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ABSTRACT

This paper discusses the use of a simulation model to predict static steering torque during parking maneuvers with the vehicle's brakes applied. Accurate prediction of static steering torque early in the product design cycle is necessary for steering component sizing as well as to assess the impact changes to wheel end geometry will have on static steering torque. The static steering torque is shown to be the sum of three main components: (1) torque required to slide the tire contact patch across the road surface, (2) torque required to overcome friction in the kingpin joints and steering linkage, and (3) torque generated by vertical force moments about the kingpin axis. Each of these components is discussed in detail, with particular attention paid to examining how their magnitude varies with steer angle. Correlation between predicted and measured static steering torque on an actual vehicle is presented for model validation. Also presented is a dimensional analysis of the static steering torque model and an example solution using dimensional analysis.

INTRODUCTION

This paper presents an approach to predicting the maximum static steering torque required for a vehicle. Static steering torque is defined here as the torque required from the steering system to turn the wheels during a parking maneuver. The maximum value of the static steering torque will occur when the vehicle's brakes are applied, which is the case considered here.

The ability to predict maximum static steering torque required for a vehicle is important for several reasons. First, it is useful for accurate design of power steering components. Some fundamental considerations involved include the piston diameter of the recirculating ball gear or rack and pinion, as well as maximum relief pressure in the system.

Second, a key tradeoff in the development of a vehicle involves the balancing of steering wheel torque required during parking maneuvers with that required during handling maneuvers. Typically, the goal is to minimize steering wheel torque during parking maneuvers while providing adequate steering wheel torque during handling maneuvers to give the driver appropriate feedback. This challenge can be better managed by minimizing the output torque required from the steering system during parking maneuvers through optimized wheel end geometry.

The purpose of this paper is to introduce a simple approach to predicting static steering torques early in a vehicle program. In particular, this approach can facilitate the design of wheel end geometry to reduce static steering torque.

SIMULATION APPROACH

In this paper, the suspension is simplified to a kingpin axis about which the wheel is steered. This is shown in Figure 1.

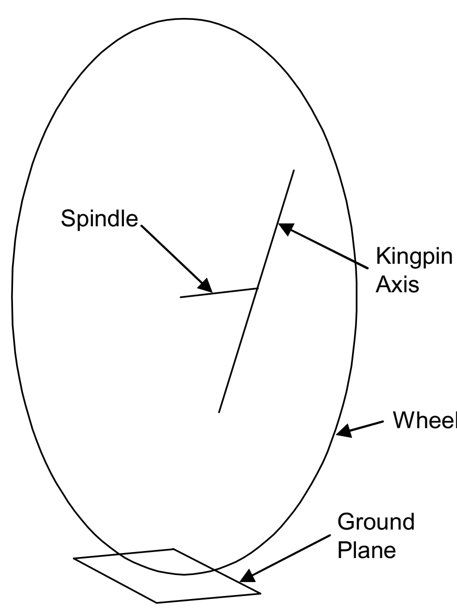

Figure 1—Wheel End Layout

The event typically used to evaluate maximum static steering torque is the dry park maneuver, which has the following procedure:

1. Vehicle in neutral, engine running
2. Brakes fully applied, no forward velocity
3. Slowly steer from straight ahead to full left lock, return to center, steer to full right lock, and return to center

The output of interest from this maneuver is the total steering gear torque or rack and pinion force required to steer the wheels. This total torque is the sum of the following components:

1. Torque required to slide the tire contact patch relative to the ground as the wheel steers about the kingpin axis
2. Torque generated by vertical force moments about the kingpin axis
3. Torque required to overcome friction in the kingpin joints and the steering linkage

The torques from tire patch sliding and vertical force moments vary with steer angle, while the friction torque is nearly constant. Figure 2 illustrates this behavior. For a symmetric vehicle, the vertical force moments from the left and right wheels are equal and opposite at zero steer. However, with steer the vertical force moments will no longer cancel each other out since the vertical force moment arm will vary from side to side. Therefore, there is a net torque due to vertical force moments as steer varies from zero. The torque due to tire patch

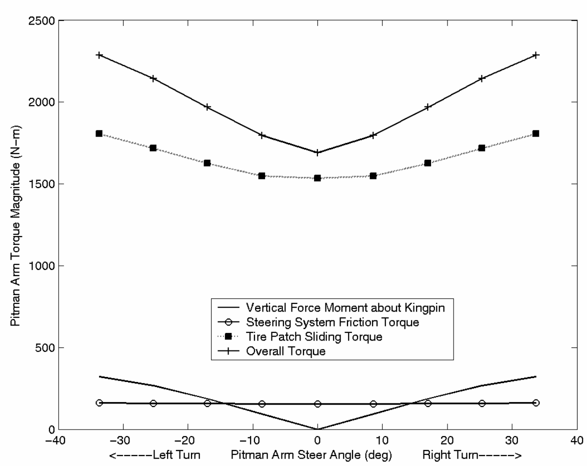

Figure 2—Components of Overall Parking Torque

sliding will also vary with steer angle, again due to the variation of the moment arm with steer.

Although the peak torque in the above plot occurs at the maximum steer angle, this isn't necessarily true for all vehicles. Therefore, the torque should be calculated at multiple steer angles to determine where the maximum value occurs.

The approach used in this paper is to calculate the three components of the total torque at multiple steer angles from zero steer to full left and right lock. Each component is calculated as follows:

Tire Patch Sliding Torque: The tire contact patch is idealized as a rectangle (Figure 3). This rectangle has as its center the geometric center of tire contact, which is found by vertically projecting the wheel center onto the ground plane, adjusting for the camber angle of the wheel relative to the ground. The friction coefficient for sliding between the tire and ground is assumed constant, and a uniform contact patch pressure distribution is assumed. The tire patch is divided into numerous small elements to facilitate calculation. Each element is assumed to rotate about the kingpin to ground intersection. The torque required to rotate the entire tire patch is the sum of the torque required to rotate each element. The torque required to rotate each element ($T_{element}$) is given by the following formula:

$$T_{element} = a\mu Pd$$

where a is the area of the element, μ is the friction coefficient between tire and ground, P is contact pressure, and d is the distance from the element to the kingpin/ground intersection. A numerical integration is performed to sum up the torque of all elements.

For each steer angle, the location of the tire contact patch is recalculated through a vertical projection of the new wheel center position onto the ground plane. This means that d for a given element will vary with steer. Therefore, the numerical integration to calculate total torque is performed at each steer angle.

Vertical Force Moments: For each steer angle, the left and right vertical force moments are calculated using the following approach. The pressure distribution across the tire patch can be idealized as a vertical force acting at the geometric center of tire contact. The position of the geometric center of tire contact is determined, then the vector from it to the kingpin/ground intersection is calculated. This is also illustrated in Figure 3. This vector is the moment arm used to calculate the moment of the vertical force about the kingpin axis.

Steering System Friction: A constant value is used that represents the combined steering linkage and kingpin friction. This value can be found experimentally by steering the vehicle on frictionless plates and measuring the hysteresis in the required steering torque.

The left and right kingpin torques are the sum of these three components. The total output torque required to steer the wheels is the sum of the left and right kingpin torques, modified by the ratio between their rotation and the pitman arm rotation or rack travel.

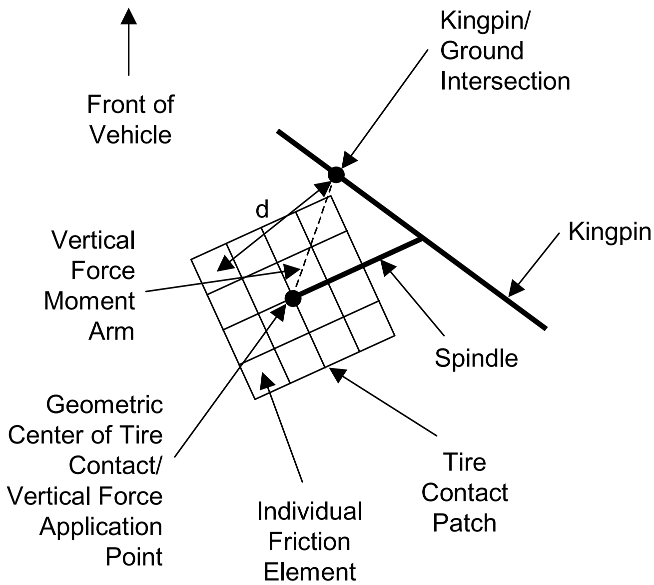

Figure 3—Detailed Top View of Tire Patch (Left Wheel Turned Left)

SIMULATION CORRELATION

To verify the simulation model, a correlation study was run comparing measured dry park data to simulation results. The measured data was from a load acquisition on a light duty pickup truck equipped with a solid axle front suspension and recirculating ball steering gear. The data acquisition provided outputs for pitman arm angle and steering gear output torque at a fully laden condition. Table 1 summarizes some basic characteristics of the measured vehicle.

Characteristic	Value
Front Axle Weight	21,100 N
Tire Patch Dimensions	0.163 m long x 0.180 m wide
Kingpin Friction per Side	51 N-m
Tire-Road Friction Coefficient	0.80

Table 1—Measured Vehicle Characteristics

The simulation was run with the above input data as well as wheel end geometry information. Measured steering gear output torque (pitman arm torque) vs. pitman angle was then compared to the simulation results. This comparison is shown in Figure 4.

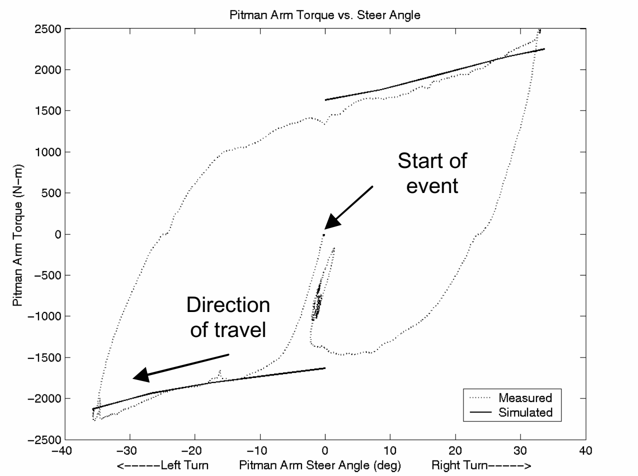

Figure 4—Correlation Between Simulation and Measurement

Correlation between simulated and measured pitman arm torque from center to left lock as well as from center to right lock is shown. The model discussed in this paper will not accurately predict the torque from left-lock to center and right lock to center. This is because the effect of deflections in the steering linkage and tire carcass are not modeled. The linkage and tire will deflect a finite amount before the tire patch actually starts to slide. For example, when the event is first initiated, the plot shows that it takes approximately eight degrees of steer before the underlying load curve is reached. Likewise, when the direction of travel is reversed after left or right lock is reached, a significant amount of steer will take place due to linkage and tire deflection before the tire patch starts sliding in the opposite direction. Since the model in this paper doesn't include linkage and tire deflections, it can't accurately model behavior in the regions of steer reversal.

The measured data shows that the regions of steer reversal will not contain the peak values. Since the simulation is intended primarily to determine peak values, steering linkage and tire carcass deflection effects were not included. This minimizes input data required while still ensuring that peak torques are accurately predicted.

DIMENSIONAL ANALYSIS

To facilitate calculation of the static steering torque, a dimensional analysis was performed. The dimensional analysis approach involves combining the variables associated with a problem into a smaller number of independent dimensionless combinations of variables. Plots of these dimensionless combinations can then be generated to facilitate problem solution. The variables involved with the prediction of static steering torque and their dimensions are:

M_v: moment about kingpin from vertical force (N-m)
M_s: moment about kingpin from tire patch sliding (N-m)
F_z: vertical force acting at tire patch (N)
s: spindle length (m)
r: tire loaded radius (m)
w: tire contact patch width (m)
l: tire contact patch length (m)
θ: kingpin inclination angle KPI (deg)
β: caster angle (deg)
δ: steer angle (deg)
μ: friction coefficient between tire and ground

These are illustrated in Figures 5 and 6. The following dimensionless quantities are found by combining variables:

$M_v/F_z s$, s/r, l/w, $M_s/F_z s\mu$

Since measures of angle are dimensionless, the following variables are already dimensionless:

μ, θ, β, δ

The above set of dimensionless combinations represents the minimum amount of independent quantities necessary to characterize static steering torque during parking maneuvers. Figure 7 shows the results of simulations performed over typical ranges of the dimensionless quantities. By using these results, the need for further simulation can be eliminated, as the following example illustrates.

CALCULATION EXAMPLE

Two potential wheel end geometries are being considered for a vehicle equipped with rack and pinion steering and independent front suspension. The designs have identical geometric characteristics with the exception of the spindle length. Design A has a spindle length of 0.080 m while Design B has a spindle length of 0.120 m. In practice, this difference might be due to different brake packages or suspension architectures. Table 2 summarizes the two designs.

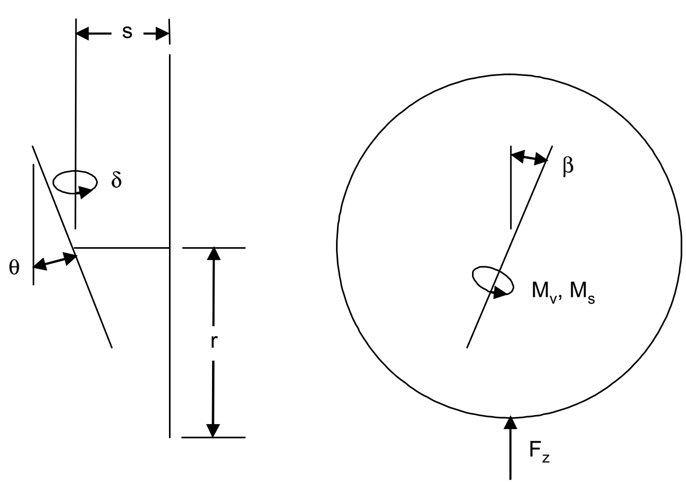

Figure 5—Front and Side View of Left Wheel End

Table 3 shows the steering linkage characteristics for both designs as quantified by inside and outside steer and steer about kingpin as a function of rack travel. The effective moment arm is also included. This is the moment about the kingpin produced by a unit rack force.

Characteristic	Symbol	Units	Design A	Design B
Spindle Length	s	m	0.080	0.120
Tire Loaded Radius	r	m	0.400	0.400
KPI	θ	deg	7.5	7.5
Caster	β	deg	5	5
Tire Patch Length	l	m	0.160	0.160
Tire Patch Width	w	m	0.160	0.160
Tire Vertical Force	F_z	N	5000	5000
Tire-Ground Friction Coeff.	μ	--	0.9	0.9

Table 2—Example Configurations

Rack Travel (m)		0	0.030	0.060	0.090
Steer (deg)	Inside	0	10.1	20.8	32.8
	Outside	0	9.7	19.2	28.8
Kingpin Steer (deg)	Inside	0	10.4	21.6	34.1
	Outside	0	10.0	19.8	29.7
Eff. Moment Arm (m)	Inside	0.169	0.165	0.154	0.137
	Outside	0.169	0.173	0.176	0.172

Table 3—Steering Characteristic for Example

	Design A	Design B
s/r	0.2	0.3
l/w	1.0	1.0
s/w	0.5	0.75

Table 4—Initial Dimensionless Quantities for Example

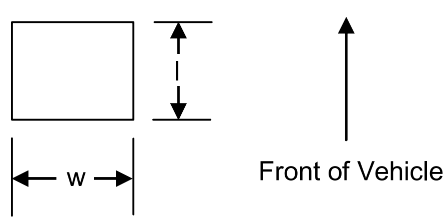

Figure 6—Top View of Tire Patch

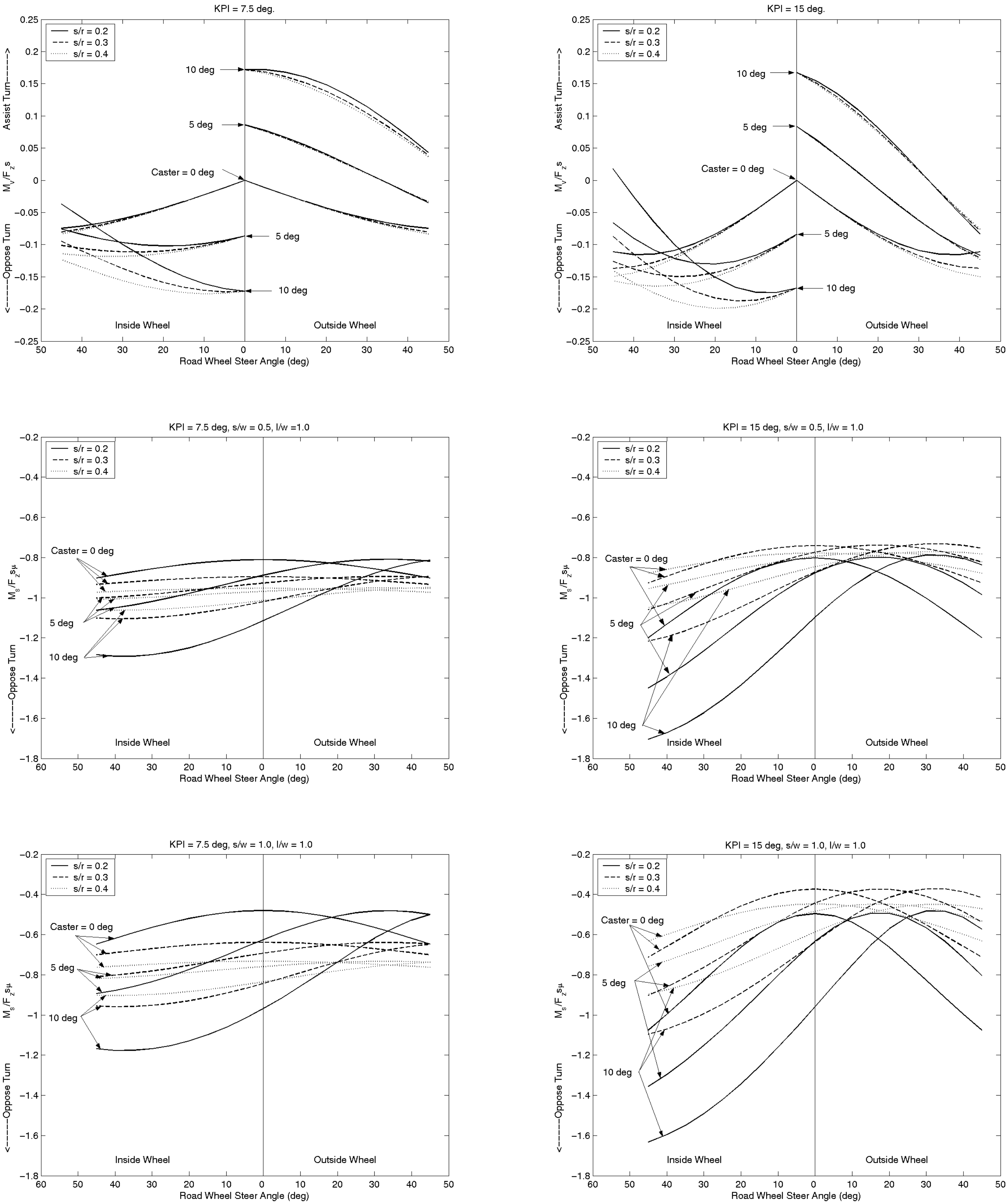

Figure 7—Calculated Dimensionless Characteristics

Rack Travel		0.00		0.030		0.060		0.090	
Design		A	B	A	B	A	B	A	B
$M_v/F_z s$	Inside Wheel	-0.086	-0.086	-0.099	-0.103	-0.103	-0.116	-0.095	-0.109
	Outside Wheel	0.086	0.086	0.065	0.065	0.046	0.046	0.015	0.015
$M_s/F_z s\mu$	Inside Wheel	-0.894	-0.840	-0.941	-0.864	-0.988	-0.887	-1.035	-0.911
	Outside Wheel	-0.894	-0.840	-0.874	-0.824	-0.824	-0.812	-0.812	-0.800

Table 5—Parking Moments for Example

The effective moment arm can be calculated readily given the steer angle about the kingpin axis as a function of rack travel with the following formula:

$a = 57.3 * dz/d\delta_k$

where:

a = effective moment arm
z = rack travel
δ_k = steer angle about kingpin axis in deg

Some dimensionless quantities are functions of the geometry alone and can be calculated directly from the above data for each design. These are shown in Table 4. By using the values from Tables 2-4 and referencing Figure 7, the information in Table 5 is obtained.

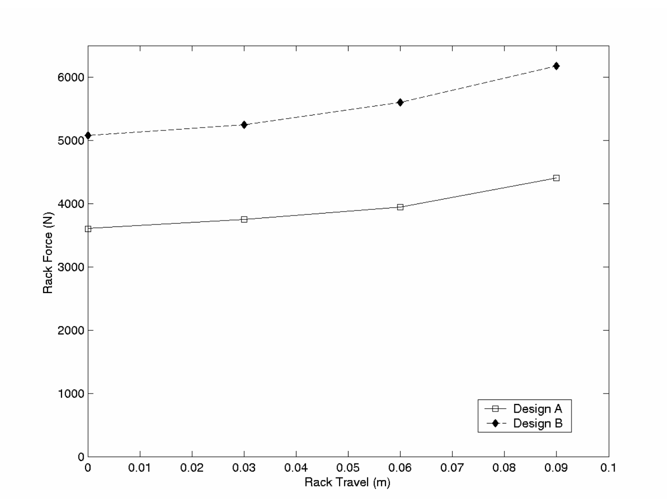

Figure 8—Rack Force vs. Rack Travel for Example

By summing up M_v and M_s and dividing by the effective moment arm for each wheel at the various rack travels, the total rack force vs. rack travel for each design can easily be calculated. The results are shown in Figure 8.

This example did not include moments due to kingpin and steering linkage friction. However, if the friction moment is included, it is given a negative value since it opposes the turn. It is then added to the sum of M_v and M_s for both the inside and outside wheel before dividing by the moment arm.

This example shows the utility of the dimensional analysis approach. The maximum static steering torque can be calculated without any additional simulation. This example also demonstrates the effect changes in spindle length will have on static steering torque.

CONCLUSION

This paper has introduced an approach to simulate static steering torque during parking maneuvers. The torque is shown to be the sum of three components: torque required to slide the tire on the ground, torque due to vertical force moments about the kingpin, and torque due to kingpin and steering linkage friction. The modeling approach for each of these components is discussed in detail. Key elements of the model include recalculation of the tire patch location relative to the kingpin for each steer angle, and numerical integration of the tire contact patch sliding friction.

Correlation results demonstrate the validity of the simulation approach. Dimensional analysis can be used to reduce the number of independent variables so that charts covering typical ranges of the dimensionless quantities can be constructed from simulation results. These charts can then be used to determine the static steering torques directly, eliminating the need for further simulation.