Operator in the Loop Simulation for Farm and Construction Vehicles

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Abstract

Models of construction and farm vehicles which are detailed enough to provide high fidelity over the full range or even a wide range of operation are very complex. Particularly demanding sub-models include tires, tire soil interaction under a wide range of conditions, and the often complex kinematics of vehicle articulations.

To address this challenge, this paper presents a model with as few input parameters as possible consistent with good fidelity while operating at low speeds and low accelerations. The key difference between the formulation presented here and more traditional vehicle dynamics models is the assumption that there is no slippage at the tire/ground interface. Furthermore, we rely on a bicycle model, so lateral load transfer is not included, and a differential cannot be modeled.

The model has been implemented at the Virtual Reality Applications Center (VRAC) at Iowa State University, and is also in use in industry. Features of the simulation setup include the vehicle controls input by way of a buck including steering, throttle, brakes, clutch, and gear shifting. Outputs are the translational coordinates and the heading of the vehicle; these are input to the real time display.

Introduction

Models of construction and farm vehicles which are detailed enough to provide high fidelity over the full range or even a wide range of operation are very complex. Particularly demanding sub-models include tires, tire soil interaction under a wide range of conditions, and the often complex kinematics of vehicle articulations. Furthermore, since these vehicles most often operate at low speeds and can shift on the fly from forward to reverse, low speed modeling of the tire/ground interface can be particularly challenging. Our experience has been that the complexity of these models is so great that the cost of parameter identification can preclude routine development of new models.

To address this challenge, this paper presents a model with as few input parameters as possible consistent with good fidelity while operating at low speeds and low accelerations. The key difference between the formulation presented here and more traditional vehicle dynamics models is the assumption that there is no slippage at the tire/ground interface. Furthermore, we rely on a bicycle model, so lateral load transfer is not included.

The power train model includes a throttle, engine map, clutch, and transmission. There is an additional degree of freedom for engine rotation in the event that user input and ambient conditions such as draw bar pull cause slippage at the clutch. Since there is a no-slip condition at the tire/ground interface, the model does not deal with the considerable complications of the zero-speed singularities associated with traditional tire models.

Subsequent sections will discuss the consequences of the no-slip assumption and provide some details of the engine/power train model. The paper will discuss the real time operation of the model in virtual reality environments at the Virtual Reality Applications Center (VRAC) at the Iowa State University.

The no-slip assumption

The motivation behind the no-slip assumption is the need for a model which is capable enough for the purposes of maneuvering around farms and construction sites, and which minimizes the need for input parameters. The simplicity of the no-slip model, which eliminates several degrees of freedom and the need for any tire parameters, yields the capability to easily create simulations of additional vehicle configurations, including front steer, rear steer, all wheel steer, and articulated vehicles. To provide some context, consider first more traditional models.

Digital simulation of vehicles started in the 1970’s and has undergone continuing refinement since then. Typically in today’s operator in the loop simulations, driver inputs control signals in real time. These controls influence the interaction between the tires and the ground, yielding time varying forces and moments on the tires. Newton’s laws relate

these forces and moments to the accelerations of the vehicle, and these accelerations are integrated to find the state of the vehicle in real time.

Commercially available real time simulators provide enough fidelity to enable good insight into vehicle braking and handling performance across a very wide range of maneuvers, including some maneuvers at high accelerations [4][5]. Passenger car models with this level of fidelity typically require well over a hundred input parameters, and articulated vehicles require many more.

This paper will provide models with applicability at low speeds and low accelerations, for example, for the study of sight lines and routine control of construction vehicles and farm equipment. The simplifying assumptions make it reasonable to use models with very few easy to measure or estimate parameters to model a single vehicle. This idea scales nicely to more complicated vehicles.

The key operational difference between traditional simulations and the simpler models presented here is that traditional tire models compute forces and moments on the tire as a function of the slip of the tire relative to the ground, and integrate those forces and moments to find the state of the vehicle. Lateral forces and aligning moments can also be computed under no-slip conditions. But a more elegant technique is writing the equations describing the total energy of the vehicle, and setting the rate of change of energy of the vehicle equal to the rate of work done on the vehicle by the forces on the tires and the implements. A consequence of approaching the modeling this way is that lateral forces on the tires need not be computed because under the assumed no-slip conditions they do no work. This is a major advantage – the parameters usually used in traditional tire models, cornering stiffness for example, are not needed for the no-slip model. Furthermore, since slip angles are assumed to be zero they need not be computed either. This is a significant advantage at low speed where calculating slip can be challenging [2][3].

The energy formulation does require forces of traction and braking. Again the model is much simpler that its high fidelity counterparts: wheel spin inertia is neglected and braking and traction forces are assumed to be just the torque applied to the wheel divided by the radius of the tire.

The energy formulation

Consider Figure 1, a yaw plane bicycle model of an articulated vehicle. The side forces on the tire are not shown because, under the zero slip angle assumption, they cannot do work.

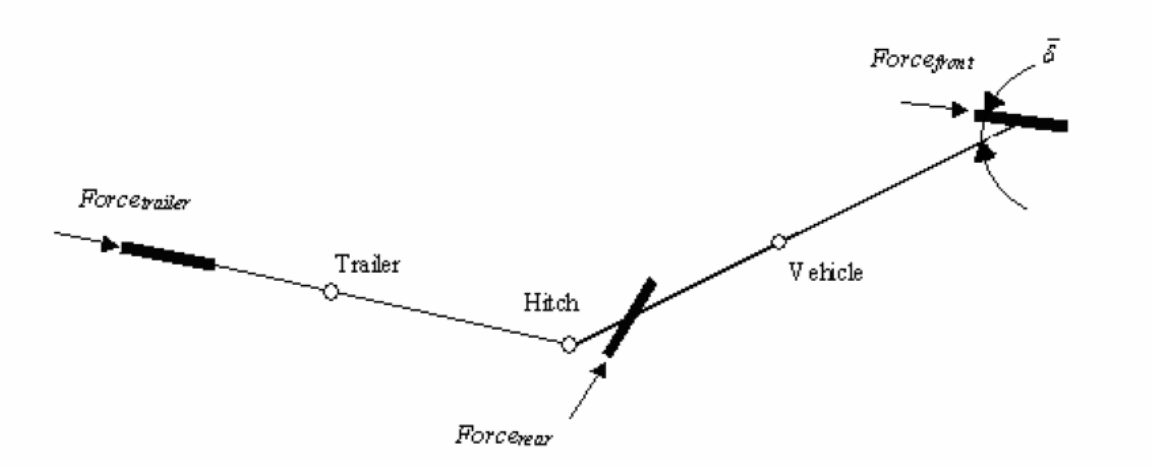

Figure 1: Forces acting on a vehicle and trailer

$$E = \frac{1}{2}M\left(u^2 + v^2\right) + \frac{1}{2}Ir^2 + M_T\left(u_T^2 + v_T^2\right) + \frac{1}{2}I_T r_T^2 \qquad (1)$$

The energy of this model is presented in Equation 1.

Nomenclature is defined in Table 1.

Table 1: Parameters for the Energy Formulation

M	Tow Vehicle Mass
I	Tow Vehicle Yaw Moment of Inertia
M_T	Trailer Mass
I_T	Trailer Yaw Moment of Inertia
U	Tow Vehicle Longitudinal Velocity
V	Tow Vehicle Lateral Velocity
R	Tow Vehicle Yaw Rate
u_T	Trailer Vehicle Longitudinal Velocity
v_T	Trailer Vehicle Lateral Velocity
r_T	Trailer Vehicle Yaw Rate

The derivative of the energy with respect to time, which is equal to the power input by the applied forces, is given by Equation 2.

$$\dot{E} = \sum_{i=1}^{n} Force_i * Speed_i + \dot{E}_{implements} = M\left(u\dot{u} + v\dot{v}\right) + Ir\dot{r} + M_T\left(u_T\dot{u}_T + v\dot{v}_T\right) + I_T r_T \dot{r}_T \qquad (2)$$

Where $Force_i$ are the tractive or braking forces on the tires, and $\dot{E}_{implements}$ signifies the work being done, likely by the terrain, on working implements. In the case where there is a force vector associated with a working implement, the associated $\dot{E}$ would be the dot

product of that force with the velocity vector of the implement at the location of the force.

Baack [2] provides the mathematical details. Note for our purposes here that *u*, *v* and *r* are algebraically related by the no-slip condition and the steer angles, so there is only one independent state variable to be integrated as a function of time to u, v, and r. Similarly, there is only one state variable to be integrated to find the rotation rate of the trailer.

The frequency content of the simulation is a function of the frequency content of the driver controls. This is not particularly demanding – we typically integrate the equations of motion at 100 Hz. Real time calculations are therefore routine.

The next sections provide an explanation of how the engine and brakes deliver torque to the wheels.

The engine and drive train model.

Since the vehicle dynamics is based on the bicycle model shown in Figure 1, we assume a bicycle model for the engine and drive train as well. Figure 2 shows the drive train model. As will be explained below, if the clutch is locked, the torque output of the engine passes directly through to the transmission. If the clutch is slipping, the torque across the clutch is a function of clutch characteristics and driver controls. This torque acts on the transmission and the engine. Note that, although the figure shows one drive wheel, the model is capable of handling multi-wheel drive vehicles. Hummel [1] presents the multi-wheel drive model.

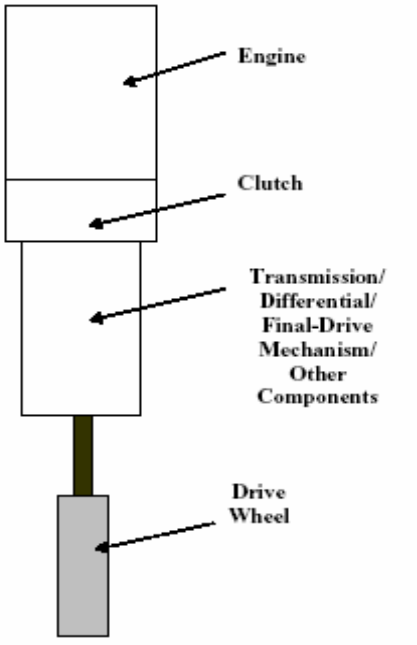

Figure 2: Drive Train Model

The engine model

The engine model computes the torque output by the engine as a function of the engine rotation rate and throttle. This model is based on an engine map typical of construction vehicles and farm vehicles. Table 2 presents the parameters needed to generate the engine map.

Table 2: Engine Map Parameters

ω_{idle}	Engine idle rotation rate	T_{idle}	Engine torque produced at engine idle rotation rate
ω_{peak}	Engine rotation rate where maximum possible torque is produced	T_{peak}	Maximum possible engine torque produced by engine
$\omega_{\max rat}$	Engine maximum rated rotation rate	$T_{\max rat}$	Engine torque produced at engine maximum rated rotation rate

Figure 3 presents a generic engine map for a tractor engine and a combine engine

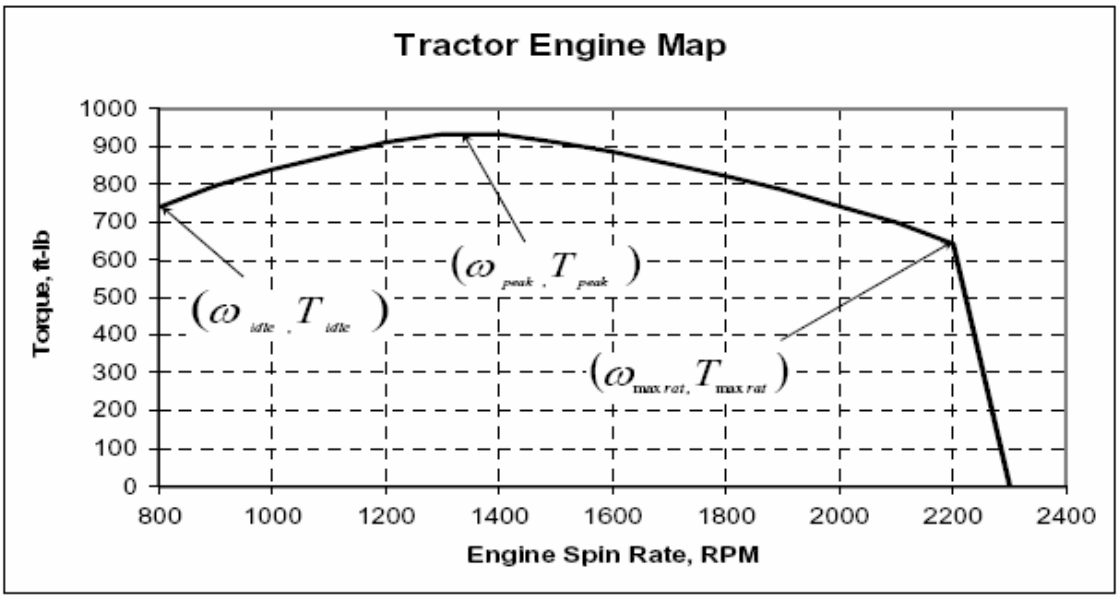

Figure 3: Generic Engine Map

In keeping with our desire to minimize input data, we model the curve between engine idle rotation rate and maximum rated engine rotation rate with a quadratic fit through the three input points, where the independent variable is the engine rotation rate, and the dependent variable is the engine torque. The maximum rated engine rotation rate, maximum rated engine torque and engine idle speed are used to generate the quadratic fit of the engine torque curve given in Eq. 3.

$$T_e = A + B\omega_e + C\omega_e^2 \qquad (3)$$

Figure 4 shows the simulated engine map of the generic tractor engine. We will refer to the line on the right of Figure 4 as the clip line. In this model, the slope of the clip line is set at 7 ft-lb/RPM.

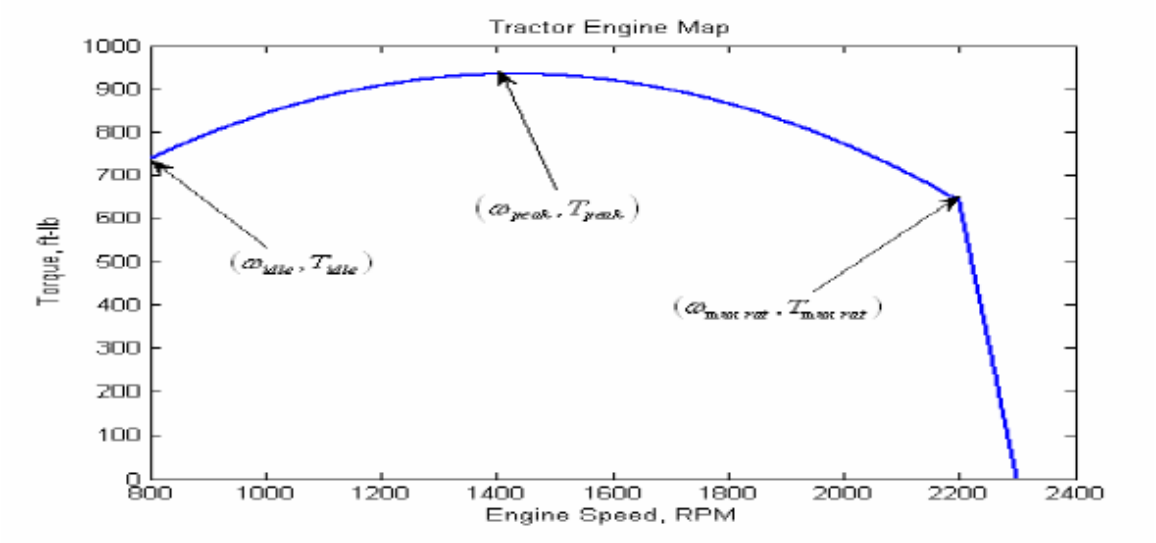

Figure 4: Simulated Tractor Engine Map

Throttle

The limiting engine rotation rate, ω_{th}, sets the maximum speed of the parabolic portion of the torque curve. The limiting engine rotation rate is set by the operator using the throttle control on the buck. The limiting engine rotation rate can be set at any point between ω_{idle} and $\omega_{\max rat}$. The throttle position set at the buck controls is *th*, a number between 0 and 1. Eq. 4 presents the algorithm used for finding ω_{th}.

$$\omega_{th} = th\left(\omega_{\max rat} - \omega_{idle}\right) + \omega_{idle} \qquad (4)$$

The limiting engine rotation rate sets the value where the clip line intersects the parabola. The slope of the clip line remains constant independent of where it is shifted.

In the case shown in Figure 4, *th* = 1.0 and therefore $\omega_{th} = \omega_{idle}$. This is the definition of full throttle. The throttle setting can adjust the clip line to any position between ω_{idle} and $\omega_{\max rat}$.

If the throttle is set at its lowest setting, then *th* = 0.0 resulting in $\omega_{th} = \omega_{idle}$. This is shown in Figure 5 with the solid line plot representing the simulated engine map for *th* = 0.0. The functioning engine map is the solid line plot, in this case a straight line.

For comparison, the full throttle curve has been superimposed onto Figure 3.9 as the dashed line plot.

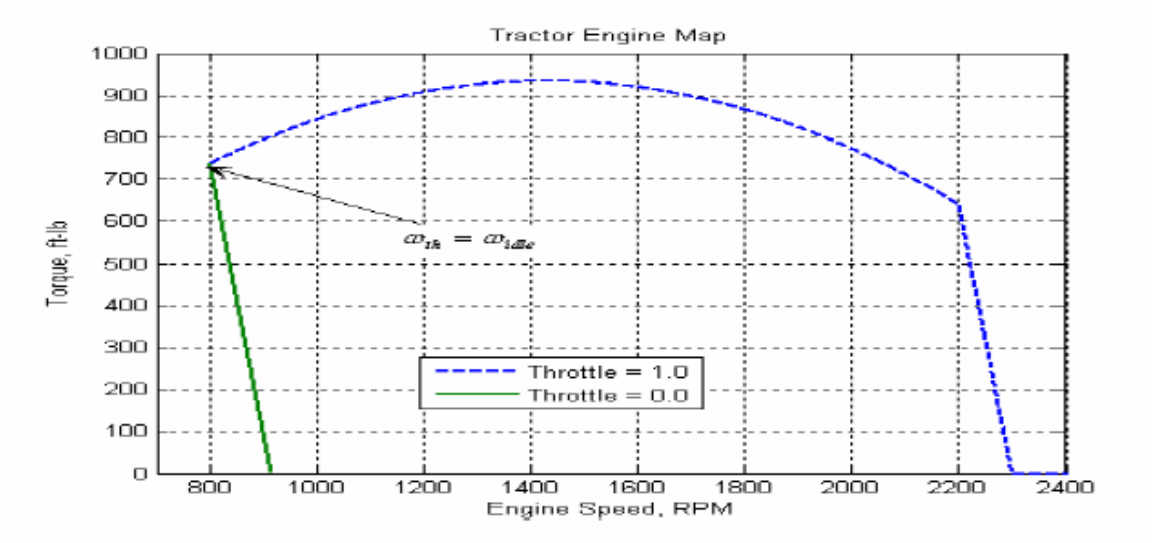

Figure 5: Simulated Tractor Engine Map,

As an additional example, Figure 6 shows the resultant tractor engine map at th = 0.5. The solid line plot is the functioning engine map. The torque curve in Figure 6 can be observed to be clipped halfway between ω_{idle} and $\omega_{\max\ rat}$. For comparison, the full throttle curve has been superimposed onto Figure 36 as a dashed line plot.

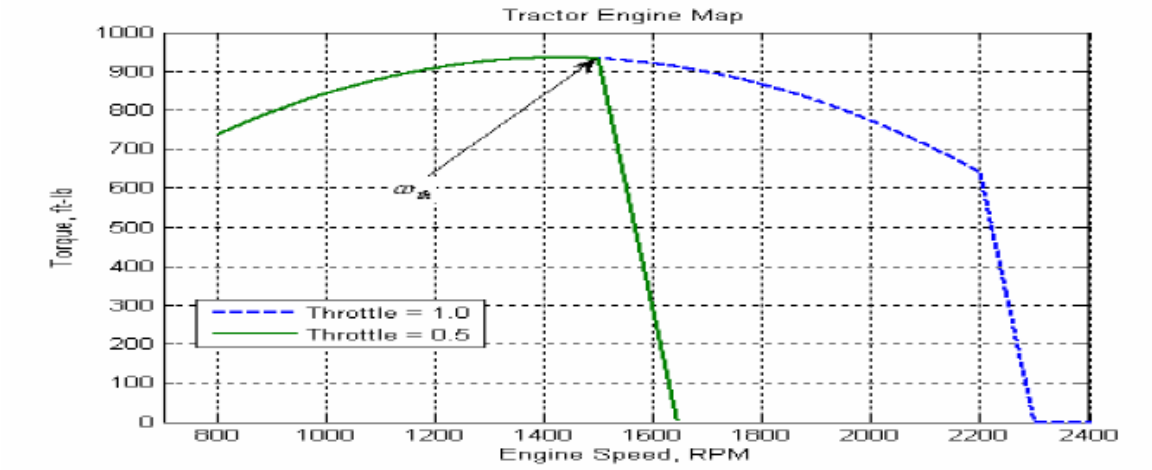

Figure 6: Simulated Tractor Engine Map,

We consider the case where the engine speed is greater than the speed where the clip line intersects zero torque in the next section. This introduces the engine braking torque phase of the engine map

Engine Braking

Farm and construction vehicle engines commonly produce braking torque when their rotation rate is above the limits imposed by the governor. We have included this feature in the model by extending the T_e vs. ω_e relationship below the intercept of the clip line with T_e = 0.

In the spirit of producing the desired effects with a simple engine model, we extend the clip line at half the positive torque clip line slope. The extension is limited by a maximum level of braking torque. The maximum level of braking torque is set equal to half the maximum rated torque.

Figure 7 shows the resultant engine map. The dashed line plot shows the engine braking torque as a function of engine rotation rate.

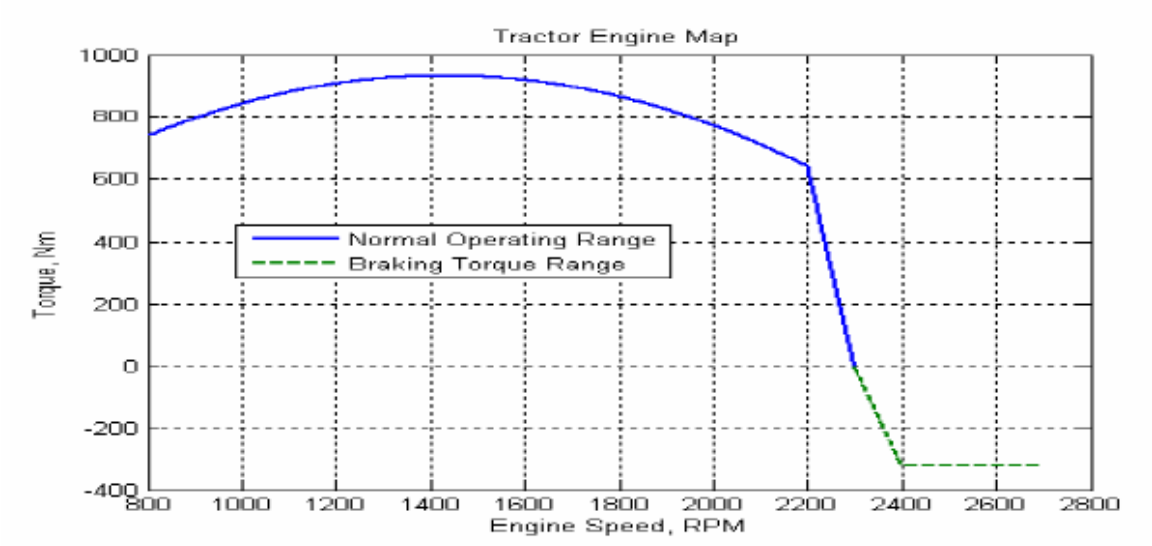

Figure 7: Tractor Engine Map with Engine Braking Torque Line

The Clutch Model

The clutch relates the engine rotation rate to the rotation rate of the transmission input shaft. The clutch model can either be locked up or slipping

In the case of a locked clutch, there is an algebraic relationship between the rotation rate of the engine and the rotation rate of the drive wheel. The drive wheel rotation rate is a direct function of the forward velocity of the tow vehicle mass center and the steer angle. For example, if the drive wheel of the model presented in Figure 1 was the front wheel, the relationship would be as presented in Eq. 5.

$$\omega_w = \frac{u}{\bar{R}\cos(\bar{\delta})} \qquad (5)$$

The speed of the engine is then just the product of the drive wheel rotation rate and the total gear ratio, including both the transmission and the differential, between the wheel and the engine. The torque applied to the drive wheel is the product of the torque output of the engine and this same gear ratio.

In the case of a slipping clutch, the engine rotation rate becomes an additional degree of freedom. The torque passing through the clutch is a function of clutch characteristics and

driver control of the clutch. That torque is applied to both the drive wheel and the engine until the no-slip condition is attained. Details are provided by Hummel[1].

The trade off between fidelity and simple input parameters

The engine/drive train model is fairly capable. Its key shortcoming in comparison with high fidelity models is the bicycle assumption that precludes modeling of a differential. The no-slip condition at the tire road interface provides an important advantage because it makes the procurement of input parameters much simpler. But its consequences are important limits on the range of validity of the model. These consequences are of course maneuver dependent. High acceleration maneuvers, by which we mean that the acceleration level in g's become significant in comparison with the ambient coefficient of friction, typically depend on a significant slip at the tire/ground interface. So the range of validity of the model is limited to accelerations (in g) far less than the ambient friction coefficients. In addition, the no-slip condition implies zero yaw rate and lateral acceleration response time. This is reasonable as the vehicle speed approaches zero, but causes significant errors as speed increases.

From a steady state point of view, the turn radius of curvature of the no-slip model is only a function of steer angle and the wheel base, the steady turn equivalent of a neutral steer vehicle. At low speeds we expect this not to be a serious impediment. From a linear analysis point of view, which seems appropriate in view of the limitation of accelerations to far less than the ambient coefficient of friction,

$$\delta = \frac{L}{R} + K\, a_y \qquad (6)$$

Where δ is the steer angle, L is the wheelbase, K is the understeer gradient and a_y is the lateral acceleration in g's. If the no-slip radius is R_0, then for a constant steer angle

$$\frac{L}{R_0} = \frac{L}{R} + K\frac{u^2}{R\,g} \qquad (7)$$

Which may be written

$$\frac{R}{R_0} = \frac{L + \frac{K u^2}{g}}{L} \qquad (8)$$

The no-slip model will always predict a steady turn radius R_0, whereas a higher fidelity model will call for the radius to change as speed increases. For example, if a vehicle with five degrees per g understeer and a wheelbase of 10 feet (3.05 meters) is modeled with the no-slip assumption in a steady turn at 10 feet/sec (3.05 m/s), the model will predict a turn radius about 3% smaller than the turn radius of a higher fidelity model with the appropriate level of understeer. (Fidelity drops off rapidly with speed; at 30 feet per second, the no-slip radius is about 25 percent smaller than the radius that would be predicted by the higher fidelity model.)

Implementation of the Model

The model has been implemented at the Virtual Reality Applications Center (VRAC) at Iowa State University, and is also in use in industry. Features of the simulation setup include the vehicle controls input by way of a buck including steering, throttle, brakes, clutch, and gear shifting. Figure 8 presents the buck in the virtual environment.
The output of the vehicle dynamics model are the translational coordinates and the heading of the vehicle. These inputs drive the real time display.



Figure 8: Farm Vehicle Simulation

Vehicles that have been modeled with the vehicle dynamics/drive train model presented in this paper include a combine, tractor, front-end loader, and backhoe.

Conclusions and comments

This paper presents a simulation suitable for routine maneuvering of farm and construction vehicles. The simulation is distinguished in its requirement of very few input parameters. These parameters are typically easy to measure or estimate, so it is inexpensive to build vehicle models.

The model is in operation at the ISU's Virtual Reality Applications Center and in industry. Features of the model that are particularly desirable include the ease of obtaining input parameters, a task which is extremely demanding for higher fidelity simulations. The model is suitable for use in low speed and low acceleration environments. The converse is also true – use in high speed and/or high acceleration environments will yield inaccurate results.

References

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