Multibody Dynamic Simulation of Off-Road Vehicles for Load Prediction, Stability, Safety, and Performance

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Introduction
Multibody Dynamic Simulation has been used to successfully simulate a wide variety of agricultural, construction, and military vehicles and predict the safety, mobility, stability, and operating loads of the complete system. The objective is to predict accurate vehicle behavior under any operating condition and provide the basis for making engineering design changes to improve performance.

Vehicles used for agriculture, construction, recreation, and military applications all involve similar modeling requirements and challenges to account for the wide range of operating conditions. Results of the multibody dynamic simulation process produce time-histories of the position, velocity, acceleration, and reaction forces on all parts. These results have been used to characterize the vehicle performance attributes.
The mobility attribute is a function of the tire or track interaction with ground, and the power map of the engine and transmission. The stability attribute is a function of the maneuver and vehicle total center of gravity, as well as the tire or track interaction with ground. The resulting loads that are predicted are used to do subsequent Finite Element Stress analysis as well as durability life prediction.

The results of simulation also provide animations of the vehicles and are used to gain insight into complex dynamic behavior and see how new designs will behave prior to actually having physical hardware to drive and test. Detailed vehicle performance knowledge is obtained from the dynamic simulations and is used to both improve concept designs and trouble-shoot existing vehicles. This paper outlines how tire forces are modeled and applied to simulation models of military, agricultural, and construction vehicle engineering. The simulation models shown were developed using LMS Virtual.Lab Motion® and have been used to predict the transient dynamic performance of vehicles in a variety of operational scenarios.

**Vehicle Dynamics**

Military, agricultural, and construction vehicles experience unusual and dramatic loads in operation. To better predict and quantify those loads, multibody simulation has become an important part of the design and engineering process. Simulation is used to understand these loads and assess the performance of the vehicle even before physical prototypes exist. The component load time histories are a direct output of solving the equations of motion. External applied loads determine of the input boundary conditions that define how the system will behave. The output loads are often used as input to subsequent finite element analysis where component stress is evaluated. A vehicle
rough terrain event shown in figure 2 is a common simulation done to predict loads, and the associated vehicle dynamic results for position, velocity, and acceleration.

Historically, commercial multibody dynamic analysis programs were developed in the late 1970’s and early 1980’s. The simulation software referenced in this paper started as DADS funded by TACOM and developed at the University of Iowa. Prof. Ed Haug and his research team developed new ways to represent, solve, and code the equations of motion for a rigid and/or flexible multibody model. The research led to more numerically stable and accurate results for handling problems involving high accelerations, high frequency degrees of freedom, and long simulation time. The software has been widely used by TARDEC, the US Army, and many military construction, and agricultural manufacturers around the world since the mid 1980’s. The current implementation has evolved from DADS into Virtual.Lab Motion ® from LMS and it is part of a comprehensive mechanical simulation environment that compliments the multibody dynamic simulation with embedded solid modeling, durability, and optimization features.

**Tire Forces**

The vehicle dynamic response to steering, braking/acceleration, and moving over terrain is dependent on the tires, suspension, and mass distribution. Agricultural, construction, military and other off-road vehicles experience extreme operating conditions as a result of both rough roads and fields, and large awkward payloads.

For vehicles operating on hard surfaces, the tire characteristics are important and for large Ag and construction tires the carcass behavior comes into play. The primary tire forces that act at the tire/road interface include lateral, longitudinal, and vertical components.

The response of a rolling tire on pavement consists of three components of force and three components of torque in the normal, longitudinal, and lateral directions. In general, these reactions are not independent, as the tangential reactions at the tire/ground
interface are dependent upon the normal force, which is assumed to be a function of deformation and deformation rate of the tire.

The torque acting on the tire consists of the aligning and overturning moments as well as the torque about the spindle axis due to tractive forces. In the lateral and longitudinal directions, empirical relations are used to relate the force in these directions to the normal force and kinematic definitions of steering and rotational slip, respectively. These two kinematic effects are not independent because their resultant is the net force acting tangent to the ground surface on the tire. The friction ellipse concept is used to limit the magnitude of the net tangential force.

**Normal Force and the Terrain Tangent Plane**

Tire normal force is calculated based on the normal deflection and velocity and is expressed as

$$F_{\text{norm}} = F_s - F_D$$

where

- $F_{\text{norm}}$: Force acting in the normal direction
- $F_s$: Stiffness Force due to normal deflection
- $F_D$: Damping Force due to normal velocity

The stiffness force due to normal deflection can be defined by one of the following:

- $$F_s = k \cdot d_{\text{norm}}$$
- $$F_s = \text{fun}(d_{\text{norm}})$$

where,

- $k$: Vertical Stiffness Coefficient
- $d_{\text{norm}}$: Normal deflection
- $\text{fun}(d_{\text{norm}})$: Vertical stiffness force as a function of normal deflection (Curve Vertical)

The damping force due to normal velocity is

$$F_D = C_D \cdot V_{\text{norm}}$$

where

- $C_D$: Vertical Damping Constant
- $V_{\text{norm}}$: The rate of change of normal deflection or normal velocity

The terrain tangent plane is defined to be the plane tangent to the terrain profile at the point of contact between tire and terrain. The longitudinal and lateral forces are computed in this plane, and are assumed to act in this plane. The terrain tangent plane
coordinate system is defined by these rules. The Z-axis of the terrain tangent plane coordinate system is normal to the tangent plane, directed upwards. The X-axis is the intersection of the terrain tangent plane and the plane of the tire disk. The Y-axis is in the terrain tangent plane, perpendicular to the X-axis, directed so that a right-handed coordinate system results.

![Figure 3. Tire axis system](image)

**Tangent Plane: Longitudinal Force**

The longitudinal force is computed based on rotational slip in the terrain tangent plane and is assumed to act in this plane. In the longitudinal direction, there are two effects, rolling resistance and traction/braking forces.

\[
F_{\text{long}} = F_\tau + F_{TB}
\]

where

- \(F_\tau\) Rolling resistance force
- \(F_{TB}\) Force due to traction/braking

Rolling resistance represents parasitic longitudinal force due to carcass deformation losses, bearing friction, etc., as a fraction of normal force.

\[
F_\tau = -C_\tau \cdot F_{\text{norm}} \cdot \text{sign} (\langle V_c \rangle_{\text{long}})
\]

where

- \(C_\tau\) Coefficient of rolling resistance

\(\langle V_c \rangle_{\text{long}}\) The forward velocity of the wheel center obtained from the model state.

Traction/braking force can be modeled when the wheel rotational inertia is included (Type Full). If the rotational inertia is not included (Type Basic and Intermediate) the traction/braking force will be zero. The ratio of the longitudinal force to the normal force (the longitudinal friction coefficient) is measured as a function of rotational slip. Using this relationship the traction/braking force can be found as

\[
F_{TB} = \mu_{\text{long}} \cdot F_{\text{norm}}
\]

where,

- \(\mu_{\text{long}}\) Longitudinal force coefficient measured as a function of rotational slip

The longitudinal friction coefficient is a piece-wise linear function of slip, based upon the nominal friction coefficient, and is shown in the following plot.
The rotational slip can be expressed as

\[ S = \frac{V_p}{\left(V_c\right)_{l\times r}} \cdot \text{sign}(V_p) \]

where
- \( S \)  Non-dimensional rotational slip
- \( V_p \)  Velocity of the bottom point of the tire

The velocity of the bottom point of the tire is defined as

\[ V_p = R_d \cdot \omega + \left(V_c\right)_{l\times r} \]

where
- \( R_d \)  Deflected tire radius
- \( \omega \)  The wheel rotational velocity obtained from the tire state.

**Tangent Plane: Lateral Force**

Lateral force is computed as a function of normal force and slip angle analogous to the longitudinal force computation with rotational slip. Lateral force experimental data is typically known as a carpet plot because it varies with both the normal force and slip angle. The lateral force is approximated by a cubic polynomial determined from the following boundary conditions:

When:

\[ \alpha = 0 \quad F_{lat} = 0 \]
\[ \alpha = 0 \quad \frac{dF_{\text{lat}}}{d\alpha} = C_{\alpha} \]

\[ \alpha = \alpha_{u} \quad F_{\text{lat}} = (F_{\text{lat}})_{\text{max}} \]

\[ \alpha = \alpha_{s} \quad \frac{dF_{\text{lat}}}{d\alpha} = 0 \]

where

- \( \alpha \) The slip angle
- \( \alpha_{u} \) The saturated slip angle
- \( F_{\text{lat}} \) Force in the lateral direction
- \( \frac{dF_{\text{lat}}}{d\alpha} \) The slope of the lateral force curve with respect to the slip angle
- \( (F_{\text{lat}})_{\text{max}} \) The maximum lateral side force
- \( C_{\alpha} \) The value known as Cornering Stiffness

This relationship can be seen in the following figure.

Figure 5. Cornering stiffness

Slip angle is defined as the angle between the tire center heading vector and the tire velocity vector projection in the terrain tangent plane (see the figure above). Since the slip angle is always acute, the sign of the slip angle is dependent on the sign of the lateral velocity component of tire center. This definition alleviates the need for logic to account for a change in direction of the tire. Thus, the slip angle is

\[ \alpha = \tan^{-1}\left(\frac{(\mathbf{v}_c)_{\text{lat}}}{(\mathbf{V}_c)_{\text{lat}}\cos\theta}\right) \]

The saturated slip angle is approximated by
\[
\alpha_n = 2.5 \cdot \frac{F_{\text{norm}}}{C_x}
\]

The maximum lateral force is the nominal friction coefficient times the normal force expressed as:

\[(F_{\text{lat}})_{n,\alpha} = \mu F_{\text{norm}}\]

where

\(\mu\) The nominal Friction Coefficient

**Tangent Plane: Friction Limiting**

In both longitudinal and lateral directions, the coefficients of proportionality between the tangential force and the normal force are considered to be functions of a kinematic representation of slip. The forces in each direction are calculated independently. However, it is well known that these two components of force are not necessarily independent, and in fact their resultant is limited by the magnitude of the friction force acting between the tire and road. The possible values for this net force are represented by the friction ellipse, with major axis of length \(\mu_{\text{long}} F_{\text{norm}}\) and minor axis equal to \(\mu_{\text{lat}} F_{\text{norm}}\)

![Friction Ellipse Diagram](image)

Figure 6. Friction Ellipse

For the case where rotational slip is computed, the frictional force applied acts in the direction opposite the velocity vector with a magnitude derived by intersecting the velocity vector with the friction ellipse. The length from the origin to the intersection gives the magnitude of this force. However, this limiting condition is active only for slips large enough that the relationships for \(\mu_{\text{long}}\) and \(\mu_{\text{lat}}\) reflect actual tire-road slippage as opposed to tire carcass stiffness. For this reason, this is a limiting condition and is not generally active for small slips. This also points out the limitations in the applicability of the neglected wheel inertia model. Since the lateral force carpet plots are measured at zero rotational slip, the presence of significant longitudinal forces tends to invalidate the use of this data except where the net vectorial slip is computed and a friction ellipse is used (i.e. case number 2 below). Thus the neglected wheel inertia models should be used for simulations with relatively small longitudinal forces only. To represent the preceding description, the following logic is implemented in the tire subroutine: 1) If the neglected inertia model is used (Type Basic or Intermediate), \(F_{\text{lat}}\) is computed from the cubic approximation, and
\[ \mu_{\text{max}} \cdot F_{\text{norm}} > \sqrt{\frac{F_{\text{long}}^2}{e} + \frac{F_{\text{lat}}^2}{e}} \quad \text{then} \quad F_{\text{lat}} < \sqrt{\left(\mu_{\text{max}} \cdot F_{\text{norm}}\right)^2 - \frac{F_{\text{long}}^2}{e}} \]

where \( \mu_{\text{max}} \) is the maximum lateral force coefficient and it is understood that

\[ F_{\text{long}} < \mu_{\text{max}} \cdot F_{\text{norm}} \]

Also, note that this is effectively a friction circle since the major axis of the friction ellipse (the longitudinal force \( F_{\text{LONG}} \)) cannot be determined without computing rotational slip.

2) If the wheel inertia model is used (Type Full), the longitudinal force is computed from the rotational slip and the lateral force from the steer slip. The following logic then imposes the friction ellipse limitation:

\[ \mu_{\text{max}} \cdot F_{\text{norm}} > \sqrt{e \cdot F_{\text{long}}^2 + F_{\text{lat}}^2} \]

\[ e = \begin{cases} \frac{\mu_{\text{long}}}{\mu_{\text{max}}} & \text{for } \mu_{\text{long}} > \mu_{\text{max}} \\ 1.0 & \text{for } \mu_{\text{long}} < \mu_{\text{max}} \end{cases} \]

then

\[ F_{\text{lat}} = -\mu_{\text{max}} \cdot F_{\text{norm}} \cdot \frac{(V_x)_{\text{lat}}}{|V|} \]

\[ F_{\text{long}} = e \cdot \mu_{\text{max}} \cdot F_{\text{norm}} \cdot \frac{V_y}{|V|} \]

where

\[ |V| = \sqrt{V_x^2 + (V_y)_{\text{lat}}^2} \]

**Verification**

One example of tire force verification comes from simulation of an agricultural vehicle driving over a discrete ditch. These tests are part of a more comprehensive set of experiments where the vehicle is driven over other obstacles like discrete bumps, and a concrete ISO test track. The results show that the simulation and test accelerations at one location on the vehicle match well. The results were found to correlate well for high-pressure settings of the tires, but not as good for very low-pressure tires. At high tire pressures, the equations used to represent the tire stiffness, damping, and friction do a good job characterizing the real physical behavior. Very low-pressure tires behave in a more complex way and the tire carcass nonlinear rubber behavior becomes important. New development effort is underway to add capability to better represent the low-pressure tire case.
Conclusion

Multibody simulation has proven to be a valuable and reliable part of the agricultural, military, and construction vehicle design and engineering process. The extensive validation done on previous projects has lead to confidence in using simulation early in the design process to compliment traditional test and prototype development methods. Simulation also offers many benefits in cases where it is too costly to build prototypes or too dangerous to do physical tests. Military, agricultural, and construction vehicles also experience much more dramatic loads than passenger cars and trucks so the benefits of predictive multibody simulation are even more important. The role of simulation also continues to grow in scope and importance as new product development time becomes even more compressed.

References


Results from Deere PEC Waterloo Iowa

Figure 7 Tractor driving on a discrete ditch simulation