering feel evaluation as demonstrated using a simulator by Mandhata *et al.* [12]. Though very useful in characterizing a given steering feel, surveys give a subjective evaluation of steering feel. This makes the iterative process of designing a good steering feel time-consuming as these experiments must be conducted repeatedly for each iteration.

In industry, objective performance measures for steering feel are used to give designers a means of characterizing steering feel in terms that drivers care about. These performance measures are used to evaluate on-center handling and are obtained by performing a weave test, in accordance with the ISO13674-1:2010 standard [19], as described by Norman [20] and Salaani *et al.* [21]. On-center handling describes the steering feel of a vehicle during nominal straight-line driving and in negotiating large radius bends at high speed but low lateral acceleration. These performance measures depend not just on steering characteristics but on vehicle dynamics as well.

The performance measures introduced in this section are obtained from data of a vehicle performing a weave maneuver, which consists of a 0.2-Hz steering input such that a maximum lateral acceleration of 0.2 g is observed at 60 m/h. The weave is illustrated in Fig. 5. Five different performance measures are obtained from the following data:

(1) returnability;
(2) on-center feel;
(3) linearity;
(4) effective torque stiffness;
(5) steering sensitivity.

These measures are obtained using three different crossplots:

(1) handwheel torque versus driver steering input (see Fig. 7);
(2) handwheel torque versus vehicle lateral acceleration (see Fig. 6);
(3) vehicle lateral acceleration versus driver steering input (see Fig. 8).

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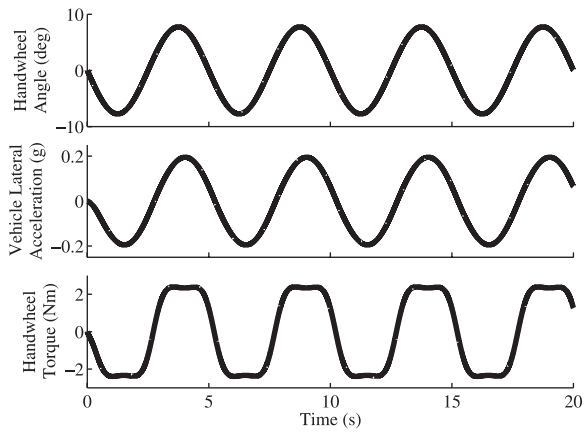

Fig. 5. Weave at 60 m/h.

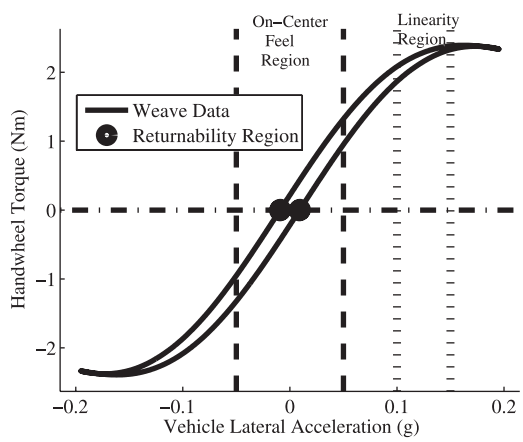

Fig. 6. Handwheel torque—vehicle lateral acceleration.

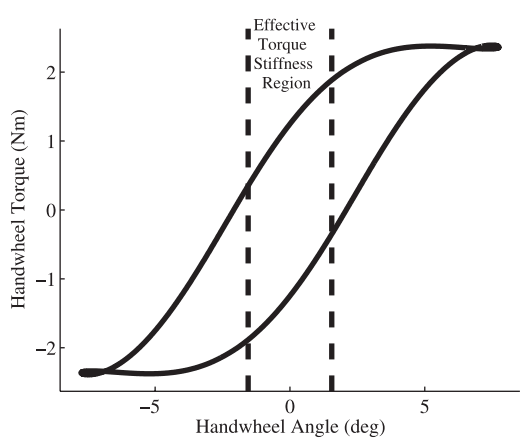

Fig. 7. Handwheel torque—driver steering input.

Since a weave is a dynamic maneuver, the vehicle's handwheel torque and vehicle lateral acceleration depend on both the current driver steering input and the past state of the vehicle. Therefore, the crossplots listed above display hysteresis. The nonlinear nature of the dynamics also means that closed-form formulas for these plots cannot be obtained.

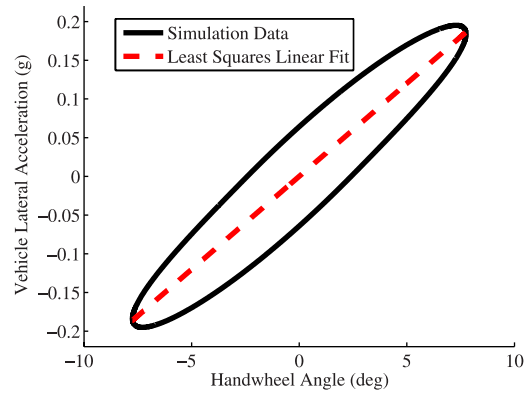

Fig. 8. Vehicle lateral acceleration—driver steering input.

A. Measures From Handwheel Torque—Vehicle Lateral Acceleration Crossplot

Fig. 6 illustrates a typical handwheel torque versus vehicle lateral acceleration plot. *Returnability, on-center feel*, and *steering torque linearity* are obtained from this plot.

1) Returnability: Returnability is the vehicle lateral acceleration at zero handwheel torque, as illustrated in Fig. 6, and represents the dynamic lag in the system. A high returnability indicates that the vehicle has persistent lateral acceleration while at zero handwheel torque and corresponds to a "heavier" steering feel.

2) On-Center Feel: On-center feel is the steering torque gradient between -0.05 and $+0.05$ g in vehicle lateral acceleration, as illustrated in Fig. 6. It represents the handling of the vehicle during highway driving which includes nominally straight-line driving and the negotiation of large radius bends at high speeds but low lateral accelerations.

3) Steering Torque Linearity: Linearity is the ratio of the steering torque gradient between $+0.1$ and $+0.15$ g (illustrated in Fig. 6) and the on-center feel described above. Linearity represents the level of power assist with a larger linearity indicating a lower power assist effect and a clearer feel of tire dynamics and road properties for the driver.

B. Measures From Handwheel Torque—Driver Steering Input Crossplot

Fig. 7 illustrates a typical handwheel torque versus driver steering input plot. Data from this plot determine the *effective torque stiffness*.

1) Effective Torque Stiffness: Effective torque stiffness is the steering torque gradient computed between -20% and $+20\%$ of the maximum steering input, as illustrated in Fig. 7, and is the stiffness felt when turning. Drivers use the change in stiffness, which depends on the effective torque stiffness, as a means of detecting the center position of the handwheel. Increasing the stiffness will make the steering feel "heavier" while also making the center position clearer to the driver.

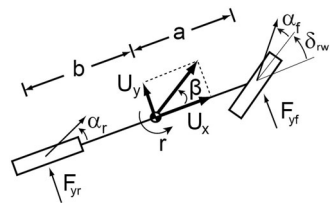

Fig. 9. Bicycle model.

C. Measures From Vehicle Lateral Acceleration Versus Driver Steering Input Crossplot

Fig. 8 illustrates a typical vehicle lateral acceleration versus driver steering input plot. *Steering sensitivity* uses data from this plot to be calculated.

1) Steering Sensitivity: Steering sensitivity is the handwheel angle gradient between -0.2 and $+0.2$ g of lateral acceleration. A least squares linear fit of the data, as illustrated in Fig. 8, gives this gradient. Increasing the sensitivity corresponds to a more "crisp" and responsive feel. Since steering sensitivity depends solely on handwheel angle and lateral acceleration, it is independent of handwheel torque. Hence, the steering feel model presented in Section II does not affect this measure. Decreasing the steering ratio or reducing the steering lag in a system will increase steering sensitivity.

IV. Simulation Structure

The objective performance measures introduced in the prior section depend on both vehicle dynamics and the steering model. Therefore, a dynamic vehicle simulation coupled with the steering model can be used to characterize and design a steering feel. Performing an ISO-standard weave in simulation with a particular set of vehicle and steering feel model parameters allows these performance measures to be obtained for a given set of model parameters. Using this simulation, different steering parameters, for any type of vehicle, can be rapidly evaluated and tuned objectively to obtain the desired steering feel.

A. Vehicle Model

Since the weave maneuver is performed at a constant speed and with low lateral acceleration, a simple bicycle model, as illustrated in Fig. 9, can be used for this simulation. The front and rear axles are lumped together and modeled as a single entity each. The dynamic states are sideslip (β) and yaw rate (r). The equations of motion are as follows:

$$\dot{\beta} = \frac{F_{yf} + F_{yr}}{mU_x} - r \tag{14}$$

$$\dot{r} = \frac{aF_{yf} - bF_{yr}}{I_{zz}} \tag{15}$$

where m is the vehicle mass, U_x is the longitudinal velocity in the body fixed frame, I_{zz} is the yaw inertia, and a and b are the distances from the center of gravity to the front and rear axles, respectively. Since this simulation represents the standard weave, U_x is constant. The tire slip angles are expressed as nonlinear functions of the dynamic states and the front steer angle, δ_{rw}

$$\alpha_f = \tan^{-1}\left(\beta + \frac{ar}{U_x}\right) - \delta_{rw} \tag{16}$$

$$\alpha_r = \tan^{-1}\left(\beta - \frac{br}{U_x}\right). \tag{17}$$

Lateral tire forces are obtained using the nonlinear brush tire model (8).

Fig. 10. X1 Steer-by-wire vehicle.

TABLE I
X1 Parameters

Parameter	Value	Units
m	1973	kg
I_{zz}	2000	$\mathrm{kg{\cdot}m^2}$
a	1.53	m
b	1.23	m
$\mathrm{C}_{\alpha f}$	110	$\mathrm{kN{\cdot}rad^{-1}}$
$\mathrm{C}_{\alpha r}$	148	$\mathrm{kN{\cdot}rad^{-1}}$
J_{system}	0.0014	$\mathrm{kg \cdot m^2}$
b_{system}	0.015	$\mathrm{N \cdot m/rad \cdot s^{-1}}$

B. Simulation Parameters

For this paper, the vehicle parameters used in simulation are from an experimental steer-by-wire vehicle, X1, illustrated in Fig. 10. The parameters are shown in Table I. X1 is a true steer-by-wire vehicle with independent steering motors on each wheel and no steering rack or torsion bar. It also has an FFB steering system allowing artificial steering feel to be generated at the handwheel.

V. Steering Feel Design

Using the simulation structure described in Section IV, the effect of the steering feel model parameters and of varying speed on steering feel can be investigated. This knowledge is crucial to understanding and designing steering feel intuitively.

A. Effect of Parameters

Section III introduced four performance measures (*returnability, on-center feel, linearity, and effective torque stiffness*) that can be changed by modifying the steering feel model

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TABLE II
STEERING FEEL ALGORITHM PARAMETER VARIATION TABLE

Increasing Parameter (↑)	Returnability	on-center	Linearity	Effective Torque stiffness
Change in damping (Δb)	↑	-	-	-
Change in inertia (ΔJ)	-	↓	-	↓
Deadband spring constant (k_{db})	↑	↑	-	↑
Jacking torque spring constant (k_{jack})	-	↑	-	-
Weighting function standard deviation (σ_{ps})	-	-	↑	↑
Weighting function lower limit (γ)	-	-	↑	-
Tire moment gain (K)	-	↑	-	↑

parameters. The final performance measure (*steering sensitivity*) does not depend on handwheel torque and cannot be varied through changing the parameters of the steering feel model. It is simply a reflection of the mapping chosen from handwheel to roadwheel angle.

Section II introduced the following seven parameters that can be varied in the steering feel model to obtain different steering feel characteristics:

(1) change in damping (Δb);
(2) change in inertia (ΔJ);
(3) deadband spring constant (k_{db});
(4) jacking torque spring constant (k_{jack});
(5) weighting function standard deviation (σ_{ps});
(6) weighting function lower limit (γ);
(7) tire moment gain (K).

By varying each of the parameters listed above in turn while the other parameters are held constant, the effect of each individual change can be investigated using the simulation described in Section III. Table II documents the results of *increasing* each model parameter in turn while holding the other parameters constant. If the effect results in an *increase* in a particular performance measure, a up-arrow (↑) is used. If the effect results in a *decrease* in that measure, a down-arrow (↓) is used. If the effect results in the measure being held mostly *constant*, a dash (−) is used. For example, in the first row concerning the change in damping (Δb), increasing this parameter results in an increase in returnability while all other measures remaining approximately constant.

1) Critical Model Parameters: Table II shows that more than one model parameter can have the same effect on a given performance measure. Hence, a reduced set of critical model parameters can be tuned in order to obtain any desired steering feel characteristics. This allows for a more intuitive tuning procedure. In order to control the four performance measures mentioned, four critical parameters are chosen. They are marked in bold in Table II and listed below:

(1) change in damping (Δb);
(2) jacking torque spring constant (k_{jack});
(3) weighting function lower limit (γ);
(4) tire moment gain (K).

The change in damping controls the returnability, the jacking torque spring constant controls the on-center feel, the weighting function lower limit controls the linearity, and the tire moment gain controls the on-center feel and effective torque stiffness. These parameters have the additional benefit that their effect is limited mainly to the performance measures that they control resulting in a targeted tuning ability. Though the tire moment gain controls both the effective torque stiffness and the on-center feel, the jacking torque spring constant can be used to tune the on-center feel without affecting the effective torque stiffness. Note that the change in inertia and weighting function standard deviation can also be similarly used in tandem with another parameter to vary effective torque stiffness. However, the change in inertia requires a noisy twice differentiated handwheel angle signal for implementation, and the weighting function standard deviation is a more abstract parameter as compared to the tire moment gain which has a physical basis. Therefore, using the tire moment gain as a means to vary effective torque stiffness results in a more intuitive tuning parameter that is simple to implement.

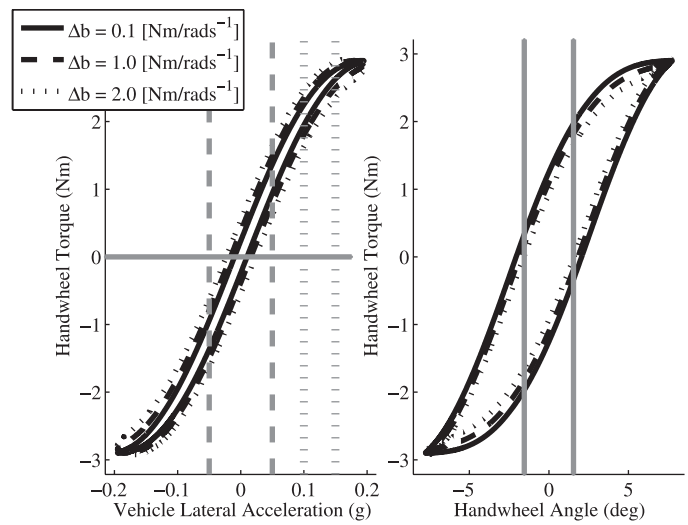

Fig. 11. Steering feel crossplots when the change in damping is varied.

The effect of each of these parameters on the steering feel crossplots and hence the performance measures can be seen in Figs. 11–14, respectively, where the relevant regions for the performance measures are marked as in Figs. 6 and 7.

B. Effect of Speed

The performance measures described in Section III represent the steering feel characteristics of a vehicle performing a fixed-speed weave maneuver. Since an ISO-standard weave can be performed at different speeds, this implies that the steering

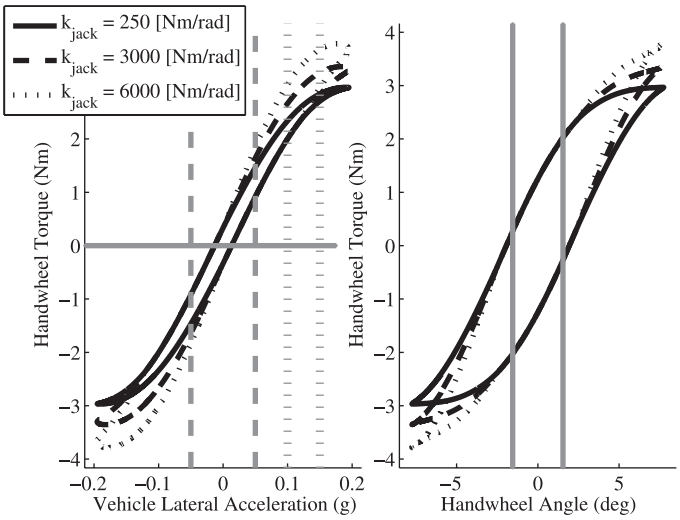

Fig. 12. Steering feel crossplots when the jacking torque spring constant is varied.

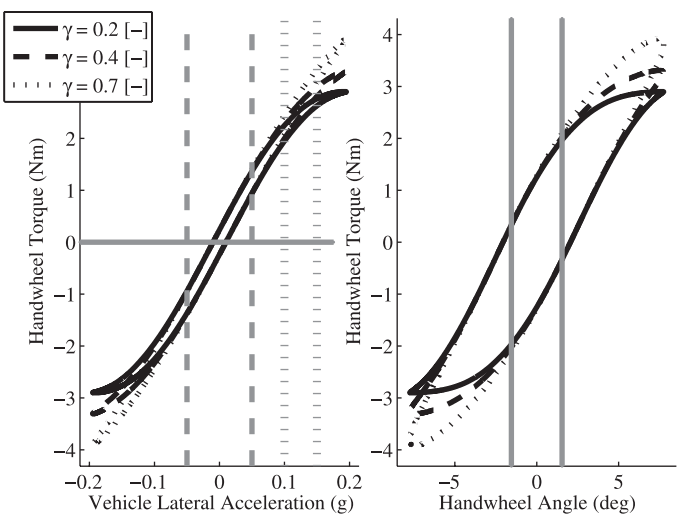

Fig. 13. Steering feel crossplots when the weighting function lower limit is varied.

feel performance measures obtained depend on speed. Using the simulation described in Section IV, Fig. 15 illustrates how the steering feel crossplots vary with speed given in meters per second.

Effective torque stiffness and *steering sensitivity* both increase with increasing speed, as seen in Fig. 15. Though the relationship between these two performance measures and speed is mathematically complex, involving many different terms, it is approximately proportional to speed when using typical steering parameters

$$\text{Effective torque stiffness} \underset{\sim}{\propto} U_x \tag{18}$$

$$\text{Steering sensitivity} \underset{\sim}{\propto} U_x. \tag{19}$$

Returnability of the feel decreases as speed increases resulting in the driver experiencing a "lighter" feel. Again, though the

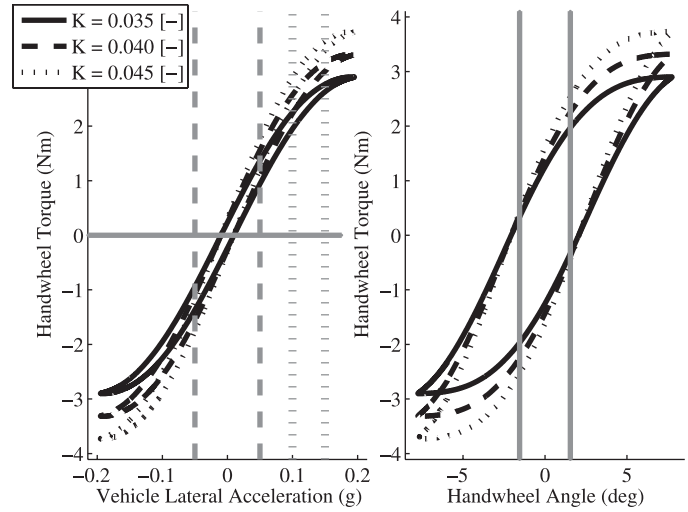

Fig. 14. Steering feel crossplots when the tire moment gain is varied.

relationship between returnability and speed is mathematically complex, it is approximately proportional to the reciprocal of speed when using typical steering parameters

$$\text{Returnability} \underset{\sim}{\propto} \frac{1}{U_x}. \tag{20}$$

On-center feel decreases with increasing speed resulting in a less pronounced centering effect of the steering.

Linearity increases with increasing speed indicating that the power assist effect decreases as well.

C. Design of Steering Feel

Using the simulation described in the Section IV, any given set of parameters for the model can, through simulation at a given speed, be mapped to the five performance measures described. This allows steering feel to be designed objectively in terms that drivers care about. Realistic ranges for the performance measures can be obtained from surveys of a variety of different vehicle models performing an ISO-standard weave at a fixed speed [21]. A steering feel can then be designed in simulation to obtain the desired performance measures using these ranges as a guide. This desired set of performance measures will then reflect the steering feel effects that individual designers deem important.

With this in mind, a realistic steering feel was designed for use in the steer-by-wire vehicle, X1. Low *returnability* and *linearity* helped to emulate the familiar "lighter" and highly assisted steering feel of modern power steering vehicles. A middle of the range *on-center feel* ensured comfortable centering properties during nominal highway driving while the appropriate *stiffness* ensured comfortable centering properties at low speeds. Finally, a higher *steering sensitivity* than the given range exploited the inherent benefit of a steer-by-wire system in reducing conventional steering lag, resulting in a highly responsive driving experience. The realistic performance measure ranges for a weave at 60 m/h as well as the measures selected for X1

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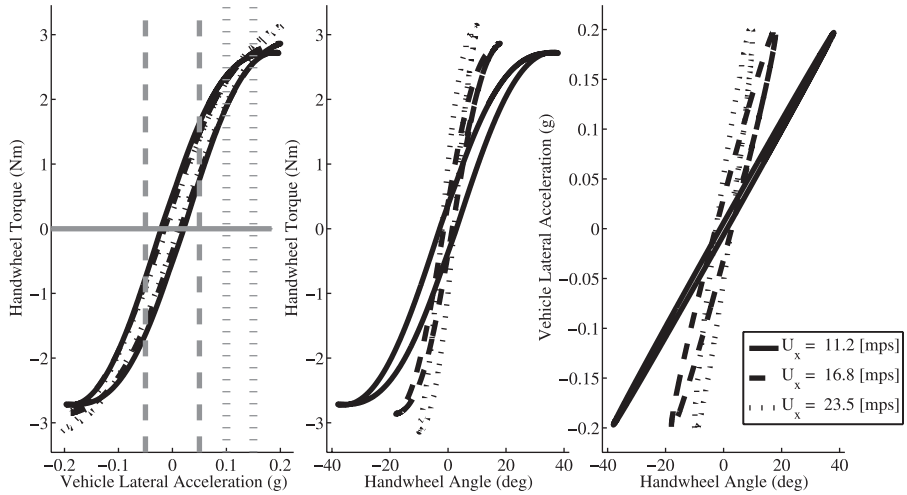

Fig. 15. Steering feel crossplots when the speed is varied.

TABLE III
STEERING FEEL DESIGN

Measure	Realistic Range (60 m/h) [21]	X1 (60 m/h)	X1 (25 m/h)
On-Center Feel (N·m/g)	7–33	17	22
Stiffness (N·m/deg)	na	0.37	0.12
Sensitivity (g/100deg)	0.27–1.42	2.33	0.52
Linearity (%)	6–121	25	22.7
Returnability (g)	0.01–0.13	0.01	0.01

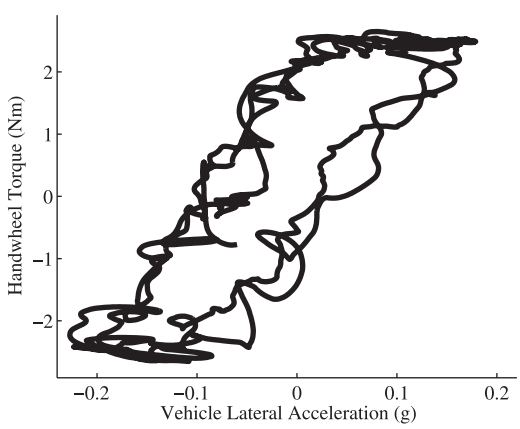

Fig. 16. Experiment: Handwheel torque—vehicle lateral acceleration.

at 60 m/h and the corresponding measures for X1 at 25 m/h are shown in Table III.

VI. ARTIFICIAL STEERING FEEL ON A STEER-BY-WIRE VEHICLE

A. Implementation on a Steer-by-Wire Vehicle

The model described in Section II-B can also be programmed directly on a steer-by-wire vehicle. The model uses the driver's steering angle (δ_{hw}), roadwheel angle (δ_{rw}), and front slip angle

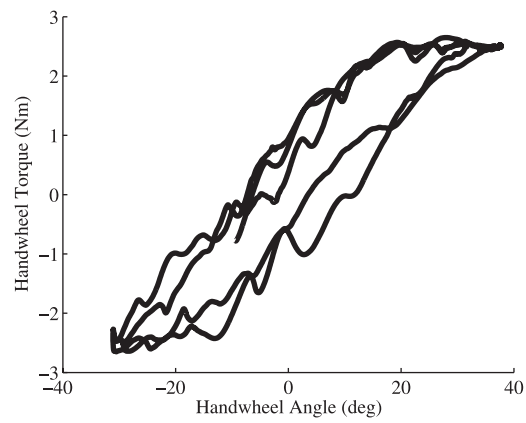

Fig. 17. Experiment: Handwheel torque—driver steering input.

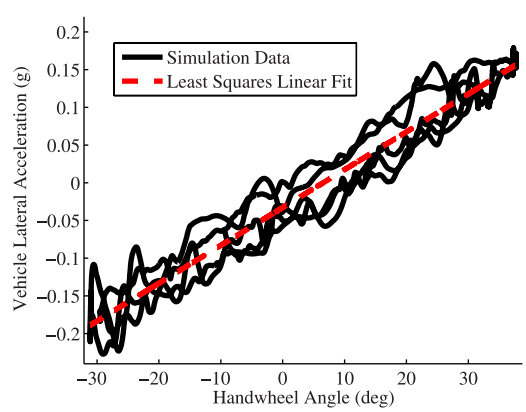

Fig. 18. Experiment: Vehicle lateral acceleration—driver steering input.

(α_f) as inputs while generating the handwheel FFB motor torque (τ_{motor}) in (1).

On the steer-by-wire vehicle X1, precise encoders obtain the handwheel angle and roadwheel angle. The roadwheel angle derivative is approximated by differencing the encoder signal.

TABLE IV
EXPERIMENTAL COMPARISON (25[M/H])

Measure	X1 (Experimental)	X1 (Simulation)
On-Center Feel (N·m/g)	23	22
Stiffness (N·m/deg)	0.11	0.12
Sensitivity (g/100deg)	0.50	0.52
Linearity (%)	23	22.7
Returnability (g)	0.02	0.01

A low-pass filter with cutoff frequency 10 Hz removes high-frequency sensor noise on this derivative. Since change in inertia is not necessary with this vehicle, the second derivative of the roadwheel angle is not calculated.

X1 uses an integrated global positioning system (GPS) and inertia measurement system (INS) to obtain the yawrate (r), sideslip (β), and longitudinal speed (U_x) of the vehicle. Based on (16), the front slip angle (α_f) can be calculated from this information. The motor torque (τ_{motor}) is then fed to the motor that is part of the FFB steering system on the vehicle. A dSPACE MicroAutoBox II (DS1401) performs all computation at a rate of 500 Hz.

B. Experimental Validation of Simulation Design Technique

Since the performance measures are dependent not just on the steering feel model but also vehicle dynamics, experimental validation verifies the validity of the simulation-based design technique. A manual weave test was conducted on the X1 test vehicle, introduced in Section IV-A, at a speed of 24–26 m/h.

C. Experimental Results

Figs. 16–18 illustrate the crossplots obtained from data of real-time artificial steering feel emulation on X1. Since the experiment was conducted at a lower speed than that in the design simulation, the steering angle required in the weave to get the maximum vehicle lateral acceleration of 0.2 g is larger in the experiment than in the design simulation. This results in Figs. 17 and 18 having a smaller range of steering input than that in Figs. 7 and 8. Also, since returnability increases with decreasing speed, the experimental crossplot in Fig. 16 is wider than the design simulation crossplot given in Fig. 6.

To compare the experimental data with simulation, a new simulation is performed to account for the change in speed and steering input. Table IV compares the performance measures obtained using this simple vehicle simulation and the experimental weave performed by X1. The five performance measures obtained through simulation match well with the measures obtained through experimental data. Since the performance measures also depend on the vehicle dynamics, this favorable comparison indicates that the dynamic vehicle simulation captures the dynamics of the weave well. Therefore, the design technique based on this simulation can be used to incorporate a desired set of objective performance measures into an actual vehicle's steering feel.

VII. CONCLUSION

This paper proposes a steering feel model of appropriate fidelity to capture the elements of steering feel used for objective evaluation. The fidelity of this model is such that while capturing all these effects, it remains intuitive to tune objectively with a small set of parameters. Using a simple dynamic vehicle simulation, coupled with the steering feel model, steering feel can be designed for an individual vehicle. Tuning a critical set of parameters of the steering feel model in simulation allows the design of a steering feel which has a desired set of performance measures. This desired steering feel can be implemented as the baseline steering feel on a steer-by-wire vehicle. In the future, additional haptic information can then be overlaid on this feel to communicate vital information (e.g., lane boundaries, oncoming obstacles, vehicle handling limits) to the driver, aiding in the development of new driver assistance systems.

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