Computational Tire Models and their Effectiveness

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Computational Tire Models and their Effectiveness

An Undergraduate Honors College Thesis

in the

Department of Mechanical Engineering
College of Engineering
University of Arkansas
Fayetteville, AR

by

Andrew Ryan Wheeler

April 23, 2013
Computational Tire Models and their Effectiveness
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April 23, 2013

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1. Abstract

This paper describes the advantages, disadvantages, and complications that arise during the modeling, simulation, and analysis of tire models using MSC’s Adams/Car software. Due to the complexity of the testing, this paper was limited to a handful of tests. This included putting a template Formula SAE vehicle through simulations on a constant radius skidpad, performing a high stress longitudinal acceleration, a fish-hook maneuver, and a step-steer maneuver. These tests were analyzed and compared to current understandings of the model’s accuracy and validity. Of the six different models tested, the FTire model proved to be the best performing. The main Pacejka model (PAC2002) was found to be the second most effective. This contradicts current claims made by MSC that state PAC2002 is the foremost model.
2. Introduction to Simulation Software and Tire Models

In recent years, there has been a tremendous push towards the simulation of complex systems. This, coupled with the growing desire by automotive manufacturers to push the limits, has created an industry devoted specifically to automotive dynamic behavior. In this industry, tires play a large role in the actions of the vehicle. As such, the accurate modeling of said tires is critically important in obtaining accurate results. With the amount of varying models out there, it is typically a difficult decision. This paper helps clarify the strengths and weaknesses of the major models.

The software used to conduct these tests is Adams. Adams is the leading multibody dynamics simulation software. There is a module within the software called Adams/Car which specifically handles vehicle dynamics. Within Adams/Car there are specific protocols that handle tire solutions. These protocols utilize various models that have been created, or at least incorporated into the software, by the software’s manufacturer, MSC.

This software can be obtained from MSC’s website (http://www.mscsoftware.com/) in both student and professional forms. The student version can be obtained for free (with software limitations) after student standing has been confirmed, or the full version can be obtained at sizable cost, but there are discounts for professors.

It is also necessary to have a computer capable of running the software. Exact requirements can be found on MSC’s website. The computer utilized during these tests was a custom made workstation specializing in demanding processing. It features 3 terabytes of hard drives, 16 gigabytes of RAM, and a 3.7 GHz quad core processor.

The vehicle used was a template provided by MSC software employees. It was not created under the direction of MSC but more as a side project to help students who participate in Formula SAE to get
started using Adams/Car in a reasonable amount of time. The link to the FSAE website can be found in the References.

The tire models used by Adams described herein include the PAC2002, PACTIME, PAC ’89, PAC ’94, 521, UA-Gim, and FTire. The first 4 of which use the same basic approach but with slight variances. The 521 and UA-Gim models use a relatively more simplistic approach, and the FTire model utilizes a completely different approach than any of these. The PAC models all work on the basic premise of research that was developed using what’s called the “Magic Formula.” This is basically a curve fit that can be used to solve for things such as the forces and moments acting on the tire. The forces and moments are the most important aspects that need to be modeled due to their impact on the vehicle performance.

Since these models vary significantly in their performance and applicability, MSC has provided a reference table designed to help decide which model to use. This can be seen below in Figure 2.1.

<table>
<thead>
<tr>
<th>Handling Event / Maneuver</th>
<th>PAC2002</th>
<th>PACTIME</th>
<th>PAC89</th>
<th>PAC94</th>
<th>FTire</th>
<th>521.1</th>
<th>UA Tire</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stand still and start</td>
<td>+</td>
<td>0/+</td>
<td>0/+</td>
<td>0/+</td>
<td>+</td>
<td>0/+</td>
<td>0/+</td>
</tr>
<tr>
<td>Parking (standing steering effort)</td>
<td>+</td>
<td>-</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Standing on tilt table</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>Steady state cornering</td>
<td>+</td>
<td>0/+</td>
<td>+</td>
<td>0/+</td>
<td>+</td>
<td>0</td>
<td>0/+</td>
</tr>
<tr>
<td>Lane change</td>
<td>+</td>
<td>0/+</td>
<td>+</td>
<td>0/+</td>
<td>+</td>
<td>0</td>
<td>0/+</td>
</tr>
<tr>
<td>ABS braking distance</td>
<td>+</td>
<td>0/+</td>
<td>0/+</td>
<td>0/+</td>
<td>+</td>
<td>0</td>
<td>0/+</td>
</tr>
<tr>
<td>Braking/power-off in a turn</td>
<td>+</td>
<td>+</td>
<td>0</td>
<td>0</td>
<td>0/+</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Vehicle Roll-over</td>
<td>+</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>+</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>On-line scaling tire properties</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

| Chassis Control                   |         |         |       |       |       |       |         |
| ABS braking control               | 0/+     | 0       | 0     | 0     | +     | 0     | 0       |
| Shimmy                              | 0/+     | 0       | 0     | 0     | +     | 0     | 0       |
| Steering system vibrations        | 0/+     | 0       | 0     | 0     | +     | 0     | 0       |
| Real-time                          | +       | -       | -     | -     | -     | -     | -       |
| Chassis control systems > 8 Hz    | 0/+     | -       | -     | -     | +     | -     | -       |
| Chassis control with ride         | -       | -       | -     | -     | +     | -     | -       |

- Not possible/Not realistic
- Possible
- 0/Better
- +/Best to use

1 wavelength road obstacles > tire diameter
2 wheel yawing vibration due to suspension flexibility and tire dynamic response

tire models assumed to be used in transient and combined slip mode

Figure 2.1 Original Reference Tire Guide
One complication that arose during the initial stages of this research was how to determine the validity of the tests. Because there is no true result that can be obtained via computation, the approach taken to determine the effectiveness of each model was to compare each individual result with the mean of all of the results. Greater deviation from the mean would therefore imply a less effective model for that given test.

Several complications were encountered during the tests. The software used, Adams/Car by MSC Software, proved to be somewhat temperamental. Sometimes simulations would run perfectly fine. The same simulation would try to be run again with no changes and an error would occur. Sometimes, certain models refused to run on certain courses. This is not due to incompatibility with the models and road, but with software inconsistencies. Examples of this include the UA-Gim tire model not running properly on the skidpad test, but being the only one to provide realistic results on the fish-hook maneuver.
3. PAC2002 Tire Model

The PAC2002 Tire Model is the industry standard when it comes to computational tire/force interaction. It is based off a book by a renowned vehicle system dynamics and tire dynamics Professor emeritus at Deft University of Technology in Deft, Netherlands named Hans Pacejka. Over the past two decades, Pacejka has been designing tire models that have little to no physical basis or structure of the formulas chosen. Because of this, they are commonly referred to as the “magic formula.” During the solving process, each tire is characterized by 15-20 different coefficients that represent different forces exerted on the tire’s contact patch [1]. Most of these can be seen below in Figure 2.1.

Generally speaking, a Magic Formula (MF) tire model describes the tire behavior for roads with surface roughness of up to eight Hz [2]. This characterizes most roads and makes the model applicable for situations including:

- Cornering at steady-state
- Single-lane change
- Braking (including split-mu and ABS)
- Most turning maneuvers
- Other common maneuvers on applicable roads

![Figure 2.1 Basic notation for the road/wheel interaction. Directions shown are positive](image-url)
For vehicles with camber angles ($\gamma$) less than 15°, the PAC2002 tire model has also proven to be valid [1].

In pure slip conditions (cornering with a free rolling tire), both the longitudinal force, $F_x$, as a function of longitudinal slip, $\kappa$, and the lateral force, $F_y$, as a function of the lateral slip, $\alpha$, have a similar sine-arctangent shape that can be represented by the general equation

$$Y(x) = D\sin\{C\arctan[Bx - E(Bx - \arctan(Bx))]\}$$

(1)

where $B, C, D,$ and $E$ are constants obtained from curve fittings and $Y(x)$ is either of the aforementioned forces with their respective independent variable[1]. The characteristic curves for these can be seen in Figure 2.2.

Since this is by no means a regularly seen curve, a little more in-depth look at it is necessary. The coefficients in the MF each affect the curve in different ways. The way the curve changes will directly change the force or moment being solved for. Figure 2.3 illustrates how these changes interact.
Equation (1) is the standard form of the Magic Formula. It can be applied to more than just the
aforementioned forces. Its other primary use is that of solving for the moments acting on the tire in all 3
directions.

The movements of the vehicle are a direct result of the road forces on the tires. These forces are directly
dependent on not only the tire’s properties, but also the road’s properties. In this model, the tire is
assumed to act as a parallel spring with damper (both linear) in the radial direction with a single point of
contact.

The inputs consist of the vertical load on the wheel \( F_z \), the longitudinal slip \( \kappa \), the lateral slip \( \alpha \), and
the previously mentioned camber angle \( \gamma \). (It should be noted that even though an input of radial
deflection, \( \rho \), is used by Pacejka, Adams does not list it as an input variable [1], [2]). The possible output
variables include the forces \( F_x \) and \( F_y \), and the moments \( M_x \), \( M_y \), and \( M_z \). To calculate these, PAC2002
utilizes a set of derived parameters acquired from testing data. On the programming side of things, the computational process typically used by Adams can be seen in Figure 2.2.

![Figure 2.4 Computation of tire forces and moments [1]](image)

In order to understand the MF, it is necessary to understand the basics of some of its inputs. One of these defines the tire slip quantity in the lateral direction, $\alpha$, and the longitudinal slip, $\kappa$. These are determined using the velocity of the contact point. As seen in Figure 2.4, the velocity of the contact patch can be broken up into several components. These are the longitudinal speed, $V_x$, the longitudinal slip speed, $V_{sx}$, and the lateral slip speed, $V_{sy}$.

![Figure 2.5 Slip velocities under lateral and longitudinal acceleration [1]](image)

The rolling and slip velocities, $V_r$ and $V_s$, can then be determined using basic geometry.
The PAC2002 model can be used to define just about every condition of slip. These include steady-state pure slip, steady-state combined slip, transient pure slip, and transient combined slip. These can all be broken up further into the longitudinal and lateral forces of each. Due to the complexity of these systems, they will not be covered in detail. For more information please refer to [3].

One of the main benefits of the transient tire model in PAC2002 is that of being able to predict tire behavior while the tire has zero speed. PAC2002 has both linear and nonlinear transient models. The linear model is valid for up to 8 Hz, whereas the nonlinear model is valid up to 15 Hz [1]. The main difference between the two is that in the linear transient model, the lateral and longitudinal stiffness of the tire while it is stopped depends on the properties of the rolling tire slip stiffness. The nonlinear model utilizes the properties of both the tire carcass itself and the slip stiffness. This produces a more accurate result.

In the rest of this paper, the PAC2002 model will be broken up into two categories. The first is the simple PAC2002. This refers to the PAC2002 model used in its steady state form with combined slip. The other category will be the complex PAC2002. This is utilizing the advanced transient form of the model with combined slip and turn-slip/parking.

4. PAC-TIME Tire Model

The PAC-TIME model is a new version of the PAC2002 model. The only modification made is in the equations for the aligning moment $M_z$ and side force $F_y$. The following is information about this new model, as stated in the paper, A New Tire Model for TIME Measurement Data [4]:

“In 1999 a new method for tire Force and Moment (F&M) testing has been developed by a consortium of European tire and vehicle manufacturers: the TIME procedure. For Vehicle Dynamics studies often a Magic Formula (MF) tire model is used based upon such F&M data. However when calculating MF
parameters for a standard MF model out of the TIME F&M data, several difficulties are observed. These are mainly due to the non-uniform distribution of the data points over the slip angle, camber and load area and the mutual dependency in between the slip angle, camber and load. A new MF model for pure cornering slip conditions has been developed that allows the calculation of the MF parameters despite of the dependency of the three input variables in the F&M data and shows better agreement with the measured F&M data points. From a mathematical point of view the optimization process for deriving MF parameters is better conditioned with the new MF-TIME, resulting in less sensitivity to starting values and better convergence to a global minimum. In addition the MF-TIME has improved extrapolation performance compared to the standard MF models for areas where no F&M data points are available. Next to the use for TIME F&M data, the new model is expected to have interesting prospects for converting ‘on-vehicle’ measured tire data into a robust set of MF parameters.”

While this model is considered by some to be technically better than the PAC2002 model, its adoption into the industry has been slow. This is due primarily to the accuracy already associated with the PAC2002. Because of the almost negligible difference between the two, switching to the new model and learning the new procedures have not been considered economical by most.

5. ’89 and ’94 Pacejka Tire Models

The ‘89 and ’94’ Pacejka models are also in the magic formula family. They are the older versions of the PAC2002 model. While the PAC2002 model has been kept up to date, the ‘89 and ’94 models have been mostly abandoned. They are still applicable and relatively accurate; So much so that they are still included in the Adams/Tire software due to their continued use in the industry.

These models use a different file format. Because of this, companies that created files using these versions are stuck using them. They could move to the PAC2002 or PAC-TIME model, but doing so is
usually not considered worth the time. MSC Software has decided to continue to include these models in their software package to not force people to switch to the newer ones.

The difference in all the models stated thus far are minute. Due to the general nature of this paper, all of the details covered for the PAC2002 model are still true for the PAC-TIME, ’89, and ’94 models. The difference can only be seen once one delves into the fine details in notation, specific formulas, computational processes, etc. Furthermore, the ’89 and ’94 models are so similar that they will be regarded as the same model. The ’89 version will be used in the testing.

6. 521-Tire Model

The 521-Tire Model is one of the more simple models in Adams. It utilizes only a handful of parameters and experimental data. One of the primary benefits of this model is its flexibility. It can solve for the moments and slip forces using two different methods. These methods are the Equation Method and the Interpolation Method. The Equation Method utilizes a set of generalized equations based off research done in the 1990s while the Interpolation Method uses an AKIMA spline to calculate the forces and moments relative to the camber, slip, or vertical load. The basic notation of this can be seen in Figure 6.1.
MSC states that this model has been superseded by newer tire models and recommends the use of these other models for more accurate work. They state that this model is only included for backwards compatibility [1].

7. UA-Gim-Tire Model

The University of Arizona Tire Model, abbreviated UA-Gim tire model, was developed based off the research of Dr. Gwangun Gim. His thesis, *Vehicle Dynamic Simulation with a Comprehensive Model for Pneumatic Tires*, originally published in 1988, prompted MSC to create a computational tire model in their Adams software. The model calculates the forces and moments at the contact point as a function of the tire’s kinematic states. One of the new concepts presented by Dr. Gim was that of the Friction Circle. Seen in Figure 7.1, it limits the total friction achieved by the wheel/ground interface but allows for different values of the longitudinal and lateral friction. For example, if $\mu_y$ is at its limit (greatest braking or acceleration in the longitudinal direction without slippage) then no lateral forces can occur without resulting in slip.
8. FTire Model

The final model to be discussed is the FTire Model. The FTire, or Flexible Ring Tire Model, varies significantly from most other models. No form of the magic formula is used. It relies almost exclusively on analytical means to solve problems using classical mechanical and thermodynamic approaches [8]. Instead of using lengthy preprocessing of the road data, it simply resolves road data as it is defined. This allows the model to be used as a more effective means for analyzing ride comfort and modeling the reactions on harsher terrain. An example of a harsh ride comfort simulation can be seen in Figure 8.1. Also, this model was created and is kept up to date by cosin scientific software. MSC Software incorporates its solving capabilities into Adams.
The approach taken by the FTire model is similar to finite element analysis. The belt of the tire is described as a ring with elasticity and stiffness properties. It is broken up into subsections and given degrees of freedom for movement. They are connected to each other by what can be represented by a spring. This can be seen in Figure 8.2.
These connections are used to represent the tire’s bending stiffness. A similar approach is used for torsion and lateral bending. Because of the complexity of this model’s approach, it takes a significantly longer time to simulate. It is also suggested by the manufacturers to use a minimum of 1000 steps per second for the integrator. This is due to the high resolution of the model and the way it was developed to deal with tire vibration and road irregularities.

9. Computational Model Testing: Skidpad

The first test undertaken was that of Constant Radius Cornering. This is commonly referred to as a Skidpad. With the vehicle inserted into Adams and all its parameters defined, the road could then be defined. An 8m turn radius was used due to the fact that it is a standard value used by most Formula SAE participants. The exact details of the test are shown in figure 8.1.
In laymen’s terms, the test was conducted over a 15 second duration. An initial acceleration of 0.5g and a final acceleration of 1.5g was specified. The acceleration would increase linearly with time over the duration of the maneuver. The vehicle would also stay in a single gear throughout to prevent any jerking movement caused by shifting.
The plot of the lateral acceleration vs. time is shown in Figure 8.2 (a full-page plot is shown in Appendix 3).

As seen in Figure 8.2, all models were in agreement until the lateral acceleration reached approximately \( \frac{3}{4}g \). At this point, the complex PAC2002, PACTIME, and FTire models stayed within 10% of each other at \( 8.9\frac{m}{s^2} \). The simple PAC2002 model converged at only \( 7.8\frac{m}{s^2} \) and the PAC89 model thought it was well over a g. During the test using the 521 tire model, the vehicle lost control at only \( 8.2\frac{m}{s^2} \) due to loss of traction. The UA-Gim tire model refused to run the test due to compatibility issues. It, for some reason, believed this maneuver was not possible.

The significance of this test can be seen in the variance of the results. The complex PAC2002, PACTIME, and FTire models can now be assumed to provide reasonable results when the tire is slowly brought up to its maximum level of grip whereas the PAC89, 521, and simple PAC2002 are known inaccurate and the UA-Gim model is not applicable.
10. Computational Model Testing: Longitudinal Acceleration

This simulation proved to be one of the more difficult ones to run. It was designed to push the vehicle to the absolute limits. Because of this, both the complex PAC2002 and the PAC89 would not solve for the entire acceleration period. They would provide decent results up until 33 and 27 seconds, respectively. At this point the solution would diverge and not be able to recover.

The test setup can be seen in Figure 9.1. The duration of the test was set to 50 seconds with 500 points of interest at which to solve. A 5km/hr velocity was assigned to the vehicle at the beginning of the test. The throttle was controlled in a linear fashion in which for the first 15 seconds of the experiment, the throttle would linearly increase until full throttle was reached. At which point the throttle position would immediately return to zero. All of this test was done with the vehicle starting in first gear and automatically shifting once the redline was reached.
The plot of the velocity of the vehicle over the testing period is shown in Figure 9.2 (a full-page plot is shown in Appendix 3). The peaks in each curve are due to the shifting of the vehicle.
This graph shows some significant aspects about the tested models. As seen above, there is quite a bit of variance in the results. Therefore, it is not possible to accurately determine the real solution. What is possible is to use the line's smoothness to characterize its accuracy. The reason for this is due to the fact that it is known that unless the tires lose traction, the lines should remain relatively smooth (this is also known by the smooth torque curve of the motor).

Based off this knowledge, the best performing model was actually the outdated 521. Its smooth parabolic curve is exactly what one would expect to see. But, it could be argued that this is only due to the simplicity of the model. The former would seem to have the most validity due to the vehicle only having to utilize 3 of its 6 gears. Most of the other models use all 6 gears i.e. there are 5 peaks.

It is also important to look at which model was able to make the vehicle accelerate the quickest. The 521 model took the lead at the beginning and then again at the end while several other models had a greater
midrange acceleration (complex PAC2002, then PAC89, then briefly PAC2002). This can be observed directly by the slopes of the lines and indirectly by the line being closest to the top of the graph at any given time.

11. Computational Model Testing: Fish-Hook Maneuver

The next test performed was that of the vehicle during a fish-hook maneuver. It is called a “fish-hook” because the path that is ultimately taken by the vehicle resembles just that. This maneuver consists of turning slightly to the right and then quickly back to the left which will cause the vehicle to oversteer in that direction and spin out. The significance of this test is that it shows how the tire models cope with sudden motions. Unfortunately, this test proved to be too much for all the models except for one: the University of Arizona model. The exact reasons for this are extremely difficult to pinpoint, but it was thought that the primary cause was that of inaccurate vehicle modeling. This model was not designed with the expectations of odd-ball maneuvers such as this and several parts of the suspension reached their breaking points. But this test is still included to further the knowledge of the reader and emphasize the capabilities of the UA-Gim tire model.

The parameters of the test are shown in Figure 10.1. Every 0.01 seconds the computer would solve for the required parameters. The vehicle was given an initial velocity of 150 km/hr in 6th gear. It would initially turn right to an angle of 2 degrees over a time of 0.2 seconds in a linear fashion. It would continue in this direction for 1 second. Then it would turn left at an angle of 5 degrees over a 0.4 second duration in a linear fashion. It would try to continue in this direction for 2 seconds but would end up losing control almost immediately after the second turn.
The plot of the lateral acceleration of the vehicle over the testing period is shown in Figure 10.2 (a full-page plot is shown in Appendix 3).
As shown in the above figure, the first 1.4 seconds of the maneuver consist of negative acceleration. This acceleration in lateral and towards the right of the vehicle. At this point, the acceleration turns positive. This time interval is due to the predefined test conditions listed in Figure 10.1.

Another point of interest is that of the oscillations. Upon further investigation, it was concluded that the combination of the speed and turn angles was great enough to cause the tires to slightly slip laterally. It was an extremely small amount of slip. This can be seen in Figure 10.3.
The limits of the graph were changed to help see the occurring oscillations. The rear had a slightly larger slip angle than the front. This implies that the vehicle was experiencing oversteer.

It should be noted that these oscillations, while possible in the real-world, are not realistic. The tires would not, under normal real-world testing conditions, be able to slip so frequently and create such large accelerations. Typically, once the tire loses traction, it cannot regain it with such frequency. The accelerations produced by this test are about 0.3g. Therefore, according to this test, the vehicle oscillated 5 times in the first second with average accelerations of about 0.25g. For this to actually happen, the vehicle would have to be driven with inhuman accuracy and control.

12. Computational Model Testing: Step-Steer Maneuver

The last test being performed is the Step-Steer Maneuver. This test consists of turning in one direction at high speed. It is typically used to measure the reaction time of the car to that of steering input, but it can also be used to measure the tire’s characteristics during the duration of the maneuver. The test
parameters are shown in Figure 11.1. It was set to last 8 seconds with a starting speed of 60 km/hr. After 1 second the vehicle would turn right to 2 degrees in a linear fashion over a 1 second interval.

![Figure 11.1 Step-Steer Test Parameters](image)

The results are seen in Figure 11.2 (a full-page plot is shown in Appendix 3). This shows the lateral acceleration of the vehicle during the maneuver.
The variance in the tests was relatively significant. With a percent difference of approximately 35% between the 521 and simple PAC2002 models, it is apparent that the choice of model selection is important. This also caused a difficulty in the analysis of the test. Due to the almost equal dispersion of all the results, it was hard to reasonably justify which is correct. It was determined that due to the extremely close proximity of the PACTIME, FTire, and complex PAC2002 models, that this was most reasonably the actual result. The fact that the FTire model was present in this group made it an easier decision.

Under this assumption, the 521 model showed a concerningly high amount of error that would potentially create a problematic situation if relied upon. The other models showed only about a 7 or 8% deviation from the assumed correct error.
13. Conclusion

Included in the documentation for the Adams software is a guide to help decide which tire model would be most appropriate for particular applications [3]. Since there was not enough tests conducted to make definitive conclusions about all aspects of these models, this provided a good starting point to make conclusions. The original version of the table has been given a numerical value system to help understand the overall effectiveness of each model. This new table was then modified to reflect the findings of this thesis. The original version of the table can be seen in Figure 12.1 and the modified version in Figure 12.2. The red marks on the modified version indicate the value has been changed.

![Figure 12.1 Original Reference Tire Guide](image-url)
All changes made to this table are based directly off the tests performed and first-hand knowledge regarding the reliability/possibility of each test. The “Stand still and start” criteria was changed due to the results of the longitudinal acceleration test, “Steady state corning” was changed to more closely reflect the results of the skidpad test, “Lane change” was altered to reflect the step steer maneuver, and the “Shimmy” was based off the fish-hook maneuver. The “Real Time” criteria was also changed due to the fact that most tests were able to run in real time or very close to it.

MSC Software states the best overall tire model is the PAC2002. Based off the findings stated within, this is not true. They are even aware of the inaccuracy of the Pacejka models at low speeds, yet indicate in the above table that it is best to use at these speeds. These incongruities roused suspicion initially. Once it was found out the creator of the PAC models is MSC, it was clear that there may be some bias.

Based on the above research, it is contended that the FTire model consistently outperformed the PAC2002 model. Every test conducted resulted in the FTire model being at or very near the average of all
the results. This conclusion is also backed up by the findings in the newly modified table in Figure 12.2.

The total score of the FTire model was 38, whereas the PAC2002 model only ended up with 33. Since a higher number is better in this case, the FTire model performed better as a whole than the PAC2002.
Appendix 1: Vehicle Description

The vehicle used in this research was a template model of a typical Formula SAE car [9]. It followed all the rules governing FSAE which can be found at reference 10. There was one change made to the vehicle. A larger 205/55R16 tire was fitted. This tire size was chosen because it could be used consistently across all the tire models.

![Figure A1.1 FSAE Vehicle Design Used During Testing](image)

The steering and braking system was also adjusted to provide more realistic solutions. These settings can be seen below in Figure A1.2.
Figure A1.2 Vehicle Parameters
Appendix 2: PAC2002 Tire Property Example

Below is the tire property file for the simple version of the PAC2002 model used. This file is available within the software’s installation files and can be edited to meet varying criteria.

$--------------------------------------------------------------------
[MDI_HEADER]
FILE_TYPE          = 'tir'
FILE_VERSION       = 3.0
FILE_FORMAT        = 'ASCII'

! : TIRE_VERSION : PAC2002
! : COMMENT : Tire 205/55 R16 90H
! : COMMENT : Manufacturer
! : COMMENT : Nom. section width (m) 0.205
! : COMMENT : Nom. aspect ratio (-) 55
! : COMMENT : Infl. pressure (Pa) 250000
! : COMMENT : Rim radius (m) 0.203
! : COMMENT : Measurement ID
! : COMMENT : Test speed (m/s) 30
! : COMMENT : Road surface
! : COMMENT : Road condition
! : FILE_FORMAT : ASCII
! : Copyright (C) 2004-2011 MSC Software Corporation
!
! USE_MODE specifies the type of calculation performed:
! 0: Fz only, no Magic Formula evaluation
! 1: Fx, My only
! 2: Fy, Mx, Mz only
! 3: Fx, Fy, Mx, My, Mz uncombined force/moment calculation
! 4: Fx, Fy, Mx, My, Mz combined force/moment calculation
! +10: including relaxation behaviour
! 15: Fx, Fy, Mx, My, Mz combined force/moment calculation, relaxation behaviour, including turn-slip torque
! +20: including advanced transient (contact mass approach)
! 25: Fx, Fy, Mx, My, Mz combined force/moment calculation, advanced transient including turn-slip torque & parking torque
! *-1: mirroring of tyre characteristics
!
! example: USE_MODE = -12 implies:
! -calculation of Fy, Mx, Mz only
! -including relaxation effects
! -mirrored tyre characteristics
!

! EXAMPLE PROPERTY FILE FOR THE TIRE DATA FITTING TOOL (TDFT)

! This tire property file contains the results when fitting
! the example tire data file: fm_data_example_tdft.txt or the 3 fm_data_example_tdft_*.tdx files
!

%----------------------------------------------------------units

[UNITS]
LENGTH = 'meter'
FORCE = 'newton'
ANGLE = 'radians'
MASS = 'kg'
TIME = 'second'
PRESSURE = 'pascal'

%----------------------------------------------------------model

[MODEL]
PROPERTY_FILE_FORMAT = 'PAC2002'
USE_MODE = 25.0 $Tyre use switch (IUSED)
VXLOW = 2.0
LONGVL = 30.0 $Measurement speed
TYRESIDE = 'LEFT' $Mounted side of tyre at test bench

%----------------------------------------------------------dimensions

[DIMENSION]
UNLOADED_RADIUS = 0.3169 $Free tyre radius
WIDTH = 0.205 $Nominal section width of the tyre
ASPECT_RATIO = 0.55 $Nominal aspect ratio
RIM_RADIUS = 0.203 $Nominal rim radius
RIM_WIDTH = 0.165 $Rim width

%----------------------------------------------------------dimensions

[TIRE_CONDITIONS]
IP = 200000.0 $Inflation Pressure

IP_NOM = 200000.0 $Nominal Inflation Pressure

$------------------------------------------parameter

[VERTICAL]

VERTICAL_STIFFNESS = 200000.0 $Tyre vertical stiffness

VERTICAL_DAMPING = 500.0 $Tyre vertical damping

BREFF = 4.9 $Low load stiffness effective rolling radius

DREFF = 0.41 $Peak value of effective rolling radius

FREFF = 0.09 $High load stiffness effective rolling radius

FNOMIN = 4700.0 $Nominal wheel load

QFZ3 = 1.0 $Variation of vertical stiffness with tire pressure

$------------------------------------------long_slip_range

[LONG_SLIP_RANGE]

KPUMIN = -1.5 $Minimum valid wheel slip

KPUMAX = 1.5 $Maximum valid wheel slip

$------------------------------------------slip_angle_range

[SLIP_ANGLE_RANGE]

ALPMIN = -1.5708 $Minimum valid slip angle

ALPMAX = 1.5708 $Maximum valid slip angle

$------------------------------------------inclination_slip_range

[INCLINATION_ANGLE_RANGE]

CAMMIN = -0.26181 $Minimum valid camber angle

CAMMAX = 0.26181 $Maximum valid camber angle

$------------------------------------------vertical_force_range
[VERTICAL_FORCE_RANGE]

FZMIN = 140.0 $Minimum allowed wheel load

FZMAX = 10800.0 $Maximum allowed wheel load

---------------------------------------------scaling

[SCALING_COEFFICIENTS]

LFZO = 1.0 $Scale factor of nominal (rated) load

LCX = 1.0 $Scale factor of Fx shape factor

LMUX = 1.0 $Scale factor of Fx peak friction coefficient

LEX = 1.0 $Scale factor of Fx curvature factor

LHX = 1.0 $Scale factor of Fx curvature factor

LVX = 1.0 $Scale factor of Fx vertical shift

LGAX = 1.0 $Scale factor of camber for Fx

LCY = 1.0 $Scale factor of Fy shape factor

LMUY = 1.0 $Scale factor of Fy peak friction coefficient

LEY = 1.0 $Scale factor of Fy curvature factor

LKY = 1.0 $Scale factor of Fy cornering stiffness

LHY = 1.0 $Scale factor of Fy horizontal shift

LVY = 1.0 $Scale factor of Fy vertical shift

LGAY = 1.0 $Scale factor of camber for Fy

LTR = 1.0 $Scale factor of Peak of pneumatic trail

LRES = 1.0 $Scale factor for offset of residual torque

LGAZ = 1.0 $Scale factor of camber for Mz

LXAL = 1.0 $Scale factor of alpha influence on Fx
<table>
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<tr>
<th>Symbol</th>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>LYKA</td>
<td>1.0</td>
<td>Scale factor of alpha influence on Fx</td>
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<tr>
<td>LVYKA</td>
<td>1.0</td>
<td>Scale factor of kappa induced Fy</td>
</tr>
<tr>
<td>LS</td>
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<td>Scale factor of Moment arm of Fx</td>
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<td>LSGKP</td>
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<td>Scale factor of Relaxation length of Fx</td>
</tr>
<tr>
<td>LSGAL</td>
<td>1.0</td>
<td>Scale factor of Relaxation length of Fy</td>
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<tr>
<td>LGYR</td>
<td>1.0</td>
<td>Scale factor of gyroscopic torque</td>
</tr>
<tr>
<td>LMX</td>
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<td>Scale factor of overturning couple</td>
</tr>
<tr>
<td>LVMX</td>
<td>1.0</td>
<td>Scale factor of Mx vertical shift</td>
</tr>
<tr>
<td>LMY</td>
<td>1.0</td>
<td>Scale factor of rolling resistance torque</td>
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<tr>
<td>LIP</td>
<td>1.0</td>
<td>Scale factor of inflation pressure</td>
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$--------------------------------------------\text{longitudinal}\$  

\text{[LONGITUDINAL\_COEFFICIENTS]}  

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Value</th>
<th>Description</th>
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<tbody>
<tr>
<td>PCX1</td>
<td>1.6410999976</td>
<td>Shape factor Cfx for longitudinal force</td>
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<tr>
<td>PDX1</td>
<td>1.17389999996</td>
<td>Longitudinal friction Mux at Fznom</td>
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<td>PDX2</td>
<td>-0.163950000368</td>
<td>Variation of friction Mux with load</td>
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<tr>
<td>PDX3</td>
<td>0.00799701044199</td>
<td>Variation of friction Mux with camber</td>
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<tr>
<td>PEX1</td>
<td>0.464029994168</td>
<td>Longitudinal curvature Efx at Fznom</td>
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<tr>
<td>PEX2</td>
<td>0.250220004787</td>
<td>Variation of curvature Efx with load</td>
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<td>PEX3</td>
<td>0.0678420315403</td>
<td>Variation of curvature Efx with load squared</td>
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<td>PEX4</td>
<td>-3.76128786192e-005</td>
<td>Factor in curvature Efx while driving</td>
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<tr>
<td>PKX1</td>
<td>22.3030000332</td>
<td>Longitudinal slip stiffness Kfx/Fz at Fznom</td>
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<td>PKX2</td>
<td>0.488934873763</td>
<td>Variation of slip stiffness Kfx/Fz with load</td>
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<td>PKX3</td>
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<td>Exponent in slip stiffness Kfx/Fz with load</td>
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<td>PHX1</td>
<td>0.00122970010737</td>
<td>Horizontal shift Shx at Fznom</td>
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</table>
PHX2 = 0.000431799634231 $Variation of shift Shx with load

PVX1 = -8.8101476682e-006 $Vertical shift Svx/Fz at Fznom

PVX2 = 1.86173102346e-005 $Variation of shift Svx/Fz with load

PPX1 = 0.0 $Variation of slip stiffness Kfx/Fz with pressure

PPX2 = 0.0 $Variation of slip stiffness Kfx/Fz with pressure squared

PPX3 = 0.0 $Variation of friction Mux with pressure

PPX4 = 0.0 $Variation of friction Mux with pressure squared

RBX1 = -8.94711590859 $Slope factor for combined slip Fx reduction

RBX2 = -13.752334467 $Variation of slope Fx reduction with kappa

RCX1 = 1.46939815445 $Shape factor for combined slip Fx reduction

REX1 = 6.36358608262 $Curvature factor of combined Fx

REX2 = -0.0510027253596 $Curvature factor of combined Fx with load

RHX1 = 3.15101790937e-011 $Shift factor for combined slip Fx reduction

PTX1 = 0.85683 $Relaxation length SigKap0/Fz at Fznom

PTX2 = 0.00011176 $Variation of SigKap0/Fz with load

PTX3 = -1.3131 $Variation of SigKap0/Fz with exponent of load

----------------------------------------------------------
\textit{overturning}

[OVERTURNING\_COEFFICIENTS]

QSX1 = 0.0 $Lateral force induced overturning moment

QSX2 = 0.0 $Camber induced overturning couple

QSX3 = 0.0 $Fy induced overturning couple

QSX4 = 0.0 $Fz induced overturning couple due to lateral tire deflection

QSX5 = 0.0 $Fz induced overturning couple due to lateral tire deflection

QSX6 = 0.0 $Fz induced overturning couple due to lateral tire deflection
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<th>Value</th>
<th>Description</th>
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<tr>
<td>QSX7</td>
<td>0.0</td>
<td>$F_t$ induced overturning couple due to lateral tire deflection by inclination</td>
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<td>QSX8</td>
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<td>$F_t$ induced overturning couple due to lateral tire deflection by lateral force</td>
</tr>
<tr>
<td>QSX9</td>
<td>0.0</td>
<td>$F_t$ induced overturning couple due to lateral tire deflection by lateral force</td>
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<td>QSX10</td>
<td>0.0</td>
<td>Inclination induced overturning couple, load dependency</td>
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<tr>
<td>QSX11</td>
<td>0.0</td>
<td>Load dependency inclination induced overturning couple</td>
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</table>

$Lateral$ coefficients:

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<th>Value</th>
<th>Description</th>
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<tr>
<td>PCY1</td>
<td>1.2675</td>
<td>Shape factor $C_{fy}$ for lateral forces</td>
</tr>
<tr>
<td>PDY1</td>
<td>0.9003</td>
<td>Lateral friction $M_{uy}$</td>
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<tr>
<td>PDY2</td>
<td>-0.1675</td>
<td>Variation of friction $M_{uy}$ with load</td>
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<td>PDY3</td>
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<td>Variation of friction $M_{uy}$ with squared camber</td>
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<td>PEY1</td>
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<td>Lateral curvature $E_{fy}$ at $F_{znom}$</td>
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<td>PEY2</td>
<td>-0.1037</td>
<td>Variation of curvature $E_{fy}$ with load</td>
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<td>PEY3</td>
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<td>Zero order camber dependency of curvature $E_{fy}$</td>
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<td>PEY4</td>
<td>-6.9535</td>
<td>Variation of curvature $E_{fy}$ with camber</td>
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<td>-25.7371</td>
<td>$Maximum$ value of stiffness $K_{fy}/F_{znom}$</td>
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<td>PKY2</td>
<td>3.2701</td>
<td>Load at which $K_{fy}$ reaches maximum value</td>
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<tr>
<td>PKY3</td>
<td>-0.0053</td>
<td>Variation of $K_{fy}/F_{znom}$ with camber</td>
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<td>PHY1</td>
<td>0.0031</td>
<td>Horizontal shift $S_{hy}$ at $F_{znom}$</td>
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<td>Variation of shift $S_{hy}$ with load</td>
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<td>Symbol</td>
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<td>Vertical shift in $Svy/Fz$ at $Fz_{nom}$</td>
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<td>PVY2</td>
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<td>Variation of shift $Svy/Fz$ with load</td>
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<td>PVY3</td>
<td>0.0126232741011</td>
<td>Variation of shift $Svy/Fz$ with camber</td>
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<td>PVY4</td>
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<td>Variation of shift $Svy/Fz$ with camber and load</td>
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<td>PPY1</td>
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<td>Variation of max. stiffness $Kfy/Fz_{nom}$ with pressure</td>
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<td>PPY2</td>
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<td>Variation of load at max. $Kfy$ with pressure</td>
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<td>Variation of friction $Muy$ with pressure</td>
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<td>Variation of friction $Muy$ with pressure squared</td>
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<td>RBY1</td>
<td>7.1433098945</td>
<td>Slope factor for combined $Fy$ reduction</td>
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<td>RBY2</td>
<td>9.19139631343</td>
<td>Variation of slope $Fy$ reduction with alpha</td>
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<td>Shift term for alpha in slope $Fy$ reduction</td>
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<td>Curvature factor of combined $Fy$</td>
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<td>8.48115547814e-005</td>
<td>Curvature factor of combined $Fy$ with load</td>
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<td>RHY1</td>
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<td>RHY2</td>
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<td>Kappa induced side force $Svyk/Muy$*$Fz$ at $Fz_{nom}$</td>
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<td>Variation of $Svyk/Muy$*$Fz$ with load</td>
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<td>-1.43033208331e-007</td>
<td>Variation of $Svyk/Muy$*$Fz$ with camber</td>
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<td>RVY5</td>
<td>1.90322384252</td>
<td>Variation of $Svyk/Muy$*$Fz$ with kappa</td>
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<tr>
<td>RVY6</td>
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<td>Variation of $Svyk/Muy$*$Fz$ with $\tan(kappa)$</td>
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<td>Peak value of relaxation length $\Sigma Alp_0/R_0$</td>
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<td>PTY2</td>
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<td>Value of $Fz/Fz_{nom}$ where $\Sigma Alp_0$ is extreme</td>
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</table>
$\-----------------------------rolling resistance$

[ROLLING_COEFFICIENTS]

QSY1 = 0.01 $Rolling resistance torque coefficient
QSY2 = 0.0 $Rolling resistance torque depending on Fx
QSY3 = 0.0 $Rolling resistance torque depending on speed
QSY4 = 0.0 $Rolling resistance torque depending on speed ^4

$\-----------------------------aligning$

[ALIGNING_COEFFICIENTS]

QBZ1 = 5.58750816051 $Trail slope factor for trail Bpt at Fznom
QBZ2 = -1.99836229829 $Variation of slope Bpt with load
QBZ3 = -0.58251165645 $Variation of slope Bpt with load squared
QBZ4 = -0.21321876888 $Variation of slope Bpt with camber
QBZ5 = 0.300130381798 $Variation of slope Bp with absolute camber
QBZ9 = 0.0 $Slope factor Br of residual torque Mzr
QBZ10 = -0.24523990733 $Slope factor Br of residual torque Mzr
QCZ1 = 1.09193928588 $Shape factor Cpt for pneumatic trail
QDZ1 = 0.0824041890797 $Peak trail Dpt" = Dpt*(Fz/Fznom*R0)
QDZ2 = -0.0116840781155 $Variation of peak Dpt" with load
QDZ3 = -0.183403226735 $Variation of peak Dpt" with camber
QDZ4 = -4.5398985756 $Variation of peak Dpt" with camber squared
QDZ6 = 0.000947597007759 $Peak residual torque Dmr" = Dmr/(Fz*R0)
QDZ7 = 0.00119626385488 $Variation of peak factor Dmr" with load
QDZ8 = 0.00662180855999 $Variation of peak factor Dmr" with camber
QDZ9 = 0.000425723501364 $Variation of peak factor Dmr" with camber and load
QEZ1  = -35.4973315774  $Trail curvature Ept at Fznom
QEZ2  = -35.1379552275  $Variation of curvature Ept with load
QEZ3  = -0.126313751836  $Variation of curvature Ept with load squared
QEZ4  = 0.635648867436  $Variation of curvature Ept with sign of Alpha-t
QEZ5  = -2.66879678871  $Variation of Ept with camber and sign Alpha-t
QHZ1  = 0.0022895106425  $Trail horizontal shift Sht at Fznom
QHZ2  = -0.00095192998837  $Variation of shift Sht with load
QHZ3  = 0.0310309315109  $Variation of shift Sht with camber
QHZ4  = 0.0579184073168  $Variation of shift Sht with camber and load
QPZ1  = 0.299317044146  $Variation of peak Dpt" with pressure
SSZ1  = 0.00975256989707  $Nominal value of s/R0: effect of Fx on Mz
SSZ2  = 0.0043617063455  $Variation of distance s/R0 with Fy/Fznom
SSZ3  = -2.09546594848e-006  $Variation of distance s/R0 with camber
SSZ4  = 1.30118688672e-006  $Variation of distance s/R0 with load and camber
QTZ1  = 0.0  $Gyration torque constant
MBELT = 0.0  $Belt mass of the wheel

$-----------------------------------------------
$turn-slip parameters

[TURNSLIP_COEFFICIENTS]
PECP1 = 0.7  $Camber stiffness reduction factor
PECP2 = 0.0  $Camber stiffness reduction factor with load
PDXP1 = 0.4  $Peak Fx reduction due to spin
PDXP2 = 0.0  $Peak Fx reduction due to spin with load
PDXP3 = 0.0  $Peak Fx reduction due to spin with longitudinal slip
PDYP1 = 0.4  $Peak Fy reduction due to spin
PDYP2 = 0.0 $Peak Fy reduction due to spin with load
PDYP3 = 0.0 $Peak Fy reduction due to spin with lateral slip
PDYP4 = 0.0 $Peak Fy reduction with square root of spin
PKYP1 = 1.0 $Cornering stiffness reduction due to spin
PHYP1 = 1.0 $Fy lateral shift shape factor
PHYP2 = 0.15 $Maximum Fy lateral shift
PHYP3 = 0.0 $Maximum Fy lateral shift with load
PHYP4 = -4.0 $Fy lateral shift curvature factor
QDTP1 = 10.0 $Pneumatic trail reduction factor
QBRP1 = 0.1 $Residual torque reduction factor with lateral slip
QCRP1 = 0.2 $Turning moment at constant turning with zero speed
QCRP2 = 0.1 $Turning moment at 90 deg lateral slip
QDRP1 = 1.0 $Maximum turning moment
QDRP2 = -1.5 $Location of maximum turning moment

$-----------------------------------------------contact patch parameters

[CONTACT_COEFFICIENTS]
PA1 = 0.35 $Half contact length dependency on sqrt(Fz/Fz0)
PA2 = 2.25 $Half contact length dependency on Fz/Fz0
PB1 = 0.9 $Half contact width dependency on sqrt(Fz/Fz0)
PB2 = 1.15 $Half contact width dependency on Fz/Fz0
PB3 = -3.0 $Half contact width dependency on Fz/Fz0*sqrt(Fz/Fz0)
ROAD_SPACING = 0.001 $Spacing of cam sections
MAX_HEIGHT = 0.1 $Maximum allowed obstacle height
PAE = 1.15 $Half ellipse length/unloaded radius
PBE = 1.05 $Half ellipse height/unloaded radius
PCE = 2.0 $Ellipse exponent
PLS = 0.8 $Shift length / contact length

N_WIDTH = 6.0 $Number of cams across tire width
N_LENGTH = 5.0 $Number of cams across tire length

$-----------------------------------------------contact patch slip model

[DYNAMIC_COEFFICIENTS]

MC = 1.0 $Contact mass
IC = 0.05 $Contact moment of inertia
KX = 409.0 $Contact longitudinal damping
KY = 320.8 $Contact lateral damping
KP = 11.9 $Contact yaw damping
CX = 435000.0 $Contact longitudinal stiffness
CY = 166500.0 $Contact lateral stiffness
CP = 20319.0 $Contact yaw stiffness
EP = 1.0
EP12 = 4.0
BF2 = 0.5
BP1 = 0.5
BP2 = 0.67
Appendix 3: Full Size Plots

Figure A.3.1 Lateral Acceleration vs. Time for the Skidpad Maneuver
Figure A.3.2 Vehicle Velocity vs. Time for the Longitudinal Acceleration Maneuver
Figure A.3.3 Vehicle Acceleration vs. Time for the Fish Hook Maneuver
Figure A.3.4 Lateral Vehicle Acceleration vs. Time for the Step-Steer Maneuver
References


