

Tire Model for Turn Slip Properties

Fan Bai, Konghui Guo, and Dang Lu
ASCL of Jilin University

ABSTRACT

In this paper a tire model for describing tire turn slip properties is derived. The tread of the contact patch is divided into many massless elastic elements in both the length and width direction. Carcass deformation is expressed by the translation, bending and twisting function. A turn slip tire model is derived by analyzing the geometric relationships among the deformation of contact patch, tread and carcass. The model is validated by experimental results of parking maneuver. The model seems capable of generating transient and steady state forces and moments for turn slip, and showing varied trend of tire force according to different turn slip velocity. It could not only describe the tread deformation, but also analyze how the tread deformation affects the tire force and moment properties.

CITATION: Bai, F., Guo, K., and Lu, D., "Tire Model for Turn Slip Properties," *SAE Int. J. Commer. Veh.* 6(2):2013, doi: 10.4271/2013-01-2371.

INTRODUCTION

Since tires are the only components contacting directly with the road surface for the vehicle, the description of tire behavior is very important for road vehicles dynamics analysis. Many studies have been carried out on tires to study the relationship between tire mechanical properties and the well-known and commonly used slip quantities which are called the lateral slip (slip angle) and longitudinal slip (slip ratio). Besides, there is a third slip input quantity: the spin. Turn slip is one possible component of spin and the other possible component is camber [1]. Tire turn slip properties play an important role in parking application and shimmy, and various tire models have been proposed to describe it. By analyzing the geometrical relations between different tire lateral inputs, Paper [2,3,4] established a theoretical model of non-steady state tire turn slip properties with small lateral inputs based on the non-steady state tire cornering model. Paper [5] derived a theoretical model of tire turn slip properties which could calculate tire force with large turn slip. And the effects of turn slip were also considered in various semi-empirical models [5,6,7,8].

In this paper a theoretical tire model is derived to better explain tire mechanical properties under turn slip, extend application range of tire model and develop a more accurate and compact semi-empirical model. First, the geometric relationships among deformations of contact patch, tread and carcass of tire under turn slip are discussed. Then a solution

calculating tire force and moment under sliding is proposed. Finally the calculated results of this model are compared with the experimental results, and the tread deformations with different turn slip ratios are analyzed.

THE DISCRETE MODEL OF TURN SLIP

The physical tire model for turn slip properties is established in this section, which can express tire force and moment under different turn slip ratios.

DEFINITIONS OF TURN SLIP AND MODEL COORDINATE SYSTEM

The condition of pure turn slip, while the sideslip angle α and longitudinal slip ratio S_x [6] remain at zero, is shown in Figure 1, where R is the radius of path curvature and Ω is the wheel speed of revolution. The turn slip ratio ϕ_t is defined as the ratio of the wheel yaw rate $\dot{\psi}$ and the updating speed V_r of the contact path, and V_r is defined as the product of the wheel speed of revolution Ω and the effective rolling radius R_e . The ϕ_t reflects variation of tire yaw angle per unit contact patch updating length. If side slip angle $\alpha=0$ and longitudinal slip ratio $S_x=0$, the value of ϕ_t is equal to the path curvature:

deformation of carcass. The angle between X_c axis and X_h axis is the carcass twisting angle θ . The ground projections of coordinate system $X_hO_hY_h$ and $X_cO_cY_c$ would coincide when the tire is rolling without any slip quantity. The positive directions of tire forces in the model are also shown in Figure 3.

DEFORMATION OF TIRE

The contact patch of the tire model is assumed to be rectangular in shape, and the tread in the contact patch is divided into numerous rectangle grids whose number of rows and columns are denoted as n and m respectively, see Figure 5.

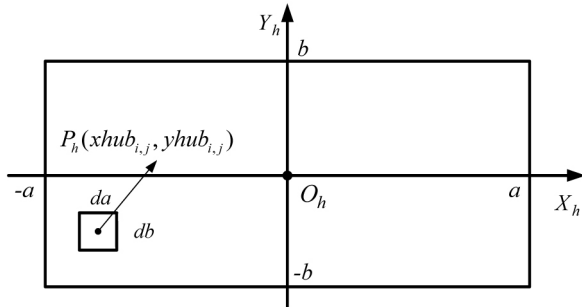


Figure 5. Contact patch discretization

Where a and b are the half length and half width of contact patch, and the length and width of grids are da and db . In the model the coordinate of the rectangle grid is substituted by the coordinate of the center of the contact path discrete element. P_h is a point on the hub corresponding to the grid in row i and column j , and its coordinate $(xhub_{i,j}, yhub_{i,j})$ in the $X_hO_hY_h$ could be expressed as:

$$\begin{cases} xhub_{i,j} = -a + \frac{2i-1}{2} \cdot da & i = 1, 2, \dots, m \quad j = 1, 2, \dots, n \\ yhub_{i,j} = -b + \frac{2j-1}{2} \cdot db & i = 1, 2, \dots, m \quad j = 1, 2, \dots, n \end{cases} \quad (4)$$

The deformations of carcass and tread in the contact patch for a certain slip quantity are shown in Figure 6. Point P_c and P_t are the corresponding points to the P_h respectively on the carcass and tread. If the model had no slip input quantity, their coordinates in the $X_hO_hY_h$ would coincide with the point P_h .

According to the geometric relationship between point P_c and P_h , the P_c coordinate $(xc_{i,j}, yc_{i,j})$ in the $X_hO_hY_h$ coordinate systems could be expressed as:

$$\begin{bmatrix} xc_{i,j} \\ yc_{i,j} \end{bmatrix} = \begin{bmatrix} \cos(\theta) & -\sin(\theta) \\ \sin(\theta) & \cos(\theta) \end{bmatrix} \times \begin{bmatrix} xhub_{i,j} \\ yhub_{i,j} + ycb_{i,j}(xhub_{i,j}, yhub_{i,j}) \end{bmatrix} + \begin{bmatrix} x_{c0} \\ y_{c0} \end{bmatrix} \quad (5)$$

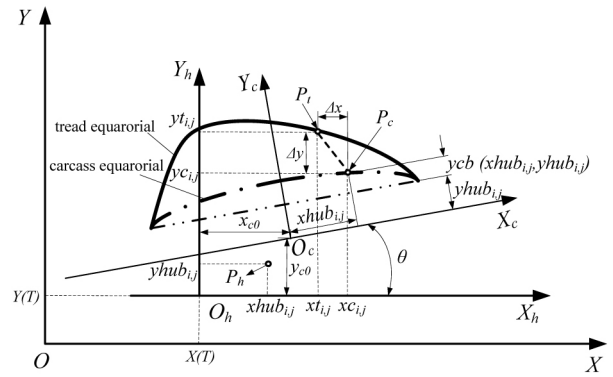


Figure 6. Deformations of carcass and tread

If the coordinate of P_t is $(xt_{i,j}, yt_{i,j})$ in the $X_hO_hY_h$, the deformation of tread grid corresponding to P_h along the X_h and Y_h axis could be expressed as:

$$\begin{cases} \Delta x_{i,j} = xt_{i,j} - xc_{i,j} \\ \Delta y_{i,j} = yt_{i,j} - yc_{i,j} \end{cases} \quad (6)$$

Then the tire force and aligning torque could be calculated as:

$$\begin{cases} F_x = \sum_{i=1}^m \sum_{j=1}^n k_{tx} \cdot \Delta x_{i,j} \cdot da \cdot db \\ F_y = \sum_{i=1}^m \sum_{j=1}^n k_{ty} \cdot \Delta y_{i,j} \cdot da \cdot db \\ M_z = \sum_{i=1}^m \sum_{j=1}^n k_{ty} \cdot \Delta y_{i,j} \cdot xc_{i,j} \cdot da \cdot db - \sum_{i=1}^m \sum_{j=1}^n k_{tx} \cdot \Delta x_{i,j} \cdot yc_{i,j} \cdot da \cdot db \end{cases} \quad (7)$$

where the k_{tx} and k_{ty} are the longitudinal and lateral stiffness of tread in per unit area. Considering the carcass twisting angle θ is very small, $\cos(\theta) \approx 1$ and $\sin(\theta) \approx \theta$. And based on the Equations (2), (3), (4), (5), (6), the longitudinal force F_x and lateral force F_y could be derived as:

$$\begin{cases} F_x = \frac{\sum_{i=1}^m \sum_{j=1}^n (k_{tx} \cdot xt_{i,j} \cdot da \cdot db)}{1 + \varepsilon_x} \\ F_y = \frac{\sum_{i=1}^m \sum_{j=1}^n (k_{ty} \cdot yt_{i,j} \cdot da \cdot db)}{1 + \varepsilon_y + \varepsilon_b} \end{cases} \quad (8)$$

with