

Development of an Electric-based Power Steering System

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ABSTRACT

In this research, a three degree-of-freedom (DOF) rack-type electric-based power steering (EPS) model is developed. The model is coupled with a three DOF vehicle model and includes EPS maps as well as non-linear attributes such as vibration and friction characteristics of the steering system. The model is simulated using Matlab's Simulink. The vibration levels are quantified using on-vehicle straight-line test data where strain-gauge transducers are placed in the tie-rod ends. Full vehicle kinematic and compliance tests are used to verify the total steering system stiffness levels. Frequency response tests are used to adjust tire cornering stiffness levels as well as the tire dynamic characteristics such that vehicle static gain and yaw natural frequency are achieved. On-center discrete sinusoidal on-vehicle tests are used to further validate the model. The validation of the model demonstrates that the interaction of the chassis and steering system can be considered during the early stages of vehicle development.

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INTRODUCTION

To gain an understanding of a vehicle's dynamic response, traditional handling simulations have used a simplified two DOF planar bicycle model [1, 2, and 3]. Of course, this model is rather basic in nature and can capture only first-order influences such as mass, weight distribution (center of gravity location), tire cornering stiffness and wheelbase. Methods exist to capture the effects of compliances due to lateral force and aligning moment and kinematic effects due to roll steer and roll camber [4]. These techniques, however, are typically used only to accurately investigate steady-state gain, not delay (phase). By including body roll, the delay effects of roll steer/camber (i.e., toe/camber change due to body roll) can be included. The delay arises because body roll lags lateral acceleration. One of the original models to include body roll is the three DOF one developed by Segel [5], which has been widely used by vehicle dynamicists.

For either of these models (2-DOF/bicycle or 3-DOF), the hand-wheel angle (δ_{SW} , controlled by the driver) is related to the road-wheel angle (δ_{RWA}) through a simple kinematic ratio, known as the on-center or steering ratio:

$$\delta_{RWA} = \frac{\delta_{SW}}{K_{SR}} \quad (1)$$

This approach is rather rudimentary and the ratio simply "scales" the overall response. A vehicle's steering system is more complex and has its own system dynamics. A one DOF steering model is shown below:

$$\delta_{RWA} = \frac{\frac{\delta_{SW}}{K_{SR}} + \frac{M_{zl} + M_{zr}}{K_{cs}}}{\frac{s^2}{\omega_n^2} + \frac{2\zeta}{\omega_n}s + 1} \quad (2)$$

Where, M_{zl} and M_{zr} are the left and right tire self-aligning moments (from the vehicle tire model), respectively, and K_{cs} is the overall effective steering stiffness. Results in Figure 1 compare the step response for a vehicle without a steering model (i.e., hand-wheel related to road-wheel using Equation 1) and a vehicle with a simple one DOF second-order model (Equation 2). The plot illustrates that having a basic steering system included in the vehicle model has significant influence on the results and the dynamics of the steering system cannot be ignored.

Including this model emulates the basic system dynamics of a steering model. The model is linear, however, and does not include non-linear elements such as friction or the assist from the power steering system. Moreover, estimating steer effort or rack load cannot be performed.

Research performed in [6 and 7] developed higher-order steering models and included important elements of a traditional hydraulic-power steering system. The model in [6] was limited in that only the power-off condition was investigated and no boost curve was included. A more comprehensive model in [7] included non-linear friction elements and a power boost curve. Using this model, steering wheel torque was estimated and exhibited reasonable correlation with experimental data.

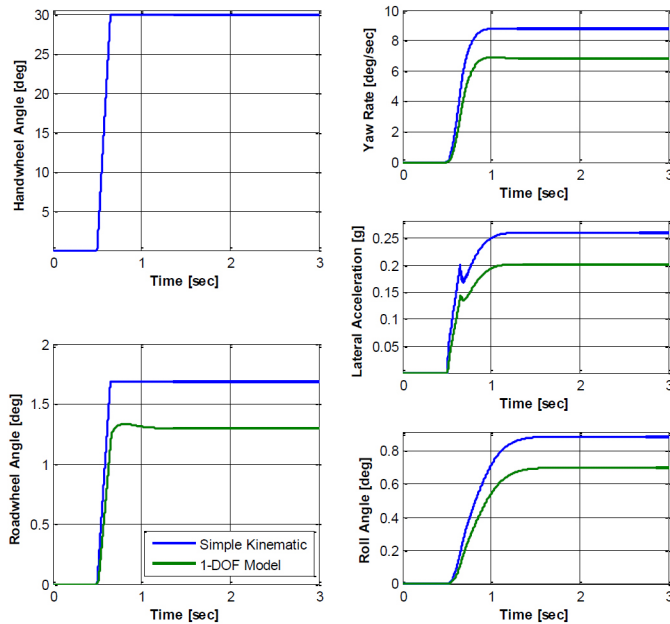


Figure 1. Simple Steering Model Results

The architecture for contemporary power steering systems has changed significantly [8]. Manufacturers are increasingly using electric-based power steering (EPS) as it is more energy efficient (increased fuel economy), easier to assemble and tends to reduce steering-related warranty claims due traditionally to hydraulic leaks from hydraulic-based power steering (HPS). Limited research or models are in the literature developing the intricacies of EPS. One such example is shown in [9], where a controller was incorporated to illustrate how system gains influenced response, such as torque effort. The details of EPS were still limited, however.

The aim of this paper is to develop a detailed steering model, with the goal of predicting on-center response (feel). A three DOF model will be presented and the implementation of a rack EPS will be included. The power steering boost curves (maps) will be shown and the model will be validated using test data. Details of how to tune the model will also be discussed.

MODEL DESCRIPTION

A schematic of the three DOF steering model used in this research is shown in Figure 2. It is coupled with a three DOF vehicle model and is similar to the one developed by [5] and is common in the literature.

For brevity, we will focus on the steering model within. The steering model consists of two rotating masses (inertias): a steering wheel and shaft and an intermediate shaft; and one translating mass: the rack and effective masses of the pinion and motor. The upper shaft (steering wheel) and intermediate shaft are connected via a column (rag) coupler; and the pinion and intermediate shaft are connected through the torsion bar.

Equations of Motion (EOM)

The equations of motion are derived using Newton's second law of motion. Equation 3 describes the motion for the steering wheel and shaft upstream of the rag coupler.

$$I_{SW}\ddot{\theta}_{SW} + B_s(\dot{\theta}_{SW} - \dot{\theta}_{IS}) + K_s(\theta_{SW} - \theta_{IS}) = T_{SW} \quad (3)$$

For this work, the input is steering wheel acceleration. After direct integration, velocity and angle are computed. In a non-traditional sense, steering torque is not an input, rather a result of Equation 3. In other words, the model is run in displacement control, not torque control. The equation for the intermediate shaft is found to be:

$$I_{IS}\ddot{\theta}_{IS} + B_s(\dot{\theta}_{IS} - \dot{\theta}_{SW}) + K_s(\theta_{IS} - \theta_{SW}) + \dots + K_{Tbar}(\theta_{IS} - \theta_p) + \text{sgn}(\dot{\theta}_{IS})f_{upper} = V_{IS} \quad (4)$$

Where, f_{upper} and V_{IS} are the friction and vibration levels in the intermediate shaft, respectively. The equation of motion for the rack is:

$$M_{eff}\ddot{x}_{rack} + B_{rack}\dot{x}_{rack} + \frac{K_{Tbar}}{r_{pin}}(\theta_p - \theta_{IS}) + \dots + \text{sgn}(\dot{x}_{rack})f_{down} = F_{PA} + V_{Trod} - F_{rack} \quad (5)$$

Where, f_{down} and V_{Trod} are the friction and vibration levels in the tie rod, respectively. F_{PA} is the power assist force from the motor and F_{rack} is the rack load. Rack displacement is related to pinion radius using Equation 6 and the total effective mass of the rack includes the rack mass, and the effective translational masses of the tire/wheel assemblies and the electric motor (Equation 7).

$$x_{rack} = \theta_p r_{pin} \quad (6)$$

$$M_{eff} = M_{rack} + 2\frac{I_{wheel-Z}}{l_{knuckle}^2} + \frac{I_{motor}}{r_{motor}^2} \quad (7)$$

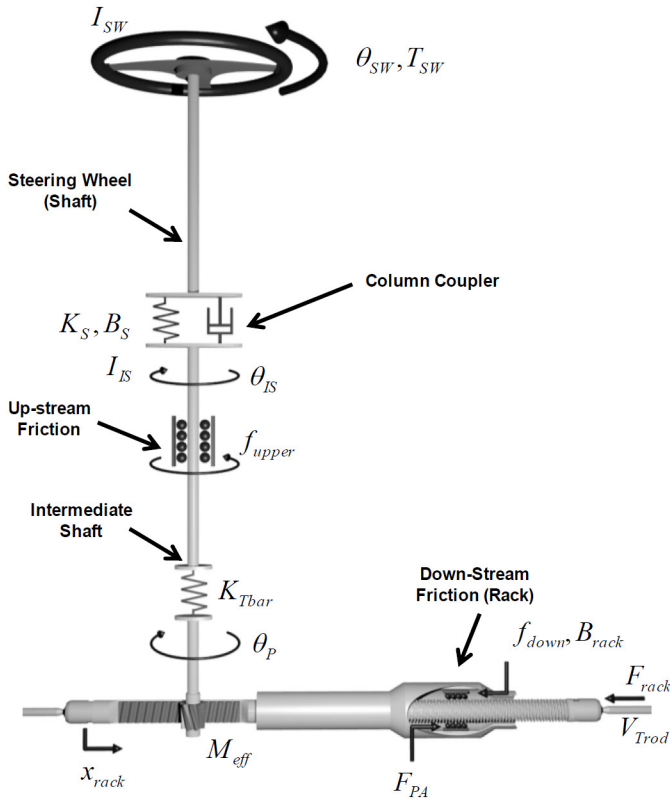


Figure 2. 3-DOF Steering Model Schematic

Rack Load

The rack load is calculated using the following equation:

$$F_{rack} = 2 \frac{M_{K/P}}{l_{knuckle}} \quad (8)$$

Where, $M_{K/P}$ is the kingpin torque and $l_{knuckle}$ is the kingpin longitudinal offset (knuckle-arm length). The two components for kingpin torque are aligning moment from the tire and the gyroscopic torque due to camber roll. See Figure 3.

$$M_{K/P} = F_Y (l_{pneu} + l_{mech}) + T_{Gyro} \quad (9)$$

Tire aligning moment is computed by the product of lateral force and the summation of pneumatic and mechanical trails, which are typically aft of the contact patch center and have negative signs. The gyroscopic torque arises from wheel camber due to vehicle body roll in turn. For a rotating wheel, overturning the wheel (camber) results in a precession (gyroscopic) torque in an axis orthogonal to the axis of wheel rotation and overturning. Equation 10 illustrates this computation:

$$\begin{aligned} T_{Gyro} &= I_{wheel-Y} \omega_y \omega_x \\ &= I_{wheel-Y} \left(\frac{v_x}{r_{wheel}} \right) (\gamma_{roll} \dot{\phi}) \end{aligned} \quad (10)$$

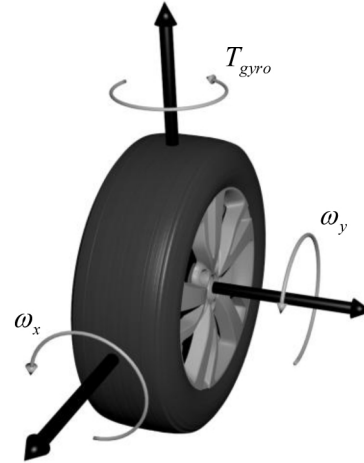


Figure 3. Gyroscopic Torque

Where, γ_{roll} is camber gain, $\dot{\phi}$ is roll rate and $I_{Wheel-Y}$ is tire/wheel spin inertia. Combining Equations 8,9 and 10 results in the total rack load:

$$F_{rack} = 2 \frac{F_Y (l_{pneu} + l_{mech}) + I_{wheel-Y} \left(\frac{v_x}{r_{wheel}} \right) (\gamma_{roll} \dot{\phi})}{l_{knuckle}} \quad (11)$$

Because the pneumatic and mechanical trails are typically negative, this equation indicates that vehicles, which exhibit greater camber gain (roll camber), would exhibit greater gyroscopic torque. Consequently, this vehicle would have lower rack load than a vehicle with similar architecture but with lower camber gain.

Tire Model

The tire model used in this research is a simple linear model where tire slip angle is related to lateral force using a constant cornering stiffness.

$$F_{Y-f,r} = C_{f,r} (F_{Z-f,r}) \alpha_{f,r} \quad (12)$$

Compliance effects, due to lateral force (δ_{Fy}) and aligning moment (δ_{Mz}) and kinematic effect, due to roll angle (δ_{ϕ}), are included in the front and rear slip angles calculations as defined in Equations 13 and 15, respectively.

$$\alpha_f = \delta_{RWA} + \delta_{Fy} + \delta_{Mz} + \delta_{\phi} - \frac{v_y + l_f r}{v_x}$$

Where, road-wheel angle is related to rack displacement by:

$$\delta_{RWA} = \frac{x_{rack}}{l_{knuckle}} \quad (14)$$

$$\alpha_r = \delta_{Fy} + \delta_{Mz} + \delta_{\phi} - \frac{v_y - l_r r}{v_x} \quad (15)$$

To capture the dynamic behavior of tire lateral force development, a second-order transfer function is applied to the slip angles computed in Equations 13 and 15. This model, described and implemented by Heydinger [10], uses the following transfer function:

$$\frac{\alpha'_{f,r}}{\alpha_{f,r}} = \frac{(v_x \omega_{f,r})^2}{s^2 + (2\zeta_{f,r} v_x \omega_{f,r})s + (v_x \omega_{f,r})^2} \quad (16)$$

Where,

$$\omega_{f,r} = \frac{K_L}{C_{f,r}} \quad (17)$$

is considered the path frequency (in rad/m) and uses the tire lateral stiffness (K_L) and cornering stiffness ($C_{f,r}$). This model also uses a damping ratio ($\zeta_{f,r}$), which can vary depending on the tire (see Heydinger [10]). For this research, damping ratio is used as a tuning parameter as will be shown in subsequent sections such that there is better correlation with test data.

Vibration Model

Figure 4 depicts the vibration model used for this research. As will be shown in the Model Validation section, the level of vibration can influence on-center steering performance. The source of vibration is typically due to tire imbalance, which is transferred through the tie-rod to the steering system, but can also be through the steering hanger beam. Because of this, the model can incorporate vibration in the intermediate shaft (column) and/or rack (see Equations 4 and 5). The frequencies and amplitudes used in the model are shown in Equations 18 and 19, respectively. The fundamental frequency (f_1) is based on the path frequency of the tire, which is a function of longitudinal speed (v_x) and static loaded radius (r_{wheel}). TF_{rms} is the experimentally measured root-mean-square (RMS) vibration level. The phase angle for each harmonic is shown in Equation 20, where Matlab's *rand* function is used to add a random component to the model; and results in the phase for each harmonic to be different. Thus, for a given RMS vibration level, each instance the model is run, the total vibration level applied to the rack and/or column is different.

$$f_n = n \frac{v_x}{r_{wheel}}; \quad n = 1, 2, 3 \quad (18)$$

$$A_1 = TF_{rms}; \quad A_2 = \frac{A_1}{3}; \quad A_3 = \frac{A_1}{6} \quad (19)$$

$$\phi_1 = \phi_2 = \phi_3 = 2 \times \pi \times rand \quad (20)$$

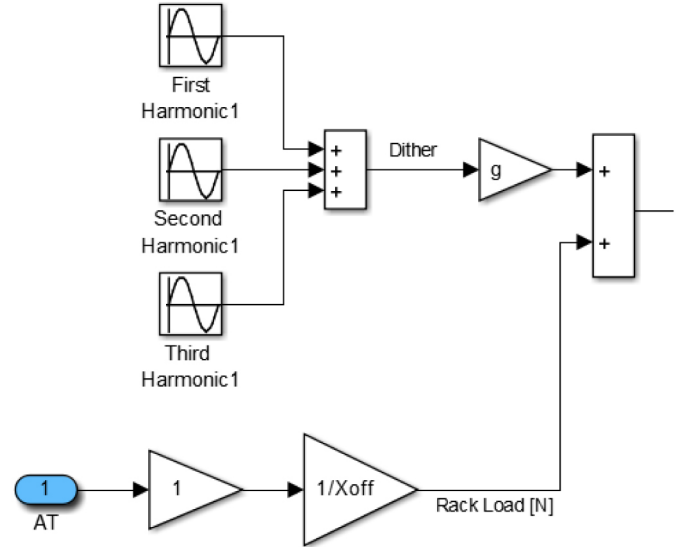


Figure 4. Vibration Model Application

EPS Model

A rack EPS model is developed in this section. The EPS functionality is depicted in Figure 22 in the Appendix. As can be seen, the basic architecture of EPS consists of the base, damping and inertial compensation maps. The base maps (Figure 5) would be the maps used by the test (performance) engineer to get the overall efforts to the appropriate levels, which are dictated by the vehicle targets.

To fine tune the steering response (feel), the test engineer utilizes the damping and inertial compensation features of the EPS system. (Example maps are shown in Figures 23 and 24 in the Appendix.) The damping maps are comprised of the base current (motor speed), velocity ratio (vehicle speed) and torque ratio (pinion torque) maps. Damping is further differentiated depending whether the driver is in a "Go" or "Return" maneuver - see Figure 7. "Go" is when the driver is turning into the maneuver; and "Return" is when the driver is turning out of the maneuver. Table 1 and Figure 8 illustrate how this logic is implemented in the model.

As an example, referring to Figure 6, for a scenario where the driver is in *Go* (the driver is turning into the maneuver) and where normalized motor speed, vehicle speed and pinion torque are 0.6, 0.4 and 0.2, respectively, the total damping compensation is: $1 \times 0.52 \times 0.71 = -0.37$ normalized Amps. This current is added to the base current dictated by map established in Figure 5.

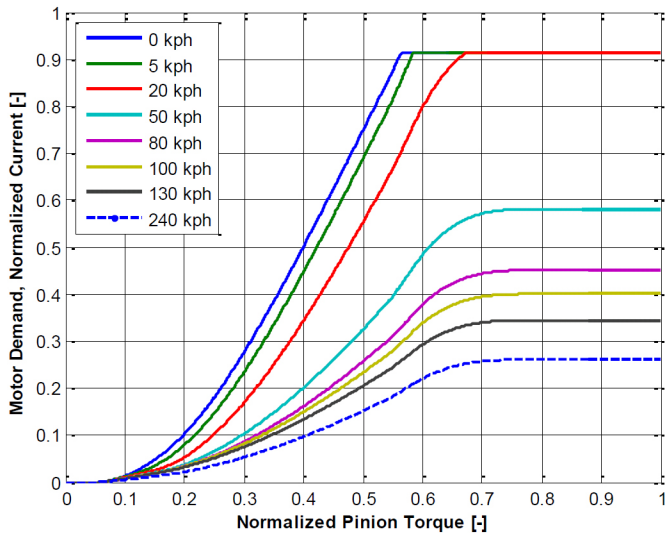


Figure 5. Base EPS Maps

Note for *Return* (the driver is turning out of the maneuver), the damping has two features: base compensation and hysteresis compensation.

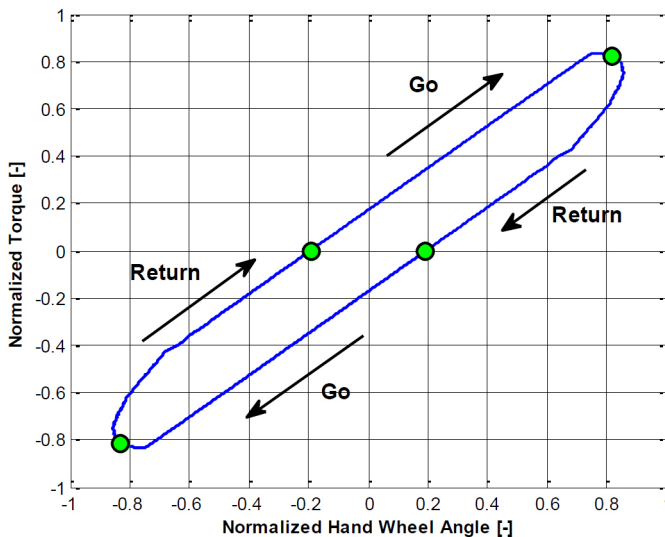


Figure 7. Go/Return Definition

For inertial compensation, the level of current applied is dependent on the base current (pinion torque rate), current ratio (vehicle speed), velocity torque ratio (vehicle speed) and torque ratio (pinion torque) - see Figure 24 in the Appendix. Inertial compensation is applied in both Go and Return, with the Return current demand having opposite sign of Go.

Table 1. Go/Return Truth Table

$sgn(T_{def})$	$sgn(\dot{x}_{rack})$	Go/Return
0	0	Go
0	1	Return
1	0	Go
1	1	Return

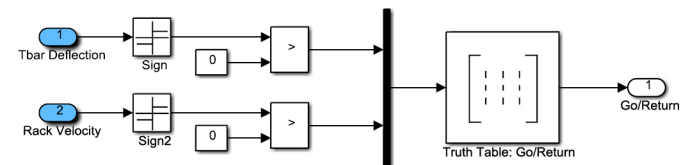


Figure 8. Go/Return Implementation

With the base, damping and inertial compensation current demands determined, the total assist applied to rack is dictated by the torque-speed curves shown in Figure 9. The motor torque output is related to rack assist by the following formula:

$$F_{PA} = \frac{T_{motor}}{r_{motor}} \quad (21)$$

This development is a simplification of the actual EPS system. Moreover, the model is absent of certain intricacies such as the system dynamics and essentially consists of a series of lookup tables and logic statements. To account for some of the system dynamics, a first-order transfer function is applied to the motor torque before it is applied to the rack. Furthermore, there is an additional tuning aspect available in the system that is not included in the model. This is called Motion Adaptive (MA) assist, which uses steering angle and velocity and yaw rate. This accounts for only approximately 4 per cent of additional assist, however.

Damping Go Subtracted (-)

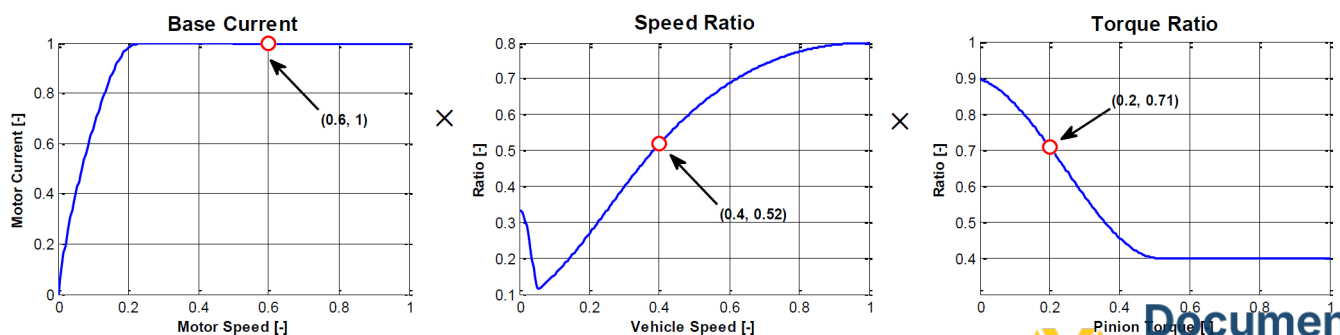


Figure 6. Example Damping Level Calculation (Map is a sub-set of EPS maps shown in Figure 23)