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I, Fredrick W Jabs, hereby submit this original work as part of the requirements for the degree of Master of Science in Mechanical Engineering.

It is entitled:
Simplified Tools and Methods for Chassis and Vehicle Dynamics Development for FSAE Vehicles

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Simplified Tools and Methods for Chassis and Vehicle Dynamics Development for FSAE Vehicles

A thesis submitted to the Graduate School of the University of Cincinnati in partial fulfillment of the requirements for the degree of Master of Science in the School of Dynamic Systems of the College of Engineering and Applied Science by

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Abstract

Chassis and vehicle dynamics development is a demanding discipline within the FSAE team structure. Many fundamental quantities that are key to the vehicle’s behavior are underdeveloped, undefined or not validated during the product lifecycle of the FSAE competition vehicle. Measurements and methods dealing with the yaw inertia, pitch inertia, roll inertia and tire forces of the vehicle were developed to more accurately quantify the vehicle parameter set.

An air ride rotational platform was developed to quantify the yaw inertia of the vehicle. Due to the facilities available the air ride approach has advantages over the common trifilar pendulum method. The air ride necessitates the use of an elevated level table while the trifilar requires a large area and sufficient overhead structure to suspend the object. Although the air ride requires more rigorous computation to perform the second order polynomial fitment of the data, use of small angle approximation is avoided during the process.

The rigid pendulum developed to measure both the pitch and roll inertia also satisfies the need to quantify the center of gravity location as part of the process. For the size of the objects being measured, cost and complexity were reduced by using wood for the platform, simple steel support structures and a knife edge pivot design. Via force balance methods, the addition of a known mass to the platform allows the computation of the center of gravity location. Measurement of the period of oscillation yields the respective inertia. Of note is the use of small angle approximations in the computation of the inertia; the magnitude of the oscillation should be kept minimal for best results.

The newest, most relevant tire information available is only in a raw data format; for the design process this fact has been another barrier to integration. Data processing scripts were developed to organize the raw data and perform fittings to the PAC2002 tire model for steady state data; an
expansion function for usage in Matlab was written to allow use of the coefficients in subsequent simulations. Estimations of the vertical spring rate and loaded radius of the tires were also developed. To quantify the transient thermal response of the tires, scripts to graph and analyze the data were prepared. Lastly, procedures for the estimation of the relaxation length properties of the tires from the raw data were executed to quantify the dynamic response of the tire’s structure. With a working tire model that can be integrated in both Matlab and Adams simulations coupled with a full quantification of the vehicle’s inertias, the capability to perform valuable chassis development and vehicle dynamics work is much more accessible.
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List of Symbols and Acronyms

FSAE – Formula SAE

SAE – Society of Automotive Engineers

FS – Formula Student

FSG – Formula Student Germany

LLTD – Later Load Transfer Distribution

FX – Tire Longitudinal Force

FY – Tire Lateral Force

FZ – Tire Normal Force

MX – Tire Overturning Moment

MY – Tire Rolling Resistance

MZ – Tire Aligning Moment

FX_o – Tire Longitudinal Force, Pure Longitudinal Slip

FY_o – Tire Lateral Force, Pure Lateral Slip

MZ_o – Tire Aligning Moment, Pure Lateral Slip

SA or $\alpha$ – Slip Angle

SL or $\kappa$ – Longitudinal Slip

IA or $\gamma$ – Inclination Angle
σ – Tire Relaxation Length

TTC – FSAE Tire Testing Consortium

Calspan TIRF – Calspan Tire Research Facility

DNF – Did Not Finish
1. Introduction

The scope of this thesis is to develop and implement methods and tools to enhance the vehicle dynamics and chassis design for FSAE teams. Focus was placed upon inertia measurement and tire data processing due to their lack of development within many FSAE organizations; summarized as follows:

- Yaw Inertia Measurement
- Pitch and Roll Inertia Measurement
- Tire Data Processing

To develop an understanding of possible points to implement these tools within the FSAE organization and design cycle, a background discussion of chassis concepts, FSAE competition dictated constraints on the vehicle, FSAE competition performance metrics and a basic logical design path for high level vehicle development are presented. The background ideas presented are meant to be an example of logical thought process applied to the problem of FSAE vehicle development, not a comprehensive guide to building a competitive FSAE entry. Vehicle dynamics and chassis development are complex subjects that demand a thorough literature search from many different sources.

1.1. Formula SAE Competition

Since the first running of the Formula SAE competition in 1981 the event has expanded from one outing in the United States to a worldwide series [1]. Backed by a relatively open rule set, FSAE emphasizes student development via hands on design, analysis, construction, testing and overall execution of a small, open-wheeled formula style race car. The problem posed to students is multi-faceted and open to interpretation. The amount of development time, resources and personnel allocated to different portions of the project is at the discretion of the students involved; this limitless approach can test the management skills of the students as much as it does their technical prowess.
during the completion of the project. It is believed that the development of common tools and methods for the development of the car’s chassis will aid Bearcat Motorsports in their future FSAE endeavors.

1.2. Bearcat Motorsports

The current FSAE team at the University of Cincinnati, Bearcat Motorsports, has been competing since 1994. Composed chiefly of engineering students, the organization faces an additional challenge due to the disconnect in personnel via the co-op education schedule. This disconnect also puts strain on transfer of knowledge and experience within the team; generally most members are only heavily involved with the design, analysis, construction and testing of the vehicle during their senior year. The carryover is very limited, as the majority of the team leaves every year. The tools and methods developed will serve to ease the implementation and analysis of the chassis design by filling a void that has existed in the knowledge utilized by the team. The more parameters of the chassis system that are defined, the less abstract the analysis and design of the car will become.

1.3. Chassis Terminology and Nomenclature

The chassis of the vehicle comprises the frame and suspension of the vehicle that serve to support all of the various subsystems of the car above the roadway and accelerate the vehicle via forces reacted by the tires to the road surface. Dependent upon the demands placed upon the chassis, parameters will be tuned differently; the base system elements of a passenger vehicle and a race car are the same although their specific forms and arrangement differs to meet the requirements of the end user. As such, knowledge and tools that originated in a passenger car environment can be utilized to meet positive ends for a race car when care is taken to adapt them to the situation.

To simplify, the chassis as a whole completes maneuvers commanded by the driver by a combination of velocity and acceleration collinear with the vehicle’s heading, the longitudinal component [2]; normal to the vehicle’s heading pointing to the vehicle’s side, the lateral component [2];
and rotation of the vehicle about the axis normal to the roadway, the yaw component [2]. These three components describe the vehicle’s motion through the global environment. A separate mass or inertia is associated with each aspect. Generally, longitudinal acceleration acts on the vehicle’s mass and rotational inertia of components such as the engine, driveline and the rotating wheel assemblies that are coupled degrees of freedom. Lateral velocity and acceleration do not act through rotating objects’ degree of freedom; as such, they are only bound to the vehicle’s mass. Finally the yaw inertia and yaw degree of freedom of the vehicle about the axis normal to the roadway acts through the yaw acceleration and velocity. In terms of Cartesian components applied to the vehicle, X, Y and Z, the longitudinal component is linear X, the lateral component is linear Y and the yaw component is rotation about Z. The origin of the system can vary based upon the desired calculations. The center of mass location in X and Y and the elevation of the ground level for Z is one possibility; the origin can be manipulated to ease the implementation of calculations. Sign convention follows the right hand rule; as a consequence choice of the direction of the Z component will flip the X and Y signs.

A frame comprises the largest role in the definition of the vehicle. It serves as the support and connection point for the vehicle subsystems and is home to the FSAE car’s larger masses; the frame, engine, engine support systems and driver make up the majority of the vehicle’s mass. Speaking in terms of system modeling, the total mass moving with the frame that is supported by the suspension is the sprung mass [3]. Generalizing the situation, if the frame is assumed to be absolutely rigid the sprung mass has 3 rigid body degrees of freedom relative to the unsprung masses in contact with the road that make up the suspension. In this case the motions are heave, pitch and roll [3]; each has its own associated mass or inertia, the sprung mass, pitch inertia and roll inertia respectively. It is noted that these degrees of freedom comprise the remainder of the components not covered by application of vehicle dynamics; these components being linear along Z, rotation around Y and rotation around X respectively. An important dynamic to include is both the frame’s and the entire chassis’ global
torsional stiffness [2]. Other frame compliances affect the chassis’ performance, such as global vertical and lateral bending as well as the local stiffness of the suspension and subsystem mounts, but lack the magnitude of impact that the torsional rigidity has on the chassis tuning options.

Adding the element of torsional stiffness of the chassis, a degree of freedom is added to the sprung body effectively splitting the roll motion into front and rear components. The sprung body thus has a front and rear roll inertia [4] and mass that are acted upon during vehicle dynamic maneuvers. Thus, the total degrees of freedom contributed by the sprung mass is four. The chassis torsional stiffness has vital implications for the design and tuning of the lateral load transfer distribution, LLTD. Sometimes referred to as a ‘Magic Number’ [4], the LLTD partly dictates the normal load distribution among the tires during dynamic maneuver accelerating the vehicle laterally. To affect the vehicle balance, understeer, oversteer and neutral steer, of the chassis, the LLTD can be shifted between the front and rear tracks. The concepts of vehicle balance describes how this yaw moment affects the direction of the vehicle as cornering speed and thus lateral acceleration is increased. As speed is increased in an understeer vehicle, a de-yawing moment is generated, thus forcing the driver to increase steering input. Oversteer results in a pro-yawing moment with a relative decrease in steering input to maintain the path. Neutral steer generates no yaw moment, thus the steer angle remains constant to maintain the path.

By the concept of normal load sensitivity [2], the fact that the tire’s effective coefficient of friction decreases with increases in normal load, the LLTD can effectively shift the cornering performance between the front and rear of the vehicle. Relative difference in lateral performance front to rear of the vehicle creates a yaw moment acting on the vehicle, steering it, which affects the vehicle balance. The key to understanding the effect of chassis torsional stiffness is approaching the front and rear of the vehicle as two separate bodies that are acted upon by the lateral acceleration at their
respective mass. If the frame has zero torsional stiffness, the roll moment at the front and rear will only have their reactions located at the end of the vehicle where they originated. The chassis will not be capable of tuning the LLTD. The roll amount of the front and rear sprung body will only be dependent on the respective roll stiffness of the suspension. Assuming an infinitely stiff torsional stiffness will dictate that the front and rear roll displacements will be equal. The LLTD will be able to be tuned very effectively due to the need to only make small changes to the suspension roll stiffness distribution. Reality lies someplace in between. High chassis torsional stiffness coupled with low roll stiffness of the suspension yields a highly effective tuning tool. As the two grow closer to each other, the needed change in roll stiffness distribution of the suspension springs to yield a similar change in LLTD will increase. When the front and rear suspension roll stiffness parameters are excessively high relative to the chassis torsional stiffness, the suspension no longer serves a function. The resulting vehicle is an approximation of a Kart in roll, with the frame satisfying the suspension needs of the vehicle. A Kart, as a vehicle, has its suspension functionality supplied by the flex of the frame.

The vehicles concepts under consideration for the FSAE competition have four wheels and thus four bodies and degrees of freedom. Otherwise either an under-constrained system would exist allowing the wheels to move in an unwanted direction when a force is applied, or the system would be over-constrained and not be capable of handling irregularities in the traversed surface. A good analogy for the need of a proper suspension is a four legged table sitting on uneven ground, thus being statically indeterminate [5]. A car with an independent suspension such as a short long arm, SLA, on each corner can be linearly generalized as having four linear degrees of the freedom in the vertical direction in total for the unsprung masses at each corner of the vehicle. Under a coupled system, such as a beam or live axle, two wheels are paired per axle. The axle, as the suspension body, has a linear and rotational degree of freedom under classic implementation of the system. Either approach still yields a total of four degrees of freedom. The common approach is to consider the four degrees of freedom as vertical
degrees at each vehicle corner. This is often expressed in quarter car models with the unsprung mass being connected to the road via the tire's vertical spring rate and to a portion of the sprung mass with a spring and damper. Another viewpoint is to consider the deflection of the 4 bodies as being a summation of the relative rigid body modes of heave, roll, pitch and twist. Utilizing both approaches will yield inferences into the system response of the vehicle to different road inputs, driver inputs and operating conditions.

Springs and dampers are installed in the suspension to control body motions with the goal of ultimately gaining favorable control of the normal force experience by the tire can be installed in three basic configurations. A spring or damper creates a force proportional to the distance or velocity between two bodies. The most common approach is the direct acting setup which acts on the difference between the states of one unsprung body and the sprung body. This can be the accomplished via direct action as on most consumer automobiles or implemented through a linkage such as a bellcrank or rocker seen on many formula cars. Barring a McPherson strut, many direct acting approaches do not act directly between the sprung and unsprung mass; as such the suspension member they act through must be built to withstand bending loads.

The second approach common to passenger and race vehicles is the anti- spring, damper or bar. An anti-element acts on the opposing difference between the states of two unsprung bodies and the sprung chassis; it can never affect the heave motion of the vehicle, but will always affect two of remaining three modes of unsprung motion, pitch, roll or twist. The most common implementation is the anti-roll bar or sway bar which adds stiffness to the roll and twist suspension rates. Consumer vehicles generally have the anti-roll bar act directly via attachment to the control arms near to the unsprung mass. When bellcranks or rockers are present, they can resolve to those elements instead.
While an anti-element resists difference in motion, a Z-element resists similar motion of two unsprung bodies with respect to the sprung body. A Z-element will always act during heave motion of the suspension plus one of the remaining modes. One example commonly seen is the third spring utilized in racing series to control the pitch attitude and ride height of vehicle under high aerodynamic loads. Synonymous to this implementation in function would be a transverse leaf spring that is allowed to pivot at its connection point to the sprung mass. The Z-element is the least utilized of the three possible implementations.

Direct acting elements affect all modes of the suspension motion, while anti-elements and Z-elements only affect two modes. It is possible to completely decouple each mode by running different anti-elements and Z-elements in series. Two differing Z-elements in series will only affect the heave mode. An anti-roll bar in series with a roll Z-bar will only affect the roll stiffness of the vehicle. Although a mechanical system composed of linkages is capable of creating such decoupling, a hydraulic analog can successfully create the same effect with less risk of hysteresis, decreased part count and reduced mass in a more flexible implementation; such an approach is seen in Tenneco’s offering of the Kinetic suspension technology [6]. As the vehicle possesses several degrees of freedom with an associated inertia, manipulation of the mechanical ratio each spring and damper is installed at, the use of various decoupled springing approaches and altering the wheelbase or vehicle tracks will allow the designer to tailor the vehicle’s dynamic response in all suspension modes. As part of this design process, increasing mechanical grip by the control of the unsprung’s oscillation on the relatively undamped spring rate of the tire is also of concern. Making compromises between the varying aspects of the design is to be expected.

The chassis designer must ultimately utilize the vehicle’s tires via the suspension design to maximize the delivered performance in response to the driver’s control inputs. The designer must be
aware of the driveline’s torque supply to the wheels, along with the dynamics of the braking force distribution and magnitude as they manipulate the *slip ratio* [2] of the tires. The type of suspension chosen, coupled with the kinematic design of the links and steering, along with the compliance setup of the suspension components and frame, will control the *inclination angle* [2] of the tires. Kinematic design and its effect on the jacking forces generated via roll centers and kinematic anti’s, the spring and damper setup, static weight distribution, weight transfer distribution and aerodynamic lift will affect the normal load on the tires. The slip angles to generate lateral force and yawing moments are generated via the design of the steering system, static toe setup, kinematic toe change and the toe compliance of the suspension and frame. The designer needs to be truly multi-disciplinary in their understanding and be prepared to make multiple iterations of the chassis design as newer, more accurate estimates of subsystem functionality become available. An organized approach to the design, development and implementation of the product is ultimately needed to successfully manage the collection and application of pertinent information and its effects on successive iterations. The chassis ends up utilizing the functionality of all the vehicle’s parts to reach the desired performance goal, as such the designer needs to be well versed in the overall system and its implementation.

**1.4. Systems Engineering Approach**

Without a plan to effectively apply and carry out the use of development tools in the engineering of the chassis, the results will be non-repeatable and of lesser quality. The plan in and of itself is one of the most important tools in the vehicle’s life. It brings order and logical progression to the engineering of the car. An organization and implementation method described in the field of System Engineering fulfills the need for a plan to execute the definition, development and deployment of the project’s life-cycle.
System Engineering is a well explored field with texts outlining various possible organization methods. The life-cycle of the vehicle design process can be represented by the ideas of research, development, test, and evaluation. To further develop this cycle a model can be utilized. The model needs not be rigid, as it should be flexible enough to fit the specific needs and intricacies of the specific product. Further break down of the definition, development and deployment concepts yields detailed steps to the overall implementation of the project.

When a simple tool has already been developed, utilizing and adapting it to the specific life-cycle at hand offers the best return on investment. Simple and useful life-cycle models are described extensively in literature [7]. Comprising the definition stage are defining the requirements and specifications for the product and then conducting a conceptual design to begin coalescing a course of action to meet the requirements. Moving onto the development of the project, three steps refine and develop the initial concept. These steps are summarized as 1) logical design, 2) detailed design and analysis, and 3) design implementation. Finally the chassis can be deployed; first it is evaluated and modified in a running state followed by the actual full operational deployment of the vehicle at competitions.

These steps are not meant to be independent entities; it is a life-cycle that feeds knowledge forward to the fruition of an operational vehicle and backward to alleviate and address problems in a closed loop iterative design process [7]. A summary of a modified system engineering life-cycle is presented below. The tools and methods presented may benefit more than one stage of the development cycle of the vehicle.

- Problem Definition
- Conceptual Design
- Logical Simulation and Design
• Detailed Analysis and Design
• Design Implementation, Testing and Validation
• Product Evaluation and Modification
• Product Deployment
2. FSAE Regulations Background

Ultimately the problem facing the participant team is to score the most points possible in the FSAE competition. Seeing that this problem is multifaceted, sub problems will be developed to decouple differing areas of performance to achieve the maximum score possible. One interpretation is that the final score can be attributed to vehicle performance, driver performance, and team knowledge; the former system being the chief focus of this text while the latter two are highly variable and difficult to quantify due to their human factor. It can be surmised that greater knowledge and development applied to the vehicle’s performance by the associated team will benefit the latter two elements. The vehicle’s function is to complete all competition events at the highest performance level possible within the confines of cost to the team, both monetary and time.

2.1. FSAE Regulation System Constraints

To enter the FSAE competition, thus gaining the possibility to earn points, the vehicle must satisfy a set of constraints laid out in the rules. A vehicle that does not meet the requirements of the rules will not compete! Rule compliance is guaranteed via the technical inspection process at each competition. The majority of regulations are safety driven details which require certain design elements and thus mass to be included in the vehicle, and will not be discussed at length. Of importance to the chassis developer are the rules placing direct limits on key vehicle parameters governing vehicle performance. These physical constraints will be the basis of the discussion of regulation based limits of the vehicle. Referencing back to the problem statement, the rules also define the point structure and the events evaluate the performance of the FSAE entry. Thus the rules will also serve as the basis for the metrics to gauge the performance of the chassis being developed.
2.2. General Chassis Limitations

Rules that affect the performance of the chassis restrict how forces are reacted into the ground and how those forces are generated. The base of the vehicle, the wheels and tires, will be the first regulations discussed. The general form of the chassis is defined by rule B2.1 [8]. The minimum rim diameter rules out the use of Kart rims and tires which generally run 5 and 6 inch diameter rims. The last excerpt is the allowance to both a dry and wet style tire to choose from. The competition events continue whether it is dry or wet on track, so the team must at least have grooved rain tires to accommodate the full wet condition. The wet tire may be utilized as a dry tire if specified at the time of technical inspection, but a properly chosen specialized dry tire should give a performance advantage.

B2.1 Vehicle Configuration

The vehicle must be open-wheeled and open-cockpit (a formula style body) with four (4) wheels that are not in a straight line.

B6.3.1 The wheels of the car must be 203.2 mm (8.0 inches) or more in diameter.

B6.4.1 Vehicles may have two types of tires as follows:

• Dry Tires – The tires on the vehicle when it is presented for technical inspection are defined as its “Dry Tires”. The dry tires may be any size or type. They may be slicks or treaded.

• Rain Tires – Rain tires may be any size or type of treaded or grooved tire... [8]

2.3. Wheelbase and Track Width Restrictions

The locations of the tires play a key role in how the vehicle behaves dynamically. Certain minimums are established in the rules to ensure stability and thus safety of the vehicle. A minimum wheelbase of 60 inches is simply stated. The minimum measurement of track width is intrinsically
defined by the rollover prevention requirements. The necessary track width is thus a function of the height of the center of gravity of the vehicle and the rollover stability requirement. To eliminate the possibility of trike concept vehicles the smaller track width must be at least 75% of the length of the larger track. There are no maximum values for wheelbase or track explicitly stated. Due to stipulations in dynamic event course layouts, as vehicle overall dimensions become larger due to increased wheelbase and track the vehicle will consume more of the allocated space between cones. This limits the possible lines to be taken by driver and vehicle. For example, a wider vehicle may have better cornering performance than a narrow vehicle due to normal load sensitivity; the narrow vehicle can attain a better racing line due to coned courses being a confined operating environment.

**B2.3 Wheelbase**

The car must have a wheelbase of at least 1525 mm (60 inches). The wheelbase is measured from the center of ground contact of the front and rear tires with the wheels pointed straight ahead.

**B6.7.2 Rollover stability will be evaluated on a tilt table using a pass/fail test. The vehicle must not roll when tilted at an angle of sixty degrees (60°) to the horizontal in either direction, corresponding to 1.7 G’s. The tilt test will be conducted with the tallest driver in the normal driving position.**

**B2.4 Vehicle Track**

The smaller track of the vehicle (front or rear) must be no less than 75% of the larger track.

**B6.7.1 The track and center of gravity of the car must combine to provide adequate rollover stability. [8]**
2.4. Suspension Functionality

With the stipulations on the position of the wheels and tires in place, rules dictating restrictions upon the suspension’s function will be highlighted. A functioning suspension with a minimum of 1 inch of travel in jounce or rebound from ride height is required. The use of shock absorbers is stipulated; the use of some form of springs to support the mass of the vehicle can also be inferred. To improve performance from the simplified perspective of linear systems, the vehicle’s mass, stiffness, and damping properties need to be assessed and tuned. A minimum ground clearance is no longer enforced, but contact with the track is forbidden, resulting in a disqualification. Safety regulations and the need to comfortably house a driver enforce certain minimum cockpit dimensions thus placing limitations on the internal pickup points of the suspension and the frame itself. Lastly the steering system utilized must be a direct mechanical link from the steering wheel to the vehicle’s wheels.

B6.1.1 The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit.

B6.2 Ground Clearance

There is no minimum ground clearance requirement. However, teams are reminded that under Rule D1.1.2 any vehicle condition which could, among other things, “... compromise the track surface” is a valid reason for exclusion from an event. Any vehicle contact that creates a hazardous condition or which could damage either the track surface or the timing system is cause for declaring a vehicle DQ.
B6.5.1  The steering wheel must be mechanically connected to the wheels, i.e. “steer-by-wire” is prohibited. [8]

2.5. Brake System Restrictions

Stipulations pertaining to the brakes ensure a reliable system that is capable of locking up all of the vehicle’s wheels. As with most other systems on the vehicle, brake by wire is prohibited. Anti-lock braking systems are allowed, but must be disabled for the brake test during technical inspection. Any active system can be implemented as long as the requirement of a dual circuit passive system is satisfied.

B7.1 Brake System - General

The car must be equipped with a braking system that acts on all four wheels and is operated by a single control.

B7.2 Brake Test

The brake system will be dynamically tested and must demonstrate the capability of locking all four (4) wheels and stopping the vehicle in a straight line at the end of an acceleration run specified by the brake inspectors. [8]

2.6. Driveline Restrictions

For safety and competitive reasons the power generating capabilities for the vehicle’s engine(s) are regulated by the use of a restrictor. Compressors are permissible but stipulated to lie after the restrictor in the intake system. The restrictor can only be fed with air metered by the throttle body directly from the atmosphere. Choked flow through a restrictor is limited to a function of the inlet pressure and temperature for compressible fluids. Along with the restrictor, the other chief engine limitation is the use of a 4 stroke Otto cycle with a maximum displacement of 610 cubic centimeters.
The power derived from the engine system can be transferred to the wheels using any drivetrain solution.

**B8.1.1** The engine(s) used to power the car must be a piston engine(s) using a four-stroke primary heat cycle with a displacement not exceeding 610 cc per cycle. Hybrid powertrains, such as those using electric motors running off stored energy, are prohibited.

Note: All waste/rejected heat from the primary heat cycle may be used. The method of conversion is not limited to the four-stroke cycle.

**B8.6.1** In order to limit the power capability from the engine, a single circular restrictor must be placed in the intake system between the throttle and the engine and all engine airflow must pass through the restrictor.

**B8.6.3** The maximum restrictor diameters are:

- Gasoline fueled cars - 20.0 mm (0.7874 inch)
- E-85 fueled cars – 19.0 mm (0.7480 inch)

**B8.7.2** The restrictor must be placed upstream of the compressor but after the carburetor or throttle valve. Thus, the only sequence allowed is throttle, restrictor, compressor, engine.

**B8.12 Transmission and Drive**

Any transmission and drivetrain may be used. [8]

Body forces generated by vehicle aerodynamic elements are also limited by the rules. Within the vertical plan view, wings must be located within boundary based off the locations of the front and rear wheels. Undertrays and diffusers are permissible as long as no powered device for the explicit
purpose of creating a pressure differential to the area under the car is utilized. Aero elements that impede driver egress are outlawed as well, whether rigidly mounted or movable. The stipulation of no track contact also has consequences for the implementation of aerodynamic devices.

B12.2 Location

B12.2.1 In plan view, no part of any aerodynamic device, wing, under tray or splitter can be:

a. Farther forward than 762 mm (30 inches) forward of the fronts of the front tires

b. No farther rearward than 305 mm (12 inches) rearward of the rear of the rear tires.

c. No wider than the outside of the front tires or rear tires measured at the height of the hubs, whichever is wider.

B12.4 Ground Effect Devices – No power device may be used to move or remove air from under the vehicle except fans designed exclusively for cooling. Power ground effects are prohibited.

B12.5 Driver Egress Requirements

B12.5.1 Egress from the vehicle within the time set in Rule B.4.8 “Driver Egress,” must not require any movement of the wing or wings or their mountings.

B12.5.2 The wing or wings must be mounted in such positions, and sturdily enough, that any accident is unlikely to deform the wings or their mountings in such a way to block the driver’s egress. [8]
3. FSAE Competition Events Background

The race car is designed to compete in the FSAE event. The performances of vehicles in the competition are evaluated utilizing a series of events in a point structure totaling 1000 points. The point distribution and event setup are specified in the rules as well. The overall point distribution is as follows. [8]

- Static Events
  - Design 150
  - Cost 100
  - Presentation 75

- Dynamic Events
  - Skid Pad 50
  - Acceleration 75
  - Autocross 150
  - Endurance 300
  - Fuel Economy 100

Detailed explanation of the methods of each event yields a set of quantifiable performance goals that vehicle characteristics directly affect.

3.1. Design Event

The design event can be scored to a maximum of 150 points. The judges make a qualitative assessment of the engineering utilized in comparison to the other teams competing and the expected level of performance in the event. A maximum score less than 150 points can be given if no team is found to be deserving of the full allotment. The engineering depth, quality, and validity are all important aspects being examined. Engineering effort can only be reasonably recognized if
corresponding documentation and data is present. The vehicle is not being judged on its quality and performance, but on whether or not it demonstrates the engineering reasoning utilized by the team; the vehicle serves as a visual aid to represent the ideas of the team. A vehicle can reflect negatively upon the team due to the physical representation of unprofessionalism and lack of preparedness. The design event points can be maximized with increases in valid engineering practices applied to the problem, supported by documentation and validation to form a cohesive and reasoned argument for a decision. Application of engineering practices such as root cause analysis, design for manufacturability, valid application of the scientific process and a robust systems engineering approach to the vehicle’s development will all bode well for the team. Not fixing problems from the past, committing common engineering mistakes and bluster to try to compensate for a lack of knowledge will not bode well for the event’s outcome; these are often signs of poor knowledge retention and transfer within the team structure, another favorite examination point of design judges are often critical of.

C5.1.1 The concept of the design event is to evaluate the engineering effort that went into the design of the car and how the engineering meets the intent of the market.

C5.1.2 The car that illustrates the best use of engineering to meet the design goals and the best understanding of the design by the team members will win the design event.

C5.11.1 The design judges will evaluate the engineering effort based upon the team’s Design Report, Spec Sheet, Student Activity Disclosure Form, responses to questions and an inspection of the car.

C5.13.1 Scoring may range from 0 to 150 points at the judge’s discretion.

C5.13.2 The judges may at their discretion award the highest placing team less than 150 points.

[8]
3.2. Cost Event

The cost event is worth 100 points total. These are divided into three categories: 40 points for total cost, 40 points for the cost report, and 20 points for the real case scenario presentation. The total cost score is based off of the corrected total cost \( P_{Your} \), the entrant with the lowest cost \( P_{Min} \), and the highest priced entrant \( P_{Max} \). To assure price parity amongst material and part suppliers utilized by entrants, common pricing tables for materials, parts and processes are utilized in the preparation of the vehicle’s total cost.

\[
Points_{TotalCost} = 40 \times \frac{(P_{Max}/P_{Your}) - 1}{(P_{Max}/P_{Min}) - 1}
\]

*Equation 1 - Cost Event - Total Cost Scoring Function [8]*

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*Table 1 – Cost Event \( P_{Min} \) and \( P_{Max} \) starting in 2009, with the new standardized costing format. [9]*

The cost report, valued at a maximum of 40 points, takes into account the accuracy of the report’s numerical figures as well as how representative the report is of the actual product. The cost derived from the report is the one utilized for the Total Cost score. Errors result in penalties to both the 40 points allocated for the report and Total Cost score.

The last portion is the real case scenario accounting for 20 points. This portion gages the team’s ability to produce possible cost reduction to specific vehicle parts and sub-systems. The topic of the scenario is announced shortly before competition to limit the time to formulate a solution. The judging is not based on presentation skills, instead upon the technical merit of the solution offered.
3.3. Presentation Event

The third static event, presentation, is valued at 75 points. The presentation seeks to procure the investment capital of the board of judges via the proposed business plan for the production of the race car. As such it must be professionally delivered, answer all of the judges’ questions, and, even though fictitious, make the judges want to invest their own money in the project. The judging is highly subjective and qualitative; quantitative elements to the presentation are very important, performance numbers, capital investment figures, return on investment and any other quantified figure must be logical, believable and have valid reasoning when questioned. Time is limited at 10 minutes for the presentation. Going over the time limit will result in the judges stopping the presentation. Preparation time, practices and revisions based upon feedback are the key to improving the Presentation score.

3.4. Skidpad

The dynamic event with the least value is Skidpad with 50 points. Skidpad seeks to measure the steady state turning capacity of the vehicle. The track is two circular paths of 3 meters in width with an inner diameter 15.25 meters joined like a figure-eight to assess turning capabilities going both directions [8]. For each direction the first lap is to settle the vehicle with the second being timed; first a right turn is executed after entry to the figure-eight, then the turn direction is reversed to the left in the other circle repeating the procedure with the vehicle exiting afterwards. \( T_{\text{Your}} \) is the average of the two timed laps. Hitting cones and going off course apply penalties to the run time. The fastest time of the day is \( T_{\text{Min}} \). A second heat can be taken immediately after to take advantage of heat in the tires. Any time greater than 6.184 seconds scores for participation and is awarded 2.5 points. For times less than or equal to 6.184 seconds the scoring takes the following form.
\[
Score_{\text{Skidpad}} = 47.5 \times \left(\frac{6.184/T_{You}}{(6.184/T_{Min})^2 - 1}\right) + 2.5
\]

Equation 2 - Skidpad Scoring Function [8]

2.5 points are automatically awarded for completion of the event successfully no matter the time needed. \( T_{You} \) is dominated by the subject of the test, steady state turning behavior. The level of performance of turning can be equated to the tire performance when moderately cold, overall mass of the vehicle, weight transfer of the vehicle, mechanical grip and vehicle balance. Maintaining a controlled steady state turn dictates the importance of off throttle response, predictable force distribution between the rear wheels via the drivetrain and controllable steering response utilizing feedback through the wheel. A symmetrical vehicle is necessary to assure that the performance of both right and left turns are similar.

Due to the setup simplicity of the Skidpad event and lack of areas of error possible in the setup, it is of interest to take a survey of the \( T_{Min} \) of the respective competitions in Table 2. Differences in the track state, weather, and temperature have an effect that requires further data to quantify. Typically a time of 5.0 seconds will yield a high placing in the dry. Minimum times appreciably higher than that range are indicative of wet running conditions. Special attention should be given to Equation 2, due to the time terms being squared, small differences in time result in a larger score difference than in other events.

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<td>4.918</td>
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Table 2 - Representative Skidpad TMin for recent competitions in seconds
3.5. Acceleration

Acceleration is an event judging the longitudinal acceleration capability of the vehicle. The length of the timed portion is 75 meters. Cars start from a standstill with the forward most part of the vehicle 0.30 meters behind the start of the timed section. 75 points are possible through completion of the event. The maximum run time for scoring is 5.8 seconds; if the time elapsed exceeds 5.8 seconds the team will still be awarded 3.5 points regardless. The calculation of the score takes into account the team’s time, \( T_{\text{Your}} \), and the minimum time of the day, \( T_{\text{Min}} \) in Equation 3.

\[
\text{Score}_{\text{Accel}} = 71.5 \times \frac{(5.8/T_{\text{Your}}) - 1}{(5.8/T_{\text{Min}}) - 1} + 3.5
\]

Equation 3 - Acceleration Scoring Function [8]

The acceleration run can be effectively divided into sections based upon the limiting factor of performance. First is the traction limited section followed by the power limited acceleration section. The location of the transition is a function of tire characteristics, effective power to the wheels, gearing utilized, and slip ratio management. Proper assessment of this point is needed for effective setup of the vehicle powertrain subsystem.

The performance in the traction limited domain is dependent on keeping the engine output limited to an amount to maintain peak slip ratio at the wheels. As traction is the limiting factor improvements to chassis dynamic response and mechanical grip of the tires will improve performance. Throttle modulation, clutch modulation, and electronic traction control aids are all available methods to manage the torque and thus slip ratio being run at the tires. The initial launch method can either lead to an effective or slow run easily.

Power limited acceleration sets in once the force reacting capabilities of the tires exceed the d’Alemberts force of the linear acceleration of the vehicle’s mass. The power of the vehicle must
accelerate both this mass and the coupled rotational inertias of the powertrain and wheels. Overcoming inefficiencies in the drivetrain, rolling resistance of the tires, and aerodynamic drag of the vehicle sap power as well. Shifting the transmission causes interruptions to transferring power from the engine to the wheels and the ground. During the shift the vehicle will actually decelerate due to the inherent inefficiencies and drags within the vehicle.

Minimum times from past events in Table 3 are representative of the performance needed to be competitive in the Acceleration test. Comparing tests from one competition to another can be misleading. Often, whether on purpose or by accident, competition organizers deviate from the procedure laid out in the rules by changing the start distance, total distance covered, and the incline of the track utilized for the run; these changes have a substantial effect on the performance measured. Expected differences such as pavement type, temperature, and weather condition effect the times as well. Generally, a time of 4.0 seconds will give a high placing and score; recent results have the competitive time dropping.

|-------|-----------|-----------|---------|-----|

Table 3 - Representative Acceleration times for TMin in seconds

3.6. Autocross

Autocross tests the vehicle’s overall dynamic performance by running the car through a series of tight obstacles. Setup for the event is driven by guidelines within the rules. The track setup can vary widely depending upon the facilities available and the track designer’s prerogative. Generally the track should be 805 meters in length with average speeds in the range of 40 to 48 kilometers per hour. Track design elements may be any of the following outlined in the rules but are not limited to these alone.
*Straights*: No longer than 60 m (200 feet) with hairpins at both ends (or) no longer than 45 m (150 feet) with wide turns on the ends.

*Constant Turns*: 23 m (75 feet) to 45 m (148 feet) diameter.

*Hairpin Turns*: Minimum of 9 m (29.5 feet) outside diameter (of the turn).

*Slaloms*: Cones in a straight line with 7.62 m (25 feet) to 12.19 m (40 feet) spacing.

*Miscellaneous*: Chicanes, multiple turns, decreasing radius turns, etc. The minimum track width will be 3.5 m (11.5 feet). [8]

Each driver gets 2 runs, which may be run in succession. The vehicle is staged 6 meters behind the starting line where it starts from a stand still. Penalties of 20 seconds and 2 seconds are applied for going off course and hitting cones respectively. Scores are determined from the team’s corrected run time, the minimum run time for the event, and the calculated maximum time. The maximum time is 125% of the minimum time; teams that exceed the max time score 7.5 points for completing the event.

Times within the scoring range receive points by the following equation.

\[
Score_{Autocross} = 142.5 \times \frac{(T_{Max}/T_{Your}) - 1}{(T_{Max}/T_{Min}) - 1} + 7.5
\]

\[
T_{Max} = 1.25 \times T_{Min}
\]

*Equation 4 - Autocross Scoring Function [8]*

Autocross course layout has similar performance affecting characteristics as the endurance event, which will host the discussion of those attributes. Special considerations that distinguish autocross from endurance are the lack of fuel economy judging and limited run time to reach steady state conditions. For vehicle performance, engine mapping should provide maximum power when
demanded without consideration for fuel economy. The transient nature of the runs, which is comparable to both skidpad and acceleration, puts an emphasis on the heat in the tires. Tire temperature and thus performance should increase throughout the lap and start at a higher temperature for the second lap. Setup changes and driver aggressiveness can induce more heat into the tires by scrubbing them; these changes will effectively take engine power to cause faster heat generation in the tires.

3.7. Endurance

The Endurance event judges the vehicle’s reliability and performance while covering 22 kilometers of an autocross style circuit. Like the Autocross event, the course is made up of elements outlined in the rules. Laps can be any length based upon the available facilities for the event. The event’s characteristics, as defined by the rules, vary slightly from the autocross setup; the organizers can alter these requirements as they see fit. Typical speeds range from 48 to 57 kilometers per hour with peak speeds of 105 kilometers per hour. Specific characteristics of the track are as follows.

Straights: No longer than 77.0 m (252.6 feet) with hairpins at both ends (or) no longer than 61.0 m (200.1 feet) with wide turns on the ends. There will be passing zones at several locations.

Constant Turns: 30.0 m (98.4 feet) to 54.0 m (177.2 feet) diameter.

Hairpin Turns: Minimum of 9.0 m (29.5 feet) outside diameter (of the turn).

Slaloms: Cones in a straight line with 9.0 m (29.5 feet) to 15.0 m (49.2 feet) spacing.

Miscellaneous: Chicanes, multiple turns, decreasing radius turns, etc. The standard minimum track width is 4.5 m (14.76 feet). [8]
A maximum of 300 points can be earned for the timed performance of the vehicle in Endurance. The same penalties apply for off courses and cones as in autocross. Penalties are also levied to punish dangerous actions by drivers. Vehicles with mechanical failures or fluid leaks can cause the vehicle to be removed from the event and assigned a DNF. Vehicles that don’t complete the event will receive zero points as a result of lap times. Post event impoundment and inspections are routine for top finishing competitors. A vehicle can be disqualified for rule technicalities found in the post event technical inspection. Teams must finish within 1.45 times the minimum run time of the event to fall within the scoring equation; this value can be modified by the organizers to cope with condition changes. Teams that do not finish all laps or fall outside the permissible timing window will be awarded 1 point for each lap completed. The scoring equation is as follows.

\[
Score_{\text{Endurance}} = 250 \times \frac{(T_{Max}/T_{Your}) - 1}{(T_{Max}/T_{Min}) - 1} + 50
\]

\[
T_{Max} = 1.45 \times T_{Min}
\]

*Equation 5 - Endurance Scoring Function [8]*

The performance of the vehicle in an autocross style course can be categorized under two types of behavior, steady state and transient. Steady state behavior can be generalized as occurring when yaw acceleration is zero and the vehicle is fully settled. Three types of maneuvers can satisfy this stipulation. Steady state acceleration under power is seen in power or traction limited acceleration down straights. This characteristic was already demonstrated in the acceleration event. After acceleration of the vehicle, braking decelerates the vehicle in a steady state behavior during limit
braking. During both acceleration and deceleration yaw rate should be zero as well. The last steady state characteristic was seen in the skidpad event, cornering. Depending upon the setup of the corner, the behavior will differ in length. During a long sweeper turn, steady state cornering will occur for an extended time. In shorter maneuvers, it will occur between transients at the peak yaw rate. An apt definition of this behavior is constant non-zero yaw rate which incurs the condition of zero yaw acceleration.

Transients make up the remainder of the track. They connect all events that are deemed steady state. Consecutive transients may occur, while steady state conditions may not. Transients can be seen as the entry and exit from a steady state cornering situation. The entry may be from either a previous turn or a braking zone. In the case of braking, the corner entry will consist of trail braking accompanied by yaw acceleration increasing in magnitude to start to build yaw rate. This is followed by mid corner entry where yaw acceleration is decreasing in magnitude and yaw rate continues to increase towards the peak yaw rate at steady state cornering. After the steady state portion, mid corner exit will begin. Yaw acceleration will start to build in magnitude with opposite sign; yaw rate will start to decrease. Finally, at corner exit, either another corner or a straight will follow. In the case of a straight, yaw rate and yaw acceleration will both become zero and power will start to be applied as traction becomes available. One transient not covered under cornering behavior is the transition from acceleration to braking in a straight, which is characterized by the pitch dynamics of the vehicle as longitudinal load transfer shifts normal force from the vehicle’s rear to its front.

Due to the constraints of the event a new set of transient aspects affecting performance will also take place. With the continuous running, tire temperatures can increase beyond the performance limits of the tire compound. A single compound must be run for the entire competition; a soft compound will be advantageous for short duration running but lose capability under the continuous
running in the Endurance event. The mass of the vehicle and its location change due to the effects of fuel burn off and the changing of the driver; these changes affect the tuning of the vehicle’s suspension and chassis. Unlike every other event during the competition, tuning changes cannot be made by the team between the two drivers. Only cockpit adjustable tuning devices accessible by the driver can be changed to adapt to the driver change. Changes to the seat and the pedal cluster position can be made by the team to allow the safe operation of the vehicle by both drivers.

3.8. Fuel Economy

Coupled with the Endurance event is Fuel Economy. The volume of fuel utilized by the vehicle over the Endurance event is measured to ascertain the Fuel Economy score. A score is possible between 100 and -100 points. The minimum combined Endurance and Fuel Economy score is zero points. Failure to complete Endurance in less than the maximum time will yield a Fuel Economy score of zero. The point that splits the positive and negative scoring equations is the calculated maximum volume. The maximum volume will equate to fuel economy of 26 liters per 100 kilometers by the rules. For a standard length endurance course this equates to consumption of 5.72 liters of petrol. Minimum volume will be determined by the team consuming the least fuel. For alternate fuels a volume correction will be applied to make its energy content equal to petrol. Teams exceeding 1.33 times the maximum volume will score -100 points. Scoring for volumes between the minimum volume and 1.33 times the maximum volume are governed by Equation 6.
Formula Student Germany has taken an alternate approach to point allocation and the scoring of events. The Fuel Economy event is supplanted by the Fuel Efficiency calculation at the German competition. The altered format takes into account the team’s endurance time and fuel volume if the maximum time and volume thresholds are met. These are then compared to minimum time within the endurance results and the minimum volume within the scoring endurance group. The time threshold is an alternate form of 1.33 times the minimum time. Taking these values into account the score is determined.

\[
Score_{Fuel} = 100 \times \frac{(V_{Max}/V_{Your})}{(V_{Max}/V_{Min})} \quad \text{for } V_{Your} \leq V_{Max}
\]

\[
Score_{Fuel} = -100 \times \left(\frac{(V_{Your}/V_{Max}) - 1}{0.33}\right)^{1.5} \quad \text{for } V_{Max} < V_{Your} \leq 1.33 \times V_{Max}
\]

\[
Score_{Fuel} = -100 \quad \text{for } V_{Your} > 1.33 \times V_{Max}
\]

Equation 6 - Fuel Economy Scoring Function [8]

Key attributes affecting fuel consumption are driveline inefficiencies, vehicle mass, endurance performance level, vehicle total inertia, rolling resistance, and aerodynamic drag. A team can make the choice to run Endurance with reduced performance, with a higher elapsed time, to get the maximum possible total score for the two events by minimizing the fuel volume. Fuel consumed anytime between
the two fillings affects the economy as well. Warming up the engine and excessive idling when not on track should be avoided. Minimum volumes from past competitions are cited below.

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</table>

Table 5 - Typical minimum fuel volumes in liters
4. FSAE Chassis Design Background

The framework of the FSAE competition is defined by a small, open set of rules limiting possibilities for vehicle development. They also set up performance metrics in the form of the competition itself. First a set of qualitative vehicle performance characteristics will be defined. These anecdotal goals will be given substance by quantifying the competition performance metrics in terms of them. The qualitative properties will then be quantified utilizing physical parameters of the vehicle. Ultimately a function of vehicle parameters will define points scored in competition metrics; although trending will only be discussed here as a simple tool, sensitivity and iterative analysis routines can be built upon these results.

4.1. Qualitative Vehicle Performance Characteristics

Qualitative statements concerning vehicle performance are simplistic goals which are important for their realization, but lack quantified characteristics that can be validated. All of these aspects are important to chassis performance and its design, as such they need to be identified and quantified. Reliability and rules compliance are simply needed to allow the vehicle to compete; they are qualifying statements that must be attained to garner an event score.

By simply going down the list of vehicle subsystems a relatively complete list of qualitative performance characteristics can be formed. Steady state braking, power limited acceleration and steady state cornering are simple to realize. These same aspects have multiple transient counterparts as well. Trail braking, corner entry, corner exit, and traction limited acceleration on corner exit can be recognized as transient behaviors due to the presence of chassis dynamics. All of these characteristics fall under the title of vehicle dynamics; they are discussed to great lengths [2] [3] [4] [5] [15] [16] [17] and serve as a key basis for all vehicle development. Application and treatment by the texts range from anecdotal evidence and concepts to analytical simulations and calculations. Key parameters are needed
to carry out vehicle dynamics calculations; emphasis on their quantification and validation is a team necessity if any in depth vehicle dynamics calculations are to be done.

Even if the car has the highest performance possible in terms of parameters, if it is uncontrollable it is of little use. To gauge the effectiveness of the subsystems, brake control, power control, and cornering control become evident as performance enabling characteristics. Vehicle efficiency effects play into the outcome of several events. Since cost has a competition event and implications on the vehicle’s performance it will be mentioned as well. A car can have the best parameters as a design basis, but if not executed to carry out its function they are for naught.

4.2. Vehicle Characteristics’ Influence upon Regulation Performance Metrics

The majority of the static event points available are not directly tied to the vehicle performance characteristics. The amount of static event points earned is an indicator of the team’s performance and organization. The design, construction and development of the vehicle serve as a visual aid that can have both negative and positive effects on the score though. Correlations between team performance and vehicle performance are evident, but are not the focus of this discussion. The total cost of the vehicle is a driving factor for 40 points of the cost event. The figures utilized in the cost event are not directly indicative of the actual price of the vehicle due to standardized pricing tables.

Reliability and rules compliance can be seen as binary modifiers of the dynamic event rule based performance metrics. An unreliable car will not be able to complete any dynamic event. One that is noncompliant with the rules will not have a chance to participate in any dynamic events. A vehicle that is unfinished lacks both reliability and rules compliance. Rule compliance makes 675 out of the 1000 points become accessible. With the addition of reliability all events can be completed. Regardless of performance level this guarantees 13.5 performance points out of the 675 available. The remainder of
the 675 will be based upon vehicle performance and efficiency assuming trained operators; though these cannot be demonstrated without reliability or rules compliance.

The first dynamic event examined, Skidpad, is fairly simple to correlate to vehicle performance characteristics. Skidpad is simply a test of steady state cornering performance and the capability to stay at that peak performance. Since both turning directions apply equally to the final time a symmetrical vehicle is desirable. To maintain the vehicle’s path, transient cornering behavior, cornering control, and power control will be exercised to some extent during the event.

The other simple dynamic event, Acceleration, is dominated initially by power control; this dominates performance until the threshold of power limited acceleration sets in. With power limited acceleration dominating the remainder of the run to termination, power control still plays a minor role in the form of shifting. This explanation of the Acceleration event illustrates the many parts of controlling the power being applied to the rear wheels. These include the throttle, clutch, engine calibration, traction control devices, gear shifting and drivetrain differential behavior.

Due to their similarity, Autocross and Endurance will be analyzed at the same time. All performance characteristics can be seen on display during an autocross style course. Establishing a breakdown of the content of a course can be accomplished with data acquisition. Utilizing logical equalities steady state portions of the track can be segmented. Since yaw rate sensor data was unavailable lateral acceleration, longitudinal acceleration, steering wheel angle, throttle position and brake pressure data were used to estimate when steady state behavior occurs. These findings are displayed in Figure 1. Transient, braking, acceleration and cornering sections are shown respectively in the histogram and track map as blue, red, green and cyan. The lap was a kilometer in length with an elapsed time of 57.5 seconds, recorded at Goodyear’s test track in Akron, Ohio with the 2007 Bearcat Motorsports FSAE car.
Estimations of braking and straight line acceleration prove to be fairly accurate with the recorded data available for analysis. By this approach these maneuvers occupy 8.5% and 12.5% of the time on track respectively. Due to the lack of yaw rate data distinguishing steady state cornering behavior from the transient portions appears to be less predictable. By viewing the portions ruled as cornering with respect to their location on the track layout, their amounts and timing are found to be logical; an estimation of 15% of the on track time is spent in steady state cornering. With only 36% of the track time quantified in steady state maneuvers, further analysis of the remaining time will yield a breakdown of the transient regimes of the vehicle’s handling.
With the data available estimations of transient braking and acceleration were found. Based upon the transient sections left by the steady state analysis, portions that exhibited any appreciable braking were highlight as transient braking zones marked by the pink section in Figure 2. Analysis of the section for positive longitudinal acceleration during any of the defined transient sections highlighted the dark green section signifying transient acceleration. With these two eliminated the remainder is comprised of transient cornering behavior marked in blue. By percentage of lap time, the transient portions total 64% of the total lap. Accounting for this total are constituent portions of 10% transient braking, 21% transient acceleration and 33% transient cornering.

The last dynamic event is Fuel Economy. Of the available qualitative vehicle parameters, vehicle efficiency is the obvious behavior dominating the event. Another possibility to improve fuel economy is to accelerate less, and get a higher elapsed time in endurance. The tradeoff between minimizing
longitudinal acceleration and the increase in elapsed time requires analysis and knowledge of the car’s systems to judge. The last portion of fuel utilized is consumption that does not occur during timed on course driving. This can be largely eliminated with planning, but in the course of entering and exiting the course some consumption will always occur.

4.3. Quantified Vehicle Parameters

The definition of basic vehicle parameters for trending analysis of effects on performance is the next valid step in developing the conceptual design of the vehicle. Ultimately, the vehicle’s absolute performance is determined by the limits of the tires. The characteristics called out should thus be first order factors in the tires’ performance levels. As defined in many tire modeling systems, these tire attributes affecting performance are normal force, inclination angle, slip angle [15], and slip ratio.

Many of these metrics can be chosen during vehicle design and validated with common resources; more detailed specifications may require new facilities and analysis methods to assess them.

4.3.1. Suspension Parameters

The following chassis parameters chiefly affect the normal load on the tires. Vehicle functionality requires components which inherently add mass. Since tires are load sensitive, higher mass leads to higher load which leads to reduced effective acceleration of the mass. The location of the lumped mass of the vehicle with respect to the four wheels yields the static weight distribution. In reality the mass is distributed thus the implications of yaw inertia on the ability to maintain stability or destabilize the vehicle must be accounted for. The height of the lumped mass and track width produce weight transfer when the vehicle is accelerated laterally. Similarly, when wheelbase is considered with longitudinal acceleration, weight transfer results as well. This will be the roll and pitch distribution respectively; from a steady state perspective this distribution is based upon installed spring rates, jacking forces [2] via the kinematics package, steering input and its effect on jacking, and overall chassis
torsional stiffness. Dynamic behavior adds effects due to the damping and masses involved in the system.

Inclination angle and slip angle are also driven chiefly by the suspension design. These aspects will be dealt with in the logical design phase, as they cover behavior beyond the mere limits in this absolute performance design stage. To evaluate the magnitudes of the absolute performance envelope of the vehicle these will be taken to be at the optimal level. At a detailed design level the logical design elements will be realized through component designs with inherent compliance; compliant structures may differ greatly from the rigid body design approaches utilized in early development. The considerations of inclination angle and slip angle at a basic level drive parts of the subsequent studies and design of the vehicle’s camber compensation and steering systems.

4.3.2. Driveline Parameters

The tire’s available traction limits the overall positive longitudinal acceleration capacity. The tractive force is limited by the power output of the engine at speeds in excess of the transition from traction to power limited acceleration. The transition speed needs to be characterized based on available power and the traction limitations of the tire. Highlighted losses affecting the effectiveness of the power are driveline inefficiencies, tire rolling resistance, and the aerodynamic drag of the vehicle. Even though a tire may be operating in the power limited regime for straight line operation, due to combined loading and weight transfer on the rear tires which occurs during cornering, the longitudinal acceleration capabilities will be significantly degraded.

Since peak longitudinal acceleration of the vehicle is of concern, total mass is only part of the applicable inertia that is accelerated. The commonplace power to weight ratio is supplanted by power to total inertia. The total inertia utilized is dependent upon the gearing utilized, peak slip ratio, component inertias and the nature of the recorded power figure in terms of test procedure. Power
figures from dynamometer runs that involve acceleration, and not steady state power, will need the total inertia corrected by the amount that was accelerated in the testing procedure.

Non-peak power and its distribution to the rear wheels are essential to cornering limits. In a cornering situation when the forces at the wheels are resolved into lateral and longitudinal components, it becomes evident that the vehicle needs tractive forces in the longitudinal direction to maintain pure lateral acceleration without slowing the vehicle. Dependent upon power available and whether the differential can react it given weight transfer characteristics, the overall cornering capacity may be limited by the whether power can be effectively reacted through the tire’s contact patch. The differential type will also lead to yaw moment generation thus affecting the heading of the vehicle.

Overall power utilization during a lap is indicative of fuel consumption. While maintaining the same acceleration performance level and thus similar lap times, the relation between power and mass is obvious. With constant power, reducing mass will decrease fuel consumption in the corners and increase levels of acceleration on straights, thus reducing overall time at peak fuel consumption.

When examining the extent of the capabilities of the driveline for the purpose of defining absolute limits it is assumed that the power train can effectively deliver continuous response to demanded power from the driver. Making improvements in the peak power, area under the power curve, linearity of the overall peak curve and predictable power response to driver inputs are a basis for the detailed design and implementation of the engine system. From the power plant to the wheels is the drivetrain’s design domain; goals include the mitigation of inefficiencies and the effective distribution of motive force to the driven wheels.

4.3.3. Braking Parameters

Absolute braking performance will be limited by tire performance, weight transfer, aerodynamic drag, driveline characteristics, and the braking distribution chosen. The tire chosen will determine the
performance limits utilized. In the suspension design, the aspects governing weight transfer were already mentioned. Brake distribution will be chosen to increase tire utilization while making allowances for stability. The design of the brake distribution must take into account the total inertia of the vehicle, rolling resistance of the tires, braking torque of the engine, driveline inefficiencies, and the aerodynamic drag acting upon the vehicle. It is assumed that the brakes will respond consistently due to precautions taken in the detailed design phase to manage the thermal state of the system.

4.3.4. Aerodynamic Parameters

Although the effects of aerodynamics are decreased considerably at the lower speeds seen at competition, this is no reason to make misguided assumptions about their implications. Any vehicle traveling through the atmosphere has some aerodynamic effects acting on it. The chief concerns are to make valid estimates of aerodynamic properties, then apply them to the vehicle and observe their effect on the absolute performance limits. Applied to the previous sections, the effects are the normal load of the tires playing into all situations and the drag on the vehicle which influences acceleration and braking. Chief among the effects on tire normal load are the overall effect, which is the coefficient of lift, and the distribution of the lift which is affected by the coefficient of pitch moment. [2] The coefficient of drag dictates the force acting on the vehicle in the longitudinal direction. The coefficients function to dictate force via a function utilizing gas properties, velocity, and the acting area of the vehicle.

4.4. Parameter Trending Effects towards Performance Metrics

Key quantifiers of vehicle characteristics have been identified as vehicle parameters. As laid out, the qualitative vehicle characteristics have ramifications on the performance in the competition metrics, while being quantified by vehicle parameters. Trending behaviors can be easily identified. An apt descriptor for the interrelationships is presentable in a table form. A summary of the effects discussed
in the initial examination of the relationships between the performance metrics and qualitative vehicle characteristics is presented in Table 6. The distributions are based upon collected track data, generalized analysis of the events and observations of the vehicle during events.

<table>
<thead>
<tr>
<th>Competition Event</th>
<th>Points</th>
<th>Steady State</th>
<th>Transient</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Acceleration</td>
<td>Braking</td>
<td>Cornering</td>
</tr>
<tr>
<td>Skidpad</td>
<td>47.5</td>
<td>95.0%</td>
<td></td>
<td>5.0%</td>
</tr>
<tr>
<td>Acceleration</td>
<td>71.5</td>
<td>95.0%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Autocross</td>
<td>142.5</td>
<td>12.5%</td>
<td>8.5%</td>
<td>15.0%</td>
</tr>
<tr>
<td>Endurance</td>
<td>300</td>
<td>12.5%</td>
<td>8.5%</td>
<td>15.0%</td>
</tr>
<tr>
<td>Fuel Economy</td>
<td>100</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Totals</td>
<td>661.5</td>
<td>123.2</td>
<td>37.6</td>
<td>111.5</td>
</tr>
</tbody>
</table>

Table 6 – Competition Points Distribution Summary

The breakdown of performance points to different vehicle aspects is an approximation. Iteration of the distribution as new information is collected is necessary to improve both the vehicle and the tools themselves. These figures, especially the autocross and endurance distribution, are rough distributions based upon simplistic breakdown of the lap into constituent parts of the total time taken. The reality of these events, where performance behaviors are stacked back to back in succession, is that the need for examination of the interdependence of the different performance segments must be addressed. A set of trending relationships for an autocross or endurance lap are illustrated in Table 7.

<table>
<thead>
<tr>
<th>Mean Velocity Gain</th>
<th>Steady State</th>
<th>Acceleration</th>
<th>Braking</th>
<th>Cornering</th>
<th>Transient</th>
<th>Acceleration</th>
<th>Braking</th>
<th>Cornering</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>-</td>
<td>+</td>
<td>-</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>+</td>
<td>-</td>
<td>-</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
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<td>-</td>
<td>-</td>
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<td>-</td>
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<td></td>
<td></td>
<td>-</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 7 – Autocross and Endurance Performance Cross Correlation
Applying the correlations of Table 7 to correct the data in Table 6 in the Autocross and Endurance event yields an improved distribution among competition point categories. The effect of max steady state cornering was increased the most. This is due to critical importance of velocity in the apex of a turn effectively increasing all other portions’ mean speeds. The improved point distribution is presented in Table 8.

<table>
<thead>
<tr>
<th>Competition Event</th>
<th>Points</th>
<th>Steady State</th>
<th>Transient</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Acceleration</td>
<td>Braking</td>
<td>Acceleration</td>
</tr>
<tr>
<td>Skidpad</td>
<td>47.5</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>95.0%</td>
</tr>
<tr>
<td>Acceleration</td>
<td>71.5</td>
<td>95.0%</td>
<td></td>
<td>5.0%</td>
</tr>
<tr>
<td>Autocross</td>
<td>142.5</td>
<td>9.0%</td>
<td>6.0%</td>
<td>27.0%</td>
</tr>
<tr>
<td>Endurance</td>
<td>300</td>
<td>9.0%</td>
<td>6.0%</td>
<td>27.0%</td>
</tr>
<tr>
<td>Fuel Economy</td>
<td>100</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Totals</td>
<td>661.5</td>
<td>107.8</td>
<td>26.6</td>
<td>164.6</td>
</tr>
</tbody>
</table>

Table 8 – Revised Competition Points Distribution Summary

With a point distribution identified amongst the simplified vehicle characteristics, changes in quantifiable vehicle parameters can be tied to trends in the effect on the total points to be scored in the competition. These trending behaviors are evaluated under a simple set of assumptions. Any change in the vehicle parameters will show no effect upon the reliability, functionality or controllability of the vehicle system; as such the enabling parameters of the system will remain unaffected. Chief among the functionality category are slip angle management, inclination angle control and maintenance of the slip ratio of the tires; such performance governing characteristics will be evaluated in the logical and detailed design portion of the process. Changes to the parameters will thus only be evaluated on their effect upon global performance of the system. Of special note is the effectiveness of Total Mass upon all points in the dynamic scoring. The vehicle parameter performance trends are summarized in Table 9.

To advance from trending to developing sensitivities, the trend will be replaced by a delta of a specified quantity. This is easily stipulated for parameters such as masses, forces, inertias and power
ratings. A quantified delta definition for effectiveness is difficult, although possible sources would be control derivatives of driver inputs. The need to move to sensitivity analysis is seen when vehicle parameters are correlated extensively; seeing that the vehicle is one system, complex interrelationships are to be expected. A classic example is the reduction of peak power to gain a decrease in the vehicle’s mass and an increase in efficiency. Caution must be taken to insure that ample quality data to define the sensitivities has been collected. Extensive sensitivity studies must be carried out through simulation and testing for this process to bear lucrative and accurate conclusions. Until these studies are carried out, trending data can fill in a large hole within the conceptual design process by providing a logical path to decision making; ultimately the decision of how to develop the car as a whole and where to allocate team resources is one of the most important in the vehicle’s engineering life cycle.
### 4.5. Vehicle and Regulation Performance Metric Relationship

An analysis only accounting for trends will yield improved results, but will be lacking in the ability to locate a local optima of the system when dependent variables are considered without the use of sensitivities. Utilization of both personal vehicle and competitor statistical information will improve on the optimization routine. The lack of quantifiable understanding of the problem that the car was designed to undertake served as an irreparable inadequacy during the design event at competition. The anecdotal reasoning utilized for the high level system design choices paled in comparison to the approaches leveraged by other competitors. Trends are better than anecdotes; as measuring validating data for an anecdote is difficult to say the least. Trends are the first step. Experimental validation of trends will yield the data points to develop sensitivities. These sensitivities can be leveraged in the next iteration and continually improved upon with procedures to guarantee knowledge retention on the

<table>
<thead>
<tr>
<th>Competition Point Allocation</th>
<th>Steady State</th>
<th>Transient</th>
<th>Parameter to Point Effectiveness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter Trends</td>
<td>Acceleration</td>
<td>Cornering</td>
<td>Acceleration</td>
</tr>
<tr>
<td>(+/-)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vehicle Parameters</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>- Total Mass</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>- Yaw Inertia</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>- Rotating Inertia</td>
<td>1</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>- CG Height</td>
<td>(1)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>+ Peak Power</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>+ Off-Peak Power Effectiveness</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>+ Driveline Efficiency</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>+ Torque Distribution Effectiveness</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>+ Brake Distribution Effectiveness</td>
<td>1</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>- Tire Rolling Resistance</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>- Aerodynamic Drag</td>
<td>1</td>
<td>(1)</td>
<td>0</td>
</tr>
<tr>
<td>- Aerodynamic Lift</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 9 – Vehicle Parameter to Competition Point Effect
team. A successful vehicle will share the same common performance characteristics while being distinctly different in the detailed design elements when compared to its predecessor. The concepts and baseline parameters that make a vehicle successful do not change drastically year to year; a system to perpetuate and iterate this knowledge within the team must be implemented to achieve continued success in spite of personnel changes.
5. Chassis Development Tools

5.1. System Parameters

Systems analysis provides a basis for the design and development of the chassis system of the vehicle. In a mechanical system, inertia, damping and springing dictate behavior and response of the state variables to forcing functions. Although not intended, many aspects of the system are ignored in the design process. They may be simplistic parameters, but due to a lack in facilities and background knowledge they are assumed incorrectly or oversimplified to the point that their contribution to the performance of the system is seriously skewed.

Certain properties get addressed year to year due to the simplicity of their existence. It is obvious to look at the vehicle and recognize components such as springs and dampers. The car can be put on a set of scales to define the total mass and its approximate location in the top plan view of the vehicle. All of these simple parameters are of the utmost importance; their intricacies are glossed over depending upon the knowledge and time available to the designer.

Many parameters are recognized and estimated, but never quantified adequately in either the design or evaluation stage of vehicle development. Lack of facilities and processes for determining many parameters leads to them being disregarded by the project team. The quantities needing to be defined are not overly complex; they may need test rigs, data collection capabilities and data processing development to accurately assess them.

A reduction in the needed parameters for chassis and suspension by eliminating redundancies can ease the work load of the designer greatly. To summarize, based upon ideas of linear systems analysis, chassis parameters can be divided into sub groups. Following classic mechanics and kinematics they can be divided into force, mass, stiffness, damping and geometrical parameters; also included are
the rotational counterparts of torque, inertia, rotational damping and rotational stiffness. The latter three subjects of stiffness, damping, and geometry have always been focused on as design and validation elements in the execution of the car each year. The former two are accounted for with rough approximations when designing the stiffness, damping and geometry of the vehicle; they are rarely validated and quantified fully with their full effect on vehicle dynamics not realized.

5.1.1. **Masses and Inertias**

When a simplified full vehicle linear model is considered, four unsprung masses with a single degree of freedom each and a sprung body with 3 degrees of freedom, pitch, roll and heave, can be considered. The concept of estimating mass is simple to accomplish with a complete model or a physical measurement. The measurement of the center of mass location needs to be accomplished as well. The associated inertias with the sprung body pitch and roll motions pose the necessity of specialized equipment and procedures to verify physically. The relatively large mass of the sprung body coupled with the need to include the driver as a major component of this mass presents a challenge to the testing method.

The measurement of the polar moment of inertia of the entire car presents a similar challenge in size and scope to the roll and pitch requirements, but in the sense of the yawing the vehicle. This parameter is essential for understanding vehicle dynamics characteristics and performance. Again, since the driver makes up a large portion of the vehicle’s mass, they must be accounted for in the measurement.

Modern suspension and vehicle dynamics development advocates the use of multi body dynamic simulations. With such an approach, the mass and inertias of each vehicle body need defined. While many components are adequately modeled to give a reasonable estimate of this quantity, the
sprung mass, due to incomplete or inaccurate modeling is often not. Quantifying different driver’s effects is also of question for a model based estimate.

5.1.2. Springs and Compliances

Commonly the greatest design emphasis is placed upon the development and implementation of springs in the chassis design. Although they serve as a mere contingent of the design as a whole, the fact that their rate is easily obtainable and explored in numerous texts brings them to the forefront of suspension development. Topics of common exploration include the rates for the sprung and unsprung degrees of freedom, the frame torsional stiffness and tire rates.

Among attributes that are acknowledged, but not quantified or compensated for, are the compliances of all components within the system. Just as design goals for wheel rates and frame torsional stiffness are chosen, so should the overall compliance stack up affecting the actual toe and camber angle of the tires. Specialized fixtures and test rigs can be utilized to quantify the compliance properties of both individual components and built up vehicle assemblies. Kinematics and compliance rigs are one example of such devices. Bearcat Motorsports collected data on Goodyear’s Suspension Parameter Measurement Machine (SPMM) in Akron, Ohio during the 2006 and 2007 vehicle revisions. Simpler fixtures could be developed for use at the University, but as with any compliance data, one must understand the implications and limitations of the method to comprehend the practical meaning of the collected parameters.

5.1.3. Damping and Friction

To continue the analogy to a second order linear system, damping within the vehicle is considered. Commonly, coil-over dampers are utilized on each corner of the vehicle; other installations are possible as well. As a key term in the vehicle’s linear system response, key characteristics such as
rise time, percent overshoot, and settling time can be calculated. Although linear analysis is utilized to estimate the system’s response, many non-linear characteristics related to friction exist.

Plain bearings are common in the suspension linkages of FSAE vehicles. Also, internal friction within dampers plays an important role in response. Generally, these non-linear effects are not quantified, nor their impact on the vehicle’s performance gaged. Design goals usually take the approach of minimalizing friction within the system, thereby making the linear system analysis more accurate when compared to the actual system behavior.

5.1.4. Forces and Moments

Forces and moments compose the most important element of the vehicle’s function, accelerating the vehicle and causing heading changes. The actual number of forces acting on the vehicle is quite limited. Body forces are imparted on the vehicle through accelerations acting on the masses via d’Alembert’s principle and aerodynamic forces. The location of the center of mass is of critical importance to these body forces. Quantifying the aerodynamic properties of a vehicle is an involved process beyond the scopes of this discussion. It must be realized that they do have a direct effect on the vehicle’s performance. Following the convention of SAE Aerodynamic Axis System [2], aerodynamics imparts a force or moment to each of the vehicle’s six degrees of freedom.

The remainder of the forces on the vehicle act through the tire contact patches. The forces and moments developed by the tires are presented via the Tire Coordinate System. Forces developed internally to the vehicle, such as braking and driving torques, ultimately resolve themselves via the tire forces. Many factors play a role in determining the force and moment makeup of each tire. A large amount of data has to be collected and processed to yield an accurate representation of the tire’s performance. The FSAE Tire Testing Consortium in cooperation with TIRF at Calspan has helped to fill this role via a distributed funding approach among member schools to carry out this testing. [18]
Although data is available, to process and comprehend its implications can be a demanding, but important, process.

5.1.5. Areas of Focus for Development

Considering the normal focuses year to year in the FSAE program, development of techniques and rigs to ascertain the vehicle’s mass and inertia properties along with development of capabilities to quantify its tire forces were chosen as areas to focus. Specifically, quantifications of the vehicle’s yaw, pitch and roll inertia will be obtained. With the release of TTC Round 4 [19] and Round 5 [20] data, no fitted models or processing was provided by sponsors of the consortium in Rounds 1 through 3 [21]. To gain insights from the data that are applicable to the vehicle’s design and prepare a fitted model for vehicle dynamics analysis, data processing scripts will be developed. In summary, the areas of focus considered for this thesis are the following:

- Yaw Inertia Measurement
- Pitch and Roll Inertia Measurement
- Tire Data Processing

5.2. Tool and Method Development Concerns

Due to design feedback in the 2008 FSAE competitions, realizations about the team’s lack of simulation and quantification capabilities came to the forefront. Students understand and can apply the principles of physics and mechanics to a vehicle from their studies. Lack of facilities and knowledge to ascertain key parameters is a barrier to entry to necessary vehicle dynamics simulation though. It was recognized that ascertaining the inertias at work within the system and quantifying the tire data into a useable model would elicit further design and simulation development within the team.
5.2.1. Yaw Inertia Measurement

Initially exploration was done to ascertain possible methods for determining the yaw inertia of both the whole vehicle and the sprung mass only in such a way that the driver could be included. It was also a goal to have the rig be capable of measuring other vehicle components’ inertias. As a measurement instrument, the rig would need to be accurate, precise, repeatable, observable and easy to set up.

To satisfy the inclusion of both the driver and complete chassis, the rig would need to support masses in excess of 700 pound mass for heavier vehicles and drivers. To accommodate the driver, the attitude of the vehicle would need to be similar to running conditions. Also, the support of the vehicle would need to be steady to ease driver changes and adjustments to the position of the mass. It is an utmost priority to not subject the driver to discomfort or unsafe conditions during the test.

Three concepts were examined as possible approaches for the test apparatus. A common approach, taking advantage of periodic motion, is the trifilar pendulum [4], [22]. This pendulum necessitates at least 3 supports above the testing area to suspend the test platform. For ease of computation, the position of these cables should form an equilateral triangle and be equidistant from the center of mass location. Positioning the cables at a large enough distance to accommodate a FSAE car results in a large platform. Also, the need for a roof or ceiling of necessary size and load capability needs to be found as well. When hanging, the rig and vehicle are not steady; thus, it is difficult to enter and exit the vehicle as a driver as well as adjust the car’s position to center its center of gravity on the rig. Exciting the rig, due to its size, is also cumbersome. The rig’s versatility suffers in the limitations of the realistic range of inertias that can be measured due to the platform’s size respective to the test object.
The next concept considered is inspired by the apparatus available on Anthony Best Dynamics Suspension Parameter Measurement Machines. As part of their kinematics and compliance rigs, the car is fixtured to a center table capable of pitching, rolling and heaving the vehicle. Since the table is instrumented to measure forces and displacement, the table can accelerate the car in different degrees of freedom. Knowing the mass of the vehicle, the accelerations and the resultant forces of the table, parameters such as yaw inertia, pitch inertia, roll inertia and center of gravity height can be extracted. To acquire this machine or something remotely similar is outside the bounds of cost and the necessary permanent facilities to house such a machine. The approach of utilizing known forces and acceleration to compute the inertia of an object led to a viable solution.

Finally, a pivoting table approach was considered. A low friction table, with a known applied torque would yield an acceleration that is easily computed to find the inertia. The same table, attached to a known rate torsional spring would yield the same information via the natural frequency of the system. This approach has distinct challenges with the need for a rotational pivoting table with minimal friction. Also, a constant force application to generate the torque with minimal or known losses is also required. A transducer is also required to measure the displacement of the table.

5.2.2. Pitch and Roll Inertia Measurement

The choices for a pitch and roll measurement apparatus were more limited due to the necessity of keeping the car or sprung body in a reasonable attitude to accommodate the driver. As discussed previously, a kinematics and compliance rig can perform these measurements. Again, this approach is not feasible.

Successful implementations have utilized a rigid pendulum approach. Such approaches have the added benefit of the capability of measuring the center of gravity height as part of the process. To ascertain both inertia and center of mass positions, design challenges include making a large, rigid
platform for the vehicle, installing an angular displacement transducer to accurately measure motion and manufacturing an accurate low friction single point pivot to swing on.

5.2.3. Tire Force and Moment Estimation

A large amount of force and moment data is collected for tires as a function of the Tire Testing Consortium [18], [25]. Examining raw data is of interest from a conceptual point, but it is of little use to vehicle simulations where the running conditions within the discrete set of points tested requires a fit or interpolation to give reasonable output data. Numerous other parameters in the data affect the scaling and phasing of the force and moment data to inputs. Realizing the differences between steady state and transient effects is of critical importance to simulation.

The data that is available for fitment consists of a test matrix for 5 normal forces and 5 inclination angles for no drive slip angle sweeps of -12 to +12 degrees. The brake drive schedule for each test consisted of 3 static slip angle settings, 4 normal forces and 3 inclination angles for slip ratio sweeps of -25% to +15%. These tests were repeated for 4 pressures on each tire and rim combination barring the combination debeading during the procedure [19], [20]. For a given run, data channels summarized in Table 14 are logged. The parameters are referenced in the TIRF capabilities report [26]; the data conforms to the SAE tire axis system.

The primary fitting goal is to produce a continuous model over the viable operating range of the tire. This model will satisfy steady state needs, providing forces and moments in the tire coordinate system for given operating conditions. The empirical model utilized should provide a proper fit to the collected experimental data. Several simple models exist [3], but they lack the flexibility to properly match the collected data. Historically, table based approaches utilizing interpolation were also used, but required large amounts of experimental data and voluminous databases [2]. For this application of
fitting, Pacejka’s ‘Magic Formula’ [15] has been shown to be capable of fitting a diverse set of data via its coefficient based model.

The base Pacejka formula is utilized by numerous models. Fitting equations for the formula are published by Pacejka [15], Stackpole Engineering Services [21] and Adams/Tire documentation [27]. To allow use of the fitted parameters in both Matlab based scripts and the Adams Car application, the ADAMS/PAC2002 tire model will be utilized [21], [27]. Advantages of the Pacejka based model include being fully continuous, easy calculation of cornering stiffness, simple calculation, scaling variables and providing fitment equations for all forces and moments of the tire.

Other key parameters can be fitted from the data as well. Utilizing the rolling radius and normal load channel, the testing provides data to fit the unloaded radius and tire vertical rate at both static and dynamic conditions. Rate testing data is also collected at multiple inclination angles and inflation pressures.

Unlike typical road tires, racing tires are sensitive to the temperatures of the surface and body of the tire. Useful anecdotal evidence and data is presented in several sources [16], [17]; the TTC data sets includes data channels to quantify these effects. At the initial warm up period of the test for each tire, a continuous slip angle sweep of the tire was performed for 8 sweeps to collect data on the thermal properties of the tire [19]. Round 5 data increased the sweep count to 12 repetitions [20] to better characterize higher temperatures. Ambient, road surface, tread inner, tread center and tread outer temperature are collected as shown in Table 14. Since the traction is temperature sensitive it is also necessary to understand the effects that occurred during the slip angle and slip ratio sweep testing. The cold to warm tests quantify this transient effect.

The final effect extracted from the data is an indication of how the tire behaves in response to transient loading. During the testing, slip angle and slip ratio are swept at a fixed rate. Thus the tire is
loaded and unloaded, and when thought of as a simple mechanical system composed of masses, springs and dampers it will introduce a phasing between the input and output of the system. This is quantified in relation to the tire’s relaxation length, $\sigma$ [15].
6. Yaw Inertia Rig Design and Implementation

6.1. Trifilar Pendulum

Initially, citing experiences in the structures and motions lab at the University of Cincinnati, a trifilar pendulum was constructed and hung. To achieve a setup that was sufficiently large and to have the needed load capacity, a location at the rear of the machine shop in 407 Rhodes was selected. Accommodating the vehicle drove the installed radius of the cable stays to be 1.22 meters (48 inches). Supports were anchored to the Unistrut standoffs in the poured ceiling. 3/32 inch nominal aircraft control cable assembled with compression sleeves supported the platform at a length of 4.11 meters (13.5 feet). Each cable assembly is rated at 4.1 kN (920 lbf) load, providing adequate load bearing capability.

![Trifilar pendulum platform design](image)

The platform consists of an aluminum spreader plate supported by a modular steel and aluminum structure tying it to the outer cable locations. The model estimated the platform inertia at 8.15 (kg*m²). The platform model is shown in Figure 3. Utilizing Equation 8 and data collected via the
rig from the 2008 chassis, the data summarized in Table 10 was calculated. The values found for the platform, empty chassis and chassis with driver are reasonable given estimates from the platform model and a 75% complete full car assembly that had a yaw inertia of 81.2 (kg*m²).

\[
J = \frac{m \cdot g \cdot r^2 \cdot T^2}{4 \cdot \pi^2 \cdot L}
\]

*Equation 8 - Trifilar Inertia [22]*

<table>
<thead>
<tr>
<th>Item</th>
<th>T (sec)</th>
<th>Mass (kg)</th>
<th>J (kg*m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Platform</td>
<td>1.487</td>
<td>46</td>
<td>9.1</td>
</tr>
<tr>
<td>2008 Car</td>
<td>2.137</td>
<td>238</td>
<td>88.4</td>
</tr>
<tr>
<td>2008 Car + Driver</td>
<td>1.984</td>
<td>350</td>
<td>114.4</td>
</tr>
</tbody>
</table>

*Table 10 - Trifilar Pendulum Results*

Overall, the trifilar pendulum approach produced satisfactory results. The downsides come as a consequence of the size and masses of the objects involved. It becomes unwieldy to introduce an excitation to the pendulum without coordination amongst multiple individuals. Also, the loading of the vehicle and driver can be trying due to the inherent instability of the platform involved. The amount of space and required support structure for the device proved to be the largest challenge. Due to the combination of these challenges, a smaller, more controlled and less facilities dependent approach was developed.
6.2. Air Ride Yaw Inertia Rig

\[ T = J\alpha \]

*Equation 9 - Simplified Angular Acceleration*

Though seemingly simple, Equation 9 forms the basis for the developed measurement method. If the torque applied to a body and the subsequent angular acceleration is known, the rotational inertia is easily calculated. The apparatus must satisfy the necessity to quantify the total torque applied to the inertia and the response to that applied torque. To achieve success, the friction forces which are not observable and capable of being isolated need to be minimized.

![Figure 4 – Air Bearing Design – Dimensions in Inches](image)

To produce a rotational pivot with enough load capacity to support the vehicle while having minimal friction an approach utilizing an air bearing was utilized. The maximum operating pressure was taken to be 552 kPa (80 psi) due to the shop air supply. From this limit, the main piston’s diameter was chosen to be 101.6 mm (4 inches) to provide a load capacity of approximately 4.45 kN (1000 pounds) for the base piston. A relatively small size was also chosen to help achieve dimensional tolerances due to the machining equipment available and to utilize material that was on hand during development. Pressure in the base piston is controlled via a valve regulating flow out of the lower manifold. Originally,
the intent was to supply pressure to both the bottom manifold as well as the radial manifold feeding the side ports. It was found that merely regulating the exit of air at the bottom manifold, with the pressure being supplied by the radial manifold, yielded satisfactory control.

Controlling the pistons radial position is the pressure feed to the radial manifold. The manifold is pressed over the base to establish the air supply to the 8 radial locating jets. The air supplied to the side of the piston exits into both the base piston area and out the upper face to atmosphere. Initial difficulties arose from the surface finish of the side walls. Polishing the surface alleviated drag issues. Also, initially the annular ring that forms the radial manifold produced an uneven top surface due to not being pressed on straight. Post-machining of the assembly yielded a flat top surface. Although not the intent of the top flange, when flattened, it introduced a self-regulating effect for the vertical position of the piston. As such, the needed precision of the pressure control was greatly reduced. The top surface also indicates mass alignment on the rig. A mass too far out of center will cause the top surface to drag; the taper of the gap of the top surface indicates which direction to shift the mass. To accommodate varying object masses, the air inlet to the radial manifold utilizes a pressure regulator.

Operating at a pressure in excess of the load distributed over the piston area causes the platform to lift uncontrollably. To constrain the platform piston, a bolt threaded into the piston through the base keeps the platform in check during over pressurization. This eliminates the possibility of damage to the rig and test specimen due to over extension of the piston. The bolt head is covered by the bottom manifold, thus sealing it within the system. Additionally, dependent upon the size of the object being tested, an additional spreader plate can be added to expand the support area of the table. The spreader is a piece of 19 mm (0.750 inch) thick aluminum with a diameter of 914 mm (36 inches); a recessed pocket centered on the part locates the plate on the piston.
To excite the system with a constant torque level, a dropping mass was utilized as a counterweight. Dependent on the inertia and the desired observed acceleration rate the dropping mass can be easily changed. During trials of different sized components, masses of 5 grams to 2 kilograms were utilized successfully. The force was applied to the table by rapping a piece of fishing line around the 7 inch machined diameter of the support table. The line then goes over a pivot at the edge of the table to support the suspended mass. The amount of tension in the string can be questionable due to the characteristics of this pivot.

Two different pivot styles were utilized. The first was a small, non-sealed deep grooved ball bearing. The second was a smooth piece of wire acting as a simple sliding pivot. Both instances have resistance that should be accounted for in order to reach an accurate estimate. To quantify the friction in the bearing pivot equal masses were hung against each other, then additional mass was added to one side to quantify the needed force to cause both breakaway and continued running. The setup can be seen in Figure 5. The test was completed for a load of 2 kilograms total. The added mass to overcome running drag was 19 grams. Further tests for other loads still need to be completed.
The sliding pivot point was more methodical in its quantification. By completing the test procedure for a known inertia, in this case the platform itself, the effect of the friction can be isolated. The platform was chosen due to it being a machined element with known dimensions and mass; the inertia was evaluated utilizing the solid model of the platform. Under the assumption of kinetic friction, that the drag from the friction is proportional to the mass utilized, a value for $\mu$ can be found. Since the force accelerating the platform is less than the attached weight, the difference is attributed to the drag of the pivot. The computed effect of sliding friction is shown in Table 11; a logical explanation for the increase in $\mu$ with increase in normal load is the alteration in the sliding contact area of the fishing line, as it conforms to a greater contact area with the pivot at higher loads. It can be surmised that once the weight reaches a level where the line is in maximum contact with the pivot the $\mu$ value should stabilize. This was not validated during testing. From the results seen, a sliding pivot was a viable and quantifiable procedure for the test, especially with the use of small excitation masses when measuring engine components.

<table>
<thead>
<tr>
<th>Dropped Mass (g)</th>
<th>$\mu$ Correction</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.191814285</td>
</tr>
<tr>
<td>10</td>
<td>0.202511861</td>
</tr>
<tr>
<td>20</td>
<td>0.213894149</td>
</tr>
</tbody>
</table>

Table 11 - Sliding pivot friction estimation

With the pivoting table and force application method determined, the last need is to record the rotational position to determine the acceleration of the platform. As a goal, the sensing method should be non-contact as to not induce drag and have reasonable resolution. For ease of use a Sharp OPIC Photointerrupter, model GP1A05, was utilized [28]. The sensor was sourced from a used printer, although it is commercially available at a low price. A small enclosure to house the sensor and provide a BNC connector was manufactured. The encoder wheel was generated via a website program [29] and saved to a simple graphics file. A laser printer was utilized to print the wheel onto a transparency. The
wheel was then trimmed and adhered to the underside of the floating table. The encoder wheel was setup with bands of 2 degrees width for a reasonable level of resolution. The installed setup as used can be seen in Figure 6.

![Figure 6 - Photointerrupter Installation](image)

![Figure 7 - Characteristic photointerrupter signal](image)
When the photo gate is interrupted a high voltage is outputted. From observations of the data, as presented in Figure 7, the transients of the rising and falling edge differ in response; when utilizing the data only the rising edge was utilized to account for this transient difference. Thus, a usable signal is recorded for every four degrees of rotation due to the encoder wheel setup. Utilizing Matlab, the time of each rising edge is extracted from the data stream and assigned an appropriate displacement. Operating under the assumption of constant acceleration, the derived displacement curve should be a simple second order function; the data should form an exact parabola. Figure 8 shows the second order fit of the data along with the numerical derivatives. The error of the second order fit is small. Also, the second numerical derivative is relatively constant, validating the earlier assumption.

![Polynomial Fit of Data](image1)

![Position Derivatives](image2)

**Figure 8 - Table displacement curve fitment**

With the input torque and associated angular acceleration known, the inertia is easily computed. To validate the setup each time, the platform can be utilized as a known inertia to validate the output. Also, since the test can be repeated easily, multiple masses can be utilized to validate if the system is behaving linearly. Example inertia estimates collected for vehicles and internal engine
components of the Honda CBR 600RR appear in Figure 9 and Figure 10 respectively. Validation tests of the platform alone were within 0.4% of the modeled estimate of 3.619 (kg·m²).

<table>
<thead>
<tr>
<th>Object Estimate (kg·m²)</th>
<th>0.62 kg</th>
<th>1.07 kg</th>
<th>1.52 kg</th>
<th>1.97 kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>2012 Sprung Mass &amp; Mass D</td>
<td>79.5</td>
<td>79.5</td>
<td>79.1</td>
<td>80.9</td>
</tr>
<tr>
<td>2005 Car (Empty)</td>
<td>116.1</td>
<td>116.1</td>
<td>116.1</td>
<td>116.2</td>
</tr>
<tr>
<td>Empty Rig with Spreader</td>
<td>3.62</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Figure 9 - Vehicle inertia estimates*

<table>
<thead>
<tr>
<th>Object Estimate (kg·m²)</th>
<th>5 g</th>
<th>10 g</th>
<th>20 g</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crank Assembly</td>
<td>0.0089</td>
<td>0.0084</td>
<td>0.0085</td>
</tr>
<tr>
<td>Clutch &amp; Primary Shaft</td>
<td>0.0108</td>
<td>0.0105</td>
<td>0.0100</td>
</tr>
<tr>
<td>Counter 1st</td>
<td>4.50E-04</td>
<td>4.05E-04</td>
<td>3.02E-04</td>
</tr>
<tr>
<td>Counter Shaft</td>
<td>6.08E-04</td>
<td>4.27E-04</td>
<td>6.22E-04</td>
</tr>
</tbody>
</table>

*Figure 10 - Engine internal component inertias*

The end user has a large degree of freedom to attain a good fit of the data. Whether an approach utilizing natural frequency or time history system response is used, high velocities of the test object should be avoided. The larger the item, the more effect that damping from aerodynamics could pose on the response. A large object has surface areas normal to the motion at greater distances to the rotation axis of the object, compared to a smaller part. Choosing areas of the response where velocity is minimal reduces this effect. Since the position measurement is discrete, as the maximum velocity observed reduces the number of points available for fitment decreases. As such, the tradeoff between velocity minimization and maximizing observed points to fit to is up to the end user.
7. Pitch and Roll Inertia Rig Design and Implementation

7.1. Rig Concept Development

Two methods for measuring the pitch and roll inertia of the vehicle at small levels of deviation from level were outlined previously. No matter the approach, the location of the center of gravity must be accurately ascertained to compute the principal inertias. As such, the integration of the procedure to measure the center of gravity into the inertia estimating equipment is prudent. Since the approach utilizing a kinematics and compliance rig is not available, a pendulum was selected as the path for development.

Figure 11 - SEA Limited Vehicle Inertia Measurement Facility [30]

To avoid similar complexities to the trifilar pendulum setup, a rigid pendulum mounted to a table or the ground was sought. One inspiration for the setup of the apparatus was drawn from a tour of SEA Limited in Columbus, Ohio [24]. The rig is shown in Figure 11, a photograph from an analysis
completed for an article in Car and Driver Magazine. The test facility incorporated air ride pivot bearings supporting a large rigid platform. To ascertain the center of gravity height, mass is added to the platform and the displacement recorded. The weights are visible in Figure 11 as dumbbell weights in the lower left corner hanging on a bar off of the side of the platform. Computing the center of gravity height is a simple force balance statics problem. Calculation of the inertia is completed via Equation 10, utilizing the period of oscillation, object mass and the length from the pivot to the center of mass.

\[ J_i = M_i \cdot g \cdot L_i \cdot \left( \frac{T_i^2}{4\pi^2} - \frac{L_i}{g} \right) \]

*Equation 10 - Inertia Computation for Simple Pendulum [4]*

Chief concerns for the execution of such a rig were the platform design, support structure, low friction pivot and the installation of the angle transducer. The platform needs to be large enough to accommodate the sprung chassis in both pitch and roll configurations, with the capability to house the entire vehicle in at least one setup to measure the total vehicle center of mass height. Additionally, the object needs to be handled in both manned and unmanned states. As such, the platform needs to take the plan view and weight distribution of the vehicle into consideration. Initially a platform consisting of a steel space frame was designed. The construction of this approach would be difficult in manufacturing, welding and fixturing. A simple approach consisting of a wooden joist structure was chosen for its ease of manufacture, low cost and ease of modification. Steel support staves to tie the platform to the tension support struts are integrated in the platform.

Connecting the platform to the pivot hubs are simple tension struts. By being bolted to the platform, the struts can be shifted between sides of the platform to accommodate changes between measurements of pitch and roll quantities. The struts are axially bolted to the platform to allow adjustments of support tension and bolted in shear at the pivot hub. The pivot is supported to the base
T-slot table via a triangulated support structure. As with the struts, the supports are made to be repositioned dependent on the testing procedure.

The initial design for the pivot sought to emulate the air bearing present on the yaw inertia rig. Due to the design of the isolation of the pivot to not induce angular misalignment, a moment was put into the pivot. The moment would cause binding of the bearing surface, thus iteration on the pivot design was needed. A simpler method, as seen in pivots of scales and clocks was utilized. A knife edge pivot is simpler to manufacture and implement. It is also tolerant of the induced moment and not as sensitive to changes in running conditions. Although the air bearing design was abandoned, most parts of the pivot hubs were utilized in the knife edge changeover with no modification.

Finally, the angular position needed to be recorded via a transducer. A potentiometer was chosen due to its availability and ease of implementation. A novotechnik SP2831 with an actual electrical angle of 308 degrees was utilized. Of importance to this installation is the low operating
torque of 0.5 Newton centimeters [31] as added friction will alter the test results. Figure 12 shows the setup of the sensor aligned with the pivot of the test apparatus. Calibration of the transducer is achieved by adding an arm to the sensor shaft that allows four calibration points, one every 90 degrees when the arm is aligned with the housing’s mount holes. The mount to the rotating assembly is stiff in torsion while allowing flexibility accommodates slight radial misalignments.

7.2. Experimental Implementation of Inertia Rig

The entire apparatus was installed on the large T-slot table that the team uses for fixturing, alignments and corner weighting. Once setup, the positions of the points where mass will be added are measured to allow the computation of the center of mass. Calibration of the fixture entails measuring the total mass of the rotating portion of the test apparatus, measurement of the mass addition points relative to the center of rotation and benching in the angular sensitivity of the potentiometer. In the pitch and roll configuration, the platform mass was, respectively, 135.6 and 132.2 kilograms. Before measuring any object, the swung platform center of mass position and inertia are found. After being quantified, the procedure is repeated for any object added to the platform. Representative results from the 2011 chassis for pitch and roll situations are listed in Table 12 and Table 13 respectively.

<table>
<thead>
<tr>
<th>Object</th>
<th>Mass (kg)</th>
<th>Inertia (kg*m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pitch Table 5/24/11</td>
<td>135.6</td>
<td>66.5</td>
</tr>
<tr>
<td>2011 Car</td>
<td>216.4</td>
<td>83.0</td>
</tr>
<tr>
<td>2011 Car &amp; Mass A</td>
<td>304.4</td>
<td>120.1</td>
</tr>
<tr>
<td>2011 Car &amp; Mass B</td>
<td>336.1</td>
<td>125.3</td>
</tr>
<tr>
<td>2011 Car &amp; Mass C</td>
<td>308.9</td>
<td>117.9</td>
</tr>
<tr>
<td>Pitch Table 6/17/11</td>
<td>135.6</td>
<td>66.3</td>
</tr>
<tr>
<td>2011 Sprung Chassis</td>
<td>155.6</td>
<td>44.5</td>
</tr>
<tr>
<td>2011 Sprung &amp; Mass B</td>
<td>274.4</td>
<td>84.0</td>
</tr>
</tbody>
</table>

Table 12 - Representative Pitch Inertia Results
<table>
<thead>
<tr>
<th>Object</th>
<th>Mass (kg)</th>
<th>Inertia (kg*m$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll Table 6/21/11</td>
<td>132.2</td>
<td>23.5</td>
</tr>
<tr>
<td>2011 Sprung Chassis</td>
<td>154.3</td>
<td>8.3</td>
</tr>
<tr>
<td>2011 Sprung &amp; Mass B</td>
<td>276.5</td>
<td>14.6</td>
</tr>
<tr>
<td>2011 Sprung &amp; Mass C</td>
<td>246.6</td>
<td>11.9</td>
</tr>
<tr>
<td>2011 Sprung &amp; Mass D</td>
<td>230.7</td>
<td>10.6</td>
</tr>
</tbody>
</table>

Table 13 - Representative Roll Inertia Results

Examination of differing test methodologies highlighted critical features of the system and their effect on the validity of the final data. Initially, the time response of a high amplitude excitation was recorded. The high amplitude response, presented in Figure 13, shows a characteristic ring down. This response has better signal to noise due to the higher amplitudes; quantifying the peaks and crossing points is more exact and repeatable due to the better signal. The decrease in the average period of oscillation with decrease in response amplitude is notable, as it continues into Figure 14. During the low amplitude response the ring down lessens at small oscillations. With reduced damping, the period of the oscillation decreases and provides a better approximation of an undamped system; thus better quantifying the period of oscillation for use in Equation 10 when computing the inertia of the system. It can also be surmised that at smaller perturbations, the small angle approximation is obviously satisfied to a greater extent. As such, the general recommendation is to use as low of an excitation as possible.
Figure 13 - High Amplitude Oscillation Test Data

Figure 14 - Low Amplitude Oscillation Test Data
8. Tire Data Processing and Presentation

The data available from the FSAE Tire Testing Consortium has varied from round to round. In rounds 1 through 3, fitted models were provided by sponsors of the consortium [21]; in the more recent rounds, 4 and 5, only the raw data has been provided to members. As such, analyzing, processing and fitting the raw data becomes necessary for teams if they want to utilize the data in vehicle dynamics, simulations and other analytical endeavors.

8.1. Pre-Processing

Raw data is provided in individual run files that consist of one test’s command file. Generally, files can contain static spring rate tests, dynamic spring rate tests, cold to warm break in cycle tests, conditioning and a test matrix of inclination angles and normal loads for a given cornering or brake-drive test [19] [20]. The test files are truncated to eliminate excess data between test runs; these areas, if available, contain extra data pertaining to the cooling of the tire between each test run to maintain a reasonably consistent tire surface temperature between each successive test in the matrix. The test procedure is a series of commands driving the testing machine to achieve a controlled series of input variables. Inflation pressure, normal load and inclination angle are maintained at a target level for each test sweep. The swept variables include slip angle for cornering testing and slip ratio for brake-drive testing; during combined cornering and brake-drive testing a constant slip angle is maintained during the slip ratio sweeps. Isolation of variables is the chief goal when executing the test procedure to allow fitment and extrapolation of the data.
The raw data for each run is a continuous time stream of at least one testing procedure. For the data in Figure 15, which comprises a full set of cornering conditions for an inflation pressure of 69 kPa (10 psi) on the given tire, the 25 runs covering the testing matrix are presented in one continuous file. For the same tire, the test procedure is repeated for all inflation pressures; 55, 69, 83 and 97 kPa (8, 10, 12 and 14 psi) inflation pressures were all tested during the cornering and brake-drive procedures in Round 4. Whether analyzing the raw data directly or fitting a linear model for a given set of inputs, the organization of data for a single tire and rim combination from the Consortium is overly complex; the data is a continuous stream with no distinction between sets and is typically spread across 12 datasets to include all inflation pressures for the cornering and brake-drive schedules. Also, included within the stream, dependent on the command file, are conditioning loops, tire rate tests and heat cycling studies as well that should only be utilized when needed.

Separating all of these data components into an easily referenced and accessed format allows both examination and processing of the raw data to be expedited. To deal with the multiple test types...
and the matrix of variables, a nested data structure was implemented to organize the data. For tire rate tests and cold to warm runs, each set of channels is recorded to a separate structure for the completed procedure. For example, individual tire rate tests are called out in the structure by ‘Structure.SpringRate.TestN’. The term ‘Structure’ is a generated unique name referencing the tire and rim width utilized for the test. In the case of repeated procedures, such as cornering and brake-drive tests, the structure is broken down by the independent variables for the respective test. These independent variables are inflation pressure, inclination angle and normal load. The base tests with this structure are cornering tests and brake-drive testing at constant slip angle of 0, 3 and 6 degrees, to fulfill the necessity for combined loading testing. The structure call out for a test of combined loading at 3 degrees slip angle, 83 kilopascals inflation pressure, 3 degrees inclination angle and 667 Newtons of normal load is ‘Structure.BrakeDrive3SA.Pa83.IA3.FZ667’. Nominal commanded inflation pressures, inclination angles and normal loads rounded to the nearest integer in Metric units were utilized for the naming scheme in the data structure.

A list of the available data channels is given in Table 14. During the data organization, all channels are transcribed into the structure. The transcription process serves as a point to perform transformations of the data in terms of units and coordinate systems. The data that is collected by TIRF is reported in the SAE tire coordinate system per standard SAE J2047 [26]. The SAE system, as shown in Figure 16, follows the conventions of a right handed coordinate system; X is the longitudinal component pointed in the forward direction, Y directed laterally to the tire’s right and positive Z oriented into the ground. For the majority of desired work for simulation purposes and model fitting, the TYDEX, Tire Data Exchange, Wheel axis system will be utilized. The W-axis system matches the ISO standard wheel coordinate system [32]. It is also the system utilized by Adams/Tire for their magic formula implementation [27]. The W-axis system follows the right hand rule as well; the system can be formed by rotating the SAE coordinate system 180 degrees about the X axis. From this rotation, Y is oriented
pointing to the left of the tire and positive Z is pointing up away from the ground. To transform from the SAE to W-axis system, the FY, FZ, MZ and SA channels must be negated. Presented in Figure 17, the W-axis system will be utilized extensively in the model development, processing and expansion.

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
<th>TIRF USCS</th>
<th>TIRF SI</th>
<th>TYDEX Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>'ET'</td>
<td>Elapsed Time</td>
<td>seconds</td>
<td>seconds</td>
<td>seconds</td>
</tr>
<tr>
<td>'testid'</td>
<td>Calspan TIRF ID Number</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>'tireid'</td>
<td>Tire Description</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>'P'</td>
<td>Inflation Pressure</td>
<td>psi</td>
<td>kPa</td>
<td>Pascal</td>
</tr>
<tr>
<td>'N'</td>
<td>Tire RPM</td>
<td>rpm</td>
<td>rpm</td>
<td>radian/sec</td>
</tr>
<tr>
<td>'V'</td>
<td>Roadway Velocity</td>
<td>mph</td>
<td>kmph</td>
<td>meter/sec</td>
</tr>
<tr>
<td>'SA'</td>
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<td>degrees</td>
<td>radian</td>
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<td>-</td>
</tr>
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<td>'SR'</td>
<td>Slip Ratio - Calspan TIRF</td>
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<td>-</td>
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<td>FY/FZ - Lateral Coefficient</td>
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<td>N*meter</td>
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<td>degree C</td>
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<td>meter</td>
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Table 14 - FSAE TTC Round 4 Data Channels [26]
8.2. Steady State Model Fitting

Included in Rounds 1-3 of the TTC were fitted models derived from the empirical data. Milliken Research Associates supplied the expansion routines and MRA Nondimensional Tire Files for the tests completed. The expansion routines are secured in a Matlab p-code, and the tire files which consist of unlabeled matrices give little clue to allow reverse engineering and thus independent team development for Round 4 and later data sets. An overview for their approach is sited in literature [2], although no details for their algorithm or present technology is given. MRA is a consulting firm that
donates the model, but maintains tight control on their algorithms; their product is appealing, the lack of documentation to allow fitting routines to be coded eliminates it from being a viable approach for development.

The other supplied models were fitted by Stackpole Engineering Services utilizing a model based upon Pacejka’s ‘Magic Formula’ [21]. Coefficients were provided utilizing both the 1996 and 2002 Adams/Tire implementations of the Magic Formula. The equations for fitting and expansion are published [15], [21], [27] and have been under development since the late 1980s. Due to the public nature of the model, integration in MSC Adams/Tire, extensive end user scaling factors and a precedent for providing an adequate fitment of empirical data, the Magic Formula was chosen as the steady state fitted model approach. The data available allows fitting of the steady state parameters; examination of transient behavior faces limitations due to the confines of the available data, which will be discussed in 8.5 Thermal Transient Property.

The PAC2002 model equations are documented in several locations to varying levels of accuracy and completeness. Rolling resistance, MY, is excluded from the fitment as the channel was not included in the data set. A simplified rolling resistance calculation is included, based upon examples in literature [3]. The accommodations for parking and turn slip will also be excluded due to lack of data and decreased influence in racing. The Stackpole documentation [21] contains an incomplete set, including the expansions for FX₀, FY₀, MZ₀, MX, FX and FY. A more complete set of equations is included in the Adams/Tire help documentation [27]; the listings of equation variables were the most complete in this document. Even from this reference, equations were found to be absent in the area of MZ. To complete the equation set, Pacejka’s original work [15] is referenced. A complete set of equations to fit FX₀, FY₀, MZ₀, MX, FX, FY and MZ was finally established as the basis for fitment. The full set, as utilized, is displayed in Appendix C - PAC2002 Model Equation Set.
The general form of the ‘Magic Formula’ is presented in Equation 11. X is representative of either slip angle or longitudinal slip for FY or FX respectively. The horizontal and vertical shifts, \( S_H \) and \( S_V \), quantify the shifting of the curve relative to the origin to cope with tire asymmetries. These effects can occur due to tire properties of conicity and ply-steer when generating cornering forces. The peak factor, D, scales the curve to the correct magnitude. C is the shape factor that determines the extent of the \( \sin \) function; a curve whose magnitude decreases from the peak magnitude will have a shape factor greater than 1, as the arctangent will approach \( \pi/2 \) as X increases. A value of C that is too small or large will produce an unrealistic curve. B is the stiffness factor that dictates the curve’s slope near the origin, while E is the curvature factor which determines the curve’s shape near the peak output. Of special importance is the relationship arctangent of BCD, which quantifies the curves slope at the origin, synonymous with the cornering stiffness of the tire for the lateral case. Graphical representations of the sensitivity of the output to the factors are presented by Pacejka [15] which can greatly aid understanding of the effects of the different parameters on curve fitment.

With the tire data already in the W-axis coordinate system from the data organization its processing can now proceed. Generally, the outputs of the model are a function of slip angle, longitudinal slip, normal load and inclination angle. In literature they are referred to, respectively, as \( \alpha \), \( \kappa \), FZ and \( \gamma \) [15], [27]. Initially the pure slip conditions are fitted, \( FX_0 \), \( FY_0 \) and \( MZ_0 \). Lateral force must be fit prior to the aligning moment due to function dependencies. Examining the data in Figure 18 reveals several processing challenges that must be overcome. Moderate levels of noise can be reduced with averaging. All of the curves exhibit a hysteresis like effect near the origin in the linear region, which can be collapsed due to fitting being steady state in nature. The effect can be averaged, but it should not be
forgotten; this behavior is indicative of the transient structural properties of the tire which will be examined later. Also, effectively averaged out is the hysteresis at the peak and sliding region of the curve. Examining the tread surface temperature at these times, while considering that the compound of race tires is thermally sensitive, reveals a component of the disparity at the peak. The thermal properties of the tire can vary during individual tests and from test to test due to varied loading, running conditions and tire change outs. Part of the design of the test program is to minimize the thermal difference from run to run by allowing time between tests to return the tread to the same starting temperature. As part of the pre-treatment of the data, overlapping and non-linear segments are truncated from the data. Ultimately a reasonably smooth curve approximating the raw data is produced. The rotating inertia of the tire will also have gyroscopic effects during the slip angle sweep procedures that will resolve into the moment channels of the tire output.

Figure 18 - Pure Slip Condition Raw Data Example
For the pure slip components, a fitting method utilizing elimination of variables is employed. Proceeding through the testing matrix of inclination angles and normal loads, the values of B, C, D, E, $S_{\mu}$, $S_v$, K and $\mu$ are isolated and stored. Although not implicitly shown in Equation 11, K is key in the formulation of B and the peak factor D is $\mu$ times the normal load, FZ. These component matrices, of the size of the testing matrix, are then fitted to the PAC2002 functions, producing the relative Pacejka coefficients for that fitment. Two other important quantities defined as part of the initial process are the nominal radius, $R_0$, and the nominal load, $F_{z0}$. The nominal load was chosen to be the middle loading value of the testing matrix to simplify fitting. Once the pure slip conditions are fit, the process can move on to fitting the response for combined slip conditions. Care must be taken to utilize the proper channel to source the longitudinal slip from, as the ubiquitously named SR channel in Table 14 is not the needed slip ratio. SR coincides with the TIRF definition for slip ratio [26], while SL corresponds to the broader definition of longitudinal slip that should be utilized for fitment [2]. Use of SL causes FX to be equal to zero when SL is zero; SR usage introduces an unwanted shift of the data that doesn’t correlate to how the model should function, causing a nonzero longitudinal force to be reported when slip ratio equals zero.

For the fitting of the FX and FY combined conditions, the respective pure slip function must already be completed. Following fits of FX and FY, fitment of the combined MZ function is possible. Combined MX is a function of FY, thus it can be completed in turn as well. The combined FX and FY formulation is effectively a transformation of the pure slip conditions to induce a reduction in magnitude due to the concurrent longitudinal slip and slip angle conditions. The transform for each is based off the opposing slip condition; combined longitudinal force is altered by slip angle and combined lateral force is altered by the longitudinal slip. The process utilized for fitment was based upon the least squares fit concept. Iterating the variable set was left to the application of random number generation times an adaptive delta window. 6, 14 and 4 variables needed to be iterated for FX, FY and MZ.
respectively. Further improvements to this fitment routine are needed to assure that the fit is the best possible, as local optima exist.

A challenge arises for the fitting if precautions are not taken to resolve the issue that combined data was only collected for one sign of slip angles. Tires can behave asymmetrically, and thus the fitting equations contain provisions for the proper fitment of asymmetries. Since only one sign of slip angles are available for the fitting of the combined data, these asymmetric elements may lead to an improper fitment. Due to the asymmetry of available slip angles, the fitment can be overly compromised to being asymmetric in an automated search for the best fit. Evidence of this behavior, to varying degrees, is present in several sets of the PAC2002 variables provided by Stackpole Engineering Services [21]. A specific indicator that allows the assessment of the possibility of an issue is \( r_{\text{total}} \), which governs the shift of the slip angle in radians. A value near to zero should be expected, as a large value will shift the trigonometric function’s center to areas far away from zero slip angle. A review of the Pacejka’s work [15] gives valuable guidelines for the proper ranges of the ‘Magic Formula’ variables that must be maintained to create a realistic fitment over the entire extrapolated range of the data.
The weighting function also includes a shape factor, $C$, which should be approximately equal to one to cause the force output to be zero at infinite slip conditions. It is not a necessity to be one, but if allowed to be other values for a better fit over a limited range the weighting function $G$ must be monitored to assure that it is always positive. Fluctuations in the sign of $G$ cause the sign of the force to be flipped as a consequence of the trigonometric functions involved. Also, due to the nature of the trigonometric functions utilized, if care is not taken in the fitting of the coefficients oscillatory conditions can show in the expansion of the model. Although the fit may be proper in the limited regions of the raw data, the response is anything but believable. Again, this condition showed up in different instances of the Stackpole coefficient sets. The expansion of the combined longitudinal force for varying longitudinal slip and slip angle in Figure 19 illustrates this point.

Finally, the determined coefficients are stored to a data structure for storage and utilization in the Matlab expansion routine. The overall fit achieved of the data was good in the case of normal
inflation pressures. When fitting the data for 55 kPa inflation pressure, the data at higher normal loads regressed from the available fit of the model. This can be attributed to the higher loads of 222 and 445 Newtons (250 and 350 lbf) being outside of the linear operating range for the lower inflation pressures. This discontinuity effects the lateral force output chiefly, as the longitudinal component was found be more tolerant of the lower pressures. A probable conclusion is that the structure and stiffness in the lateral direction is much more susceptible to variations in inflation pressure; logically this makes sense from the orientation of the structure in the sidewall and the outer belt by the tread.

Although the standard PAC2002 equations must be performed as dictated to integrate with Adams/Tire, the implementation within Matlab can be altered to provide a better fit to the data, as the expansion routine is alterable. Most of the model fits were performed adequately bar the pure slip aligning moment model for some tires. For this portion of the model, the pneumatic trail factor, t, has a peak magnitude stipulated by Dt. In the equation set it is effectively factored by the normal load twice due to being a function of FZ and the normalized change in normal load dFz. By examining the fitting of the variable, it became evident that eliminating the FZ factor would provide a dramatically better fit across the load ranges tested. The difference in the fit of Dt is show in Figure 20 and Figure 21.
Generally, the fitment of the combined slip aligning moment was less than satisfactory in many instances. Low inflation pressures often made the fit quality unsatisfactory. When utilizing the fitted data, care must be taken to understand the fitting process and thus the errors inherent to the model, otherwise erroneous results may be arrived at. The fact that the combined equations rely on several layers of fitted data helps explain the large errors. Error is propagated in the combined aligning moment from all prior fits.
Most tires, consisting of the Goodyear and Hoosier offerings, in Round 4 fit well to the overturning moment model. The radial offerings from Dunlop and Michelin showed behaviors consistent with phenomenon described by Pacejka [15]. In these cases, deformation of wide tires causes the overturning moment to reverse in sign as lateral load increases. Although the solution existed, it was not implemented as it was not the standard PAC2002 approach as well as the tires were of little interest for the development of future vehicles.

Even though rolling resistance was unavailable as a data channel in the TTC offering, inclusion of a simplified estimate of MY is prudent for the model. An approach consistent with examples in literature [3] is applied within the model to at least give a place holder for a quantity that should be accounted for. The rolling resistance is a constant factor, in this case chosen to be ~0.0165, times the normal load on the tire. Taking this quantity times the nominal radius gives the quantity in terms of a moment.

The following is a summary of the process used to fit the data as part of the PAC2002 Fitting Script presented in Appendix D.

- Fit Pure Slip Lateral Force $F_Y_0$ and Longitudinal Force $F_X_0$
  - Fit Initial Parameter Set at 0 degrees Inclination Angle and $F_Z_0$
  - Fit Shape Factors B, C, D, E and K and Shift Factors $S_{n_l}$ and $S_{v}$ for each run in test matrix
  - Fit the Complete PAC2002 Parameter Set to the Shape and Shift Factor Matrices
- Fit Pure Slip Aligning Moment $M_Z_0$ ($M_Z_0$ Function is Dependent on $F_Y_0$)
  - Fit Initial Parameter Set at 0 degrees Inclination Angle and $F_Z_0$
  - Fit Shape Factors B, C, D, E and K and Shift Factors $S_{n_l}$ and $S_{v}$ for each run in test matrix
  - Fit the Complete PAC2002 Parameter Set to the Shape and Shift Factor Matrices
• Fit Combined Slip Lateral Force FY and Longitudinal Force FX (Dependent on FY\(_0\) and FX\(_0\) respectively) utilizing a least squares fitting procedure coupled with a random input.

• Fit Combined Slip Aligning Moment MZ (Dependent on FY and FX) utilizing a least squares fitting procedure coupled with a random input.

• Fit Combined Slip Overturning Moment MX (Dependent on FY) utilizing a least squares fitting procedure coupled with a random input.

8.3. Model Expansion and Usage

With the PAC2002 model fitting complete, the next logical step is to utilize the coefficients to aid in understanding and improve the accuracy of simulations. By choosing to fit to the PAC2002 model, implementation into Adams/Tire simulation tool is already satisfied. For usage within Matlab a simple expansion function was programmed. It is capable of matrix computation, although a full rigorous set of failsafes and error checking have not been implemented in the programming. For a given set of slip angle, longitudinal slip, normal load and inclination angle running conditions, combined with a supplied structures containing the Pacejka coefficients and the user scaling factors, all remaining forces and moments of the tire axis system along with their pure slip counterparts are returned.

A valuable part of the implementation of the model is the utilization of the user scaling variables. This is of particular interest due to the nature of the flat belt testing performed. The surface of the belt is very consistent and possesses higher amounts of grip compared to a roadway. From examination at TIRF, the surface consisted of 120 grit sandpaper adhered to the roadway. The surface was then ‘stoned’ to normalize the surface and reduce its grip. Even with this preparation, the tire testing returns results that indicate a normalized lateral and longitudinal force, FY/FZ and FX/FZ, between 2 and 3. With logged on-track lateral acceleration indicating maximum levels in the range of 1.7 g with no aerodynamic downforce, any use of the data to realistically simulate and estimate vehicle
performance requires the scaling of the peak force. Capability for this is delivered via the scaling parameters \( \lambda_{ux} \) and \( \lambda_{uxx} \) by giving them a value less than unity. When the tire model is utilized in a vehicle dynamics simulation, the scaling factors should be iterated until the simulated performance level matches the on track performance. Other parameters are available to modify the model, adding needed flexibility to alter the results. Further discussion of the scaling parameters is presented in literature [15]. Plotting different sets of applied scaling factors back to back can aid in their understanding and application as well.

Simple plotting of the model data allows inferences into the relative performance levels between the tires in the TTC round. The varieties of plotting are only limited by the user. One plotting method that aided understanding of the performance limits of the tire was to expand the model over a matrix of slip angles and longitudinal slip values within the range of the tire. The points are then plotted in mesh according to their lateral and longitudinal force component. The resulting graph shown in Figure 22, for a given normal load and inclination angle, shows the performance envelope of the tire in a manner similar to a ‘g-g’ diagram. Producing a set of subplots for a matrix of normal loads and inclination angles graphically presents the respective sensitivities. Another possibility is to plot internal components of the model, such as the peak factor D and cornering stiffness K. An implementation of this approach in Figure 23 and Figure 24 illustrates the impact of inflation pressure and rim width on the peak factor and cornering stiffness. This data usage gives inferences to the impacts of inflation pressure and rim widths succinctly.
Figure 22 - Tire Performance Envelope

Figure 23 - Rim Width and Inflation Pressure Effect on $\mu_y$
Integrating the fitted model into various vehicle calculations and simulations pertaining to vehicle dynamics is the most valuable use of the data. Depending on the complexity of the simulation, only small portions of the model may be needed. In the case computing the stability derivatives of the vehicle and subsequent vehicle factors as presented by Milliken [2], only the cornering stiffness needs to be calculated leading to simple model implementation comprised of only 3 parameters to compute $K_y$. Simple sensitivity simulations to estimate maximum vehicle performance may only need the peak factor and vertical shift components expanded to be effective. The limitations of the fitted model must be observed, chiefly the steady state requirement and the range of the data that was fit, to get proper results. Excessive extrapolation of the model beyond the data sets’ range of slip angles, longitudinal slip, normal loads and inclination angles is a questionable practice.

Moving to a more complex, full implementation of the data, specifically a simulation of a 4 wheeled vehicle with inclination angle effects, requires certain precautions. The tire coordinate system for each wheel has the same orientation and sign convention. As such, for a vehicle with static negative camber (wheel tilted inwards toward vehicle), the inclination angle for the left tire will be positive while
the right tire will be negative. Negative inclination angles were unavailable for fitment. Also, even if data was available, the likelihood that FY(IA)=FY(-IA) is improbable. One implementation strategy is to deal with the right side tires as mirrored coordinate systems. This implementation negates the slip angle and inclination angle inputs; the outputs FY, MZ and MX must then be negated to return to the normal coordinate system.

**8.4. Tire Vertical Rate**

The tire’s vertical rate for several different conditions was estimated via a first order polynomial fit of the provided test data. Along with the vertical spring rate, the tire’s effective unloaded radius is known via the intercept of the linear function. Spring rate tests were carried out for both static and dynamic conditions, at the tested inclination angles and inflation pressures. The results of the analysis were stored to a data structure. The outcome of this analysis fulfills design needs for determining the loaded spindle height and spring rates utilized in the suspension development.

**8.5. Thermal Transient Property Observations**

A key racing tire property that is explored specifically in the testing procedure of the FSAE TTC is the thermal effect on the tire’s performance. It doesn’t take long from driving a car on track to realize that tires have a performance envelope due to the tire’s running temperature. Too cold or too hot, and anecdotally, a sharp decrease in performance can be felt. Smith gives a maximum traction temperature range of 190 to 200 degrees Fahrenheit for dry tires and 140 to 160 for rain tires [16], [17]. Due to the low car weights and the demands of the competition in the cool Michigan spring, successful tires for FSAE are more akin to rain tire compounds. Quantifying this effect is a distinct challenge when merely considering collected track data. As part of the Round 4 procedure [19], when a tire was initially started on the machine a continuous slip angle sweeps covering 8 repetitions were completed at a high slip rate
of 8 degrees per second to study the transient thermal and break-in effects of the tire. Analyzing and plotting this data in different ways allows inferences useful for comparing tire alternatives at the least.

Figure 25 - Tread Temperature Time History and Clipping Illustration

Initially, the entire time stream was simply plotted for the temperature data in Figure 25. The simple stream of temperature data shows that temperatures increase on average, but due to the cyclic nature of the slip angle sweep the local value of the inner, center, and outer tread temperatures vary between gaining and losing heat dependent on the current part of the sweep. The test was conducted with 0 degrees of inclination angle at a normal load of 1112 Newtons (250 lbf), as such it would be expected that the inner and outer tread for a symmetrical tire would behave similarly. From observation of the data this is not the case. As lateral force builds the carcass deflects, this naturally shifts the area that the infrared sensor sees on the tread. For very compliant tires or poor aiming of the sensor, the focal point can even leave the tread surface that contacts the roadway causing a sudden clipping of the temperature data of the inner or outer channels. An effective method for spotting this
effect is to plot the tread temperatures versus normalized lateral force, as seen in the second subplot of Figure 25. Generally, the inner tread temperature was the most likely to be effected throughout all Round 4 tests. The most consistent and reliable data to work with was observed from the center channel due to this effect; the center channel will be the primary focus of the examination of thermal properties.

![Graphs showing tread temperatures and lateral force](image1)

Figure 26 - Quantification of Change of Lateral Force Curve for the Cold to Warm Test – Green Tire

When plotting lateral force versus slip angle for the entire run in Figure 26, several important aspects are noticeable. From a simple data presentation in the top-left subplot, the general trend is for the linear region of the force curve, where the cornering stiffness stipulates performance, to remain constant in behavior. This statement is affirmed when the cornering stiffness values were extracted as...
the test progressed, as shown in the bottom-right subplot. Although there is trending to indicate a change over temperature, other non-quantifiable effects tied to the fact that this is the first running of a green tire cannot be ruled out. Select used tires were re-run over the warm-up and initial conditioning segments, which illustrates this effect. The corresponding used tire run for Figure 26 is presented in Figure 27. The conditioned tire delivers a more linear and predictable response. This difference is a quantifiable reasoning behind the actions of ‘scrubbing in’ and conditioning new, green tires [33]. It should be noted that the used tire may have more normalized performance, but the peak grip available is reduced.

![Lateral Force vs. Slip Angle](image1.png)

![Lateral Force vs. Slip Angle](image2.png)

![Cornering Stiffness Variation](image3.png)

**Figure 27 - Quantification of Change of Lateral Force Curve for the Cold to Warm Test – Used Tire**

The other subplots of Figure 26 and Figure 27 show the trends of where the response curve enters the transition region and the peak force location. The delineation point is fairly consistent in its
associated slip angle, but the lateral force level of the points make a small gain during the elapsed time of the test. The conditions for the definition of delineation were that the response deviated more than 100 Newtons (22.5 lbf) from the last cornering stiffness fit. As the tire warms and gains more peak lateral performance in the Bottom-Left subplot the associated slip angle also increases. The tire is a spring in the lateral sense, thus, as load is increased deformation does as well.

![Graph](image)

Figure 28 - Temperature Time Derivative Correlation to Lateral Load and Slip Angle

Another implementation of the data is the examination of the temperature derivative with respect to time. This derivative, scaled properly, would also equate to the absolute value of the temperature derivative with respect to slip angle, due to slip angle being applied at a constant rate of 8 degrees per second. With this in mind, the data presented in Figure 28 has interesting implications. Based upon the derivative versus normalized lateral force, it can be seen that the only time a net gain in temperature is made is when operating at peak force levels. Also, the peak area allows operation in either the positive or negative region while maintaining roughly the same performance level. Operating within the linear region causes a loss of tread surface temperature.
This lesson is applicable to defining good driver behavior. A person not pushing the limits of the current temperature constraint will never build substantial heat in the tires. A person overdriving the vehicle will get heat into the tires at the cost of probable over heating during approximations of extended high performance steady state maneuvers, thus reducing the performance envelope. An experienced person driving the vehicle will be able to manipulate the applied controls to stay in the peak performance range, while managing the thermal equilibrium of the tires to effectively increase the performance envelope. The driver must realize what the performance limit is for the current temperature, then, via manipulation of location in the transition, peak and sliding region of the performance curve, actively control the thermal state of the tire to get the highest performance possible. The driver is likely not managing the behavior in this logical manner, but, instead, by the way the car’s feedback changes while performing different maneuvers through the course of the event.

The other subplot, showing the relation between slip angle and temperature derivative can be taken as having the form of ‘V’ in the approximated linear and transition regions, then approaching a steady value in the sliding region. A possible use of this behavior would be to develop a slip angle estimating procedure for on track performance when utilizing data acquisition with tire temperature sensors. The temperature derivative magnitude does vary with the loading on the tire, as observed in the general cornering tests; with measurement of the tread surface temperature and spring displacement to estimate normal load, an examination of the temperature derivative would allow an estimated observability of the tire’s slip angle running condition.

A final observation based upon the cold to warm testing is presented in Figure 29; the normalized peak lateral force is isolated from each peak of the response then plotted against the center tread temperature. The data presentation is a straightforward presentation of how quickly the tire comes to performance temperatures, and the difference between cold and hot performance levels.
With a change of only 25 degrees Celsius, the lateral force output increases by a factor of 1.25 then proceeds to drop off at a similar rate as temperatures continue to climb. Unfortunately, due to the test length of only 8 slip angle sweeps, the diminishing portion of the curve is often poorly observed in Round 4. With the execution of Round 5, the test length was increased to 12 sweeps to better quantify this region. The analysis kept the differing curves separate as contact length based transient characteristics caused a discrepancy between the loading conditions.

Figure 29 - Variation of Peak Force Mu with Change in Center Tread Temperature
8.6. Tire Structural Properties Fitting

The tire is a composite structure, possessing properties indicative of a mechanical system: mass, damping and stiffness. As a simplification, when examining the lateral response of the tire during a slip angle sweep the force levels rise loading the structure and inducing a deflection. Dependent on the system and the nature of the loading, phase and magnitude changes are imparted on the response. This response characteristic, possibly identified as a hysteresis like condition in the linear region, can be seen in Figure 30. Quantifying the structural characteristics of the tire in different deformation modes is a complex task; estimates for the general scaling of these parameters are given in Pacejka. For a rolling tire, the structure’s damping is a relatively low quantity, typically 1-6% of the critical damping value [15]. An assumption of a lightly damped structure is thus not out of the realm of possibility.

![Figure 30 - Effect on Transient Response with Different Slip Velocities]
A simplified approach to induce this transient characteristic into the simulation results is the utilization of a low pass filter to approximate the tire’s mechanical system. Equation 12 from Pacejka is utilized to replace the steady state cornering stiffness in linear simulations with a transient stiffness to simulate the lag of the tire’s cornering force generation in response to changes in slip angle [15]. The same principle is utilized with Equation 13. \( \sigma_i \), the variable for relaxation length, and \( u \), the longitudinal velocity of the tire aligned with the X axis of the tire coordinate system make up the time constant of the system. Iterative utilization of Matlab’s LTI model linear simulation allows the relaxation length to be found. Applying the filter to the PAC2002 fitted data gives a reasonable match of the phasing of the experimental data in Figure 31. Irregularities in the transition, peak and sliding regions can be partially attributed to the previous discussion of thermal effects. The transient properties of the lateral force and longitudinal force are easily computed. Pacejka advises that at the nominal loading of the tire, the relaxation length should be on the order of magnitude of the loaded radius [15].

\[
a = p_{a1} \left( \sqrt{\frac{\rho_{z}}{r_{o}}} + \frac{\rho_{z}}{r_{o}} \right) r_{o} \text{ for } (p_{a1} = 0.35) (p_{a2} = 2.25)
\]

Equation 14 - Contact Length Estimate: \( a \) [15]

\[
c_{x,y} = \frac{C_{F,k,a}}{\sigma_{x,y}} - a
\]

Equation 15 - Carcass Stiffness Calculation [15]
Representative findings of the relaxation length properties for a tire are presented in Table 15 and Table 16 for lateral and longitudinal properties respectively. These findings are applications of the SWIFT fitting methods in Pacejka [15]. As part of the application, the fitted PAC2002 stiffness coefficients, dynamic tire unloaded radius and dynamic vertical rate of the tire were utilized to determine an estimate for the contact length, \( a \), and the carcass stiffness at the ground, \( c \). The estimation for the contact length \( a \), Equation 14, utilizes the tire vertical deflection, \( \rho_z \), the nominal radius and a set of two parameters derived from the testing method. With the contact length known, the PAC2002 fitted cornering or longitudinal slip stiffness coupled with the measured lateral and longitudinal relaxation length allow the computation of the carcass stiffness via Equation 15. The respective carcass stiffness can be distributed to the various structural elements of the tire, per the final steps laid out in the SWIFT implementation of Pacejka [15]. The implementation will not be performed, but the constituent components are supplied for reference and future use.
<table>
<thead>
<tr>
<th>Normal Load</th>
<th>Tire Deflection</th>
<th>Contact Length / 2</th>
<th>Cornering Stiffness</th>
<th>$\sigma_y$ (m)</th>
<th>$c_y$ (N/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_z$ (N)</td>
<td>$\rho_z$ (m)</td>
<td>$\alpha$ (m)</td>
<td>$C_{\tau\alpha}$ (N/rad)</td>
<td>4 deg/sec</td>
<td>8 deg/sec</td>
</tr>
<tr>
<td>222</td>
<td>0.0016</td>
<td>0.0337</td>
<td>-15540</td>
<td>0.1956</td>
<td>-96011</td>
</tr>
<tr>
<td>445</td>
<td>0.0033</td>
<td>0.0484</td>
<td>-30109</td>
<td>0.2794</td>
<td>-130349</td>
</tr>
<tr>
<td>667</td>
<td>0.0049</td>
<td>0.0599</td>
<td>-42756</td>
<td>0.3353</td>
<td>-155264</td>
</tr>
<tr>
<td>1112</td>
<td>0.0082</td>
<td>0.0786</td>
<td>-60997</td>
<td>0.4191</td>
<td>-179156</td>
</tr>
<tr>
<td>1557</td>
<td>0.0114</td>
<td>0.0943</td>
<td>-70208</td>
<td>0.4750</td>
<td>-184422</td>
</tr>
</tbody>
</table>

Table 15 - Selected Lateral Structural Characteristics, Partial SWIFT Implementation

<table>
<thead>
<tr>
<th>$F_z$ (N)</th>
<th>$\rho_z$ (m)</th>
<th>$\alpha$ (m)</th>
<th>$C_{\tau\alpha}$ (N/%)</th>
<th>$\sigma_x$ (m)</th>
<th>$c_x$ (N/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>222</td>
<td>0.0016</td>
<td>0.034</td>
<td>22348</td>
<td>0.140</td>
<td>210871</td>
</tr>
<tr>
<td>667</td>
<td>0.0049</td>
<td>0.060</td>
<td>60297</td>
<td>0.363</td>
<td>198790</td>
</tr>
<tr>
<td>1112</td>
<td>0.0082</td>
<td>0.079</td>
<td>89147</td>
<td>0.531</td>
<td>197128</td>
</tr>
<tr>
<td>1557</td>
<td>0.0114</td>
<td>0.094</td>
<td>108942</td>
<td>0.699</td>
<td>180304</td>
</tr>
</tbody>
</table>

Table 16 - Selected Longitudinal Structural Characteristics, Partial SWIFT Implementation

Utilization of the transient properties outlined gives the capability to accomplish extension of the steady state PAC2002 model into the transient realm of lateral and longitudinal slip velocity as well as dynamic load changes. Anytime there is a change in reacted force by the tire over a time based analysis, the transient property should be utilized. Without accounting for the property, key information relating to the transient response of the system to control inputs and tire operating conditions is lost.
9. Conclusions

Overall, the development of the tools and methods for both the inertia measurement and tire data processing was successful, yielding reasonable and repeatable results. Although many developments and improvements were implemented into the design of the tools, many revisions are still possible to attain the better results. An understanding of the problem posed by the competition requirements to the chassis designer is the first step to the implementation of the presented chassis development tools. The tools and their presented data are multifaceted and fit into many steps of the vehicle’s product life cycle and should be leveraged as such.

9.1. Yaw Inertia Rig

The implementation of the yaw inertia measurement rig gave satisfactory results. The photo gate sensor utilized had a difference in rising and falling edge characteristics, thus disallowing utilization of both edges. An improved sensor, such as any photo eye, could replace it with better results, assuming that its transient response characteristics were well explored to guarantee quality of the signal. The tone wheel would be better with a higher angular resolution, but the limitation of the sensing method hinders this. Ideally, the data analyzed should be from the zero velocity region collected by actuating the test rig with an initial velocity winding up the weight. The angular resolution doesn’t allow this method to be carried out properly. The transparency material utilized for the trigger wheel printing was difficult to trim accurately, lacked robustness and the laser printing would smudge when exposed to cleaning chemicals. A different approach utilizing a laser cut tone wheel or a rapid prototyping process may yield more desirable results. When the pressure was run at a higher level than needed, the table, when not utilizing the spreader plate, would self-actuate to one direction. This was attributed to the poor trimming and installation of the tone wheel. Through the different pivot designs
for the string actuation, the sliding method was found to be the most predictable. It requires more calculation to determine the correction factors, but they are quantified each time a test is carried out.

9.2. Pitch and Roll Inertia Rig

Making measurements on the pitch and roll rig is a much more involved process than desired. Although the switch from an air pivot to a knife edge worked well, many elements to the test setup and procedure are not as simple as intended. The initial setup of the rig, particularly the alignment of the rig on the table and the measurement of the hard points on the hanging rig can be difficult and prone to human error. As the rig is modular and meant to be disassembled and stored between uses, it is a necessary complexity. It should be noted that just as much care should be taken in taking the setup distance measurements as the actual dynamic test data. With poor setup, one will receive poor results in turn. Error analysis is encouraged to ascertain critical aspects of the setup. Also, concerning the setup of the points that the actuation mass will be hung on, permanent, purpose installed points are advised to be added to the rig, as hanging masses from the ends of the cross tubes in the platform lacks accuracy and repeatability. To ease loading of test objects onto the rig, an overhead winch on a sliding rail would be a great addition to the shop; this winch would also allow loading vehicles onto and off of the T-slot table. Currently the length of chassis that can be measured during roll measurements is limited due to the utilization of the same upright supports in both pitch and roll configurations. For the roll configuration, the support structure for the pivot should be a gantry style, positioned outside of the support struts, to allow the nose of longer vehicles to protrude beyond the edge of the platform between the struts.

Ascertaining the center of mass height involves adding and removing the test mass from the rig in a controlled manner, then damping out the motion of the rig at each point. It should be expected to repeat the test a couple times to gain the experience needed to complete it with good repeatability.
Assuring that the specimen is prepared properly, with the removal of all fluids that can slosh in their tanks is important as well. Also, if a driver is to be part of the test, they need to exercise control to not move while seated as to not disrupt the test. The higher the actuated angle of the table during the test, the more accurate the center of mass estimate will be. The position of the added mass should be as far as possible from the pivot as well, which is difficult to implement during roll inertia measurements. A possible solution would be to add a long tube fastened to the top of the platform that would move the application points beyond the edge of the T-slot table to a more definite point, in both the pitch and roll setups. This would have the added benefit of reducing the test mass; during the tests of the vehicle, a test mass of approximately 23 kg (50 lbm) was utilized which proved to be bulky and difficult to hang in close proximity to the edge of the T-slot table. It is also suggested that the test mass be added to actuate the rig in both directions to see if the results are repeatable for both directions and side to side.

In comparison to other methods of center of mass determination for the entire vehicle, such as hanging the car or rolling it onto two tires to balance it at an angle, the table proved to be a controlled and accurate method while also serving the need to estimate the inertias of the car and sprung mass.

Measuring the period of oscillation for the swing of the pendulum is the easiest step of the procedure. There is a definite difference between the period of low and high amplitude oscillations. This was attributed to discrepancies due to small angle approximation and the large surface area of the vehicle and platform being damped by air movement at higher angles of actuation which led to higher table velocities. The difficulty in attaining quality low amplitude data is the signal to noise ratio. Monitoring of the signal and the power supplied to the sensor indicated that the supply was not providing clean power, which was not out of the question due to it not being a lab grade piece of equipment. Improvements to the power supply would have the largest impact on the quality of the signal. Also, the sensor utilized had a viable measurement range of 318 degrees, although the actual rig only needs to utilize a maximum of 30 degrees of travel. Replacing the sensor with one equipped with a
smaller measurement range should significantly improve the results for both the center of mass and period measurement. The fact that the sensor adds resistance to the motion of the pendulum, although small, is significant and should be eliminated if possible. A non-contact sensing device with high accuracy would be ideal, but commercially available devices that were surveyed were cost prohibitive. Another possibility to improve the measurement resolution would be the utilization of a mechanical gain device such as a follower linkage, step up pulleys or gearing. Proper implementation to assure minimal addition of friction to the device and hysteresis to the measurement would be challenging to achieve using this method.

Post-processing of the data for all of the inertia rigs was straightforward and logical, assuming that all elements of the test were calibrated and carried out appropriately. As with any testing method, care and diligence during the setup of the test will reward the user with viable data. Sampling rates in the range of 500 to 1000 Hz were utilized to properly characterize the rising edge of the yaw inertia rig data while the pitch and roll inertia tests produced satisfactory results at a lower sampling rate of 50 Hz over a long testing period of 120 seconds; this was done to capture as many oscillations as possible to aid in the averaging of the oscillation period. Also, the long test period allows the application and removal of the test mass to the system for center of mass measurements in a continuous time stream.

9.3. Tire Data Processing

Tire data processing is part science and part art. Interpreting all that the data has to offer can be difficult and overwhelming, especially in the voluminous raw form presented by the TTC. The pre-processing scripts developed to organize the data into a data structure for easy reference and analysis was a great enabler for the subsequent processing of the data. It is strongly recommended that the end user familiarize themselves with Matlab’s string manipulation capabilities to allow the dynamic allocation and execution of complex commands in a looped environment; the ability to dynamically
allocate variable names, data arrays and script commands is necessary to efficiently work with the volumes of data. With the data sorted into the structure, it is easily called and manipulated for the purposes of raw data plotting, model fitment and any other analysis. The present version disregards certain portions of the data such as the conditioning and warm up loops of the test. These can be preserved easily by editing the script to save out the wanted sections to a different segment of the data structure.

The PAC2002 fitment of the tire data was a learning experience. The associated script is rather long and rambling to some extents, and could be extensively consolidated through the use of called functions and simpler fitting algorithms. As such, it served its purpose as a learning and development tool to develop the fitting procedure. Often, a better fit to the supplied data set can be achieved via manipulation of certain parameters, but this fit may end up resulting in an asymmetric fit with poor characteristics when expanded to regions beyond the running conditions that the raw data was collected under. This effect is evident in several of the models supplied by Stackhouse Engineering Services [21]. Care must be taken when utilizing any model, whether fitted in house or supplied by a third party, to assure that it accurately represents the behaviors exhibited in the raw data. Future work includes a major revision of the script to make it more modular via the use of called functions. Also, incorporation of the methods developed in the thermal and transient force analyses into the fitting script would yield improved initial raw data curves for fitting. The data would be corrected at the peak value for thermal scaling effects and run through a filter to effectively collapse the dynamic characteristic of the curve in the linear region. Inclusion of a load range for model utilization based upon the quality of the fit at different loadings should be considered. At low inflation pressures coupled with high loads, collapse of the lateral force response curve to one that is not well characterized by the model is evident. Ideally the end user should fit the data themselves or plot the fitted model versus the raw data to understand the shortcomings of the fitted model before utilizing it in simulations or analysis.
Also, the scaling of the force from the Calspan TIRF testing facility is greater than what will be seen on track. Experimentation with the user scaling parameters of the Pacejka model is encouraged.

Problems that may arise with the script as written occur occasionally, dependent on the tire being analyzed. In the pure slip coefficient determination the fitment for the curvature factor \( E \) can exceed the valid region of \( E \) being less than or equal to 1; which generally causes a nonlinear fit at higher loads due to parameters factoring in the effect of normalized load difference. Another issue that should be noted, again, is the difference between the data channels SR and SL; SL, or longitudinal slip, is the desired channel for most analysis of longitudinal force. When examining the combined operating conditions for lateral and longitudinal force, often the raw data curves do not align with the value in the pure slip curves, \( F_{X_0} \) and \( F_{Y_0} \). The fitment compensates for this characteristic, but it is unknown if the raw data could or should be shifted to align with the pure slip conditions. Probable reason for the disparity, that should be explored, is the transient effect of the relaxation length property. The fitment of the MZ combined data can be very good to very poor; it is suggested to the end user to check this fitment to judge whether it should be utilized in simulation. This disparity was not tire specific. Often, one inflation pressure would fit adequately while the next would have severe differences in shape and magnitude for the same tire. Radial tires such as the Dunlop, Michelin and Continental offerings exhibited a reversal of the overturning moment at higher lateral loading due to displacement and lifting of the tread element. This effect was not corrected for due to lack of interest in these tires as options for the car. If they were to be utilized in simulations, an alteration of the fitment and expansion code in accordance with Pacejka’s recommendation [15] may provide a more realistic expansion of the model. Ultimately, poor fittings of the pure slip conditions will cascade into larger problems when fitting the combined loading data.
Another design element extracted from the data set is the vertical spring rate and unloaded radius of the tire at varying inflation pressures and inclination angles. At the moment, the script only sources data from the purpose run spring rate runs in the data. Implementation in the PAC2002 fitting script to extract the rolling radius and normal load for each zero crossing of slip angle and slip ratio would yield a more comprehensive data set, with spring rates at more combinations of inclination angle and inflation pressure. Variation of the rolling radius with increased slip ratio and slip angle is also of interest. The simplest implementation of the data in design would be utilizing the unloaded radius and spring rate of the tire in conjunction with the design corner weights of the vehicle to yield a better estimate of the spindle height for suspension design.

Some of the most interesting data came from the thermal transient analysis. With the quantification of the thermal sensitivity of the tire, the issue becomes even more important. In the past, the team has collected probe temperatures at the track when the vehicle is pitted, which should continue. Adding an on track acquisition of the dynamic surface temperatures would greatly enhance the understanding of the performance of the vehicle and how the vehicle’s design coupled with the driver’s actions are utilizing the tires’ available performance. From sensitivity of the tire’s lateral force to temperature, judgments can be made whether the car is being over driven or underutilized from examining the on track temperature data. Monitoring temperature time derivatives may serve as a tool to indirectly monitor the slip angle that the tire is operating at. Considering, understanding and quantifying the thermal sensitivity of the tire should have a larger impact on gaging and improving the performance of the vehicle’s design due to the large thermal transients present in most competition events. Transient thermal properties also serve as important criteria for the selection of the car’s tires.

The last property explored were characteristics related to the relaxation length of the tire. The relaxation length is a quantification of the tire’s physical structure, thus it induces aspects of the tire’s
performance analogous to characteristics in linear system modeling and structural dynamics. Through use of Matlab’s LTI simulation capabilities, the hysteresis like condition in the linear region of the lateral force curves was effectively quantified. Understanding this effect is vital to properly implementing chassis analysis involving dynamic application of driver controls and vehicle response as phasing will be induced dependent on the rate of changes in the system.

9.4. Future Work

Future work includes all of the applications where the data collected by these chassis development tools can be utilized. The vehicle inertias and tire model are vital inputs to making a functioning dynamic simulation in Adams/Car. MSC Adams is a powerful and diverse multibody dynamics package capable of modeling and simulating nonlinear systems with compliant elements. Although useful, the inherent complexities of Adams to deal with these complex systems introduces many variables that are unneeded and confusing for simplified analyses that will garner the biggest returns in both vehicle performance and student learning. Without a full understanding of the analysis being done the student’s understanding of the results, meaningful application of the analysis and the validity of the results will be in question.

Approaches in Matlab utilizing the collected data are advocated due to how the complexity of the work will scale with the student’s understanding as well as how the program provides an open framework in a simple programming environment that the student will implement themselves. In the realm of LTI systems, simple quarter or half vehicle models of the sprung and unsprung masses can be implemented for individual corner, pitch and roll dynamics. A full vehicle implementation with four unsprung degrees of freedom and four sprung degrees of freedom, pitch, heave, front roll and rear roll, will utilize the inertias and tire spring rates collected. Different springing and damping implementations can be modeled and simulated.
Bicycle linear models [15] can also be implemented utilizing the yaw inertia and cornering stiffness modeled from the tire data to simulate vehicle response and stability in the linear regime of low slip angles. The system equations for this approach are well documented [2], [3], [15]. Utilizing the full PAC2002 model implementation can move the bicycle model concept into quantifying the full vehicle performance in both linear and limit steady state conditions with the application of the MRA Moment Method [2] or Gough Plot [15]. A bicycle model is simpler to initially implement and can eventually lead to a full vehicle implementation accounting for track widths, static toe values, weight transfer, aerodynamic loading and inclination angle effects. With development being done in Matlab, the extent of the analysis is only limited by the amount of scripting done. With proper implementation, the horizontal axis of the analysis, which characterizes steady state turning, can be validated via instrumented ISO or SAE constant radius skip pad testing procedures.

With the realization of the transient tire structural properties, the concept of measuring the vehicle’s response to varied controlled inputs becomes interesting. With the vehicle speed limited by an engine rev limiter, controlled inputs to the steering system would be recorded while monitoring the vehicles dynamic response. As a driver would have less than optimal control of the inputs, a servo motor setup could be attached to the steering shaft to drive the system at varying magnitudes and frequencies; this test would necessitate a controlled setting possessing adequate runoff room, with the driver only exercising control to override the system in case of an emergency. The outcome of such an experiment would be interesting and indicative of the vehicles dynamic response. A FRF of the vehicle’s system response to steering input might be realizable at varying steer magnitudes and vehicle speeds. As part of a parameter driven vehicle development process, a desirable system response to design the vehicle to could be defined during development and validated during testing.
A final project for implementation based upon the measurements of the engine component inertias via the yaw inertia rig would be an inertial dynamometer for the engine. A purely transient method of tuning, the apparatus would allow quantification and tuning of the acceleration and deceleration throttle position compensations outside of the car in addition to the normal tuning that is done on the dynamometer. As the load control is purely inertial, partial throttle pulls and tuning would be easily achieved. The system would also allow quantification of the full torque curve of the engine, from 0% to 100% throttle position. Pulls accelerating the inertia would occur at higher throttle positions. After these pulls the throttle would be closed, to decelerate the inertia and thus measure the engine braking curve. The rig would also provide an excellent platform to do development of shifting strategies. Data acquisition needs for the dynamometer would be a tachometer on the inertia and ideally a load cell on a pivoting carriage to isolate the torque reacted into the support bearings. Safety would demand a sturdy enclosure and a braking system. With knowledge of the inertia of the engine and dynamometer, the acceleration rate of the dynamometer and the torque being reacted to the bearings, an accurate measurement of torque output could be obtained. The largest challenge to the project is acquiring a balanced inertia of sufficient size and speed capability. Ultimately, the inertial dynamometer operates on the same principles as the yaw inertia rig; with known rotational inertias, parasitic losses and position data of the rotary position the torque can be easily computed. Even without knowing the inertias involved, the test would still be useful, as only the proper scaling for the measured torque would be absent.
Bibliography


Appendix A - Yaw Inertia Test Documentation

A.1. Test Procedure

1. Locate the storage crate and spreader plate. They should be stored in the back closet of the office, with the crate on the shelf and the circle plate on edge behind the cabinet. Locate a lab station with data acquisition. Structures and motions lab equipment was utilized with success.

2. Clear the large T-slot table in the shop. Assure that the surfaces are clean to allow the rig to sit flat on the table. The lower manifold block should fit into the middle slot of the table, allowing the air hoses to extend out to the edge of the table as seen in Figure 32. Maintain easy access to the air controls.

3. Attach the regulator to the shop air supply. Dry shop air with no oil should be used.

4. Validate free motion of table. Open the bottom valve and back off the regulator. Turn on the air supply. Slowly increase the pressure supplied by the regulator until the platform is floating freely. Too much pressure will bind the system; to alleviate this issue back off the regulator and turn off the air supply if necessary. Validate that the table is floating freely. The table should spin down very slowly once rotating. Also, a very small force should be necessary to initiate
motion. No sticking should be felt. If necessary, remove the lower manifold and retaining bolt to allow disassembly to clean the internal surfaces. Damage, such as dents, scratches and scrapes need to be dressed and polished to maintain performance. Once complete, turn off the air supply.

5. Place the tone wheel sensor in the T-slot opposite of the air connection as Figure 33 shows. Place it over the tone wheel as close as possible without touching. Route the cables out the T-slot, connecting them to power and the BNC connection of the data acquisition equipment respectively. Attach an oscilloscope in parallel to validate signal from the sensor. A small prop to support the cables may be necessary to keep the sensor upright.

6. Place the spreader plate on the rig if necessary. The plate has a recess that sits over the table’s upper diameter. Care should be taken to not disturb the sensor. If bumped, simply nudge it back into position.

7. Procure a piece of fishing line and attach it to the rig. Under light loads, without the use of the spreader plate, it can be taped to the top edge of the platform. When utilizing the spreader
plate there are numerous holes to anchor the line to. Assure that the line is long enough to reach over the edge of the table to the floor.

8. Position the pivot, either the ball bearing or a wire loop, at the edge of the table. With a light load on the line, slowly wind it up. The line should wind around the outside of the 7 inch diameter of the platform, above the tone wheel and below the spreader plate. Assure that the pivot is aligned and that the line is not making contact with the tone wheel or spreader plate. The pivot may need shimmed up to accomplish this. Keep tension on the line to maintain the wrap of the line as shown in Figure 34.

9. Once a mass is attached, position a chock to keep the table from rotating. Turn on the air supply and adjust the regulator until the table is free floating. Develop familiarity for the mass drop procedure while validating that the rig is working properly prior to testing. A small drop mass can accelerate the table quickly.

10. Perform validation/calibration runs on a known inertia. The table can satisfy this role, but adding another inertia can also be done. For all planned drop mass variations, perform a calibration run. This is especially important for the looped wire pivot to quantify the friction of the pivot, so its effect can be isolated from the response data.

11. Turn off the air supply, if on. Place the test specimen on the platform. For best results make sure there are no shifting liquids or masses in the specimen. If possible, center it as closely as
possibly initially based upon the static weight distribution. Once positioned, activate the air supply. Adjust the regulator until the object is supported, or the pressure is maxed. At this point the bottom relief valve can be closed slowly until support is attained. If the mass is adequately centered, there will be no binding of the pivot, making rotation very easy with no noticable damping. If there is binding, look at the air gap between the rotor and base beneath the tone wheel. This will give an indication of the direction to shift the mass. Turn off the air supply, shift the mass, and repeat the procedure until free motion is observed. ‘Shuddering’ and ‘shimmying’ are also signs of need to realign or adjust the support pressure.

12. Perform the wanted test runs. Multiple dropping masses should be utilized for each specimen.

Validation, calibration and tests all utilize the same general procedure, as documented below:

a. Setup data acquisition for long time testing. A testing period of 60 seconds at 500 Hz sampling rate was found to be satisfactory to capture the pertinent information and allow time to make adjustments. Manual logging was utilized, although triggers could be utilized.

b. Start data acquisition.

c. Remove chock block. Let the specimen start to rotate. One method to guarantee a good start is to initial apply a spin to lift the mass; doing so will assure that the zero velocity point was not effected by the release.

d. Watch the test and do not interfere. Low speed data is the best; at most, 10 seconds of rotational data should fulfill all needs.

e. Stop the rotation gently and rewind the rig; lock it in place with the chock block.

f. Stop data collection.

g. Do a quick check of the data. If satisfied, save out the data to an external drive.

h. Repeat the test for different drop masses.
i. Repeat the test for opposite spin direction.

j. Do a quick analysis check to validate the data. If a processing script is utilized during the testing, the goodness of the polyfit can give indications of the quality of the results.

13. Clean up equipment, place in the storage container and put it back in the closet for the next use.

A.2. Data Processing Scripts

clear all;
close all;

dragvalue=.019;
HangerMass=.615;

% The timethreshold is the limit placed on the fitted data.
% Limit is a fitting threshold to keep multiple points from being picked at
% a single rising edge.

% Sprung Chassis Measured 343 lbm

% % 2011 Sprung, Empty, Hanger Mass Only
load '2011_Unsprung_HangerDrop.mat';
actmass=HangerMass-dragvalue;
timethreshold=12;
limit=1.2;

tachsig=Data.data.t_domain(:,1);
timedat=Data.data.t;

h=1;

% Pull out rising edge data of the tach signal.
for n=2:length(timedat)
    if (tachsig(n)-tachsig(n-1))>limit
        neotime(h)=timedat(n-1);
        h=h+1;
    end;
end;

% Plot limits.
StartTime=min(neotime)-1;
EndTime=StartTime+2+timethreshold;

% Loop to determine the best time offset of the data set to optimize
% the fitment of the polyfit.
for toffset=1:101
    toff(toffset)=((toffset-1)/50*(neotime(2)-neotime(1)));
    neotime=neotime-neotime(1)+toff(toffset);
    neoangl=0:2*pi/180:(2*pi/180*(length(neotime)-1));

    h=1;
    for n=1:length(neotime)
        if neotime(n)<=timethreshold
            timecrit(h)=neotime(n);
            h=h+1;
        end
    end

    anglcrit=0:2*pi/180:(2*pi/180*(length(timecrit)-1));
polycrit=polyfit(timecrit,anglcrit,2);

actrad=((6.998/2)+(.013/2))*0.0254;
gravity=9.80665;
tableinert=3.619;

polytime=timecrit;
polyvect=polycrit(1).*polytime.^2+polycrit(2).*polytime+polycrit(3);

polyerror(toffset)=sum(abs((polyvect-anglcrit)*100))/length(polyvect);
end

% Plot of polyfit error.
figure(4);
plot(toff,polyerror);
title('Polyfit error');
xlabel('Time Offset (sec)');

% Identify minimum error
[polymin, polyindex]=min(polyerror);

% Shift time vector by best possible offset.
neotime=neotime-neotime(1)+toff(polyindex);
neoangl=0:2*pi/180:(2*pi/180*(length(neotime)-1));

h=1;
% Truncate data to within the time threshold.
for n=1:length(neotime)
    if neotime(n)<=timethreshhold
        timecrit(h)=neotime(n);
        h=h+1;
    end
end

anglcrit=0:2*pi/180:(2*pi/180*(length(timecrit)-1));

polycrit=polyfit(timecrit,anglcrit,2);

% Constants - Global and System
% Radius of string's action (m)
actrad=((6.998/2)+(.013/2))*0.0254;
% Gravity (m/sec^2)
gravity=9.80665;
% Table and spreader plate inertia (kg*m^2)
tableinert=3.619;

% Polyfit computation for comparison to actual data.
polytime=timecrit;
polyvect=polycrit(1).*polytime.^2+polycrit(2).*polytime+polycrit(3);

% Numerical derivative of position - omega
m=3;
for n=m:length(timecrit)
    omega(n-m+1)=(anglcrit(n)-anglcrit(n-m+1))/(timecrit(n)-timecrit(n-m+1));
    omegatime(n-m+1)=(timecrit(n)+timecrit(n-m+1))/2;
end
% Numerical derivative of omega - alpha
m=3;
for n=m:length(omegatime)
    alpha(n-m+1)=(omega(n)-omega(n-m+1))/(omegatime(n)-omegatime(n-m+1));
    alphatime(n-m+1)=(omegatime(n)+omegatime(n-m+1))/2;
end

% Validation plots
figure(2);
plot(timedat,tachsig); axis([StartTime EndTime...
    min(tachsig)-.25 max(tachsig)+.25]);
title('Photointerrupter Signal');
xlabel('Time (seconds)');
ylabel ('Signal (Volts)');
figure(1); subplot(1,2,1);
plot(timecrit,anglcrit,'r-*',polytime,polyvect,'go',polytime,...
    (polyvect-anglcrit)*100,'b+');
axis([0 timethreshhold+.5 min((polyvect-anglcrit)*100)-.02...
    max(anglcrit)+.05]); title('Polynomial Fit of Data');
xlabel('Time (seconds)');
ylabel('Table Angle (radians)');
legend('Experimental Data','2^n^d Order Polyfit','Polyfit Error * 100',...
    'Location','NorthWest');
grid on;
subplot(1,2,2); hold on;
plot(timecrit,anglcrit,'r-*',omegatime,omega*10,'b-',...
    alphatime,alpha*100,'g-o');
title('Position Numerical Derivitives');
xlabel('Time (seconds)'); ylabel('Position Derivatives');
legend('Theta (radians)','Omega (10*(rad/sec))','Alpha (100*(rad/sec^2))');
figure(1);

% Car/Object Inertia
carinertia=((actrad*gravity*actmass)/(abs(polycrit(1))*2)-tableinert);

% Calcs for table only test:
tableinertia=carinertia+tableinert;
tableerror=abs(tableinertia-tableinert)/tableinert;

% Maximum speed reached by rig for fitted data.
caromega=max(omega);

% Print results to command window.
disp(carinertia);
disp(actmass+dragvalue);
A.3. Rig Component Drawings
\[ \text{Diagram showing engineering specifications.} \]
Appendix B - Pitch and Roll Inertia Documentation

B.1. Test Procedure

1. Locate all measurement rig components. The majority are stored in a tote in the FSAE office closet. The long support poles are also stored in the closet, inside of a piece of PVC pipe. The rig platform is stored either in the shop or outside the office. Its large size should make it easy to locate.

2. The setup and experiment procedure requires in excess of 4 individuals to easily move and locate all components due to their mass and size. Organizing manpower should also take into consideration which drivers are needed to take part in the measurements. The manpower required to complete the procedure can be reduced if facilities such as an overhead lift are available to lift and locate heavy objects onto the T-slot table.

3. Measure the mass of the swung table in both pitch and roll configurations. The mass includes all items that move in that mode of motion. This includes the table, support struts, central pivot, knife edge, sensor offset piece and all associated hardware. Assembly is not required. Initial testing performed during the rig’s development found a swung mass of 135.6 kg and 132.2 kg for the pitch and roll configurations respectively.

4. Clear the T-slot table. Place the stabilizer blocks, which are wooden 4x4 blocks measuring 5 and 5/16 inch long, on the corners of the table. Place the Pitch Table assembly on top of the blocks. The blocks are the proper setup height for the table. Square, center and align the Pitch Table with the T-slot table. Rulers and carpenter squares are useful for this process. The Pitch Table should be flipped so that the roll pickup points, circled in Figure 35, are towards the bottom of the platform. If misplaced, switching between pitch and roll measurements will be impossible without flipping the table mid process.
5. Assemble the 2 upright structures utilizing ¾-28 fasteners. The lower ends of the supports are assembled with Allen head cap screws threaded into the 2x2 bottom bars. Connecting the two bars is an aluminum cross member. The upper portion of the upright is assembled with nuts and bolts. Two plates sandwich the support tubes and upper aluminum receiver block. All
joints were spray painted as a unit to provide witness marks for assembly. The assembly is illustrated in Figure 36.

6. Attach the upright assemblies to the T-slot table in the desired pitch or roll arrangement. For the pitch setup, utilize the red circled holes in Figure 36 inserted into the highlighted blue T-slots in Figure 35. For the roll setup, the green slots of Figure 35 can utilized with any of the holes. The roll setup is aligned properly via the T-slots as the upright gets slid all the way against the T-slot table. During the pitch setup, the upright is offset from the edge of the T-slot table, which is dictated by the T-slot spacing. To properly place and align the uprights along the length of the T-slots during the pitch configuration utilize a long bar clamped to the base bars to align the uprights to each other, as shown in Figure 37. Measure from the end of the T-slot table to the edge of the bottom bar; both sides should measure 36.25 inches from the end of the table when centered.

Figure 37 - Upright alignment utilizing a straight bar
7. Install the pivot heads on top of the upright assemblies. The joint is pinned to allow rotation to relieve misalignment. Do not bolt the connection. The piece of the pivot head that is pinned to is asymmetric; the long side of the piece should be oriented to the inside of the test rig. Assure that the knife edge pivot is aligned within the rotating head. The alignment of the pivot may need fixed at different times during testing to keep parts from rubbing. The installed pivot head and knife edge setup are shown Figure 38.

8. Install the support struts between the table mounts and central hub. The best method was found to be to insert the bolts in the central pivot first, followed by starting the threads of the half inch bolts in the other end. With the strut held in place, attach the nuts at the central pivot and tighten them. For the pitch setup, utilize the holes circled in red in Figure 38; for roll use the blue circled holes. Assure that the pivot does not rotate excessively during this process. With the center pivot fixed, double check that the table is still aligned with the T-slot table. Tighten all half inch bolts until they contact the table mount point. Draw all of them up evenly until the blocks are only lightly loaded.

![Figure 38 - Pivot installation and setup](image)
9. Remove the support blocks. They should be tight fitting still, but should slide out with some force. Remove the blocks on one end, and then pivot the table to remove the other two. Check the setup by swinging the rig. The pendulum should oscillate and not damp out quickly. Check for points of contact between the pendulum and support structure. Assure that the bolted joints from the struts to the pivot are not flexing. Once verified, let the rig settle out to its free-hanging position.

10. Gather structure data for calculation purposes. Ultimately the origin of the system is the pivot point of the knife edge. With the table free-hanging, measure the height of the pivot above the T-slot table. Use a plumb bob to transfer the pivot’s position along the length of the table to both the platform and T-slot table. Utilizing the T-slot table as a datum, measure the longitudinal position and height from the table of each point where mass will be added to determine the center of mass position. During testing, the strut mount position farthest from the pivot was utilized, as seen in Figure 41. Utilizing previous data is not advised, as the position of the table can vary with the setup and installation.

11. Calibrate the angle sensor before mounting. Setup the data acquisition system. A filter applied to the sensor output is advised. Center the sensor in its measurement range by aligning the slot on the shaft with the output wire of the sensor. Attach the calibration arm to the sensor, so
that it is pointed at the output wire as well. Utilizing a ¼ inch alignment pin or reamer, sweep
the arm to the four mount holes on the sensor housing while logging the signal. This will give 4
data points covering 270 degrees of sensor output. The general setup utilized is show in Figure
39.

12. Attach the sensor to the pivot point as shown in Figure 40. The offset bracket will attach to the
green circled holes in Figure 38, while the center mount hole will be attached via a set screw to
the angle sensor. Do not alter the set screw position, sensor setup or data acquisition setup mid
test without recalibrating the system.

![Figure 40 - Angle sensor installation](image)

13. Measure the properties of only the table as the first test to attain the rig’s center of mass
position and rotational inertia. Without proper quantification of the rig’s properties,
subsequent tests will not give satisfactory results. The test is common with or without a
specimen on the table, described as follows:

a. Quantify the center of mass location by measuring the free hanging angle followed by
the angle with a known mass added at a known location, as shown in Figure 41. When
adding or removing masses, damp the resulting oscillatory motion by hand to the get to
the new equilibrium point. The larger the angle of the table attained, the better the
data. For table only measurements, adding a mass of 6.235 kg was utilized. With a vehicle sized object added, an applied mass in the range of 23 kg was satisfactory. From the collected data, the positions of the rig setup and the table-only properties, a simple force balance yields the center of mass position of the new object.

b. Perform the inertia test. Start the table oscillating and record the position signal. Averaging the period over multiple oscillations gives better results. The center of mass position is necessary to calculate the inertia, so both tests are required to quantify inertia. There is no reason not to measure the inertia after the center of mass, as the test is simple. Record the oscillation of both low and high magnitude initial displacement.

14. Replace the support blocks to stabilize the table when making any adjustments or additions to the device. Load the test vehicle onto the table. Any vehicle that is being tested should have all
fluids topped off or drained to eliminate shifting masses. Measure the mass of the object as tested at a point during the test.

15. Center the vehicle on the table and block to prevent rocking. Remove the support blocks on the end of the table where they are unloaded. Slowly move the test object towards that end until the other blocks are unloaded. Full vehicles can have their brakes locked with a tie-strap. Slop in the floating disc mount points can be taken up, once the brakes are locked, by pulling the front tire contact patch away from the rear tire. This will keep the vehicle from rocking during testing. Also, it should be done as a safety precaution to prevent the vehicle from rolling off the platform. Non-rolling vehicles and sprung masses pose fewer issues.

16. Perform the test on the vehicle to ascertain its parameters.

17. Switches between pitch and roll setups can be performed without a full teardown. Support the table with the spacer blocks and change the orientation of the uprights from the sides to the end of the platform.
B.2. Data Processing Scripts

% Pitch/Roll Rig Oscillation Period Extraction

close all
clear all

% Experimental Run Selection
Analysis='Pitch';
Type='HiOsc';
Driver='Table';
eval(['load ' strcat(Analysis,'_','Driver','_','Type','.mat')]);

% Number of oscillations to average the period value out of.
N=30;
Tvec=Data.data.t;
Tsig=smooth(Data.data.t_domain(:,1),19);

[MAX MXind]=max(Tsig);
VecL=length(Tsig);
SigM=mean(Tsig(MXind:VecL));

% Truncate data to complete oscillations
for k=1:VecL
    if Tsig(VecL-k)<Tsig(VecL-k+1) && Tsig(VecL-k)>SigM
        ENDind=VecL-k+1;
        break;
    end
end

% Truncate and perform a vertical shift of data to setup zero-crossing
% point. Effectively, remove DC component of signal.
TruncVec=Tsig(MXind:ENDind)-mean(Tsig(MXind:ENDind));
TruncTime=Tvec(MXind:ENDind);

m=1;
P=1;

% Extract zero-crossing and peak points of signal.
for k=2:length(TruncVec)
    if TruncVec(k)<TruncVec(k-1) && sign(TruncVec(k))==-1 && ...
        sign(TruncVec(k-1))==1 && (k-P>=70 || P==1)
        ZeroT(m)=TruncTime(k);
        LocalVec=TruncVec(P:k);
        [MAX MXind]=max(LocalVec);
        clear LocalVec;
        PeakT(m)=TruncTime(MXind+P-1);
        PeakV(m)=TruncVec(MXind+P-1);
        P=k;
        m=m+1;
    elseif TruncVec(k)<TruncVec(k-1) && sign(TruncVec(k))==-1 && ...
        sign(TruncVec(k-1))==0 && (k-P>=70 || P==1)
        ZeroT(m)=TruncTime(k);

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LocalVec=TruncVec(P:k);
[MAX MXind]=max(LocalVec);
clear LocalVec;
PeakT(m)=TruncTime(MXind+P-1);
PeakV(m)=TruncVec(MXind+P-1);
P=k;
m=m+1;
end
end

% Plotting operations
if strcmp(Type,'LoOsc')
    GraphTitle='Low Amplitude Motion';
else
    GraphTitle='High Amplitude Motion';
end

figure(1); hold on;
subplot(1,2,1);
plot(TruncTime,TruncVec./0.007260753);
hold on;
plot(ZeroT,zeros(size(ZeroT)),'ro',PeakT,PeakV./0.007260753,'go');
PlotTitle=[Analysis ' - ' GraphTitle ' - Time Response'];
title(PlotTitle);
xlabel('Time (seconds)'); ylabel('Table Angle (degrees)');
legend('Time Response','Zero (Falling)','Peak Time');

% Average period calculation.
for k=N+1:length(ZeroT)
    AverageT(k-N)=(ZeroT(k)-ZeroT(k-N))/N;
    PeakAveT(k-N)=(PeakT(k)-PeakT(k-N))/N;
end

subplot(1,2,2); hold on;
plot(AverageT,'b-x');
plot(mean(AverageT).*ones(size(AverageT)),'r-');
plot(PeakAveT,'g*');
plot(mean(PeakAveT).*ones(size(PeakAveT)),'c-');
PlotTitle=[Analysis ' - Average Period - N = ' int2str(N) ' Samples'];
title(PlotTitle);
xlabel('Average Number'); ylabel('Average Period T (seconds)');
legend('Zero Crossing Point','Zero - Mean','Peak Time Point','Peak - Mean');
B.3. Rig Component Drawings
Appendix C - PAC2002 Model Equation Set

[15] [27] [21]

C.1. Common Load Equations

Nominal Load:

$$F'_{zo} = \lambda_{Fzo} \cdot F_{zo}$$

Normalized Load Change:

$$df_z = \frac{F_z - F'_{zo}}{F'_{zo}}$$

C.2. Longitudinal Force – Pure Longitudinal Slip Conditions

$$F_{xo} = D_x \sin[C_x \tan^{-1}(B_x \kappa_x - E_x (B_x \kappa_x - \tan^{-1} B_x \kappa_x))] + S_{Vx}$$

$$\kappa_x = \kappa + S_{Hx}$$

$$\gamma_x = \gamma \cdot \lambda_{Vx}$$

$$C_x = p_{Cx1} \cdot \lambda_{Cx}$$

$$D_x = \mu_x \cdot F_z$$

$$\mu_x = (p_{DX1} + p_{DX2} \cdot df_z) \cdot (1 - p_{DX3} \cdot \gamma_x^2) \cdot \lambda_{ux}$$

$$E_x = (p_{EX1} + p_{EX2} \cdot df_z + p_{EX3} \cdot df_z^2) \cdot (1 - p_{EX4} \cdot sgn(\kappa_x)) \cdot \lambda_{Ex}$$

$$K_x = F_z \cdot (p_{KX1} + p_{KX2} \cdot df_z) \cdot \exp(p_{KX3} \cdot df_z) \cdot \lambda_{Kx}$$

$$K_x = B_x \cdot C_x \cdot D_x = \frac{\delta F_{xo}}{\delta \kappa_x} \text{ at } \kappa_x = 0 (= C_{Fx})$$

$$B_x = \frac{K_x}{C_x \cdot D_x}$$

$$S_{Hx} = (p_{Hx1} + p_{Hx2} \cdot df_z) \cdot \lambda_{Hx}$$

$$S_{Vx} = (p_{Vx1} + p_{Vx2} \cdot df_z) \cdot \lambda_{Vx} \cdot \lambda_{ux}$$

C.3. Lateral Force – Pure Side Slip Conditions

$$F_{yo} = D_y \sin[C_y \tan^{-1}(B_y \alpha_y - E_y (B_y \alpha_y - \tan^{-1} B_y \alpha_y))] + S_{Vy}$$

$$\alpha_y = \alpha + S_{Hy}$$
\( \gamma_y = \gamma \cdot \lambda_{\gamma y} \)

\( C_y = p_{C\gamma 1} \cdot \lambda_{C\gamma} \)

\( D_y = \mu_y \cdot F_z \)

\( \mu_y = (p_{DY 1} + p_{DY 2} \cdot d_f) \cdot (1 - p_{DY 3} \cdot \gamma_y^2) \cdot \lambda_{\mu y} \)

\( E_y = (p_{E\gamma 1} + p_{E\gamma 2} \cdot d_f) \cdot [1 - (p_{E\gamma 3} + p_{E\gamma 4} \cdot \gamma_y) \cdot \text{sgn}(\alpha_y)] \cdot \lambda_{E\gamma} \)

\( K_{yo} = p_{K\gamma 1} \cdot F_{zo} \cdot \sin \left( 2 \cdot \tan^{-1} \left( \frac{F_z}{p_{K\gamma 2} \cdot F_{zo}} \right) \right) \cdot \lambda_{K\gamma} \)

\( K_y = B_y \cdot C_y \cdot D_y = \frac{\delta F_{yo}}{\delta \alpha_y} \) at \( \alpha_y = 0 \) \( (= C_F\alpha) \)

\( K_y = K_{yo} \cdot (1 - p_{K\gamma 3} \cdot |\gamma_y|) \)

\( B_y = \frac{K_y}{C_y \cdot D_y} \)

\( S_{Hy} = (p_{Hy 1} + p_{Hy 2} \cdot d_f) \cdot \lambda_{Hy} + p_{Hy 3} \cdot \gamma_y \)

\( S_{Vy} = F_z \cdot \left\{ (p_{V\gamma 1} + p_{V\gamma 2} \cdot d_f) \cdot \lambda_{V\gamma} + (p_{V\gamma 3} + p_{V\gamma 4} \cdot d_f) \cdot \gamma_y \right\} \cdot \lambda_{\mu x} \)

\( K_{vyo} = \left\{ p_{Hy 3} \cdot K_{yo} + F_z \cdot (p_{V\gamma 3} + p_{V\gamma 4} \cdot d_f) \right\} \left( = \frac{\delta F_{yo}}{\delta y} \right) \) at \( \alpha = \gamma = 0 \) \( (= C_F\gamma) \)

**C.4. Aligning Moment – Pure Side Slip Conditions**

\( M_{zo} = -t_o \cdot F_{yo} + M_{zr} \)

\( t_o = D_t \cos[C_t \tan^{-1}(B_t \alpha_t - E_t(B_t \alpha_t - \tan^{-1} B_t \alpha_t))] \cdot \cos(\alpha) \)

\( \alpha_t = \alpha + S_{Ht} \)

\( M_{zr} = D_r \cdot \cos[C_r \cdot \tan^{-1}(B_r \cdot \alpha_r)] \cdot \cos(\alpha) \)

\( \alpha_r = \alpha + S_{Hf} \)

\( S_{Hf} = S_{Hy} + \frac{S_{Vy}}{K_y} \)

\( \gamma_z = \gamma \cdot \lambda_{\gamma z} \)

\( B_t = (q_{Bz1} + q_{Bz2} \cdot d_f + q_{Bz3} \cdot d_f^2) \cdot (1 + q_{Bz4} \cdot \gamma_z + q_{Bz5} \cdot |\gamma_z|) \cdot \frac{\lambda_{Ky}}{\lambda_{\mu y}} \)
\[ C_t = q_{C_1} \]

\[ D_t = F_z \cdot (q_{Dz1} + q_{Dz2} \cdot d_{f_z}) \cdot (1 + q_{Dz3} \cdot y_z + q_{Dz4} \cdot y_z^2) \cdot \frac{R_o}{F_{zo}} \cdot \lambda_t \]

Modified \( D_t \) = \( q_{Dz1} + q_{Dz2} \cdot d_{f_z} \cdot (1 + q_{Dz3} \cdot y_z + q_{Dz4} \cdot y_z^2) \cdot \frac{R_o}{F_{zo}} \cdot \lambda_t \)

\[ E_t = (q_{Ez1} + q_{Ez2} \cdot d_{f_z} + q_{Ez3} \cdot d_{f_z}^2) \cdot \left\{ 1 + (q_{Ez4} + q_{Ez5} \cdot y_z) \cdot \frac{2}{\pi} \tan^{-1}(B_t \cdot C_t \cdot \alpha_t) \right\} \]

\[ B_r = q_{Br9} \cdot \frac{\lambda_{xy}}{\lambda_{xy}} + q_{Br10} \cdot B_y \cdot C_y \]

\[ D_r = F_z \cdot ((q_{Dz6} + q_{Dz7} \cdot d_{f_z}) \cdot \lambda_r + (q_{Dz8} + q_{Dz9} \cdot d_{f_z}) \cdot y_z) \cdot R_o \cdot \lambda_{xy} \]

C.5. **Longitudinal Force – Combined Slip**

\[ F_x = G_{xa} \cdot F_{x0} \]

\[ G_{xa} = \frac{\cos[C_{xa} \tan^{-1}(B_{xa} \alpha_s - E_{xa}(B_{xa} \alpha_s - \tan^{-1} B_{xa} \alpha_s))] \cos[C_{xa} \tan^{-1}(B_{xa} S_{Hxa} - E_{xa}(B_{xa} S_{Hxa} - \tan^{-1} B_{xa} S_{Hxa}))]} \]

\[ \alpha_s = \alpha + S_{Hxa} \]

\[ B_{xa} = r_{Bx1} \cdot \cos[\tan^{-1}(r_{Bx2} \cdot \kappa)] \cdot \lambda_{xa} \]

\[ C_{xa} = r_{Cx1} \]

\[ E_{xa} = r_{Ex1} + r_{Ex1} \cdot d_{f_z} \]

\[ S_{Hxa} = r_{Hx1} \]

C.6. **Lateral Force – Combined Slip**

\[ F_y = G_{yk} \cdot F_{y0} + S_{VyK} \]

\[ G_{yk} = \frac{\cos[C_{yk} \tan^{-1}(B_{yk} \kappa_s - E_{yk}(B_{yk} \kappa_s - \tan^{-1} B_{yk} \kappa_s))] \cos[C_{yk} \tan^{-1}(B_{yk} S_{HyK} - E_{yk}(B_{yk} S_{HyK} - \tan^{-1} B_{yk} S_{HyK}))]} \]

\[ \kappa_s = \kappa + S_{HyK} \]

\[ B_{yk} = r_{By1} \cdot \cos[\tan^{-1}(r_{By2} \cdot (\alpha - r_{By3}))] \cdot \lambda_{yk} \]

\[ C_{yk} = r_{Cy1} \]

\[ E_{yk} = r_{Ey1} + r_{EY2} \cdot d_{f_z} \]
\[ S_{Hy} = r_{Hy1} + r_{Hy2} \cdot df \]
\[ S_{Vyk} = D_{Vyk} \cdot \sin[\gamma_{Vyk} \cdot \tan^{-1}(\gamma_{Vyk} \cdot \kappa)] \]
\[ D_{Vyk} = \mu_y \cdot F_z \cdot (\gamma_{Vyk1} + \gamma_{Vyk2} \cdot df + \gamma_{Vyk3} \cdot \gamma) \cdot \cos[\tan^{-1}(\gamma_{Vyk4} \cdot \alpha)] \]

C.7. Overturning Moment

\[ M_x = F_z \cdot R_o \cdot (q_{Sx1} - q_{Sx2} \cdot \gamma + q_{Sx3} \cdot \frac{F_y}{F_{zo}}) \cdot \lambda_{Mx} \]

C.8. Rolling Resistance Moment – Simple Estimate

\[ M_y = -F_z \cdot R_o \cdot f \quad (f \cong 0.0165) \quad [3] \]

C.9. Aligning Moment – Combined Slip

\[ M_z = -t \cdot F_y' + M_{zr} + s \cdot F_x \]
\[ t = t(\alpha_{t,eq}) = D_t \cos[C_t \cdot \tan^{-1}(B_t \cdot \alpha_{t,eq} - E_t (B_t \cdot \alpha_{t,eq} - \tan^{-1} B_t \cdot \alpha_{t,eq}))] \cdot \cos(\alpha) \]
\[ F_y' = F_y - S_{Vyk} \]
\[ M_{zr} = M_{mr}(\alpha_{r,eq}) = D_r \cdot \cos[C_r \cdot \tan^{-1}(B_r \cdot \alpha_{r,eq})] \cdot \cos(\alpha) \]
\[ s = R_o \cdot \left\{ s_{Sx1} + s_{Sx2} \cdot \left( \frac{F_y}{F_{zo}} \right) + (s_{Sx3} + s_{Sx4} \cdot df) \cdot \gamma \right\} \cdot \lambda_s \]
\[ \alpha_{t,eq} = \sqrt{\alpha_t^2 + \left( \frac{K_x}{K_y} \right)^2 \cdot \kappa^2 \cdot \text{sgn}(\alpha_t)} \]
\[ \alpha_{r,eq} = \sqrt{\alpha_r^2 + \left( \frac{K_x}{K_y} \right)^2 \cdot \kappa^2 \cdot \text{sgn}(\alpha_r)} \]
Appendix D - Matlab Tire Data Processing and Utilization

D.1. Tire Data Pre-Processing Script

clear all;

% Altered version to modify TTC data in SAE convention to ISO standard
% compatible with Adams tire modeling.

% SA=-SA
% FY=-FY
% FZ=-FZ
% MZ=-MZ

TireName='Goodyear D2696';
TireSize='20.0x7.0-13';
RimWidth=7;

% Prefix for generation of variables
FileName='GoodyearD2696_7inch';
% TIRF test run numbers associated with tire.
Tests=[14 15 16 17 18 92 93 94 95 96 97 155];

% Defined Test Types by the TTC Summary Document
% 1   Cornering
% 2   w + Cornering Cold
% 3   BrakeDrive
% 4   BrakeDrive Cold
% 5   w Only

% Test types for the given test run numbers.
TestTypes=[2 1 1 1 1 5 4 3 3 3 3 5];

% Component breakdown of the different test types.
% 11  Tire Rate
% 12  Warm Up Test
% 13  Scrub In
% 14  Conditioning
% 15  SA Sweep, follow with total sweeps
% 16  SR Sweep, follow with total sweeps
% 17  0 -> 25 MPH

% Cornering Schedule Definition:
% SA: -4:+12:-12:+3
% FZ: 350 150 50 250 100
% IA: 0 2 4 1 3
% Pressure: 12 10 14 8 12

% BrakeDrive Schedule Definition:
% SR: +20:+20
% FZ: 350 150 250 50
% IA: 0 2 4
% SA: 0 3 6
% Pressure: 12 10 14 8 12

% Test schedule for each test type.
% For test type 15, the next vector element dictates the number of
% repetitions of test type 15, which is the test matrix size (5 IA x 5 FZ).
% For test type 16, the next vector element dictates the number of
% repetitions of test type 16, which is the test matrix size (3 IA x 4 FZ
% x 3 SA).
TestType1=[14 15 25 11];
TestType2=[11 11 17 11 11 12 11 11 13 15 25 11];
TestType3=[14 16 36];
TestType4=[14 16 36];
TestType5=[11 11 17 11 11 12 11 11 11 13];

% Test number declaration.
WarmIndex=1;
SpringIndex=1;

% Commanded nominal test velocities, pressures, and normal loads. These
% are compared to, to establish the fields for the data structure.
Velocities=[0 40];
Pressures=[55 69 83 97];
NormalLoads=[-222 -445 -667 -1112 -1557];

for n=1:length(Tests)
    clear AMBMP ET FX FY FZ IA MX MZ N NFX NFY P RE RL SA SL SR TSTC TSTI
    TSTO V testid tirid
    matfile=sprintf('B1320run%d.mat',Tests(n));
    eval(['load ' matfile]);
    switch TestTypes(n)
        case 1
            TestOrder=TestType1;
        case 2
            TestOrder=TestType2;
        case 3
            TestOrder=TestType3;
        case 4
            TestOrder=TestType4;
        case 5
            TestOrder=TestType5;
    end
    m=1;
    for k=1:length(TestOrder)
        switch TestOrder(k)
            case 11
                while m<=length(ET)
                    if round(FZ(m)/-222)==1
                        Start=m;
                        break;
                    else m=m+1;
                end;
            end
            while m<=length(ET)
                if FZ(m+1)-FZ(m)>200
                    Last=m;
                end
            end
        end
    end
end

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m=m+1;
break;
elseif m-Start>=110
  Last=m-11;
  m=m-10;
  break;
else m=m+1;
end;
end;
MainTxt=[FileName '.SpringRate.Test' int2str(SpringIndex) '.'];
.eval([MainTxt
  'PAVG=databin(mean(P(Start:Last)),Pressures);'
  ]);
.eval([MainTxt 'IAVG=round(mean(IA(Start:Last)));'
  ]);
.eval([MainTxt
  'VAVG=databin(mean(V(Start:Last)),Velocities);'
  ]);
.eval([FileName '.SpringRate.SpringIndex='
  int2str(SpringIndex) ';']);
SpringIndex=SpringIndex+1;
case 12
  Start=m;
  while m<=length(ET)
    if ET(m+1)-ET(m)>1
      Last=m;
      m=m+1;
      break;
    else m=m+1;
  end;
  while m<=length(ET)
    if ET(m+1)-ET(m)>1
      m=m+1;
      break;
    else m=m+1;
  end;
MainTxt=[FileName '.Cold2Warm.Test' int2str(WarmIndex) '.'];
WarmIndex=WarmIndex+1;
case {13, 14}
  while m<=length(ET)
    if FZ(m)+1557<150
      break;
    else m=m+1;
  end;
end;
case 15
  for p=1:TestOrder(k+1)
    while m<=length(ET)
      if ET(m+1)-ET(m)>1
        Start=m+1;
        m=m+1;
        break;
      else m=m+1;
    end;
  end;
m=Start;
  while m<=length(ET)
    if ET(m+1)-ET(m)>1
      Last=m+11;
      m=m-10;
      break;
    else m=m+1;
  end;
end;
end;
if ET(m+1) - ET(m) > 1
    Last = m;
    m = m + 1;
    break;
else
    m = m + 1;
end;
end;
while m <= length(ET)
    if ET(m+1) - ET(m) > 1
        m = m + 1;
        break;
    else
        m = m + 1;
    end;
end;
while SA(Start) <= 3
    Start = Start + 1;
end;
while SA(Last) >= 2
    Last = Last - 1;
end;
switch TestTypes(n)
    case 1
        int2str(databin(round(mean(P)), Pressures)) '.IA'
        int2str(round(mean(IA(Start:Last)))) '.FZ'
        int2str(abs(databin(round(mean(FZ(Start:Last))), NormalLoads))) '.]
    case 2
        int2str(databin(round(mean(P)), Pressures)) 'COLD.IA'
        int2str(round(mean(IA(Start:Last)))) '.FZ'
        int2str(abs(databin(round(mean(FZ(Start:Last))), NormalLoads))) '.]
end
eval([MainTxt 'testid=testid;']);
 eval([MainTxt 'tireid=tireid;']);
 eval([MainTxt 'SOURCE=matfile;']);
 eval([MainTxt 'AMBTMP=AMBTMP(Start:Last);']);
 eval([MainTxt 'ET=ET(Start:Last);']);
 eval([MainTxt 'FX=FX(Start:Last);']);
 eval([MainTxt 'FY=-FY(Start:Last);']);
 eval([MainTxt 'FZ=-FZ(Start:Last);']);
 eval([MainTxt 'IA=IA(Start:Last);']);
 eval([MainTxt 'MX=MX(Start:Last);']);
 eval([MainTxt 'MZ=-MZ(Start:Last);']);
 eval([MainTxt 'N=N(Start:Last);']);
 eval([MainTxt 'NFX=NFX(Start:Last);']);
 eval([MainTxt 'NFY=NFY(Start:Last);']);
 eval([MainTxt 'P=P(Start:Last);']);
 eval([MainTxt 'RE=RE(Start:Last);']);
 eval([MainTxt 'RL=RL(Start:Last);']);
 eval([MainTxt 'RST=RST(Start:Last);']);
 eval([MainTxt 'SA=SA(Start:Last);']);
 eval([MainTxt 'SL=SL(Start:Last);']);
 eval([MainTxt 'SR=SR(Start:Last);']);
 eval([MainTxt 'TSTC=TSTC(Start:Last);']);
 eval([MainTxt 'TSTI=TSTI(Start:Last);']);
 eval([MainTxt 'TSTO=TSTO(Start:Last);']);
 eval([MainTxt 'V=V(Start:Last);']);
end;
k=k+1;
case 16
for p=1:TestOrder(k+1)
    Start=m;
    while m<=length(ET)
        if m==length(ET)
            Last=m;
            m=1;
            break;
        elseif ET(m+1)-ET(m)>1
            Last=m;
            m=m+1;
            break;
        else m=m+1;
    end;
end;
switch TestTypes(n)
    case 3
        MainTxt=[FileName '.BrakeDrive'
            int2str(abs(round(mean(SA(Start:Last))))) 'SA.Pa'
            int2str(databin(round(mean(P)),Pressures)) '.IA'
            int2str(round(mean(IA(Start:Last)))) '.FZ'
            int2str(abs(databin(round(mean(FZ(Start:Last))),NormalLoads))) '.']
    case 4
        MainTxt=[FileName '.BrakeDrive'
            int2str(abs(round(mean(SA(Start:Last))))) 'SA.Pa'
            int2str(databin(round(mean(P)),Pressures)) 'COLD.IA'
            int2str(round(mean(IA(Start:Last)))) '.FZ'
            int2str(abs(databin(round(mean(FZ(Start:Last))),NormalLoads))) '.']
end
eval([MainTxt 'testid=testid;'])
eval([MainTxt 'tireid=tireid;'])
eval([MainTxt 'SOURCE=matfile;'])
eval([MainTxt 'AMBTMP=AMBTMP(Start:Last);'])
eval([MainTxt 'ET=ET(Start:Last);'])
eval([MainTxt 'FX=FX(Start:Last);'])
eval([MainTxt 'FY=-FY(Start:Last);'])
eval([MainTxt 'FZ=-FZ(Start:Last);'])
eval([MainTxt 'IA=IA(Start:Last);'])
eval([MainTxt 'MX=MX(Start:Last);'])
eval([MainTxt 'Mz=-Mz(Start:Last);'])
eval([MainTxt 'N=N(Start:Last);'])
eval([MainTxt 'NFX=NFX(Start:Last);'])
eval([MainTxt 'NFY=NFY(Start:Last);'])
eval([MainTxt 'P=P(Start:Last);'])
eval([MainTxt 'RE=RE(Start:Last);'])
eval([MainTxt 'RL=RL(Start:Last);'])
eval([MainTxt 'RST=RST(Start:Last);'])
eval([MainTxt 'SA=SA(Start:Last);'])
eval([MainTxt 'SL=SL(Start:Last);'])
eval([MainTxt 'SR=SR(Start:Last);'])
eval([MainTxt 'TSTC=TSTC(Start:Last);'])
eval([MainTxt 'TSTI=TSTI(Start:Last);'])
eval([MainTxt 'TSTO=TSTO(Start:Last);'])
eval([MainTxt 'V=V(Start:Last);'])
end
k=k+1;
case 17
  while m<=length(ET)
    if round(FZ(m)/-222)==1 && ET(m+1)-ET(m)>1
      m=m+1;
      break;
    else
      m=m+1;
    end;
  end;
end;

switch TestOrder(k)
case {11, 12}
  eval([MainTxt 'testid=testid;']);
  eval([MainTxt 'tireid=tireid;']);
  eval([MainTxt 'SOURCE=matfile;']);
  eval([MainTxt 'AMBTMP=AMBTMP(Start:Last);']);
  eval([MainTxt 'ET=ET(Start:Last);']);
  eval([MainTxt 'FX=FX(Start:Last);']);
  eval([MainTxt 'FY=-FY(Start:Last);']);
  eval([MainTxt 'FZ=-FZ(Start:Last);']);
  eval([MainTxt 'IA=IA(Start:Last);']);
  eval([MainTxt 'MX=MX(Start:Last);']);
  eval([MainTxt 'MZ=-MZ(Start:Last);']);
  eval([MainTxt 'N=N(Start:Last);']);
  eval([MainTxt 'NFX=NFX(Start:Last);']);
  eval([MainTxt 'NFY=NFY(Start:Last);']);
  eval([MainTxt 'P=P(Start:Last);']);
  eval([MainTxt 'RE=RE(Start:Last);']);
  eval([MainTxt 'RL=RL(Start:Last);']);
  eval([MainTxt 'RST=RST(Start:Last);']);
  eval([MainTxt 'SA=-SA(Start:Last);']);
  eval([MainTxt 'SL=SL(Start:Last);']);
  eval([MainTxt 'SR=SR(Start:Last);']);
  eval([MainTxt 'TSTC=TSTC(Start:Last);']);
  eval([MainTxt 'TSTI=TSTI(Start:Last);']);
  eval([MainTxt 'TSTO=TSTO(Start:Last);']);
  eval([MainTxt 'V=V(Start:Last);']);
  end;
end;
end;

disp('Data processing completed successfully');
eval(['save ' FileName '.mat ' FileName ';']);
OutputTxt=['Data structure ' FileName ' saved successfully to ' FileName '.mat.'];
disp(OutputTxt);
clear all;

function [Index]=databin(X,Y)
% DATABIN Returns the index of the nearest value in Y to X
% [INDEX] = databin(X,Y) Returns the index of the value in the vector Y
% that is closest to the value of X.

[MIN Ind]=min(abs(Y-X));
D.2. PAC2002 Fitting Script

```matlab
clear all;
close all;

% Pacejka 2002 Adams Curve Fitting Utility
% Round 4 FSAE TTC data sectored utilizing data wrangler.

pFileName='GoodyearD2696_7inch'; D=20.0;
pFileName='GoodyearD2696_8inch'; D=20.0; % No Pa55
pFileName='Hoosier20x6_13_6inch'; D=20.5;
pFileName='Hoosier20x6_13_7inch'; D=20.5;
pFileName='Hoosier20x7_13_6inch'; D=20.5;
pFileName='Hoosier20x7_13_7inch'; D=20.5;
pFileName='Hoosier20x75_13_6inch'; D=20.5; % No Pa55
pFileName='Hoosier20x75_13_7inch'; D=20.5; % No Pa55
pFileName='Hoosier20x75_13_8inch'; D=20.5; % No Pa55

% Size is 175/505R13 for Dunlop Tires
% pFileName='Dunlop_7inch'; D=(2*175*.505+13*25.4)/1000*2/.0254;
% pFileName='Dunlop_8inch'; D=(2*175*.505+13*25.4)/1000*2/.0254;
% Size is 160/53R13. The section width given in the documentation does not work. 160 is a better guess.
% pFileName='Michelin_7inch'; D=(2*160*.53+13*25.4)/1000*2/.0254;
% pFileName='Michelin_8inch'; D=(2*160*.53+13*25.4)/1000*2/.0254;

eval(['load ' pFileName '.mat;'])

% Pressure run for fitting
% Possibilities are [Pa55 Pa69 Pa83 Pa97 Pa83COLD]
pressure='Pa83';

% Nominal Wheel Load and Unloaded Radius
pR0=D*.0254/2; % Radius in m
%pR0=; % Radius in m for Dunlop
%pR0=(2*160*.53+13*25.4)/1000; % Radius in m for Michelin

% Debug graph reference counter
DaBug=100;

% Plotting Options - Graphs for validation and debug, not presentation
%pureside=1;
%pureside graphs for Longitudinal Force - Pure Long Slip data.
purelong=1;
%pureside graphs for Aligning Torque - Pure Lateral Slip data.
purezmom=1;
```
% Plots validation graphs for Longitudinal Force - Combined Slip data.
comblong=1;
% Plots validation graphs for Lateral Force - Combined Slip data.
combside=1;
% Plots validation graphs for Aligning Moment - Combined Slip data.
combzmom=1;
% Plots validation graphs for Overturning Moment, the same parameters are
% utilized for pure and combined data.
combxmom=1;
% Debug plots for Fy0 determination
pebugFy0=0;
% Debug plots for Fx0 determination
pebugFx0=0;
% Plots debug plots for Aligning Torque coefficient loop.
pebugzmo=0;
% Plot thermal curves for raw Fy0 data.
thermFy0=0;
% Plot thermal curves for raw Fx0 data.
thermFx0=0;

% Modified D_t fitment procedure - generates alternate file!
pDmztMod=0;

writefile=1; % Record finished coefficients to mat file for storage.
% eval(['save ' wFileName '.mat ' wVariable]);

pColors=get(gca,'ColorOrder'); % Add standard colors to plots.

% Fitting Method Switches
% Alternate Fx0 - pEx3 Fitment - To fix exhorbitant pEx3 magnitude
pEx3Switch=0;

%% Lateral Force - Pure Side Slip - Initial Coefficient Definition
% Data analysis done for Pure Side Slip runs at pFz0 to ultimately determine
% pFz0 and shape factor, C, which is characterized solely by pCy1.

% Cornering load averaged from the test given will be used.
% The middle loading of 667 N (150 lbf) will be utilized as pFz0.
FZN=[pFileName '.Cornering.' pressure '.IA0.FZ667.FZ'];
SAN=[pFileName '.Cornering.' pressure '.IA0.FZ667.SA'];
FYN=[pFileName '.Cornering.' pressure '.IA0.FZ667.FY'];
TIREN=[pFileName '.Cornering.' pressure '.IA0.FZ667.tireid'];

eval(['FZ=' FZN '']);
eval(['SA=' SAN '']);
eval(['FY=' FYN '']);
eval(['tireid=' TIREN '']);
k=1;
bnd=[1,length(SA)];
for n=1:length(SA)
    if SA(n)>=0 && SA(n+1)<0 && k==1
        bnd(k,1)=n;
k=k+1;
    elseif SA(n)>0 && SA(n+1)<=0 && k==2
        bnd(k,1)=n;
k=k+1;
    else
        bnd(k,1)=n;
k=k+1;
    end
end
elseif k==3
    break;
end

% Test plot for truncation check.
if pebugFy0==1
    figure(DaBug); DaBug=DaBug+1; hold on;
    plot(SA(bnd(1,1):bnd(2,1)),FY(bnd(1,1):bnd(2,1)),'');
    title('Initial Fy Truncation');
end

% Compute pFz0 as the mean of the FZ for this data set.
pFz0=(mean(FZ(bnd(1,1):bnd(2,1))));

% Fit curve to FY data to collapse hysteresis-like condition. The
% difference in curves is possibly due to turn-slip effects as proposed by
% Pacejka. Only one point of data is available to fit to though.
SmoothSpan=19;
[SAfit,Sord]=sort(SA(bnd(1,1):bnd(2,1)));
SArad=SAfit*pi/180;
FYtrunc=(FY(bnd(1,1):bnd(2,1)));
FYfit=FYtrunc(Sord);
FYsmth=smooth(FYfit,SmoothSpan,'lowess');

[SAU Umat]=unique(SArad);
FYU=FYsmth(Umat);

% Data smoothing test plot.
if pebugFy0==1
    figure(DaBug); DaBug=DaBug+1; hold on;
    plot(SA(bnd(1,1):bnd(2,1))*pi/180,FY(bnd(1,1):bnd(2,1)),'...',
    SArad,FYfit,'r*',SArad,FYsmth,'g-');
    title('Fy Initial Smoothing Check');
end

pDy1=(max(FYU)+abs(min(FYU)))/2/pFz0;
pVy1=-((max(FYU)-((max(FYU)+abs(min(FYU)))/2))/pFz0;
FYU=(FYU+pFz0*pVy1);
FY1=FYU/(pDy1*pFz0);

% Test plots of Unique data and Normalized Data
if pebugFy0==1
    figure(DaBug); DaBug=DaBug+1; hold on;
    plot(SAU,FYU); title('Fy Unique Data Check');
    figure(DaBug); DaBug=DaBug+1; hold on;
    plot(SAU,FY1); title('Fy Normalized Data by Factor D');
end

% Set bound region around Fy=0 to define region to fit the cornering
% stiffness, Ky, over. The shorter the range generally the more severe the
% slope found.
k=1;
FYbound=.25*max(FYU);
for n=1:length(FYU)
if FYU(n)>FYbound && FYU(n+1)<=FYbound
    bnd(1,1)=n;
    k=k+1;
elseif FYU(n)>-FYbound && FYU(n+1)<=-FYbound
    bnd(2,1)=n+1;
    k=k+1;
elseif k==3
    break;
end

% A first order linear fit of the Fy=0 region yields the cornering
% stiffness and horizontal shift of the Fy curve.
PolyOrig=polyfit(SAU(bnd(1,1):bnd(2,1)),FYU(bnd(1,1):bnd(2,1)),1);
Ky=PolyOrig(1);
pHy1=PolyOrig(2)/PolyOrig(1);

% Apply horizontal shift to the slip angle vector.
SA1=SAU+pHy1;

% Define Shape Factor C and Curvature Factor E concurrently for best
% possible fit. pCy1 will not be iterated any further than this initial
% setup.
pCy1=2.0;
E=1;
DpCy1=.15;
DE=1;

pCy1t=pCy1;
Et=E;
DpCy1t=DpCy1;
DEt=DE;
Byt=Ky/((pCy1t*DpCy1t)*Dy1pFz0)*SA1;
MCy1=sum(abs((abs(FY1)-abs(sin(pCy1t*atan(Byt-Et.*(Byt-atan(Byt))))))));
while 1
    MCy2=sum(abs((abs(FY1)-abs(sin((pCy1t-DpCy1t)*atan(Byt-(Et-DEt).*...))-(Byt-atan(Byt))))));
    if MCy2<MCy1
        Et=Et DEt;
        MCy1=MCy2;
    elseif MCy2>MCy1
        Et=Et-DEt;
        DEt=-DEt/2;
        MCy11=MCy1;
    end
    if abs(DEt)<.00000001
        break;
    end
end

MCy10=MCy11;
while 1
    Et=E;
    DEt=DE;
    Byt=Ky/((pCy1t-DpCy1t)*Dy1pFz0)*SA1;
    MCy1=sum(abs((abs(FY1)-abs(sin((pCy1t-DpCy1t)*atan(Byt-(Et).*...))));
(Byt-atan(Byt))));

MCy0=MCy1;
while 1
    MCy2=sum(abs((abs(FY1)-abs(sin((pCy1t-DpCy1t)*atan(Byt-...
    (Et-DEt).*(Byt-atan(Byt)))))));)
    if MCy2<MCy1
       Et=Et-DEt;
       MCy0=MCy1;
       MCy1=MCy2;
    elseif MCy2>=MCy1
       Et=Et-DEt;
       DEt=-DEt/2;
       MCy12=MCy1;
       MCy1=MCy0;
    end
    if abs(DEt)<.0000001
       break;
    end
end
if MCy12<MCy11
    pCy1t=pCy1t-DpCy1t;
    MCy10=MCy11;
    MCy11=MCy12;
elseif MCy12>=MCy11
    pCy1t=pCy1t-DpCy1t;
    DpCy1t=-DpCy1t/2;
    MCy11=MCy10;
end
if abs(DpCy1t)<.0000001
    pCy1=pCy1t;
    break;
end
end
if pebugFy0==1
    figure(DaBug); DaBug=DaBug+1; hold on;
    plot(SA1,FY1,‘’,SA1,sin((pCy1t+DpCy1t)*atan(Byt-(Et-DEt).*...
        (Byt-atan(Byt))));,’');
    title(‘Factors C and E Effectiveness - Overall Fit’);
end

%% Lateral Force - Pure Side Slip - Full Coefficient Definition
% Curve fit for the full test matrix of loads and inclination angles.

% Inclination Angle Runs
IAP=[0 1 2 3 4];
% Normal Load Runs
FZP=[222 445 667 1112 1557];

% Declare variables
Fzy=zeros(length(FZP),length(IAP));
dFzy=zeros(size(Fzy));
Gammay=zeros(size(Fzy));
Dy=zeros(size(Fzy));
Mewy=zeros(size(Fzy));
SVy=zeros(size(Fzy));
SHy=zeros(size(Fzy));
Ky=zeros(size(Fzy));
By=zeros(size(Fzy));
EyP=zeros(size(Fzy));
EyN=zeros(size(Fzy));

% The stiffness bounds determines the portion of max(Fy) to perform the
% linear fit for the cornering stiffness K. A larger ratio lessens the
% fitted stiffness for the particular run. Useful for tuning in full
% charts for linearity to measured data.
StiffnessBnd=[...
  .35 .35 .35 .35 .45;...
  .30 .30 .30 .30 .30;...
  .25 .25 .25 .25 .45;...
  .20 .20 .20 .20 .20;...
  .10 .10 .10 .10 .10];

for n=1:length(IAP)
  for m=1:length(FZP)
    clear FZN SAN FYN FZ SA FY k p SAfit Sord SARad FYtrunc...
    FYfit FYsmth SAU Umat FYU
    FZN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
         '.FZ' int2str(FZP(m)) '.FZ'];
    SAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
         '.FZ' int2str(FZP(m)) '.SA'];
    FYN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
         '.FZ' int2str(FZP(m)) '.FY'];
    IAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
         '.FZ' int2str(FZP(m)) '.IA'];
    eval(['FZ=' FZN ';']);
    eval(['SA=' SAN ';']);
    eval(['FY=' FYN ';']);
    eval(['IA=' IAN ';']);
    if thermFy0==1
      TSTCN=[pFileName '.Cornering.' pressure '.IA'...
             int2str(IAP(n)) '.FZ' int2str(FZP(m)) '.TSTC'];
      TSTIN=[pFileName '.Cornering.' pressure '.IA'...
             int2str(IAP(n)) '.FZ' int2str(FZP(m)) '.TSTI'];
      TSTON=[pFileName '.Cornering.' pressure '.IA'...
             int2str(IAP(n)) '.FZ' int2str(FZP(m)) '.TSTO'];
      RSTN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
             '.FZ' int2str(FZP(m)) '.RST'];
      AMBTMPN=[pFileName '.Cornering.' pressure '.IA'...
                int2str(IAP(n)) '.FZ' int2str(FZP(m)) '.AMBTMP'];
      ETN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
           '.FZ' int2str(FZP(m)) '.ET'];
      eval(['TSTC=' TSTCN ';']);
      eval(['TSTI=' TSTIN ';']);
      eval(['TSTO=' TSTON ';']);
      eval(['RST=' RSTN ';']);
      eval(['AMBTMP=' AMBTMPN ';']);
      eval(['ET=' ETN ';']);
    end
  end
end
% Truncate data to one full sweep minimizing start and end effects
k=1;
bnd=[1:length(SA)];
for p=1:length(SA)
    if SA(p)>=0 && SA(p+1)<0 && k==1
        bnd(k,1)=p;
k=k+1;
    elseif SA(p)>0 && SA(p+1)<=0 && k==2
        bnd(k,1)=p;
k=k+1;
    elseif k==3
        break;
end
end
if thermFy0==1
    k=15;
    figure(300+n); hold on;
    plot(smooth(FY(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...bnd(2,1)),k),TSTC(bnd(1,1):bnd(2,1)),'Color',pColors(m,:));
    figure(310+n); hold on;
    plot(smooth(FY(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...bnd(2,1)),k),TSTI(bnd(1,1):bnd(2,1)),'Color',pColors(m,:));
    figure(320+n); hold on;
    plot(smooth(FY(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...bnd(2,1)),k),TSTO(bnd(1,1):bnd(2,1))-'Color',pColors(m,:));
    figure(330+n); hold on;
    plot(smooth(FY(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...bnd(2,1)),k),RST(bnd(1,1):bnd(2,1)),'Color',pColors(m,:));
    figure(340+n); hold on;
    figure(350+n); hold on;
    figure(360+n); hold on;
    DFy0=diff(smooth(FY,25))./diff(ET);
    figure(370+n); hold on;
    plot3(smooth(FY(1:length(DFy0)),k)./smooth(FZ(1:length(DFy0)...),k),smooth(DFy0./FZ(1:length(DFy0))...),k),TSTC(1:length...(DFy0)),'Color',pColors(m,:));
end

% Compute dFz for the given run
% dFz = (Fz - pFz0)/pFz0
Fzy(m,n)=mean(FZ(bnd(1,1):bnd(2,1)));
dFzy(m,n)=(Fzy(m,n)-pFz0)/pFz0;

% Record IA for given run
Gammay(m,n)=(mean(IA(bnd(1,1):bnd(2,1)))*pi/180);
% Smooth data to allow fitting
% Fit curve to FY data to eliminate hysteresis.
SmoothSpan=25;
[SAfit,Sord]=sort(SA(bnd(1,1):bnd(2,1)));  
SArad=SAfit*pi/180;
FYtrunc=(FY(bnd(1,1):bnd(2,1)));  
FYfit=FYtrunc(Sord);
FYsmth=smooth(FYfit,SmoothSpan,'lowess');

[SAU Umat]=unique(SArad);
FYU=FYsmth(Umat);
FYR=FYU;
SAR=SAU;

% Compute Peak Value factor D and mew
Dy(m,n)=(max(FYU)+abs(min(FYU)))/2;
Mewy(m,n)=Dy(m,n)/Fzy(m,n);

% Compute Vertical shift factor SVy and shift data curve
SVy(m,n)=max(FYU)-((max(FYU)+abs(min(FYU)))/2);
FYU=FYU-SVy(m,n);

% Compute Stiffness K (dFy/dSA @ SA=0)
% Define the range over which the linear fit will occur
k=1;
FYbound=StiffnessBnd(m,n)*max(FYU);
for p=1:length(FYU)
    if FYU(p)>FYbound && FYU(p+1)<=FYbound
        bnd(1,1)=p;
        k=k+1;
    elseif FYU(p)>-FYbound && FYU(p+1)<=-FYbound
        bnd(2,1)=p+1;
        k=k+1;
    elseif k==3
        break;
    end;
end;

% Perform 1st order linear fit over the defined range to define K
PolyOrig=polyfit(SAU(bnd(1,1):bnd(2,1)),FYU(bnd(1,1):bnd(2,1)),1);
Ky(m,n)=PolyOrig(1);

% Utilizing K and the y-intercept compute the horizontal shift SHy
SHy(m,n)=PolyOrig(2)/PolyOrig(1);
% Shift curve to origin
SAU=SAU+SHy(m,n);

% Utilizing C, D, and K compute stiffness factor By
By(m,n)=Ky(m,n)/(pCy1*Dy(m,n));

% Plot of shifted and unshifted smoothed curves.
if puriside==1
    figure(n);
    hold on

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plot(SAU,FYU,'b-');
figure(n+80);
hold on
plot(SAR,FYR,'b-');
end;

% Compute Curvature Factor E
% E is dependent on the sign of the SA, it is thus necessary to fit
% E seperately to positive and negative SA.
FYU=FYU/Dy(m,n);
[SAmmin Eymnin]=min(abs(SAU));
SApos=SAU(Eymnin:length(SAU));
SAneg=SAU(1:Eymnin);
FYUpos=FYU(Eymnin:length(FYU));
FYUneg=FYU(1:Eymnin);

% Begin positive side E definition
Etest=-10;
teststep=1;
SAtest=SApos;
FYUtest=FYUpos;
FYP=sin(pCy1.*atan(By(m,n).*SApos-(Etest.*(By(m,n).*SApos-
   atan(By(m,n).*SApos)))))
while 1
   if Etest>=-2 && teststep==1
      teststep=.1;
   end;
   FYP1=sin(pCy1.*atan(By(m,n).*SAtest-(Etest+teststep).*(By(m,n).*SAtest-
       atan(By(m,n).*SAtest))));
delFYP=mean(abs(FYP-FYUtest));
delFYP1=mean(abs(FYP1-FYUtest));
   if delFYP1<delFYP
      Etest=Etest+teststep;
      FYP=FYP1;
   elseif delFYP1>=delFYP
      Etest=Etest+teststep;
      FYP=FYP1;
      teststep=-.25*teststep;
   end;
   if abs(teststep)<=.000000001
      if pureside==1
         figure(n);
         hold on
         plot(SAtest,FYP.*Dy(m,n),'g--');
      end;
      EyP(m,n)=Etest;
      break;
   end;
end;
% Begin negative side E definition
Etest=-10;
teststep=1;
SAtest=SAneg;
FYUtest=FYUneg;
FYP=sin(pCy1.*atan(By(m,n).*SAtest-(Etest.*(By(m,n).*SAtest-
   atan(By(m,n).*SAtest)))))
while 1
if Etest>=-2 \&\& teststep=1
    teststep=.1;
end;

FYP1=sin(pCyl.*atan(By(m,n).*SAtest-(Etest+teststep).*...
    (By(m,n).*SAtest-atan(By(m,n).*SAtest))));
delFYP=mean(abs(FYP-FYUtest));
delFYP1=mean(abs(FYP1-FYUtest));
if delFYP1<delFYP
    Etest=Etest+teststep;
    FYP=FYP1;
elseif delFYP1>=delFYP
    Etest=Etest+teststep;
    FYP=FYP1;
    teststep=-.25*teststep;
end
if abs(teststep)<.000000001
    if pureside==1;
        figure(n);
        hold on
        plot(SAtest,FYP.*Dy(m,n),'g--');
    end;
    EyN(m,n)=Etest;
    break;
end;
end;
end

%% Lateral Force - Pure Side Slip - Final Coefficient Definition
%% Lateral Force (Pure Side Slip) Coefficients
%% User scaling factors not listed in equations
%% Nominal conditions are matrix element (3,1)

% Fitting Controls
if strcmp(pFileName,'Michelin_7inch')
    PEY34t=[0 1];
elseif strcmp(pFileName,'Hoosier20x75_13_8inch') \&\& strcmp(pressure,'Pa69')
    PEY34t=[1 0];
else
    PEY34t=[0 0];
end

% Raw Fit Data Storage
pRAWFY0Fit.Fzy=Fzy;
pRAWFY0Fit.Gammay=Gammay;

% SHy
% SHy = (pHy1+pHy2*dFz)+pHy3*gamma
if pureside==1
    figure(1001);
    mesh(IAP,FZP,SHy.*180./pi); title('SH_y (deg)'); xlabel('IA (deg)'); ylabel('FZ (N)');
end
pRAWFY0Fit.SHy=SHy;
pHy1=SHy(3,1);
for k=1:length(FZP)
PHY2(k,:) = polyfit(dFzy(:,k), SHy(:,k), 1);
end
pHy2 = mean(PHY2(:, 1));
for k = 1:length(IAP)
    PHY3(k,:) = polyfit(Gammay(k,:), SHy(k,:), 1);
end
pHy3 = mean(PHY3(:, 1));

% SVy
% SVy = Fz*(pVy1 + pVy2*dFz + (pVy3 + pVy4*dFz)*gamma)
if pureside == 1
    figure(1002);
    mesh(IAP, FZP, SVy); title('SV_y'); xlabel('IA (deg)'); ylabel('FZ (N)');
end
pRAWFY0Fit.SVy = SVy;
SVy0 = SVy;
SVy = SVy ./ Fzy;
pVy1 = SVy(3,1);
[Slope Offset] = polyfit(dFzy(:,1), SVy(:,1), 1);
pVy2 = Slope(1);
[Slope Offset] = polyfit(Gammay(3,:), SVy(3,:), 1);
pVy3 = Slope(1);
SVy = (SVy-pVy1-pVy2.*dFzy-pVy3.*Gammay);
for k = 2:length(IAP)
    PVY34(k,:) = polyfit(dFzy(:,k).*Gammay(:,k), SVy(:,k), 1);
end
pVy4 = mean(PVY34(:, 1));
SVy = SVy ./ (Gammay.*dFzy);
% pVy4 = 0;

% Shape Factor C was determined manually using curve at (3,1)

% Peak Factor D
% Determine coefficients for Mewy
% Mewy = (pDy1 + pDy2*dFz)*(1 - pDy3*gamma^2)
% Method 2
pRAWFY0Fit.Mewy = Mewy;
if pureside == 1
    figure(1003);
    mesh(IAP, FZP, Mewy); title('Mu_y'); xlabel('IA (deg)'); ylabel('FZ (N)');
end
for k = 1:length(FZP)
    PDY12(k,:) = polyfit(dFzy(:,k), Mewy(:,k), 1);
end
pDy1 = PDY12(3,2);
pDy2 = mean(PDY12(:,1));
Mewy1 = Mewy;
Mewy = (Mewy ./ (pDy1+pDy2*dFzy)) - 1;
for k = 1:length(FZP)
    PDY3(k,:) = polyfit(Gammay(k,:), Mewy(k,:), 2);
end
pDy3 = -1 * mean(PDY3(:,1));
Mewy2 = Mewy;
Mewy3 = Mewy + pDy3 * Gammay.^2;
% Curvature Factor E
% \( E_y = (pE_{y1} + pE_{y2} \cdot dFz) \cdot (1 - (pE_{y3} + pE_{y4} \cdot \gamma) \cdot \text{sgn}(SA)) \)

\[
E_{yM} = (E_{yP} + E_{yN}) / 2;
\]

\[
p_{RAWFY0Fit.EyM} = E_{yM};
\]

\[
p_{RAWFY0Fit.EyP} = E_{yP};
\]

\[
p_{RAWFY0Fit.EyN} = E_{yN};
\]

if pureside == 1
    figure(1004);
    mesh(IAP, FZP, E_{yM});
    title('E_y Mean'); xlabel('IA (deg)'); ylabel('FZ (N)');
    figure(1005);
    mesh(IAP, FZP, E_{yP});
    title('E_y Positive SA'); xlabel('IA (deg)'); ylabel('FZ (N)');
    figure(1006);
    mesh(IAP, FZP, E_{yN});
    title('E_y Negative SA'); xlabel('IA (deg)'); ylabel('FZ (N)');
end

p_{Ey1} = E_{yM}(3,1);
for k = 1:length(FZP)
    PEY2(k,:) = polyfit(dFzy(:,k), E_{yM}(:,k), 1);
end

p_{Ey2} = mean(PEY2(:,1));
E_{yPD} = (E_{yP} ./ (p_{Ey1} + p_{Ey2} \cdot dFzy) - 1) / (-1); % Divide by -\text{sgn}(SA)
E_{yND} = (E_{yN} ./ (p_{Ey1} + p_{Ey2} \cdot dFzy) - 1);
E_{yD} = (E_{yPD} + E_{yND}) / 2;

E_{y34Confidence} = 2;
E_{y34Criteria} = max([std(E_{yPD}, 0, 2) std(E_{yND}, 0, 2)], [], 2);
E_{y34Test} = mean(mean([std(E_{yND}, 0, 2) std(E_{yPD}, 0, 2)])) .* E_{y34Confidence};
m = 1;
for k = 1:length(FZP)
    if E_{y34Criteria}(k) <= E_{y34Test}
        E_{yDt}(m,:) = E_{yD}(k,:);
        m = m + 1;
    end
end
for k = 1:m-1
    PEY34(k,:) = polyfit(Gammay(k,:), E_{yDt}(k,:), 1);
end

p_{Ey4} = mean(PEY34(:,1));
p_{Ey3} = mean(PEY34(:,2));

% Stiffness Factor B
% Stiffness Ky
p_{RAWFY0Fit.Ky} = Ky;
Ky2 = Ky;
if pureside == 1
    figure(1007);
    mesh(IAP, FZP, Ky); title('K_y'); xlabel('IA (deg)'); ylabel('FZ (N)');
end

Ky = Ky / p_{Fz0};
if pureside == 1
    figure(1008);
    mesh(IAP, FZP, Ky); title('K_y / F_z_0'); xlabel('IA (deg)'); ylabel('FZ (N)');
end

[p_{Ky1p} p_{Ky1i}] = max(abs(Ky(:,1)));
pKy1=pKy1^2/Ky(pKy1,1);
pKy2=Fzy(pKy1,1)/pFz0;
if pKy1<length(Ky(:,1)) && pKy1>1
    if (abs(Ky(pKy1+1))-abs(pKy1))<=(abs(Ky(pKy1-1))-abs(pKy1))
        DpKy1=1*pKy1/abs(pKy1);
        DpKy2=(Fzy(pKy1+1)/pFz0-pKy2)/10;
    elseif (abs(Ky(pKy1-1))-abs(pKy1))<=(abs(Ky(pKy1+1))-abs(pKy1))
        DpKy1=1*pKy1/abs(pKy1);
        DpKy2=(Fzy(pKy1-1)/pFz0-pKy2)/10;
    end
else pKy1==length(Ky(:,1))
    DpKy1=1*pKy1/abs(pKy1);
    DpKy2=(pKy2-Fzy(length(Fzy(:,1))-1)/pFz0)/10;
elseif pKy1==1
    DpKy1=1*pKy1/abs(pKy1);
    DpKy2=(pKy2-(Fzy(2)-1)/pFz0)/10;
end

% Curve fitment for pKy1 and pKy2
MpKy1=mean(abs(Ky(:,1)-pKy1*sin(2*atan(Fzy(:,1)/(pKy2*pFz0)))));
MpKy0=MpKy1;
pKy2t=pKy2;
pKy1t=pKy1;
DpKy2t=DpKy2;
while 1
    MpKy2=mean(abs(Ky(:,1)-pKy1t*sin(2*atan(Fzy(:,1)/((pKy2t+DpKy2t)*pFz0)))));
    if MpKy2<=MpKy1
        pKy1t=pKy1t+DpKy1t;
        MpKy0=MpKy1;
        MpKy1=MpKy2;
    elseif MpKy2>MpKy1
        pKy2t=pKy2t-DpKy2t;
        DpKy2t=DpKy2t/4;
        MpKy11=MpKy1;
        MpKy1=MpKy0;
    end
    if abs(DpKy2t)<.00000001
        break;
    end
end

MpKy10=MpKy11;
DpKy1t=DpKy1;
while 1
    pKy2t=pKy2;
    DpKy2t=DpKy2;
    MpKy1=mean(abs(Ky(:,1)-(pKy1t+DpKy1t)*sin(2*atan(Fzy(:,1)/((pKy2t+DpKy2t)*pFz0)))));
    MpKy0=MpKy1;
    while 1
        MpKy2=mean(abs(Ky(:,1)-(pKy1t+DpKy1t)*sin(2*atan(Fzy(:,1)/((pKy2t+DpKy2t)*pFz0)))));
        if MpKy2<MpKy1
            pKy2t=pKy2t+DpKy2t;
            MpKy0=MpKy1;
            MpKy1=MpKy2;
        end
    end
end
elseifMpKy2>=MpKy1
    pKy2t=pKy2t-DpKy2t;
    DpKy2t=DpKy2t/4;
    MpKy12=MpKy1;
    MpKy1=MpKy0;
end
if abs(DpKy2t)<.00000001
    break;
end
end
if MpKy12<MpKy11
    pKy1t=pKy1t+DpKy1t;
    MpKy10=MpKy11;
   MpKy11=MpKy12;
elseif MpKy12>=MpKy11
    pKy1t=pKy1t-DpKy1t;
    DpKy1t=DpKy1t/4;
    MpKy11=MpKy10;
end
if abs(DpKy1t)<.00000001
    pKy1=pKy1t;
pKy2=pKy2t;
    break;
end
end

clear pKyli pKylp pKylt pKy2t
Ky=(Ky./(pKy1*sin(2*atan(Fzy/(pKy2*pFz0))))) -1;
for k=1:length(IAP)
    PKY3(k,:) = polyfit(abs(Gammay(k,:)),Ky(k,:),1);
end
pKy3=-1*mean(PKY3(:,1));

% Plot Derived Curves vs. Data Curves
if pureside==1
    Dy1=Fzy.*(pDy1+pDy2*dFzy).*(1-pDy3*Gammay.^2);
    Kyl=pKy1.*sin(2*atan(Fzy/(pKy2*pFz0))).*(1-pKy3*abs(Gammay));
    By1=Kyl./(pCy1*Dy1);
    SAD=-12*pi/180:.01*pi/180:12*pi/180;
    SHy1=(pHy1+pHy2.*dFzy)+pHy3.*Gammay;
    SVy1=Fzy.*((pVy1+pVy2.*dFzy)+(pVy3+pVy4.*dFzy).*Gammay);
    for n=1:length(IAP)
        for m=1:length(FZP)
            FYD=Dy1(m,n).*sin(pCy1.*atan(By1(m,n).*SAD-(pEy1+pEy2.*dFzy(m,n).*Gammay(m,n)))*abs(SAD)./SAD).
            *(By1(m,n).*SAD-atan(By1(m,n)*SAD));
        figure(n);
        hold on
        plot(SAD,FYD,'r-');
        figure(n+80);
        hold on
        plot(SAD-SHy1(m,n),FYD+SVy1(m,n),'r--');
    end
end
end

%Perform some cleanup!
pDaBug=DaBug;
clear B* D* E* F* G* I* K* M* O* P* R* S* U* d* j* k* m* n* p...
    pCylt teststep;
DaBug=pDaBug;
% close all;

%%% Longitudinal Force - Pure Longitudinal Slip - Initial Coefficient Definition
% Data analysis done for Pure Longitudinal Slip runs at pFz0 to ultimately % determine shape factor, C, which is characterized solely by pCx1.

eval(['load ' pFileName '.mat;']);

% Cornering load averaged from the test given will be used.
% The middle loading of 667 N (150 lbf) will be utilized.
FZN=[pFileName '.BrakeDrive0SA.' pressure '.IA0.FZ667.FZ'];
SRN=[pFileName '.BrakeDrive0SA.' pressure '.IA0.FZ667.SL'];
FXN=[pFileName '.BrakeDrive0SA.' pressure '.IA0.FZ667.FX'];

eval(['FZ=' FZN ';']);
eval(['SR=' SRN ';']);
eval(['FX=' FXN ';']);

k=1;
bnd=[1;length(SR)];
for n=1:length(SR)
    if SR(n)>=.15 && SR(n+1)<=.15 && k==1
        bnd(k,1)=n;
        k=k+1;
    elseif n==length(SR) && k==2
        bnd(k,1)=n;
        break
    elseif SR(n)<.12 && SR(n+1)>=.12 && k==2 && n-bnd(1,1)>100
        bnd(k,1)=n;
        k=k+1;
    elseif k==3
        break;
    end
end

% Test plot for truncation check.
if pebugFx0==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SR(bnd(1,1):bnd(2,1)),FX(bnd(1,1):bnd(2,1)),'');
end

% Compute pFz0 as the mean of the FZ for this data set.
Fzx=(mean(FZ(bnd(1,1):bnd(2,1))));

% Fit curve to FY data to eliminate hysteresis.
SmoothSpan=35;
[SRfit,Sord]=sort(SR(bnd(1,1):bnd(2,1))); 
FXtrunc=(FX(bnd(1,1):bnd(2,1)));
FXfit=FXtrunc(Sord);
FXsmth=smooth(FXfit,SmoothSpan,'lowess');

[SRU Umat junk]=unique(SRfit);
FXU=FXsmth(Umat);

%% Data smoothing test plot.
if pebugFx0==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SR(bnd(1,1):bnd(2,1)),FX(bnd(1,1):bnd(2,1)),',',SRfit,FXfit,'r*...',
         SRU,FXU,'g-');
end

pDx1=(max(FXU)+abs(min(FXU)))/2/Fzx;
pVx1=-((max(FXU)-((max(FXU)+abs(min(FXU)))/2))/Fzx;

FXU=(FXU+Fzx*pVx1);
FX1=FXU/(pDx1*Fzx);

if pebugFx0==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SRU,FXU);
    figure(DaBug); DaBug=DaBug+1; % Debug graphs
    plot(SRU,FX1);
end

k=1;
FXbound=.4*max(FXU);
for n=1:length(FXU)
    if FXU(n)<-FXbound && FXU(n+1)>=-FXbound
        bnd(1,1)=n;
        k=k+1;
    elseif FXU(n)<FXbound && FXU(n+1)==FXbound
        bnd(2,1)=n+1;
        k=k+1;
    elseif k==3
        break;
    end;
end;

PolyOrig=polyfit(SRU(bnd(1,1):bnd(2,1)),FXU(bnd(1,1):bnd(2,1)),1);
Kx=PolyOrig(1);
pHx1=PolyOrig(2)/PolyOrig(1);
SR1=SRU+pHx1;

if pebugFx0==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SR1,FX1);
end

pCx1=1.8;
E=1;
DpCx1=.05;
DE=.1;

pCx1t=pCx1;
Et=E;
DpCx1t=DpCx1;
DEt=DE;
\[ B_{xt} = \frac{K_x}{(p_{Cx1}t + pDx1 * Fzx) * SR1}; \]
\[ MCx1 = \text{sum}\left( \text{abs}\left( (\text{abs}(F_{x1}) - \text{abs}(\sin(p_{Cx1}t \cdot \text{atan}(B_{xt} - \text{Et} \cdot (B_{xt} - \text{atan}(B_{xt}))))) \right) \right); \]
\[ MCx0 = MCx1; \]
\[ \text{while } 1 \]
\[ \quad MCx2 = \text{sum}\left( (\text{abs}(F_{x1}) - \text{abs}(\sin(p_{Cx1}t \cdot \text{atan}(B_{xt} - \text{Et} - \text{DEt}) \cdot (B_{xt} - \text{atan}(B_{xt})))) \right); \]
\[ \quad \text{if } MCx2 < MCx1 \]
\[ \quad \quad \text{Et} = \text{Et} - \text{DEt}; \]
\[ \quad \quad MCx0 = MCx1; \]
\[ \quad \quad MCx1 = MCx2; \]
\[ \quad \text{elseif } MCx2 \geq MCx1 \]
\[ \quad \quad \text{Et} = \text{Et} - \text{DEt}; \]
\[ \quad \quad \text{DEt} = -\text{DEt}/2; \]
\[ \quad \quad MCx12 = MCx1; \]
\[ \quad \quad MCx1 = MCx0; \]
\[ \quad \text{end} \]
\[ \quad \text{if } \text{abs}(\text{DEt}) < .00000001 \]
\[ \quad \quad \text{break}; \]
\[ \quad \text{end} \]
\[ \text{end} \]
\[ MCx10 = MCx11; \]
\[ \text{while } 1 \]
\[ \quad \text{Et} = \text{E}; \]
\[ \quad \text{DEt} = \text{DE}; \]
\[ \quad B_{xt} = \frac{K_x}{(p_{Cx1}t - Dp_{Cx1}t) + pDx1 * Fzx) * SR1}; \]
\[ \quad MCx1 = \text{sum}\left( (\text{abs}(F_{x1}) - \text{abs}(\sin((p_{Cx1}t - Dp_{Cx1}t) \cdot \text{atan}(B_{xt} - \text{Et}) \cdot (B_{xt} - \text{atan}(B_{xt})))) \right); \]
\[ \quad MCx0 = MCx1; \]
\[ \quad \text{while } 1 \]
\[ \quad \quad MCx2 = \text{sum}\left( (\text{abs}(F_{x1}) - \text{abs}(\sin((p_{Cx1}t - Dp_{Cx1}t) \cdot \text{atan}(B_{xt} - \text{Et} - \text{DEt}) \cdot (B_{xt} - \text{atan}(B_{xt})))) \right); \]
\[ \quad \quad \text{if } MCx2 < MCx1 \]
\[ \quad \quad \quad \text{Et} = \text{Et} - \text{DEt}; \]
\[ \quad \quad \quad MCx0 = MCx1; \]
\[ \quad \quad \quad MCx1 = MCx2; \]
\[ \quad \quad \text{elseif } MCx2 \geq MCx1 \]
\[ \quad \quad \quad \text{Et} = \text{Et} - \text{DEt}; \]
\[ \quad \quad \quad \text{DEt} = -\text{DEt}/2; \]
\[ \quad \quad \quad MCx12 = MCx1; \]
\[ \quad \quad \quad MCx1 = MCx0; \]
\[ \quad \quad \text{end} \]
\[ \quad \quad \text{if } \text{abs}(\text{DEt}) < .00000001 \]
\[ \quad \quad \quad \text{break}; \]
\[ \quad \quad \text{end} \]
\[ \quad \text{end} \]
\[ \text{if } MCx12 < MCx11 \]
\[ \quad p_{Cx1}t = p_{Cx1}t - Dp_{Cx1}t; \]
\[ \quad MCx10 = MCx11; \]
\[ \quad MCx11 = MCx12; \]
\[ \text{elseif } MCx12 \geq MCx11 \]
\[ \quad p_{Cx1}t = p_{Cx1}t - Dp_{Cx1}t; \]
\[ \quad Dp_{Cx1}t = -Dp_{Cx1}t/2; \]
\[ \quad MCx11 = MCx10; \]
\[ \text{end} \]
\[ \text{if } \text{abs}(Dp_{Cx1}t) < .00000001 \]
\[ \quad p_{Cx1} = p_{Cx1}t; \]

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break;
end
end

if pebugFx0==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SR1,FX1,','SR1,sin((pCxlt+DpCxlt)*atan(Bxt-(Et-DEt)).*... (Bxt-atan(Bxt))))','
end
clear pCxlt;

%% Longitudinal Force  
- Pure Longitudinal Slip  
- Full Coefficient Definition
% Curve fit for the full test matrix of loads and inclination angles.

% Inclination Angle Runs
IAP=[0 2 4];
% Normal Load Runs
FZP=[222 667 1112 1557];

% Declare variables
Fzx=zeros(length(FZP),length(IAP));
dFzx=zeros(size(Fzx));
Gammax=zeros(size(Fzx));
Dx=zeros(size(Fzx));
Mewx=zeros(size(Fzx));
SVx=zeros(size(Fzx));
SHx=zeros(size(Fzx));
Kx=zeros(size(Fzx));
Bx=zeros(size(Fzx));
ExP=zeros(size(Fzx));
ExN=zeros(size(Fzx));

% The stiffness bounds determines the portion of max(Fy) to perform the
% linear fit for the cornering stiffness K.  A larger ratio lessens the
% fitted stiffness for the particular run.  Useful for tuning in full
% charts for linearity to measured data.
StiffnessBnd=[...
    .65 .45 .35;...
    .50 .55 .55;...
    .40 .40 .30;...
    .15 .15 .25];

for n=1:length(IAP)
    for m=1:length(FZP)
        clear FZN SRN FXN FZ SR FX k p SRfit Sord FXtrunc FXfit FXsmth SRU
        Umat FXU
        FZN=['pFileName '.BrakeDrive0SA.' pressure '.IA' int2str(IAP(n))... '.FZ' int2str(FZP(m)) '.FZ'];
        SRN=['pFileName '.BrakeDrive0SA.' pressure '.IA' int2str(IAP(n))... '.FZ' int2str(FZP(m)) '.SL'];
        FXN=['pFileName '.BrakeDrive0SA.' pressure '.IA' int2str(IAP(n))... '.FZ' int2str(FZP(m)) '.FX'];
        IAN=['pFileName '.BrakeDrive0SA.' pressure '.IA' int2str(IAP(n))... '.FZ' int2str(FZP(m)) '.IA'];
        eval(['FZ=' FZN ' ;']);

eval(['SR=' SRN ';']);
eval(['FX=' FXN ';']);
eval(['IA=' IAN ';']);

if thermFx0==1
  TSTCN=[pFileName '.BrakeDriveOSA.' pressure '.IA' int2str...
        (IAP(n)) '.FZ' int2str(FZP(m)) '.TSTC'];
  TSTIN=[pFileName '.BrakeDriveOSA.' pressure '.IA' int2str...
        (IAP(n)) '.FZ' int2str(FZP(m)) '.TSTI'];
  TSTON=[pFileName '.BrakeDriveOSA.' pressure '.IA' int2str...
        (IAP(n)) '.FZ' int2str(FZP(m)) '.TSTO'];
  RSTN=[pFileName '.BrakeDriveOSA.' pressure '.IA' int2str...
        (IAP(n)) '.FZ' int2str(FZP(m)) '.RST'];
  AMBTMPN=[pFileName '.BrakeDriveOSA.' pressure '.IA' int2str...
        (IAP(n)) '.FZ' int2str(FZP(m)) '.AMBTMP'];
  ETN=[pFileName '.BrakeDriveOSA.' pressure '.IA' int2str...
        (IAP(n)) '.FZ' int2str(FZP(m)) '.ET'];
  eval(['TSTC=' TSTCN ';']);
  eval(['TSTO=' TSTON ';']);
  eval(['TSTI=' TSTIN ';']);
  eval(['RST=' RSTN ';']);
  eval(['AMBTMP=' AMBTMPN ';']);
  eval(['ET=' ETN ';']);
end

if thermFx0==1
  k=15;
  figure(400+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...
      bnd(2,1)),k),TSTC(bnd(1,1):bnd(2,1)), 'Color', pColors(m,:));
  figure(410+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...
      bnd(2,1)),k),TSTI(bnd(1,1):bnd(2,1)), 'Color', pColors(m,:));
  figure(420+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...
      bnd(2,1)),k),TSTO(bnd(1,1):bnd(2,1)), 'Color', pColors(m,:));
  figure(430+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...

%% Truncate data to one full sweep minimizing start and end effects
k=1;
bnd=[1;length(SR)];
for p=1:length(SR)
  if SR(p)>.15 && SR(p+1)<=.15 && k==1
    bnd(k,1)=p;
    k=k+1;
  elseif p==length(SR)
    break;
  elseif SR(p)<.12 && SR(p+1)>=.12 && k==2 && p-bnd(1,1)>100
    bnd(k,1)=p;
    k=k+1;
  elseif k==3
    break;
  end
end

if thermFx0==1
  k=15;
  figure(400+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...
      bnd(2,1)),k),TSTC(bnd(1,1):bnd(2,1)), 'Color', pColors(m,:));
  figure(410+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...
      bnd(2,1)),k),TSTI(bnd(1,1):bnd(2,1)), 'Color', pColors(m,:));
  figure(420+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...
      bnd(2,1)),k),TSTO(bnd(1,1):bnd(2,1)), 'Color', pColors(m,:));
  figure(430+n); hold on;
  plot(smooth(FX(bnd(1,1):bnd(2,1)),k)./smooth(FZ(bnd(1,1):...
end

% Compute dFz for the given run
% dFz = (Fz - pfz0)/pfz0
Fzx(m,n)=mean(FZ(bnd(1,1):bnd(2,1)));
dFzx(m,n)=(Fzx(m,n)-pfz0)/pfz0;

% Record IA for given run
Gammax(m,n)=(mean(IA(bnd(1,1):bnd(2,1)))*pi/180);

SmoothSpan=35;
[SRfit,Sord]=sort(SR(bnd(1,1):bnd(2,1)));% SRfit = sorted SRfit
FXtrunc=(FX(bnd(1,1):bnd(2,1)));% FXtrunc = FX trunct
FXfit=FXtrunc(Sord);% FXfit = FXfit
FXsmth=smooth(FXfit,SmoothSpan,'lowess');% FXsmth = smooth FXfit

[SRU Umat junk]=unique(SRfit);% SRU = unique SRfit
FXU=FXsmth(Umat);
SRR=SRU;
FXR=FXU;

% Compute Peak Value factor D and mew
Dx(m,n)=max(FXU)+abs(min(FXU))/2;
Mewx(m,n)=Dx(m,n)/Fzx(m,n);

% Compute Vertical shift factor SHy and shift data curve
SVx(m,n)=max(FXU)-((max(FXU)+abs(min(FXU)))/2);
FXU=FXU-SVx(m,n);

% Compute Stiffness K (dFy/dSA @ SA=0)
% Define the range over which the linear fit will occur
k=1;
FXbound=StiffnessBnd(m,n)*max(FXU);
% bnd=[1; length(FXU)];
for p=1:length(FXU)
    if FXU(p)<-FXbound && FXU(p+1)>=-FXbound
        bnd(1,1)=p;
        k=k+1;
    elseif FXU(p)<FXbound && FXU(p+1)>FXbound
        bnd(2,1)=p+1;
        k=k+1;
    elseif k==3
        break;
    end;
end;

% Perform 1st order linear fit over the defined range to define K
PolyOrig=polyfit(SRU(bnd(1,1):bnd(2,1)),FXU(bnd(1,1):bnd(2,1)),1);
Kx(m,n)=PolyOrig(1);

% Utilizing K and the y-intercept compute the horizontal shift SHy
SHx(m,n)=PolyOrig(2)/PolyOrig(1);
% Shift curve to origin
SRU=SRU+SHx(m,n);

% Utilizing C, D, and K compute stiffness factor By=Ky/(Cy*Dy)
Bx(m,n)=Kx(m,n)/(pCx1*Dx(m,n));

if purelong==1
    figure(n+10);
    hold on
    plot(SRU,FXU, 'b-');
    figure(n+15);
    hold on
    plot(SRR,FXR, 'b-');
end;

% Compute Curvature Factor E
% E is dependent on the sign of the SA, it is thus necessary to fit E seperately to positive and negative SA.
FXU=FXU/Dx(m,n);
[SRmin Exmin]=min(abs(SRU));
SRpos=SRU(Exmin:length(SRU));
SRneg=SRU(1:Exmin);
FXUpos=FXU(Exmin:length(FXU));
FXUneg=FXU(1:Exmin);

% Begin positive side E definition
Etest=-10;
teststep=1;
SRtest=SRpos;
FXUtest=FXUpos;
FXP=sin(pCx1.*atan(Bx(m,n).*SRpos-(Etest.*(Bx(m,n).*SRpos-atan(...
    (Bx(m,n).*SRpos))))));
while 1
    if Etest>=-2 && teststep==1
        teststep=.1;
    end;
    FXP1=sin(pCx1.*atan(Bx(m,n).*SRtest-((Etest+teststep).*...
\[(Bx(m,n) \cdot SRtest - \text{atan}(Bx(m,n) \cdot SRtest)))\];
defFXP=mean(abs(FXP-FXUtest));
defFXP1=mean(abs(FXP1-FXUtest));
if defFXP1<defFXP
    Etest=Etest+teststep;
    FXP=FXP1;
elseif defFXP1>defFXP
    Etest=Etest-teststep;
    FXP=sin(pCx1.*atan(Bx(m,n).*SRtest-(Etest.*(Bx(m,n).*SRtest-atan(Bx(m,n).*SRtest)))))
end
defstep=.25*defstep;
if defstep<=.000001
    if purelong==1
        figure(n+10);
        hold on
        plot(SRtest,FXP.*Dx(m,n),'k--');
    end;
    ExP(m,n)=Etest;
    break;
end;
end

% Begin negative side E definition
Etest=-10;
teststep=1;
SRtest=SRneg;
FXUtest=FXUneg;
FXP=sin(pCx1.*atan(Bx(m,n).*SRtest-(Etest.*(Bx(m,n).*SRtest-atan(Bx(m,n).*SRtest)))));
while 1
    if Etest>=-2 && teststep=1
        teststep=.1;
    end;
    FXP1=sin(pCx1.*atan(Bx(m,n).*SRtest-((Etest+teststep).*SRtest-atan(Bx(m,n).*SRtest))))
end
defFXP=mean(abs(FXP-FXUtest));
defFXP1=mean(abs(FXP1-FXUtest));
if defFXP1<defFXP
    Etest=Etest+teststep;
    FXP=FXP1;
elseif defFXP1>defFXP
    Etest=Etest-teststep;
    FXP=sin(pCx1.*atan(Bx(m,n).*SRtest-(Etest.*(Bx(m,n).*SRtest-atan(Bx(m,n).*SRtest)))));
end
defstep=.25*defstep;
if defstep<=.0000001
    if purelong==1
        figure(n+10);
        hold on
        plot(SRtest,FXP.*Dx(m,n),'k--');
    end;
    ExN(m,n)=Etest;
    break;
end;
end;
%% Longitudinal Force (Pure Longitudinal Slip) Final Coefficient Definition
% User scaling factors not listed in equations
% Nominal conditions are matrix element (2,1)

%% Fx0 Raw Data Callout
pRAWFX0Fit.Fzx = Fzx;
pRAWFX0Fit.Gammax = Gammax;

%% SHx
% SHx = (pHx1 + pHx2 * dFz)
pRAWFX0Fit.SHx = SHx;
if purelong == 1
    figure(1101);
    mesh(IAP, FZP, SHx); title('SH_x (SR)'); xlabel('IA (deg)'); ylabel('FZ (N)');
end
for k = 1 : length(IAP)
    PHX12(k, :) = polyfit(dFzx(:, k), SHx(:, k), 1);
end
pHx1 = mean(PHX12(:, 2));
pHx2 = mean(PHX12(:, 1));

%% SVx
% SVx = Fz * (pVx1 + pVx2 * dFz)
pRAWFX0Fit.SVx = SVx;
if purelong == 1
    figure(1102);
    mesh(IAP, FZP, SVx); title('SV_x (Newton)'); xlabel('IA (deg)'); ylabel('FZ (N)');
end
SVx = SVx ./ Fzx;
for k = 1 : length(IAP)
    PVX12(k, :) = polyfit(dFzx(:, k), SVx(:, k), 1);
end
pVx2 = mean(PVX12(:, 1));
pVx1 = mean(PVX12(:, 2));

%% Shape Factor C was determined using curve at position (2,1)

%% Peak Factor D
% Determine coefficients for Mewx
% Mewx = (pDx1 + pDx2 * dFz) * (1 - pDx3 * gamma^2)
% Method 2
pRAWFX0Fit.Mewx = Mewx;
if purelong == 1
    figure(1103);
    mesh(IAP, FZP, Mewx); title('Peak Mu_x (FX/FZ)'); xlabel('IA (deg)'); ylabel('FZ (N)');
end
for k = 1 : length(IAP)
    PDX12(k, :) = polyfit(dFzx(:, k), Mewx(:, k), 1);
end
pDx1 = PDX12(2, 2);
pDx2 = mean(PDX12(:, 1));
Mewx1=Mewx;
Mewx=(Mewx./(pDx1+pDx2*dFzx)) - 1;
for k=1:length(FZP)
    PDX3(k,:) = polyfit(Gammax(k,:),Mewx(k,:),2);
end
pDx3=-1*mean(PDX3(:,1));
Mewx2=Mewx;
Mewx3=Mewx+pDx3*Gammax.^2;

% Curvature Factor E
% Ex = (pEx1 + pEx2*dFz + pEx3*dFz^2)*(1 - pEx4*sgn(SR))
ExM=(ExP+ExN)/2;
pRAWFX0Fit.ExM=ExM;
pRAWFX0Fit.ExN=ExN;
pRAWFX0Fit.ExP=ExP;
if purelong==1
    figure(1104);
    mesh(IAP,FZP,ExM); title('E_x Mean');
    xlabel('IA (deg)'); ylabel('FZ (N)');
    figure(1105);
    mesh(IAP,FZP,ExP); title('E_x Positive SA');
    xlabel('IA (deg)'); ylabel('FZ (N)');
    figure(1106);
    mesh(IAP,FZP,ExN); title('E_x Negative SA');
    xlabel('IA (deg)'); ylabel('FZ (N)');
end
for k=1:length(IAP)
    PEX123(k,:) = polyfit(dFzx(:,k),ExM(:,k),2);
end
pEx1=PEX123(2,3);
pEx2=mean(PEX123(:,2));
pEx3=mean(PEX123(:,1));
switch pEx3Switch
    case 0
        ExPD=(ExP./(pEx1+pEx2*dFzx+pEx3*dFzx.^2) - 1)/(-1);
        ExND=(ExN./(pEx1+pEx2*dFzx+pEx3*dFzx.^2) - 1);
    otherwise
        ExPD=(ExP./(ExM) - 1)/(-1); % Divide by -sgn(SA)
        ExND=(ExN./(ExM) - 1);
end
ExD=(ExPD+ExND)/2;
Ex4Confidence=2;
Ex4Criteria=[std(ExPD,0,2) std(ExND,0,2)];
Ex4Test=mean(mean([std(ExND,0,2),std(ExPD,0,2)]).*Ex4Confidence);
m=1;
for k=1:length(FZP)
    if Ex4Criteria(k,1)<=Ex4Test & Ex4Criteria(k,2)<=Ex4Test
        ExPDt(m,:) = ExPD(k,:);
        ExNDt(m,:) = ExND(k,:);
        m=m+1;
    end
end
pEx4=mean([mean(ExPDt) mean(ExNDt)]);

% Stiffness Factor B
% Stiffness Ky
% Kx = Fz*(pKx1 + pKx2*dFz)*exp(pKx3*dFz)
Kx0=Kx;
if purelong==1
    figure(1107);
    mesh(IAP,FZP,Kx); title('Kx (Newton/SR)');
    xlabel('IA (deg)'); ylabel('FZ (N)');
end
Kx=Kx./Fzx;
for k=1:length(IAP)
    PKX12(k,:)=polyfit(dFzx(:,k),Kx(:,k),1);
end
pKx1=mean(PKX12(:,2));
pKx2=mean(PKX12(:,1));

Kx=log((Kx./pKx1+pKx2*dFzx));
for k=1:length(IAP)
    PKX3(k,:)=polyfit(dFzx(:,k),Kx(:,k),1);
end
pKx3=mean(PKX3(:,1));

% Plot Derived Curves vs. Data Curves
if purelong==1
    Dx1=Fzx.*(pDx1+pDx2*dFzx).*(1-pDx3*Gammax.^2);
    Kx1=Fzx.*(pKx1+pKx2*dFzx).*exp(pKx3*dFzx);
    Bx1=Kx1./(pCx1*Dx1);
    SVx1=Fzx.*(pVx1+pVx2.*dFzx);
    SHx1=pHx1+pHx2.*dFzx;
    SRD=-.25:.01:.15;
    for n=1:length(IAP)
        for m=1:length(FZP)
            FXD=Dx1(m,n).*sin(pCx1.*atan(Bx1(m,n).*SRD-((pEx1+pEx2.*dFzx(m,n).^2).*pEx3.*dFzx(m,n).^2).*(1-pEx4.*abs(SRD)./SRD)).*Bx1(m,n).*SRD-atan(Bx1(m,n)*SRD));
            SRS=SRD+SHx1(m,n);
            FXS=Dx1(m,n).*sin(pCx1.*atan(Bx1(m,n).*SRD-((pEx1+pEx2.*dFzx(m,n).^2).*pEx3.*dFzx(m,n).^2).*(1-pEx4.*abs(SRS)/SRS)).*Bx1(m,n).*SRD-atan(Bx1(m,n)*SRD))+SVx1(m,n);
            figure(n+10);
            hold on;
            plot(SRD,FXD,'r-');
            figure(n+15);
            hold on;
            plot(SRD,FXS,'r-');
            clear SRS FXS FXD
        end
    end
end

%Perform some cleanup!
pDaBug=DaBug;
clear B* D* E* F* G* I* K* M* O* P* R* S* U* d* b* j* k* m* n* p teststep;
DaBug=pDaBug;
% close all;
% Aligning Torque - Pure Side Slip - Initial Coefficient Definition
% Initial analysis done at pFz0 to verify procedure and identify shape
% factor qCz1.

eval(['load ' pFileName '.mat;']);

% Corning load averaged from the test given will be used.
% The middle loading of 667 N (150 lbf) will be utilized as pFz0.
FZN=[pFileName '.Cornering.' pressure '.IA0.FZ667.FZ'];
SAN=[pFileName '.Cornering.' pressure '.IA0.FZ667.SA'];
MZN=[pFileName '.Cornering.' pressure '.IA0.FZ667.MZ'];

eval(['FZ=' FZN '']);
eval(['SA=' SAN '']);
eval(['MZ=' MZN '']);

k=1;
bnd=[1,length(SA)];
for n=1:length(SA)
    if SA(n)>=0 && SA(n+1)<0 && k==1
        bnd(k,1)=n+1;
        k=k+1;
    elseif SA(n)>=0 && SA(n+1)<0 && k==2
        bnd(k,1)=n;
        k=k+1;
    elseif k==3
        break;
    end
end

% Presmooth Raw Data before reorder
if pebugzmo==1
    figure(DaBug);
    hold on;
    plot(MZ,'b');
end
SmoothSpan=61;
MZ=smooth(MZ,SmoothSpan,'lowess');
if pebugzmo==1
    figure(DaBug); DaBug=DaBug+1;
    hold on;
    plot(MZ,'r');
end

% Test plot for truncation check.
% plot(SA(bnd(1,1):bnd(2,1)),FY(bnd(1,1):bnd(2,1)),'');

% Fit curve to MZ data to eliminate hysteresis.
SmoothSpan=21;
EndTrunc=50;
[Shfit,Sord]=sort(SA(bnd(1,1):bnd(2,1)));
SArad=Shfit*pi/180;
SArad=SArad(EndTrunc:length(SAfit)-EndTrunc);
MZtrunc=(MZ(bnd(1,1):bnd(2,1)));
MZfit=MZtrunc(Sord);
MZfit=MZfit(EndTrunc:length(MZfit)-EndTrunc);
MZsmth = smooth(MZfit, SmoothSpan, 'lowess');

[SAR Umat junk] = unique(SArad);
MZU = MZsmth(Umat);
% SAU = tan(SAU);

% Data smoothing test plot.
if pebugzmo == 1
   figure(DaBug); DaBug = DaBug + 1;
   plot(SA(bnd(1,1):bnd(2,1))*pi/180, MZ(bnd(1,1):bnd(2,1)), 'r.', SArad, MZfit, 'r.', SArad, MZsmth, 'k-');
end

% Compute SHf
% SHf = SHy + (SVy / Ky)
% SHy = (pHy1 + pHy2*dFz) + pHy3*gamma
% SVy = Fz * (pVy1 + pVy2*dFz) + (pVy3 + pVy4*dFz)*gamma
% Ky = pKy1*Fz0*sin(2*atan(Fz/(pKy2*Fz0)))*(1 - pKy3*abs(gamma))
% At this data sample dFz = 0, gamma = 0, and Fz = Fz0
SHf = pHy1 + (pFz0*pVy1/(pKy1*pFz0*sin(2*atan(1/pKy2))));
SHy = pHy1;
SAr = SAU + SHf;

% Compute Residual Torque Peak Factor Dr
Dr = interp1q(SAU, MZU, SHf);

% Compute Br
% Br = (qBz9 + qBz10*By*Cy) (qBz9 is suggested to be 0 by Pacejka)
% Cy = pCy1;
% By = Ky/(Cy*Dy)
% Dy = Mewy / Fz
% Mewy = (pDy1 + pDy2*dFz)*(1 - pDy3*gamma^2)
ByCy = pKy1*pFz0*sin(2*atan(1/pKy2))/(pFz0*pDy1);
Br1 = 500;
Brd = 100;
By0 = (pKy1*pFz0*sin(2*atan(1/pKy2)))/(pCy1*pFz0*pDy1);
SAy = SAU + pHy1;
Fy0 = pFz0*pDy1*sin(pCy1*atan(By0*SAy - (pEy1*(1 - pEy3*SAy./abs(SAy)))).*...
     (By0*SAy-atan(By0*SAy)))+(pFz0*pVy1);
Mzr1 = Dr*cos(atan(Br1*SAr)).*cos(SAU);
Mzt1 = (MZU - Mzr1)./(1*Fy0).mist(SAU);
Mzt = Mzt1;
LoAng = .03; HiAng = .15;
TestAng = LoAng:.01:HiAng;
Shtl = 0;
Shtd = .002;
ShtThresh = .00001;
Cdiff1 = 0;
for n = 1:length(TestAng)
   Cdiff1 = Cdiff1 + abs(interp1q(SAU, Mzt1, TestAng(n) + Shtl) - interp1q(SAU, ...
                          Mzt1, TestAng(n) + Shtl));
end
Cdiff0 = Cdiff1;
while 1
CDiff2=0;
for n=1:length(TestAng)
    CDiff2=CDiff2+abs(interp1q(SAU,MZt1,TestAng(n)+SHt1+SHtd)-
    interp1q(SAU,MZt1,-1*TestAng(n)+SHt1+SHtd));
end
if CDiff2<CDiff1 && abs(SHtd)>SHtThresh
    CDiff1=CDiff2;
    SHt1=SHt1+SHtd;
elseif CDiff2>=CDiff1 && abs(SHtd)>SHtThresh
    CDiff1=CDiff2;
    SHt1=SHt1+SHtd;
    SHtd=-SHtd/2;
else
    CDiff01=CDiff1;
    SHt=-SHt1;
    break;
end
end

BrdThresh=1;
Br1Limit=2000;
while 1
    MZr2=Dr*cos(atan((Br1+Brd)*SAr)).*cos(SAU);
    MZt2=((MZU-MZr2)/(-1*Fy0))./cos(SAU);
    SHt1=0;
    SHtd=.002;
    SHtThresh=.00001;
    CDiff1=0;
    for n=1:length(TestAng)
        CDiff1=CDiff1+abs(interp1q(SAU,MZt2,TestAng(n)+SHt1)-interp1q...
        (SAU,MZt2,-TestAng(n)+SHt1));
    end
end
while 1
    CDiff2=0;
    for n=1:length(TestAng)
        CDiff2=CDiff2+abs(interp1q(SAU,MZt2,TestAng(n)+SHt1+SHtd)-
        interp1q(SAU,MZt2,-1*TestAng(n)+SHt1+SHtd));
    end
if CDiff2<CDiff1 && abs(SHtd)>SHtThresh
    CDiff1=CDiff2;
    SHt1=SHt1+SHtd;
elseif CDiff2>=CDiff1 && abs(SHtd)>SHtThresh
    CDiff1=CDiff2;
    SHt1=SHt1+SHtd;
    SHtd=-SHtd/2;
else
    CDiff02=CDiff1;
    break;
end
if CDiff02<CDiff01 && abs(Brd)>BrdThresh && abs(Br1)<Br1Limit
    Br1=Br1+Brd;
    CDiff01=CDiff02;
else
    Br1=Br1+Brd;
    CDiff01=CDiff02;
end
else
    Br=Br1;
    SHT=-SHT1;
    MZt=MZt2;
    MZr=MZr2;
    break
end
end

if pebugzmo==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SAU,MZr,'r--');
    figure(DaBug); DaBug=DaBug+1;
    plot(SAU,MZU-MZr,'r--',SAU,MZU,'g--');
end

% Factor pneumatic trail out of MZ utilizing Fy0 and MZr.
% Conditions met: IA=0; FZ=Fz0; dFz=0;

% By0=(pKy1*pFz0*sin(2*atan(1/pKy2)))/(pCyl*pFz0*pDy1);
% SAy=SAU+pHy1;
% Fy0=pFz0*pDy1*sin(pCyl*atan(By0*SAy-(pEy1*(1-pEy3*SAy./abs(SAy))).*(By0*SAy-atan(By0*SAy))))+(pFz0*pVy1);
% MZt=((MZU-MZr).*(-1*Fy0))./cos(SAU);

if pebugzmo==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SAU,Fy0);
    figure(DaBug); DaBug=DaBug+1;
    plot(SAU,MZt);
end

Cut=31;
[SAJ MinInd]=min(abs(SAU-SHf));
MZtBot=MZt(1:MinInd-Cut);
SAUBot=SAU(1:MinInd-Cut);
MZtTop=MZt(MinInd+Cut:length(MZt));
SAUTop=SAU(MinInd+Cut:length(MZt));

MZtT=MZtBot; MZtT(length(MZtT)+1:length(MZtTop)+length(MZtT))=MZtTop;
SAUT=SAUBot; SAUT(length(SAUT)+1:length(SAUTTop)+length(SAUT))=SAUTop;
MZtT=smooth(MZtT,9,'rloess');

if pebugzmo==1
    figure(DaBug);
    hold on;
    plot(SAUT,MZtT,'r--');
end

SAUT=SAUT+SHT;

if pebugzmo==1
    figure(DaBug); DaBug=DaBug+1;
    hold on;
    plot(SAUT,MZtT,'g--');
end
% Calculate Peak Factor Dt
[Dt MidInd]=max(MZtT);
MZtT1=MZtT/Dt;

if pebugzmo==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SAUT,MZtT1);
end

BtB=6.3;
EtB=1.4;
CtB=2.65;
DB=.1;
DE=.1;
DC=.02;
DBt=DB;
DEt=DE;
DCl=DC;

Mt111=sum(abs(cos(CtB*atan(BtB*SAUT-EtB*(BtB*SAUT-atan(BtB*SAUT))))-MZtT1));
Mt110=Mt111;

while 1
    DEt=DE;
    Mt110=sum(abs(cos((CtB-DCt)*atan((BtB+DBt)*SAUT-((EtB-DEt)*((BtB+DBt)*SAUT)-atan((BtB+DBt)*SAUT))))-MZtT1));
    Mt100=Mt110;
    while 1
        DBt=DB;
        Mt100=sum(abs(cos((CtB-DCt)*atan((BtB+DBt)*SAUT-((EtB-DEt)*((BtB+DBt)*SAUT)-atan((BtB+DBt)*SAUT))))-MZtT1));
        Mt000=Mt100;
        while 1
            Mt200=sum(abs(cos((CtB-DCt)*atan((BtB+DBt)*SAUT-((EtB-DEt)*((BtB+DBt)*SAUT)-atan((BtB+DBt)*SAUT))))-MZtT1));
            if Mt200<Mt100
                BtB=BtB+DBt;
                Mt000=Mt100;
                Mt100=Mt200;
            elseif Mt200>=Mt100
                BtB=BtB+DBt;
                DBt=-DBt/4;
                Mt100=Mt200;
            end
            if abs(DBt)<.00001
                Mt120=Mt200;
                break;
            end
        end
    end
    if Mt120<Mt110
        EtB=EtB-DEt;
        Mt100=Mt110;
        Mt110=Mt120;
    elseif Mt120>=Mt110
        EtB=EtB-DEt;
        Mt110=Mt120;
end
DEt=-DEt/4;
end
if abs(DEt)<.00001
    Mt112=Mt120;
    break;
end
if Mt112<Mt111
    CtB=CtB-DcT;
    Mt110=Mt111;
    Mt111=Mt112;
elseif Mt112>=Mt111
    CtB=CtB-DcT;
    DcT=-DcT/4;
    Mt111=Mt112;
end
%     figure(99);
%     plot(SAUT,MZtT1,'',SAUT,cos(CtB*atan(BtB*SAUT-
                     EtB*(BtB*SAUT-
                     atan(BtB*SAUT)))),'');
if abs(DcT)<.00001
    break;
end
end
if pebugzmo==1
    figure(DaBug); DaBug=DaBug+1;
    plot(SAUT,MZtT1,'',SAUT,cos(CtB*atan(BtB*SAUT-
                     EtB*(BtB*SAUT-
                     atan(BtB*SAUT)))),'');
end
qCz1=CtB;

% Aligning Moment - Pure Side Slip - Full Coefficient Definition
% Curve fit for the full test matrix of loads and inclination angles.
% Inclination Angle Runs
IAP=[0 1 2 3 4];
% Normal Load Runs
FZP=[222 445 667 1112 1557];

% Declare variables
Fzmz=zeros(length(FZP),length(IAP));
dFzmz=zeros(size(Fzmz));
Gammanz=zeros(size(Fzmz));
Bmzr=zeros(size(Fzmz));
Bmzt=zeros(size(Fzmz));
Dmzr=zeros(size(Fzmz));
Dmzt=zeros(size(Fzmz));
Emzt=zeros(size(Fzmz));
SHf=zeros(size(Fzmz));
SHt=zeros(size(Fzmz));
SHy=zeros(size(Fzmz));
SVy=zeros(size(Fzmz));
Ky=zeros(size(Fzmz));
Dy=zeros(size(Fzmz));
By=zeros(size(Fzmz));
clear n m;

MZP=zeros([length(FZP) length(IAP) 800]);
SAP=zeros(size(MZP));
Fy0P=zeros(size(MZP));
MATSIZE=size(SAP);

for n=1:length(IAP)
    for m=1:length(FZP)
        clear FZ FZN SAN MZ MZU SAU SA Mt* IAN IA k bnd ByCy By0 p...
        FZ=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
            '.FZ' int2str(FZP(m)) '.FZ'];
        SAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
            '.FZ' int2str(FZP(m)) '.SA'];
        MZ=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
            '.FZ' int2str(FZP(m)) '.MZ'];
        IAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
            '.FZ' int2str(FZP(m)) '.IA'];
        eval(['FZ=' FZ ';
        eval(['SA=' SAN ';
        eval(['MZ=' MZ ';
        eval(['IA=' IAN ';

        k=1;
        bnd=[1,length(SA)];
        for p=1:length(SA)
            if SA(p)>=0 && SA(p+1)<0 && k==1
                bnd(k,1)=p+1;
                k=k+1;
            elseif SA(p)>=0 && SA(p+1)<0 && k==2
                bnd(k,1)=p;
                k=k+1;
            elseif k==3
                break;
            end
        end

        % Presmooth Raw Data before reorder
        SmoothSpan=61;
        MZ=smooth(MZ,SmoothSpan,'lowess');

        % Fit curve to MZ data to eliminate hysteresis.
        SmoothSpan=21;
        EndTrunc=50;
        [SAfit,Sord]=sort(SA(bnd(1,1):bnd(2,1)));
        SArad=SAfit*pi/180;
        SArad=SArad(EndTrunc:length(SAfit)-EndTrunc);
        MZtrunc=(MZ(bnd(1,1):bnd(2,1)));
        MZfit=MZtrunc(Sord);
        MZfit=MZfit(EndTrunc:length(MZfit)-EndTrunc);
        MZsmth=smooth(MZfit,SmoothSpan,'lowess');

        % Assure uniqueness
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[SAR Umat junk]=unique(SArad);
MZU=MZsmth(Umat);
MZP(m,n,1:length(MZU))=MZU;
SAP(m,n,1:length(SAU))=SAU;
VECTL(m,n)=length(MZU);

% Calculate Fzmz dFzmz
Fzmz(m,n)=mean(FZ);
dFzmz(m,n)=(Fzmz(m,n)-pFz0)/pFz0;

% Calculate Gammamz
Gammamz(m,n)=mean(IA)*pi/180;

% Compute SHf, SHy, SVy, Ky
% SHf = SHy + (SVy / Ky)
% SHy = (pHy1 + pHy2*dFz) + pHy3*gamma
% SVy = Fz * ((pVy1 + pVy2*dFz) + (pVy3 + pVy4*dFz)*gamma)
% Ky = pKy1*Fz0*sin(2*atan(Fz/(pKy2*Fz0)))*(1-pKy3*abs(gamma))
% At this data sample dFz = 0, gamma = 0, and Fz = Fz0
SHy(m,n)=(pHy1+pHy2*dFzmz(m,n))+pHy3*Gammamz(m,n);
SVy(m,n)=Fzmz(m,n)*((pVy1+pVy2*dFzmz(m,n))+(pVy3+pVy4*dFzmz(m,n)))*Gammamz(m,n);
Ky(m,n)=pKy1*pFz0*sin(2*atan(Fzmz(m,n)/(pKy2*pFz0)))*(1-pKy3*abs(Gammamz(m,n)));
SHf(m,n)=SHy(m,n)+(SVy(m,n)/Ky(m,n));

% Plot treated data curves to be fitted.
if purezmom==1
    figure(n+30);
    hold on;
    plot(SAU,MZU,'b-');
end

% Apply SHf to residual torque slip angle SAR
SAR=SAU+SHf(m,n);

% Compute Residual Torque Peak Factor Dr
Dmzr(m,n)=interp1q(SAU,MZU,-SHf(m,n));

% Compute Br
% Br = (qBz9 + qBz10*By*Cy) (qBz9 is suggested to be 0 by Pacejka)
% Cy = pCy1;
% By = Ky/(Cy*Cy)
% By*Cy=Ky/Dy
% Dy=Mewy*Fz
% Mewy = (pDy1 + pDy2*dFz)*(1 - pDy3*gamma^2)
Dy(m,n)=Fzmz(m,n)*(pDy1+pDy2*dFzmz(m,n))*(1-pDy3*Gammamz(m,n)^2);
By(m,n)=Ky(m,n)/(pCy1*Dy(m,n));
ByCy=By(m,n)*pCy1;
Br1=50; % Previous final Br value utilized as initial point.
Brd=-10;
By0=By(m,n);
SAY=SAU+SHy(m,n);
% Compute Fy0 for operation to isolate pneumatic trail.
Fy0=Dy(m,n)*sin(pCy1*atan(By(m,n))*SAY-((pEy1+pEy2*dFzmz(m,n)))*...
    (1-(pEy3+pEy4*Gammamz(m,n)))*SAY./abs(SAY))]*(By0*SAY-atan...
(By0*SAy)))+SVy(m,n);
Fy0P(m,n,1:length(SAy))=Fy0;
M2r1=Dmzr(m,n)*cos(atan(Br1*SAr)).*cos(SAU);
M2t1=((MZU-M2r1)./(-1*Fy0))./cos(SAU);
M2t=M2t1;
LoAng=.03; HiAng=.15;
TestAng=LoAng:.01:HiAng;
SHT1=0; % Previous final SHT value utilized as initial point.
SHTd=.002;
SHTThresh=.00001;
CDiff1=0;
for k=1:length(TestAng)
    CDiff1=CDiff1+abs(interp1q(SAU,MZt1,TestAng(k)+SHT1)-
                                       interp1q(SAU,MZt1,-TestAng(k)+SHT1));
end
CDiff0=CDiff1;
while 1
    CDiff2=0;
    for k=1:length(TestAng)
        CDiff2=CDiff2+abs(interp1q(SAU,MZt1,TestAng(k)+SHT1+...
                                    SHTd)-interp1q(SAU,MZt1,-1*TestAng(k)+SHT1+SHTd));
    end
    if CDiff2<CDiff1 && abs(SHTd)>SHTThresh
        CDiff1=CDiff2;
        SHT1=SHT1+SHTd;
    elseif CDiff2>=CDiff1 && abs(SHTd)>SHTThresh
        CDiff1=CDiff2;
        SHT1=SHT1+SHTd;
        SHTd=-SHTd/2;
    else
        CDiff0=CDiff1;
        SHT(m,n)=-SHT1;
        break;
    end
end
BrdThresh=1;
Br1Limit=2000;
while 1
    M2r2=Dmzr(m,n)*cos(atan((Br1+Brd)*SAr)).*cos(SAU);
    M2t2=((MZU-M2r2)./(-1*Fy0))./cos(SAU);
    SHT1=0;
    SHTd=.002;
    SHTThresh=.00001;
    CDiff1=0;
    for k=1:length(TestAng)
        CDiff1=CDiff1+abs(interp1q(SAU,MZt2,TestAng(k)+SHT1)-
                                   interp1q(SAU,MZt2,-TestAng(k)+SHT1));
    end
    while 1
        CDiff2=0;
        for k=1:length(TestAng)
            CDiff2=CDiff2+abs(interp1q(SAU,MZt2,TestAng(k)+SHT1+...
                                        SHTd)-interp1q(SAU,MZt2,-1*TestAng(k)+SHT1+SHTd));
        end
        if CDiff2<CDiff1 && abs(SHTd)>SHTThresh
            CDiff1=CDiff2;
        else
            CDiff0=CDiff1;
            SHT(m,n)=-SHT1;
            break;
        end
    end
\[ S_{H1} = S_{H1} + S_{Htd}; \]

\[
\text{elseif } \text{CDiff2} \geq \text{CDiff1} \land \text{abs}(S_{Htd}) > S_{HThresh} \\
\quad \text{CDiff1} = \text{CDiff2}; \\
\quad S_{H1} = S_{H1} + S_{Htd}; \\
\quad S_{Htd} = -\frac{S_{Htd}}{2}; \\
\text{else} \\
\quad \text{CDiff02} = \text{CDiff1}; \\
\quad \text{break}; \\
\end{align*} \]

\[
\text{end} \\
\text{if} \ \text{CDiff02} < \text{CDiff01} \land \land \text{abs}(\text{Brd}) > \text{BrdThresh} \land \text{abs}(\text{Brd}) < \text{BrdLimit} \\
\quad \text{Br1} = \text{Br1} + \text{Brd}; \\
\quad \text{CDiff01} = \text{CDiff02}; \\
\text{elseif} \ \text{CDiff02} \geq \text{CDiff01} \land \text{abs}(\text{Brd}) > \text{BrdThresh} \land \text{abs}(\text{Brd}) < \text{BrdLimit} \\
\quad \text{Br1} = \text{Br1} + \text{Brd}; \\
\quad \text{CDiff01} = \text{CDiff02}; \\
\quad \text{Brd} = \frac{\text{Brd}}{-2}; \\
\text{else} \\
\quad \text{Bmr}(m,n) = \text{Br1}; \\
\quad S_{Ht}(m,n) = -S_{H1}; \\
\quad M_{Zt} = M_{Zt2}; \\
\quad M_{Zr} = M_{Zr2}; \\
\quad \text{break} \\
\end{align*} \]

\[
\text{end} \\
\text{end} \\
\text{if} \ \text{pebugzmo} == 1 \\
\quad \text{figure}(200+n); \\
\quad \text{hold on}; \\
\quad \text{plot}(S_{AU}, M_{Zt}, 'b-'); \\
\quad \text{figure}(230+n); \\
\quad \text{hold on}; \\
\quad \text{plot}(S_{AU}, M_{Zr}/\max(\text{abs}(M_{Zr}))*\max(\text{abs}(F_{y0})), 'b-', S_{AU}, F_{y0}, 'r-'); \\
\text{end} \\
\text{end} \\
\text{ShaveSetting} = \ldots \\
\quad \begin{bmatrix} 20 & 24 & 24 & 24 & 18; \\
\quad 18 & 24 & 18 & 18 & 18; \\
\quad 18 & 18 & 18 & 18 & 18; \\
\quad 18 & 21 & 21 & 24 & 31; \\
\quad 24 & 21 & 28 & 21 & 33 \end{bmatrix}; \ % 24 21 21 21
\]

\[
\text{Cut} = \text{ShaveSetting}(m,n); \\
\text{[SAJ MinInd]} = \min(\text{abs}(S_{AU} - S_{HF}(m,n))); \\
\text{MZtBot} = \text{MZt}(1:\text{MinInd} - \text{Cut}); \\
\text{SAUBot} = \text{SAU}(1:\text{MinInd} - \text{Cut}); \\
\text{MZtTop} = \text{MZt}(\text{MinInd} + \text{Cut} : \text{length}(\text{MZt})); \\
\text{SAUTop} = \text{SAU}(\text{MinInd} + \text{Cut} : \text{length}(\text{MZt})); \
\]
MZtT = MZtBot; MZtT(length(MZtT)+1:length(MZtTop)+length(MZtT)) = MZtTop;
SAUT = SAUBot; SAUT(length(SAUT)+1:length(SAUTop)+length(SAUT)) = SAUTop;
MZtT = smooth(MZtT, 9, 'rloess');

SAUt = SAUT;
SAUT = SAUT + SHt(m, n);

if pebugzmo == 1
    figure(210+n);
    hold on;
    plot(SAUt, MZtT, 'b-', SAUT, MZtT, 'r-');
end

% Calculate Peak Factor Dt
[Dt MidInd] = max(MZtT);
Dmzt(m, n) = Dt;
MZtT1 = MZtT/Dt;

DB = .1;
DE = .1;
DBt = DB;
DEt = DE;
BtB = 1.0;
EtB = 0.5;

Mt110 = sum(abs(cos(qCz1*atan(BtB*SAUT-EtB*(BtB*SAUT-atan(BtB*SAUT)))-MZtT1));

while 1
    DBt = DB;
    Mt100 = sum(abs(cos(qCz1*atan((BtB+DBt)*SAUT-(EtB-DEt)*(BtB+DBt)*SAUT)-atan((BtB+DBt)*SAUT)))-MZtT1);
    Mt000 = Mt100;
    while 1
        Mt200 = sum(abs(cos(qCz1*atan((BtB+DBt)*SAUT-(EtB-DEt)*(BtB+DBt)*SAUT)-atan((BtB+DBt)*SAUT)))-MZtT1);
        if Mt200 < Mt100
            BtB = BtB + DBt;
            Mt000 = Mt100;
            Mt100 = Mt200;
        elseif Mt200 >= Mt100
            BtB = BtB + DBt;
            DBt = -DBt/4;
            Mt100 = Mt200;
        end
        if abs(DBt) < .00001
            Mt120 = Mt200;
            break;
        end
    end
    if Mt120 < Mt110
        EtB = EtB - DEt;
        Mt100 = Mt110;
    end
end

if Mt120 < Mt110
    EtB = EtB - DEt;
    Mt100 = Mt110;

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Mt110=Mt120;
elseif Mt120>=Mt110
EtB=EtB-DEt;
Mt110=Mt120;
DEt=-DEt/4;
end
if abs(DEt)<.00001
Bmzt(m,n)=BtB;
Emzt(m,n)=EtB;
break;
end

if pebugzmo==1
figure(220+n);
hold on;
plot(SAUT,MZtT1,'b-',SAUT,cos(qCz1.*atan(BtB.*SAUT-EtB.*...BtB.*SAUT-atan(BtB.*SAUT))),'r-');
end

if purezmom==1
clear SAUT;
SAUT=SAU+SHt(m,n);
MZR0=Dmzr(m,n).*cos(atan(Bmzr(m,n).*SAr)).*cos(SAU);
t0=Dmzt(m,n).*cos(qCz1.*atan(Bmzt(m,n).*SAUT-Emzt(m,n).*...Bmzt(m,n).*SAUT-atan(Bmzt(m,n).*SAUT))).*cos(SAU);
MZ0=-t0.*Fy0+MZR0;
figure(n+30);
hold on;
plot(SAU,MZ0,'r-');
end
end

%% Aligning Moment - Pure Side Slip - Final Coefficient Definition

% Aligning Moment (Pure Side Slip) Coefficients
% User scaling factors not listed in equations
% Nominal conditions are matrix element (3,1)

% Raw Fitted Coefficients
pRAWMZO.Fzmz=Fzmz;
pRAWMZO.Gammamz=Gammamz;

% Residual Torque Coefficient Definition

% Stiffness Factor - Br
% Br = qBz9 + qBz10*ByCy
if purezmom==1
    figure(1201);
    mesh(IAP,FZP,Bmzr); title('Br_m_z');
    xlabel('IA (deg)'); ylabel('FZ (N)');
end
pRAWMZO.Bmzr=Bmzr;
k=mean(mean(Bmzr));
for m=1:length(FZP)
    for n=1:length(IAP)
if Bmzr(m,n)<=k-100
  Bmzr(m,n)=-50;
end
end

gBz9=mean(mean(Bmzr));
Bmzr1=(Bmzr-qBz9)./(By*pCy1);
gBz10=mean(mean(Bmzr1));
Bmzr0=qBz9+gBz10*By*pCy1;

% Peak Factor - Dr
% Dr = Fz*R0*[(qDz6 + qDz7*dFz) + (qDz8 + qDz9*dFz)*Gamma ]
if purezmom==1
  figure(1202);
  mesh(IAP,FZP,Dmzr); title('Dr_m_z');
  xlabel('IA (deg)'); ylabel('FZ (N)');
end

pRAWMZ0Fit.Dmzr=Dmzr;
Dr=Dmzr./(Fzmz*pR0);
Slope=polyfit(Gammamz(3,:),Dr(3,:),1);
qDz6=Slope(2);
qDz8=Slope(1);
Dr=(Dr-qDz6-qDz8*Gammamz)./dFzmz;
MqDz7=mean(Dr,2);
qDz7=(sum(MqDz7)-MqDz7(3))/(length(MqDz7)-1);
clear Slope
for k=1:length(FZP)
  Slope(k,:)polyfit(Gammamz(k,:),Dr(k,:),1);
end
qDz9=(sum(Slope(:,1))-Slope(3,1))/(length(FZP)-1);
Dmzr0=Fzmz.*pR0.*((qDz6+qDz7.*dFzmz)+(qDz8+qDz9.*dFzmz).*Gammamz);

% Pneumatic Trail Coefficient Definition

% Horizontal Shift Factor - SHt
% SHt = qHz1 + qHz2*dFz + (qHz3 + qHz4*dFz)*Gamma
if purezmom==1
  figure(1203);
  mesh(IAP,FZP,SHt); title('SHt_m_z');
  xlabel('IA (deg)'); ylabel('FZ (N)');
end

pRAWMZ0Fit.SHt=SHt;
clear Slope
Slope=polyfit(Gammamz(3,:),SHt(3,:),1);
qHz1=Slope(2);
qHz3=Slope(1);
SHt1=SHt-qHz1-qHz3.*Gammamz./dFzmz;
MqHz2=mean(SHt1,2);
qHz2=(sum(MqHz2)-MqHz2(3))/(length(MqHz2)-1);
clear Slope
for k=1:length(FZP)
  Slope(k,:)polyfit(Gammamz(k,:),SHt1(k,:),1);
end
qHz4=(sum(Slope(:,1))-Slope(3,1))/(length(FZP)-1);
SHt0=qHz1+qHz2*dFzmz+(qHz3+qHz4*dFzmz).*Gammamz;
% Stiffness Factor - Bt
% \( Bt = (qBz1 + qBz2*dFz + qBz3*dFz^2)*(1 + qBz4*Gamma + qBz5*abs(Gamma)) \)

if purezmom==1
    figure(1204);
    mesh(IAP,FZP,Bmzt); title('Bt_m_z');
    xlabel('IA (deg)'); ylabel('FZ (N)');
end

pRAWMZ0Fit.Bmzt=Bmzt;
clear Slope
Slope=polyfit(dFzmz(:,1),Bmzt(:,1),2);
qBz3=Slope(1);
qBz2=Slope(2);
qBz1=Slope(3);
Bmzt1=Bmzt./(qBz1+qBz2.*dFzmz+qBz3.*dFzmz.^2);
qBz5=0; % No data to fit to.
clear Slope
for k=1:length(FZP)
    Slope(k,:)=polyfit(Gammamz(k,:),Bmzt1(k,:),1);
end
qBz4=mean(Slope(:,1));
Bmzt0=(qBz1+qBz2.*dFzmz+qBz3.*dFzmz.^2).*...
    (1+qBz4.*Gammamz+qBz5.*abs(Gammamz));

% Peak Factor - Dt
% \( Dt = Fz*(qDz1 + qDz2*dFz)*(1 + qDz3*Gamma + qDz4*Gamma^2)*(R0/Fz0) \)

pRAWMZ0Fit.Dmzt=Dmzt;
if purezmom==1
    figure(1205);
    mesh(IAP,FZP,Dmzt); title('Dt_m_z');
    xlabel('IA (deg)'); ylabel('FZ (N)');
end
if pDmztMod==0
    Dmzt1=Dmzt./(Fzmz.*(pR0./pFz0));
clear Slope
    for k=1:length(IAP)
        Slope(k,:)=polyfit(dFzmz(:,k),Dmzt1(:,k),1);
    end
    qDz2=mean(Slope(:,1));
    qDz1=mean(Slope(:,2));
    Dmzt2=Dmzt1./(qDz1+qDz2.*dFzmz);
elseif pDmztMod==1
    % Modified Dmzt Fit - No FZ multiplication!
    Dmzt1=Dmzt./(pR0./pFz0);
clear Slope
    for k=1:length(IAP)
        Slope(k,:)=polyfit(dFzmz(:,k),Dmzt1(:,k),1);
    end
    qDz2m=mean(Slope(:,1));
    qDz1m=mean(Slope(:,2));
    Dmzt2=Dmzt1./(qDz1m+qDz2m.*dFzmz);
end

clear Slope
for k=1:length(FZP)
    Slope(k,1:3)=polyfit(Gammamz(k,1:length(IAP)),...
                   Dmzt2(k,1:length(IAP)),2);
end
qDz3=mean(Slope(:,2));
qDz4=mean(Slope(:,1));
if pDmztMod==0
    Dmzt0=Fzmz.*(qDz1+qDz2.*dFzmz).*((1+qDz3.*Gammamz+qDz4.*
    Gammamz.^2).*pR0./pFz0);
elseif pDmztMod==1
    Dmzt0=(qDz1m+qDz2m.*dFzmz).*((1+qDz3.*Gammamz+qDz4.*
    Gammamz.^2).*pR0./pFz0);
end
if purezmom==1
    figure(1206);
    mesh(IAP,FZP,Dmzt0); title('Dt_m_z Fitted');
    xlabel('IA (deg)'); ylabel('FZ (N)');
end

% Curvature Factor - Et
% Et = {1 + (qEz4 + qEz5*Gamma)*(2/pi)*atan(Bt*Ct*SAt)} for (Et<=1)
if purezmom==1
    figure(1207);
    mesh(IAP,FZP,Emzt); title('Et_m_z');
    xlabel('IA (deg)'); ylabel('FZ (N)');
end
pRAWMZ0Fit.Emzt=Emzt;
qEz1=.2; qEz2=-.6; qEz3=1;
qEz4=-.1; qEz5=1.7;
clear TEST DTEST SAMP
TEST=[qEz1 qEz2 qEz3 qEz4 qEz5];
DTEST=[1 1 1 1 1];
SampSize=150;
TESTc=TEST;
Mz0=zeros(MATSIZE);
for k=1:MATSIZE(3)
    Emzt0=(TESTc(1)+TESTc(2).*dFzmz+TESTc(3).*dFzmz.^2).*((1+(TESTc(4)...
    +TESTc(5).*Gammamz).*((2/pi).*atan(qCz1.*Bmzt0.*SAP(;;,k))));
    SAt=SAP(;;,k)+SHt0;
    SAR=SAP(;;,k)+SHf;
    tP=Dmzt0.*cos(qCz1.*atan(Bmzt0.*SAt-Emzt0.*(Bmzt0.*SAt-atan...
    (Bmzt0.*SAt)))).*cos(SAP(;;,k));
    Mzr=Dmzr0.*cos(atan(Bmzr0.*SAr)).*cos(SAP(;;,k));
    Mz0(;;,k)=-tP.*Fy0P(;;,k)+Mzr;
end
BEST=sum(sum(abs(Mz0-MZP),3)./VECTL)./Fzmz);
disp(BEST);
disp(TEST);
m=0;
kthresh=25;
while 1
    SAMP=randn(SampSize,length(TEST));
    SAMP=ABS(SAMP);
    RESU=zeros(SampSize,1);
    for p=1:SampSize
        TESTc=TEST+SAMP(p,:).*DTEST;
        for k=1:MATSIZE(3)
            Emzt0=(TESTc(1)+TESTc(2).*dFzmz+TESTc(3).*dFzmz.^2).*((1+(TESTc(4)+TESTc(5).*Gammamz).*((2/pi).*atan(qCz1.*...,
            Bmzt0.*SAP(;;,k))));
            SAt=SAP(;;,k)+SHt0;
        end
    end
end
SAr=SAP(:,:,k)+SHf;
tP=Dmzt0.*cos(qCz1.*atan(Bmzt0.*SAt-Emzt0.*\(Bmzt0.*SAt-\atan(Bmzt0.*SAt)))).*cos(SAP(:,:,k));
Mzr=Dmzr0.*cos(atan(Bmzr0.*SAr)).*cos(SAP(:,:,k));
Mz0(:,:,k)=-tP.*Fy0P(:,:,k)+Mzr;
end
RESU(p,1)=sum(sum((sum(abs(Mz0-MZP),3)./VECTL)./Fzmz));
end
m=m+1;
[TBEST TInd]=min(RESU);
if TBEST<BEST && m<kthresh
BEST=TBEST;
TEST=TEST+DTEST.*SAMP(TInd,);
m=0;
disp(BEST);
disp(TEST);
else m>=kthresh
break
end
end
qEz1=TEST(1);
qEz2=TEST(2);
qEz3=TEST(3);
qEz4=TEST(4);
qEz5=TEST(5);

% Plot finalized fitted curves for comparison
if purezmom==1
    for n=1:length(IAP)
        for m=1:length(FZP)
            clear SAt SAR tP Mzr Fy0 SAs Mz0
            SAt=SAU+SHt0(m,n);
            SAR=SAU+SHf(m,n);
            if qEz4==0 && qEz5==0
                tP=Dmzt0(m,n).*cos(qCz1.*atan(Bmzt0(m,n).*SAt-\(SAt-\atan(Bmzt0(m,n).*SAt\))).)*cos(SAU);
            else
                Emzt0=(qEz1+qEz2.*dFzmz(m,n)+qEz3.*dFzmz(m,n).^2)*\(1+(qEz4+qEz5.*Gammamz(m,n)).*(2/pi).*atan(qCz1.*\(Bmzt0(m,n).*SAt\)).\);
                tP=Dmzt0(m,n).*cos(qCz1.*atan(Bmzt0(m,n).*SAt-\(Bmzt0(m,n).*SAt\))).)*cos(SAU);
            end
            Mzr=Dmzr0(m,n).*cos(atan(Bmzr0(m,n).*SAr)).*cos(SAU);
            SAs=SAU+SHy(m,n);
            Fy0=Dy(m,n)*sin(pCyl*atan(By(m,n)*SAy-((pEy1+pEy2*dFzmz(m,n)...)(1-(pEy3+pEy4*Gammamz(m,n))*SAy./abs(SAy))).)*\(By0*\(SAY-\atan(By0*SAy)\)).+SVy(m,n);
            Mz0=-tP.*Fy0+Mzr;
            figure(n+30);
            hold on;
            plot(SAU,Mz0,'g-');
        end
    end
end
%% Longitudinal Force - Combined Slip - Coefficient Determination

% Slip Angle Runs
SAP=[3 6];
% Inclination Angle Runs
IAP=[0 2 4];
% Normal Load Runs
FZP=[222 667 1112 1557];

FXU=zeros(length(FZP),length(SAP)*length(IAP),600);
FYU=zeros(length(FZP),length(SAP)*length(IAP),600);
MZU=zeros(length(FZP),length(SAP)*length(IAP),600);
SRU=zeros(length(FZP),length(SAP)*length(IAP),600);
Fx0=zeros(length(FZP),length(SAP)*length(IAP),600);
Fzxa=zeros(length(FZP),length(SAP)*length(IAP));
dFzxa=zeros(length(FZP),length(SAP)*length(IAP));
Gammaaxa=zeros(length(FZP),length(SAP)*length(IAP));
SAxa=zeros(length(FZP),length(SAP)*length(IAP));
VECTL=zeros(length(FZP),length(SAP)*length(IAP));

% Load, treat, and store data for fitting process of combined FX, FY, and MZ.
for p=1:length(SAP)
    for n=1:length(IAP)
        for m=1:length(FZP)
            cn=(p-1)*length(IAP)+n;
            clear FZN SRN FXN IAN SAN FYN MZN FZ SR FX IA SA FY MZ
            FZN=[pFileName '.BrakeDrive' int2str(SAP(p)) 'SA.' pressure...
                 'IA' int2str(IAP(n)) 'FZ' int2str(FZP(m)) '.FZ'];
            SRN=[pFileName '.BrakeDrive' int2str(SAP(p)) 'SA.' pressure...
                 'IA' int2str(IAP(n)) 'FZ' int2str(FZP(m)) '.SL'];
            FXN=[pFileName '.BrakeDrive' int2str(SAP(p)) 'SA.' pressure...
                 'IA' int2str(IAP(n)) 'FZ' int2str(FZP(m)) '.FX'];
            IAN=[pFileName '.BrakeDrive' int2str(SAP(p)) 'SA.' pressure...
                 'IA' int2str(IAP(n)) 'FZ' int2str(FZP(m)) '.IA'];
            SAN=[pFileName '.BrakeDrive' int2str(SAP(p)) 'SA.' pressure...
                 'IA' int2str(IAP(n)) 'FZ' int2str(FZP(m)) '.SA'];
            FYN=[pFileName '.BrakeDrive' int2str(SAP(p)) 'SA.' pressure...
                 'IA' int2str(IAP(n)) 'FZ' int2str(FZP(m)) '.FY'];
            MZN=[pFileName '.BrakeDrive' int2str(SAP(p)) 'SA.' pressure...
                 'IA' int2str(IAP(n)) 'FZ' int2str(FZP(m)) '.MZ'];
            eval(['FZ=' FZN ';']);
            eval(['SR=' SRN ';']);
            eval(['FX=' FXN ';']);
            eval(['IA=' IAN ';']);
            eval(['SA=' SAN ';']);
            eval(['FY=' FYN ';']);
            eval(['MZ=' MZN ';']);
        end
    end
end

% Truncate data to one full sweep minimizing start and end effects
k=1;
bd=[1:length(SR)];
for o=1:length(SR)-1
    if k<=1 && SR(o)>=.15 && SR(o+1)<.15
        bd(k,1)=o;
    end
end
$$k = k + 1;$$

**elseif** SR(1) < .15 && k == 1
  $$bnd(k,1) = 1;$$
  $$k = k + 1;$$
**elseif** o == length(SR) && k == 2
  $$bnd(k,1) = o;$$
  **break;**
**elseif** SR(o) < .15 && SR(o+1) >= .15 && k == 2 && ...
  $$(o - bnd(1,1)) >= 100$$
  $$bnd(k,1) = o;$$
  $$k = k + 1;$$
**elseif** k == 3
  **break;**
end

% Compute dFz for the given run
% dFz = (Fz - pFz0)/pFz0
Fzxa(m,cn) = mean(FZ(bnd(1,1):bnd(2,1)));
$$dFzxa(m,cn) = (Fzxa(m,cn) - pFz0)/pFz0;$$

% Record IA for given run
GammaXa(m,cn) = (mean(IA(bnd(1,1):bnd(2,1)))*pi/180);

% Record SA for given run
SAXa(m,cn) = (mean(SA(bnd(1,1):bnd(2,1)))*pi/180);

SmoothSpan = 35;
[SRfit, Sord] = sort(SR(bnd(1,1):bnd(2,1)));
FXtrunc = (FX(bnd(1,1):bnd(2,1)));
FYtrunc = (FY(bnd(1,1):bnd(2,1)));
MZtrunc = (MZ(bnd(1,1):bnd(2,1)));
FXfit = FXtrunc(Sord);
FYfit = FYtrunc(Sord);
MZfit = MZtrunc(Sord);
FXsmth = smooth(FXfit, SmoothSpan, 'lowess');
FYsmth = smooth(FYfit, SmoothSpan, 'lowess');
MZsmth = smooth(MZfit, SmoothSpan, 'lowess');

% Store the smoothed unique data to 3-d matrices
[SRUi Umat junk] = unique(SRfit);
VECTL(m,cn) = length(SRU1);
SRU(m,cn,1:length(SRU1)) = SRUi;
FXU(m,cn,1:length(SRU1)) = FXsmth(Umat);
FYU(m,cn,1:length(SRU1)) = FYsmth(Umat);
MZU(m,cn,1:length(SRU1)) = MZsmth(Umat);

% Graph Raw Data
if comblong == 1
  figure(40+cn);
  hold on;
  plot(SRU1, FXsmth(Umat), 'b-');
end

% Compute Fx0 for given conditions
Kx = Fzxa(m,cn) * (pKx1 + pKx2 * dFzxa(m,cn)) * exp(pKx3 * dFzxa(m,cn));
\begin{align*}
\text{Cx} &= \text{pCx1}; \\
\text{Dx} &= \text{Fzxa(m, cn)} \cdot (\text{pDx1} + \text{pDx2} \cdot \text{dFzxa(m, cn)}) \cdot (1 - \text{pDx3} \cdot \text{Gammaxa(m, cn)}^2); \\
\text{Bx} &= \text{Kx} / (\text{Cx} \cdot \text{Dx}); \\
\text{SHx} &= \text{pHx1} + \text{pHx2} \cdot \text{dFzxa(m, cn)}; \\
\text{SVx} &= \text{Fzxa(m, cn)} \cdot (\text{pVx1} + \text{pVx2} \cdot \text{dFzxa(m, cn)}); \\
\text{SRx} &= \text{SRUi} + \text{SHx}; \\
\text{Fx0i} &= \text{Dx} \cdot \sin(\text{Cx} \cdot \text{atan} (\text{Bx} \cdot \text{SRx} - ((\text{pEx1} + \text{pEx2} \cdot \text{dFzxa(m, cn)} + \text{pEx3} \cdot \text{dFzxa(m, cn)}^2) \cdot (1 - \text{pEx4} \cdot \text{abs(SRx)}/\text{SRx}) \cdot (\text{Bx} \cdot \text{SRx})))) + \text{SVx}; \\
\text{Fx0} &= \text{Fx0i}(:,1);
\end{align*}

end

end

\% Compute Fy0 for given conditions
\text{clear SHy SVy By Ey Dy Ky Cy SAy}
\text{SHy} = (\text{pHy1} + \text{pHy2} \cdot \text{dFzxa}) + \text{pHy3} \cdot \text{Gammaxa};
\text{SVy} = \text{Fzxa} \cdot ((\text{pVy1} + \text{pVy2} \cdot \text{dFzxa}) + (\text{pVy3} + \text{pVy4} \cdot \text{dFzxa}) \cdot \text{Gammaxa});
\text{SAy} = \text{SAxa} + \text{SHy};
\text{Cy} = \text{pCy1};
\text{Dy} = \text{pDy1} + \text{pDy2} \cdot \text{dFzxa} \cdot (1 - \text{pDy3} \cdot \text{Gammaxa}^2); \\
\text{Ky} = \text{pKy1} \cdot \text{pFz0} \cdot \sin(2 \cdot \text{atan(Fzxa}/(\text{pKy2} \cdot \text{pFz0}))) \cdot (1 - \text{pKy3} \cdot \text{abs(Gammaxa)}); \\
\text{By} = \text{Ky} / (\text{Cy} \cdot \text{Dy});
\text{Fy0} = \text{Dy} \cdot \sin(Cy \cdot \text{atan(By} \cdot \text{SAy} - ((\text{pEy1} + \text{pEy2} \cdot \text{dFzxa}) \cdot (1 - (\text{pEy3} + \text{pEy4} \cdot \text{Gammaxa}) \cdot \text{abs(SAxa)} / \text{SAxa})) \cdot (By \cdot \text{SAy})) + \text{SVy};

\% Initial Coefficient Values and Search Deltas
\text{Bx1i} = -16.5;
\text{Bx1d} = 8;
\text{Bx2i} = -9;
\text{Bx2d} = 5;
\text{Cx1i} = .99;
\text{Cx1d} = .01;
\text{Ex1i} = -.7;
\text{Ex1d} = .5;
\text{Ex2i} = 1;
\text{Ex2d} = .5;
\text{Hx1i} = 0;
\text{Hx1d} = .01;
\text{SampSize} = 150;

\% Mirror data to provide better fitment. Otherwise variables will be skewed heavily to an asymmetric situation due to the lack of negative slip angles. As such the assumption of mirroring is better than the consequences of having no mirroring at all.
\text{MATSz= size(SRU)};
\text{DMATSz = [2*MATSz(1) MATSz(2) MATSz(3)];}
\text{SRUf = zeros(DMATSz);}
\text{dFzxaF = zeros(DMATSz(1:2));}
\text{Fx0f = zeros(DMATSz);}
\text{FXUf = zeros(DMATSz);}
\text{SAxaf = zeros(DMATSz(1:2));}
\text{VECTlf = zeros(DMATSz(1:2));}
\text{SRUf(1:MATSz(1),1:MATSz(2),1:MATSz(3)) = SRU;}
\text{SRUf(MATSz(1)+1:MATSz(1),1:MATSz(2),1:MATSz(3)) = SRU;}
\text{Fx0f(1:MATSz(1),1:MATSz(2),1:MATSz(3)) = Fx0;
Fx0f(MATSIZE(1)+1:DMATSIZE(1),1:MATSIZE(2),1:MATSIZE(3))=Fx0;
FXUf(1:MATSIZE(1),1:MATSIZE(2),1:MATSIZE(3))=FXU;
FXUf(MATSIZE(1)+1:DMATSIZE(1),1:MATSIZE(2),1:MATSIZE(3))=FXU;
dFzxaf(1:MATSIZE(1),1:MATSIZE(2))=dFzxaf;
dFzxaf(MATSIZE(1)+1:DMATSIZE(1),1:MATSIZE(2))=dFzxaf;
SAxa(1:MATSIZE(1),1:MATSIZE(2))=SAxa;
SAxa(MATSIZE(1)+1:DMATSIZE(1),1:MATSIZE(2))=SAxa.*(-1);
VECLf(1:MATSIZE(1),1:MATSIZE(2))=VECLf;
VECLf(MATSIZE(1)+1:DMATSIZE(1),1:MATSIZE(2))=VECLf;

% Make initial computation based off initial guess for comparison in loop.
Bxa=Bxli*cos(atan(Bx2i.*SRUf));
BxaHx1=Bxa*Hx1i;
BxaSAxa=zeros(DMATSIZE);
Exa=zeros(DMATSIZE);
for bean=1:DMATSIZE(3)
    BxaSAxa(:,:,bean)=Bxa(:,:,bean).*(SAxaf+Hx1i);
    Exa(:,:,bean)=Ex1i+Ex2i.*dFzxaf;
end
Gxi=cos(Cx1i.*atan(BxaSAxa-Ex1i.*(BxaSAxa-atan(BxaSAxa))))./cos(Cx1i.*...
          atan(BxaHx1-Ex1i.*(BxaHx1-atan(BxaHx1))));
Fxi=Fx0f.*Gxi;
BEST=sum(sum((sum(abs(Fxi-FXUf).*abs(SRUf+abs(SRUf)./3),3)./VECLf));

% Set up test matrices and counters.
TEST=[Bxli Bx2i Cxli Exli Ex2i Hxli];
DTEST=[Bx1d Bx2d Cx1d Ex1d Ex2d Hx1d];
SAMP=zeros(SampSize,length(TEST));
RESU=zeros(SampSize,1);
k=0;
kthresh=60;
disp(BEST);
disp(TEST);

% Curve fitting is done with a randomized 'cloud' format. This could be
% improved quite a bit to deal with situations where the best lies outside
% of the search region at another localized optima.
while 1
    SAMP=randn(SampSize,length(TEST));
    SAMP=SAMP.*abs(SAMP);
    for n=1:SampSize
        TESTc=TEST+DTEST.*SAMP(n,:);
        if TESTc(3)>=.99
            TESTc(3)=.99;
        elseif TESTc(3)<=.85
            TESTc(3)=.85;
        end
        if TESTc(6)>=.1
            TESTc(6)=.1;
        elseif TESTc(6)<=-.1
            TESTc(6)=-.1;
        end
        Bxa=TESTc(1)*cos(atan(TESTc(2).*SRUf));
        BxaHx1=Bxa*TESTc(6);
        for bean=1:MATSIZE(3)
            BxaSAxa(:,:,bean)=Bxa(:,:,bean).*(SAxaf+TESTc(6));
        end
        Gxi=cos(Cx1i.*atan(BxaSAxa-Ex1i.*(BxaSAxa-atan(BxaSAxa))))./cos(Cx1i.*...
              atan(BxaHx1-Ex1i.*(BxaHx1-atan(BxaHx1))));
        Fxi=Fx0f.*Gxi;
        BEST=sum(sum((sum(abs(Fxi-FXUf).*abs(SRUf+abs(SRUf)./3),3)./VECLf)));
Exa(:,:,bean)=TESTc(4)+TESTc(5).*dFzxaf;
end
Gxi=cos(TESTc(3).*atan(BxaSAxa-Exa.*(BxaSAxa-atan(BxaSAxa))))./cos(TESTc(3).*atan(BxaHx1-Exa.*(BxaHx1-atan(BxaHx1))));
Fxi=Fx0f.*Gxi;
RESU(n,1)=sum(sum((sum(abs(Fxi-FXUf).*abs(SRUf+abs(SRUf)/3),... 3)./VECTLf)));}
end
k=k+1;
[TBEST TInd]=min(RESU);
if TBEST<BEST && k<kthresh
BEST=TBEST;
TESTc=TEST+DTEST.*SAMP(TInd,:);
if TESTc(3)>=.99
TESTc(3)=.99;
elseif TESTc(3)<=.80
TESTc(3)=.80;
end
if TESTc(6)>=.1
TESTc(6)=.1;
elseif TESTc(6)<=-.1
TESTc(6)=-.1;
end
TEST=TESTc;
k=0;
disp(BEST);
disp(TEST);
elseif k>=kthresh
break
end
end
rBx1=TEST(1); %rBx1=0;
rBx2=TEST(2); %rBx2=0;
rCx1=TEST(3);
rEx1=TEST(4);
rEx2=TEST(5);
rHx1=TEST(6);

% Compute Fx for Combined Z Moment Calculation
TESTc=TEST;
Bxa=TESTc(1)*cos(atan(TESTc(2).*SRU));
BxaHx1=Bxa*TESTc(6);
BxaSAxa=zeros(MATSIZE);
Exa=zeros(MATSIZE);
for bean=1:MATSIZEx(3)
  BxaSAxa(:,:,bean)=Bxa(:,:,bean).*(SAxa+TESTc(6));
  Exa(:,:,bean)=TESTc(4)+TESTc(5).*dFxa;
end
Gxi=cos(TESTc(3).*atan(BxaSAxa-Exa.*(BxaSAxa-atan(BxaSAxa)))./cos(... (TESTc(3).*atan(BxaHx1-Exa.*(BxaHx1-atan(BxaHx1)))));
Fxi=Fx0f.*Gxi;

% Plot Combined Longitudinal Data
if comblong==1
  for p=1:length(SAP)
for n=1:length(IAP)
    for m=1:length(FZP)
        clear Fx0i SRUi
        cn=(p-1)*length(IAP)+n;
        for k=1:VECTL(m,cn)
            Fx0i(k)=Fx0(m,cn,k);
            SRUi(k)=SRU(m,cn,k);
        end
        Bxa=rBx1*cos(atan(rBx2.*SRUi));
        BxaHx1=Bxa*rHx1;
        BxaSxAx=Bxa.*(SxAx(m,cn)+rHx1);
        Exa=rEx1+rEx2.*dFzxa(m,cn);
        Gxi=cos(rCx1.*atan(BxaSxAx-Exa.*(BxaSxAx)))/cos(rCx1.*atan(BxaHx1-Exa.*(BxaHx1-atan(BxaHx1))));
        Fxi=Fx0i.*Gxi;
        figure(40+cn);
        hold on;
        plot(SRUi,Fxi,'g-');
    end
end
end

%% Lateral Force - Combined Slip - Coefficient Determination
% The combined slip lateral force coefficients are fitted from the same
% data as the longitudinal combined features. As such the data has already
% been defined during the longitudinal fitting.

% Plot raw data curves.
if combside==1
    for p=1:length(SAP)
        for n=1:length(IAP)
            for m=1:length(FZP)
                clear Fyi SRUi
                cn=(p-1)*length(IAP)+n;
                Fyi=zeros(VECTL(m,cn),1);
                for k=1:VECTL(m,cn)
                    Fyi(k)=FYU(m,cn,k);
                    SRUi(k)=SRU(m,cn,k);
                end
                figure(50+cn);
                hold on;
                plot(SRUi,Fyi,'b-');
                % figure(60+cn);
                % hold on;
                % plot(SRUi,Fyi./Fy0(m,cn),'b-');
            end
        end
    end
end

% Initial Coefficient Data
rBy1=11.718;
rBy2=7.988;
rBy3=-.05;
clear TEST DTEST SAMP
TEST=[rBy1 rBy2 rBy3 rCy1 rEy1 rEy2 rHy1 rHy2 ...
    rVy1 rVy2 rVy3 rVy4 rVy5 rVy6];
DTEST=[2 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1 .1];
SampSize=200;
TESTc=TEST;

% Create Clearing Matrix
CLEAR=zeros(MATSIZE);
for m=1:MATSIZE(1)
    for n=1:MATSIZE(2)
        CLEAR(m,n,1:VECTL(m,n))=1;
    end
end

% Initial fitment check for comparison to in loop.
Byk=TESTc(1).*cos(atan(TESTc(2).*SAxa-TESTc(3)));
Cyk=TESTc(4);
Eyk=TESTc(5)+TESTc(6).*dFzxa;
SHyk=TESTc(7)+TESTc(8).*dFzxa;
Mewy=(pDy1+pDy2.*dFzxa).*(1-pDy3.*Gammaxa.^2);
DVyk=Fzxa.*(TESTc(9)+TESTc(10).*dFzxa+TESTc(11).*Gammaxa).*...
    cos(atan(TESTc(12).*SAxa)).*Mewy;
BykSHyk=Byk.*SHyk;
SRs=zeros(MATSIZE);
SVyk=zeros(MATSIZE);
BSH=zeros(MATSIZE);
EYK=zeros(MATSIZE);
FY0=zeros(MATSIZE);
BYK=zeros(MATSIZE);
for k=1:MATSIZE(3)
    SRs(:,:,k)=SRU(:,:,k)+SHyk;
    SVyk(:,:,k)=DVyk.*sin(atan(TESTc(13).*atans (14).*SRU(:,:,k)));
    BSH(:,:,k)=BykSHyk;
    EYK(:,:,k)=Eyk;
    BYK(:,:,k)=Byk;
    FY0(:,:,k)=FY0;
end
BYKSRs=SRs.*BYK.*CLEAR;
SRs=SRs.*CLEAR;
SVyk=SVyk.*CLEAR;
BSH=BSH.*CLEAR;
EYK=EYK.*CLEAR;
FY0=FY0.*CLEAR;

Gy=cos(Cyk.*atan(BYKSRs-EYK.* (BYKSRs-atan(BYKSRs))))./cos(Cyk.*atan...(BSH-EYK.* (BSH-atan(BSH))));
Fy=Gy.*FY0+SVyk;
BEST=sum(sum(abs(Fy-FYU),3)./VECTL)./Fzxa);
disp(BEST);
disp(TEST);

% Fitting loop is similar in process to longitudinal fitment and could see % serious gains in performance with some work.
m=0;
while 1
   SAMP=randn(SampSize,length(TEST));
   SAMP=SAMP.*abs(SAMP);
   for n=1:SampSize;
      TESTc=TEST+DTEST.*SAMP(n,:);
      Byk=TESTc(1).*cos(atan(TESTc(2).* (SAxa-TESTc(3))));
      Cyk=TESTc(4);
      Eyk=TESTc(5)+TESTc(6).*dFzxa;
      SHyk=TESTc(7)+TESTc(8).*dFzxa;
      Mewy=(pDy1+pDy2.*dFzxa).*(1-pDy3.*Gammaxa.^2);
      DVyk=Fzxa.*(TESTc(9)+TESTc(10).*dFzxa+TESTc(11).*Gammaxa).*... 
          cos(atan(TESTc(12).*SAxa)).*Mewy;
      BykSHyk=Byk.*SHyk;
      for k=1:MATSIZE(3)
         SRs(:,:,k)=SRU(:,:,k)+SHyk;
         SVyk(:,:,k)=DVyk.*sin(atan(TESTc(13).*SRU(:,:,k)));
         BSH(:,:,k)=BykSHyk;
         EYK(:,:,k)=Eyk;
         BYK(:,:,k)=Byk;
         FY0(:,:,k)=FY0;
      end
      BYKSRs=SRs.*BYK.*CLEAR;
      SRs=SRs.*CLEAR;
      SVyk=SVyk.*CLEAR;
      BSH=BSH.*CLEAR;
      EYK=EYK.*CLEAR;
      FY0=FY0.*CLEAR;
      Gy=cos(Cyk.*atan(BYKSRs-EYK.* (BYKSRs-atan(BYKSRs))))./cos(Cyk.*... 
         atan(BSH-EYK.* (BSH-atan(BSH))));
      Fy=Gy.*FY0+SVyk;
      RESU(n,1)=sum(sum(abs(Fy-FYU),3)./VECTL)./Fzxa);
   end
   m=m+1;
   [TBEST TInd]=min(RESU);
   if TBEST<BEST && m<kthresh
      BEST=TBEST;
      TEST=TEST+DTEST.*SAMP(TInd,:);
      m=0;
      disp(BEST);
      disp(TEST);
   elseif m>=kthresh
      break
   end
end
% Computation of fitted graph data
Byk=TESTc(1).*cos(atan(TESTc(2).*(SAxa-TESTc(3))));
Cyk=TESTc(4);
Eyk=TESTc(5)+TESTc(6).*dFzxa;
SHyk=TESTc(7)+TESTc(8).*dFzxa;
Mewy=(pDy1+pDy2.*dFzxa).*(1-pDy3.*Gammaxa.^2);
DVyk=Fzxa.*(TESTc(9)+TESTc(10).*dFzxa+TESTc(11).*Gammaxa).*cos(atan... 
(TESTc(12).*SAxa)).*Mewy;
BykSHyk=Byk.*SHyk;
for k=1:MATSIZEx(3)
    SRs(:,;,k)=SRU(:,;,k)+SHyk;
    SVyk(:,;,k)=DVyk.*sin(TESTc(13).*atan(TESTc(14).*SRU(:,;,k)));
    BSH(:,;,k)=BykSHyk;
    EYK(:,;,k)=Eyk;
    BYK(:,;,k)=Byk;
    FY0(:,;,k)=FY0;
end
BYKSRs=SRs.*BYK.*CLEAR;
SRs=SRs.*CLEAR;
SVyk=SVyk.*CLEAR;
BSH=BSH.*CLEAR;
EYK=EYK.*CLEAR;
FY0=FY0.*CLEAR;

Gy=cos(Cyk.*atan(BYKSRs-EYK.*(BYKSRs-atan(BYKSRs)))./cos(Cyk.*atan... 
(BSH-EYK.*(BSH-atan(BSH)))));
Fy=Gy.*FY0+SVyk;

% Graphs of fitted data for combined lateral force.
if combside==1
    for p=1:length(SAP)
        for n=1:length(IAP)
            for m=1:length(FZP)
                clear Fyi SRUi
                cn=(p-1)*length(IAP)+n;
                Fyi=zeros(VECTL(m,cn),1);
                for k=1:VECTL(m,cn)
Fyi(k)=Fy(m,cn,k);
SRUi(k)=SRU(m,cn,k);
end
figure(50+cn);
hold on;
plot(SRUi,Fyi,'r*');
figure(60+cn);
hold on;
plot(SRUi,Fyi./Fy0(m,cn),'r*');
end
end
end
end

%% Aligning Moment - Combined Slip - Coefficient Determination

% Plot original combined slip aligning moment data.
if combzmom==1
    for p=1:length(SAP)
        for n=1:length(IAP)
            for m=1:length(FZP)
                clear Fyi SRUi
                cn=(p-1)*length(IAP)+n;
                Mzi=zeros(VECTL(m,cn),1);
                for k=1:VECTL(m,cn)
                    Mzi(k)=MZU(m,cn,k);
                    SRUi(k)=SRU(m,cn,k);
                end
                figure(70+cn);
                hold on;
                plot(SRUi,Mzi,'b-');
            end
        end
    end
    plot(SRUi,Mzi,'b-');
end

% Compute Aligning Torque SAT and SAR
SHt=qHz1+qHz2.*dFzxa+(qHz3+qHz4.*dFzxa).*Gammaxa;
SAT=SAxa+SHt;
SHy=(pHy1+pHy2.*dFzxa)+pHy3.*Gammaxa;
SVy=Fzxa.*((pVy1+pVy2.*dFzxa)+(pVy3+pVy4.*dFzxa).*Gammaxa);
Ky=pKy1.*pFz0.*sin(2.*atan(Fzxa./(pKy2.*pFz0))).*(1-pKy3.*abs(Gammaxa));
SAr=SAxa+SHy+SVy./Ky;

% Compute KxsR and Kysa
% K=BCD
Ky=pKy1.*pFz0.*sin(2.*atan(Fzxa./(pKy2.*pFz0))).*(1-pKy3.*abs(Gammaxa));
Kx=Fzxa.*(pKx1+pKx2.*dFzxa).*exp(pKx3.*dFzxa);

% Compute Equivalent SAT and SAR
SAT=zeros(MATSIZE); SAR=zeros(MATSIZE); 
KY=zeros(MATSIZE); KX=zeros(MATSIZE);
for k=1:MATSIZE(3)
    SAT(:,:,k)=SAT;
    SAR(:,:,k)=SAR;
    KY(:,:,k)=KY;
    KX(:,:,k)=KX;
% Compute Fy prime
Fyp=Fy-SVyK;

% Compute Residual Torque
Dr=Fzxa.*(qDz6+qDz7.*dFzxa)+(qDz8+qDz9.*dFzxa).*Gammaxa).*pR0;
Dy=Fzxa.*(pDy1+pDy2.*dFzxa).*(1-pDy3.*Gammaxa.^2);
Ky=pKy1.*pFz0.*sin(2.*atan(Fzxa./(pKy2.*pFz0))).*(1-pKy3.*abs(Gammaxa));
Br=qBz9+qBz10.*(Ky./pCyl.*Dy).*pCyl;
Mzr=zeros(MATSIZE);
for k=1:MATSIZE(3)
    Mzr(:,:,k)=Dr.*cos(atan(Br.*SArEQ(:,:,k))).*cos(SAxa);
end

% Compute Pneumatic Trail t(SAtEQ)
if pDmztMod==0
    Dt=Fzxa.*(qDz1+qDz2.*dFzxa).*((1+qDz3.*Gammaxa+qDz4.*Gammaxa.^2).*pR0./pFz0;
elseif pDmztMod==1
    Dt=(qDz1m+qDz2m.*dFzxa).*((1+qDz3.*Gammaxa+qDz4.*Gammaxa.^2).*pR0./pFz0;
end
Ct=qCz1;
Bt=(qBz1+qBz2.*Fzxa+qBz3.*dFzxa.*^2).*((1+qBz4.*Gammaxa+qBz5.*abs(Gammaxa));
if qEz4==0 & qEz5==0
    Et=((qEz1+qEz2.*Fzxa+qEz3.*dFzxa.*^2);
tEQ=zeros(MATSIZE);
    for k=1:MATSIZE(3)
        tEQ(:,:,k)=Dt.*cos(Ct.*atan(Bt.*SAtEQ(:,:,k))-Et.*(Bt.*StAtEQ...((:,k)-atan(Bt.*StAtEQ(:,,:,k)))).*cos(SAxa);
    end
else
    for k=1:MATSIZE(3)
        Et((:,k)=(qEz1+qEz2.*Fzxa+qEz3.*dFzxa.*^2).*((1+(qEz4+qEz5.*...Gammaxa.^2).*2/pi).*atan(Bt.*Ct.*StAtEQ(:,:,k));
    end
    tEQ=zeros(MATSIZE);
    for k=1:MATSIZE(3)
        tEQ(:,:,k)=Dt.*cos(Ct.*atan(Bt.*SAtEQ(:,:,k)-Et(:,:,k).*...((Bt.*StAtEQ(:,:,k)-atan(Bt.*StAtEQ(:,:,k))).*cos(SAxa);
    end
end

% Compute Mzt prime
Mzp=-tEQ.*Fyp;

% Compute Pneumatic Scrub Radius sEQ
% Initial Coefficient Values
sSz1=-.01;
sSz2=-.001;
sSz3=.83;
sSz4=-.09;
clear TEST TESTc BEST
% Test variable and test delta values
TEST=[sSz1 sSz2 sSz3 sSz4];
DTEST=[.01 .01 .1 .01];
TESTc=TEST;
sEQ=zeros(MATSIZE);
for k=1:MATSIZE(3)
    sEQ(:,:,k)=pR0.*(TESTc(1)+TESTc(2).*(Fy(:,:,k)./pFz0)+(TESTc(3)+
        TESTc(4).*dFzxa).*Gammaxa);
end

Mz=Mzp+Mzr+sEQ.*Fx;
BEST=sum(sum(sum(abs(Mz-MZU),3)./VECTL));
disp(BEST);
disp(TEST);

clear sampsize k kthresh RESU SAMP
sampsize=100;
k=0;
kthresh=50;

% Curve fitment loop
while 1
    SAMP=randn(sampsize,length(TEST));
    SAMP=SAMP.*abs(SAMP);
    for m=1:sampsize
        clear Mzi sEQ
        TESTc=TEST+DTEST.*SAMP(m,:);
        for p=1:MATSIZE(3)
            sEQ(:,:,p)=pR0.*(TESTc(1)+TESTc(2).*(Fy(:,:,p)./pFz0)+
                (TESTc(3)+TESTc(4).*dFzxa).*Gammaxa);
        end
        Mzi=Mzp+Mzr+sEQ.*Fx;
        RESU(m,1)=sum(sum(sum(abs(Mzi-MZU),3)./VECTL));
    end
    k=k+1;
    [TBEST TInd]=min(RESU);
    if TBEST<BEST && k<kthresh
        BEST=TBEST;
        TEST=TEST+DTEST.*SAMP(TInd,:);
        k=0;
        disp(BEST);
        disp(TEST);
    elseif k>=kthresh
        break
    end
end

% Graph fitted data
if combzmom==1
    for p=1:MATSIZE(3)
        sEQ(:,:,p)=pR0.*(TESTc(1)+TESTc(2).*(Fy(:,:,p)./pFz0)+
            (TESTc(3)+TESTc(4).*dFzxa).*Gammaxa);
    end
    Mz=Mzp+Mzr+sEQ.*Fx;
    for p=1:length(SAP)
        for n=1:length(IAP)
            for m=1:length(FZP)
clear Mzi SRUi
cn=(p-1)*length(IAP)+n;
Mzi=zeros(VECTL(m,cn),1);
for k=1:VECTL(m,cn)
    Mzi(k)=Mz(m,cn,k);
    SRUi(k)=SRU(m,cn,k);
end
figure(70+cn);
hold on;
plot(SRUi,Mzi,'r-');
end
end

%% Overturning Moment - Pure and Combined Slip - Full Coefficient Determination
%% Overturning Moment Mx is simply defined via one equation and 3 coefficients. Scaling factors are not shown.
%% Mx = R0*Fz*(qSx1 - qSx2*Gamma + qSx3*Fy/Fz0)
%% Fitting will occur over the pure side slip data range thus dictating
%% Fy=Fy0. qSx1 - qSx2*Gamma will equal the value of Mx/(R0*Fz) at the
%% zero crossing point of Fy similar to the residual torque concept of the
%% aligning moment.

%% Inclination Angle Runs
IAP=[0 1 2 3 4];
%% Normal Load Runs
FZP=[222 445 667 1112 1557];
MATSIZE=[length(FZP) length(IAP)];
MAXL=800;

%% Initialize matrices
clear VECTL
Fzmx=zeros(MATSIZE);
dFzmx=zeros(MATSIZE);
Gammamx=zeros(MATSIZE);
SAU=zeros([MATSIZE MAXL]);
FYU=zeros([MATSIZE MAXL]);
MXU=zeros([MATSIZE MAXL]);
VECTL=zeros(MATSIZE);

%% Load in and preprocess data
for n=1:length(IAP)
    for m=1:length(FZP)
        clear FZN SAN MXN IAN FZ SA MX IA k p SAfit Sord SARad MXtrunc MXfit
        MXsmth Umat
        FZN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
             '.FZ' int2str(FZP(m)) '.FZ'];
        SAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
             '.FZ' int2str(FZP(m)) '.SA'];
        MXN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
             '.FZ' int2str(FZP(m)) '.MX'];
        IAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n))...
             '.FZ' int2str(FZP(m)) '.IA'];
        end
    end
end
eval(['FZ=' FZN ';']);

eval(['SA=' SAN ';']);

eval(['MX=' MXN ';']);

eval(['IA=' IAN ';']);

% Truncate data to one full sweep minimizing start and end effects
k=1;
bnd=[0;0];

for p=1:length(SA)
    if SA(p)>=0 && SA(p+1)<0 && k==1
        bnd(k,1)=p;
        k=k+1;
    elseif SA(p)>0 && SA(p+1)<=0 && k==2
        bnd(k,1)=p;
        k=k+1;
    elseif k==3
        break;
    end
end

% Compute dFz for the given run
% dFz = (Fz - pFz0)/pFz0
Fzmx(m,n)=mean(FZ(bnd(1,1):bnd(2,1)));
dFzmx(m,n)=(Fzmx(m,n)-pFz0)/pFz0;

% Record IA for given run
Gammamx(m,n)=(mean(IA(bnd(1,1):bnd(2,1)))*pi/180);

% Smooth data to allow fitting
% Fit curve to FY data to eliminate hysteresis.
SmoothSpan=25;
[SArad,Sord]=sort(SA(bnd(1,1):bnd(2,1)));
[SArad,Sord]=sort(SA(bnd(1,1):bnd(2,1)));
MXtrunc=(MX(bnd(1,1):bnd(2,1)));
MXfit=MXtrunc(Sord);
MXsmth=smooth(MXfit,SmoothSpan,'lowess');

[SArad,Umat,junk]=unique(SArad);
MX1=MXsmth(Umat);

if combxmom==1
    figure(60+n);
    hold on;
    plot(SA1,MX1,'b-');
end

VECTL(m,n)=length(SA1);

SHy=(pHy1+pHy2.*dFzmx(m,n))+(pHy3.*Gammamx(m,n));

SVy=Fzmx(m,n).*((pVy1+pVy2.*dFzmx(m,n))+(pVy3+pVy4.*dFzmx... 
(m,n).*Gammamx(m,n)));

Ky=pKy1.*pFz0.*sin(2.*atan(Fzmx(m,n)./(pKy2.*pFz0))).*... 
(1-pKy3.*abs(Gammamx(m,n))));

Dy=Fzmx(m,n).*((p Dy1+p Dy2.*dFzmx(m,n)).*(1-pDy3.*Gammamx(m,n).^2));

Cy=pCy1;
By = Ky. / (Cy.*Dy);
SAY = SA1 + SHy;
FY1 = Dy.*sin(Cy.*atan(By.*SAY - ((pEy1 + pEy2.*dFzmx(m,n)).*(1 - ...
  (pEy3 + pEy4.*Gammamx(m,n)).*(abs(SAY) / SAY))).*(By.*SAY -
  (By.*SAY)))) + SVy;
for k = 1:VECTL(m,n)
  SAU(m,n,k) = SA1(k);
  MXU(m,n,k) = MX1(k);
  FYU(m,n,k) = FY1(k);
end
end

% Initial Coefficient Values
qSx1 = 0.02;
qSx2 = 1;
qSx3 = 0.02;
clear TEST TESTc DTEST

% Test Matrix and Delta Values
TEST = [qSx1 qSx2 qSx3];
TESTc = TEST;
DTEST = [0.1 0.1 0.1];
clear CLEAR

% Create Clearing Matrix
CLEAR = zeros([MATSIZE MAXL]);
for m = 1:MATSIZEx(1)
  for n = 1:MATSIZEx(2)
    CLEAR(m,n,1:VECTL(m,n)) = 1;
  end
end

Fz1 = zeros([MATSIZE MAXL]);
Gammal = zeros([MATSIZE MAXL]);
for k = 1:MAXL
  Fz1(1:k,1) = Fzmx(:,k);
  Gammal1(1:k,1) = Gammamx(:,k);
end
Fz1 = Fz1.*CLEAR;
Gammal = Gammal1.*CLEAR;

Mxi = pR0.*Fz1.*(TESTc(1) - TESTc(2).*Gammal + TESTc(3).*((FYU / pFz0));
BEST = sum(sum(abs(Mxi - MXU), 3) / VECTL);
disp(BEST);
disp(TESTc);
clear sampsize k kthresh
sampsize = 200;
k = 0;
kthresh = 100;
clear RESU SAMP
% Curve fitting loop for overturning moment
while 1
    SAMP=randn(sampsize,length(TEST));
    SAMP=SAMP.*abs(SAMP);
    for m=1:sampsize
        clear Mxi
        TESTc=TEST+DTEST.*SAMP(m,:);
        Mxi=pR0.*Fz1.*(TESTc(1)-TESTc(2).*Gamma1+TESTc(3).*(FYU./pFz0));
        RESU(m,1)=sum(sum((sum(abs(Mxi-MXU),3)./VECTL)));
    end
    k=k+1;
    [TBEST TInd]=min(RESU);
    if TBEST<BEST && k<kthresh
        BEST=TBEST;
        TEST=TEST+DTEST.*SAMP(TInd,:);
        k=0;
        disp(BEST);
        disp(TEST);
    elseif k>=kthresh
        break
    end
end

qSx1=TEST(1);
qSx2=TEST(2);
qSx3=TEST(3);

clear SA1 FY1

% Plot fitted overturning moment data.
if combxmom==1
    for n=1:length(IAP)
        for m=1:length(FZP)
            clear SA1 FY1 MX1
            SA1=zeros(VECTL(m,n));
            FY1=zeros(VECTL(m,n));
            for k=1:VECTL(m,n)
                SA1(k)=SAU(m,n,k);
                FY1(k)=FYU(m,n,k);
            end
            MX1=pR0.*Fzmx(m,n).*((qSx1-qSx2.*Gammamx(m,n)+qSx3.*(FY1./pFz0));
            figure(60+n);
            hold on
            plot(SA1,MX1,'r-');
        end
    end
end

% Store and Save Coefficients to Mat file
% Create a data structure to store pertinent information and the variables
% themselves in a simple fashion.

% Variable Structure Name and File Name to be written to.
if pDmztMod==0
    wVariable=[pFileName '_Pac2002_' pressure];
elseif pDmztMod==1
wVariable=[pFileName '_Pac2002mod_' pressure];
wFileName=[pFileName '_Pac2002mod'];
end

% Date and Script Information
DateRun=datestr(now);
eval([wVariable '.DateRun=DateRun;']);
ScriptVersion='PacejkaFit_20120326';
eval([wVariable '.ScriptVersion=ScriptVersion;']);

% Fitting Method Utilized, General Measurement and Coordinate System Info
FitMethod='ADAMS/PAC2002 Modified Pacejka Equations';
eval([wVariable '.FitMethod=FitMethod;']);
FitScript='Pacejka Fitting Script by Fred Jabs - 03/26/2012';
eval([wVariable '.FitScript=FitScript;']);
BaseLength='Meter'; eval([wVariable '.BaseLength=BaseLength;']);
BaseForce='Newton'; eval([wVariable '.BaseForce=BaseForce;']);
BaseAngle='Radian'; eval([wVariable '.BaseAngle=BaseAngle;']);
BaseSR='Unitless'; eval([wVariable '.BaseSR=BaseSR;']);
CoordinateSystem='ISO - W-Axis System';
eval([wVariable '.CoordinateSystem=CoordinateSystem;']);
NominalRoadSpeed='25 MPH';
eval([wVariable '.NominalRoadSpeed=NominalRoadSpeed;']);
NominalSlipRate='4 Degrees/Second';
eval([wVariable '.NominalSlipRate=NominalSlipRate;']);

% Tire Identification and TTC Information
eval([wVariable '.TireID=tireid;']);
eval([wVariable '.TirePressure=pressure;']);
TTCRound='Round 4 - 10/2009'; eval([wVariable '.TTCRound=TTCRound;']);
TTCWarning='Consortium members are free to use this data in the design and construction of their FSAE entries, other school projects and related academic activities. Any publication or presentation of the tire data must acknowledge Calspan and the FSAE TTC. Individuals and teams are prohibited from donating or selling the data to any other individual, group, team or university, or posting it on the internet. [...] The data may not be used in any commercial application.';
eval([wVariable '.TTCWarning=TTCWarning;']);
TTCWebsite='www.millikenresearch.com/fsaettc.html';
eval([wVariable '.TTCWebsite=TTCWebsite;']);
TTCForum='http://sae.wsu.edu/ttc/';
eval([wVariable '.TTCForum=TTCForum;']);

% Basic Tire Properties
eval([wVariable '.Fz0=pFz0;']);
eval([wVariable '.R0=pR0;']);

% Longitudinal Force - Pure Slip Coefficients
eval([wVariable '.pCx1=pCx1;']);
eval([wVariable '.pDx1=pDx1;']);
eval([wVariable '.pDx2=pDx2;']);
eval([wVariable '.pDx3=pDx3;']);
eval([wVariable '.pEx1=pEx1;']);
eval([wVariable '.pEx2=pEx2;']);
eval([wVariable '.pEx3=pEx3;']);
eval([wVariable '.pEx4=pEx4;']);
eval([wVariable '.pKx1=pKx1;']);
eval([wVariable '.pKx2=pKx2;']);
eval([wVariable '.pKx3=pKx3;']);
eval([wVariable '.pHx1=pHx1;']);
eval([wVariable '.pHx2=pHx2;']);
eval([wVariable '.pVx1=pVx1;']);
eval([wVariable '.pVx2=pVx2;']);
%
Longitudinal Force - Pure Slip Raw Coefficients
eval([wVariable '.RAWFX0Fit=pRAWFX0Fit;']);
%
Longitudinal Force - Combined Slip Coefficients
eval([wVariable '.rBx1=rBx1;']);
eval([wVariable '.rBx2=rBx2;']);
eval([wVariable '.rCx1=rCx1;']);
eval([wVariable '.rEx1=rEx1;']);
eval([wVariable '.rEx2=rEx2;']);
eval([wVariable '.rHx1=rHx1;']);
%
Overturing Moment Coefficients
eval([wVariable '.qSx1=qSx1;']);
eval([wVariable '.qSx2=qSx2;']);
eval([wVariable '.qSx3=qSx3;']);
%
Lateral Force - Pure Slip Coefficients
eval([wVariable '.pCy1=pCy1;']);
eval([wVariable '.pDy1=pDy1;']);
eval([wVariable '.pDy2=pDy2;']);
eval([wVariable '.pDy3=pDy3;']);
eval([wVariable '.pEy1=pEy1;']);
eval([wVariable '.pEy2=pEy2;']);
eval([wVariable '.pEy3=pEy3;']);
eval([wVariable '.pEy4=pEy4;']);
eval([wVariable '.pKy1=pKy1;']);
eval([wVariable '.pKy2=pKy2;']);
eval([wVariable '.pKy3=pKy3;']);
eval([wVariable '.pHy1=pHy1;']);
eval([wVariable '.pHy2=pHy2;']);
eval([wVariable '.pHy3=pHy3;']);
eval([wVariable '.pVy1=pVy1;']);
eval([wVariable '.pVy2=pVy2;']);
eval([wVariable '.pVy3=pVy3;']);
eval([wVariable '.pVy4=pVy4;']);
%
Lateral Force - Pure Slip Raw Coefficients
eval([wVariable '.RAWFY0Fit=pRAWFY0Fit;']);
%
Lateral Force - Combined Slip Coefficients
eval([wVariable '.rBy1=rBy1;']);
eval([wVariable '.rBy2=rBy2;']);
eval([wVariable '.rCy1=rCy1;']);
eval([wVariable '.rEy1=rEy1;']);
eval([wVariable '.rEy2=rEy2;']);
eval([wVariable '.rHy1=rHy1;']);
eval([wVariable '.rHy2=rHy2;']);
eval([wVariable '.rVy1=rVY1;']);
eval([wVariable '.rVy2=rVY2;']);
eval([wVariable '.rVy3=rVY3;']);
eval([wVariable '.rVy4=rVY4;']);
eval([wVariable '.rVy5=rVY5;']);
eval([wVariable '.rVy6=rVY6;']);

% Aligning Moment - Pure Slip Coefficients
eval([wVariable '.qBz1=qBz1;']);
eval([wVariable '.qBz2=qBz2;']);
eval([wVariable '.qBz3=qBz3;']);
eval([wVariable '.qBz4=qBz4;']);
eval([wVariable '.qBz5=qBz5;']);
eval([wVariable '.qBz9=qBz9;']);
eval([wVariable '.qBz10=qBz10;']);
eval([wVariable '.qCz1=qCz1;']);
if pDmztMod==0
  eval([wVariable '.qDz1=qDz1;']);
  eval([wVariable '.qDz2=qDz2;']);
elseif pDmztMod==1
  eval([wVariable '.qDz1=qDz1m;']);
  eval([wVariable '.qDz2=qDz2m;']);
end
eval([wVariable '.qDz3=qDz3;']);
eval([wVariable '.qDz4=qDz4;']);
eval([wVariable '.qDz6=qDz6;']);
eval([wVariable '.qDz7=qDz7;']);
eval([wVariable '.qDz8=qDz8;']);
eval([wVariable '.qDz9=qDz9;']);
eval([wVariable '.qEz1=qEz1;']);
eval([wVariable '.qEz2=qEz2;']);
eval([wVariable '.qEz3=qEz3;']);
eval([wVariable '.qEz4=qEz4;']);
eval([wVariable '.qEz5=qEz5;']);
eval([wVariable '.qHz1=qHz1;']);
eval([wVariable '.qHz2=qHz2;']);
eval([wVariable '.qHz3=qHz3;']);
eval([wVariable '.qHz4=qHz4;']);

% Aligning Moment - Pure Side Slip Raw Coefficients
eval([wVariable '.RAWMZ0Fit=pRAWMZ0Fit;']);

% Aligning Moment - Combined Slip Coefficients
eval([wVariable '.ssZ1=ssZ1;']);
eval([wVariable '.ssZ2=ssZ2;']);
eval([wVariable '.ssZ3=ssZ3;']);
eval([wVariable '.ssZ4=ssZ4;']);

% Save Coefficient Data to Mat File
if writefile==1
  K=dir;
  Kname={K(:).name};
  OutputName=[wFileName '.mat'];
  FileExist=max(strcmp(OutputName,Kname));
if FileExist==1
    eval(['save ' wFileName '.mat ' wVariable ' -append']);
elseif FileExist==0
    eval(['save ' wFileName '.mat ' wVariable]);
end
end

% Close out script
disp('So long, and thanks for all the fish!');
% clear all

D.3. PAC2002 Scaling Structure Script

% Script to set up standard scaling mat files for use with pac2002
% function. Experiment with the scaling factors to understand their effect
% on the data.

% See 'Tire and Vehicle Dynamics' by Pacejka (2002), pg 186-190 for further
% further explanation of the scaling parameters and their effect on the
% 'Magic Formula' equation set.

clear all;

% Enter the name that the variable and mat file names will derive from
TireName='Hoosier20x6_13_6inch';

% Longitudinal Pure Force Scaling Factors
SCALE.Gammax=1;
SCALE.Cx=1.3;
SCALE.Mewx=.7;
SCALE.Ex=1;
SCALE.KxKappa=1;
SCALE.Hx=1;
SCALE.Vx=1;
%
% Lateral Pure Force Scaling Factors
SCALE.Gammay=1;
SCALE.Cy=1.3;
SCALE.Mewy=.7;
SCALE.Ey=1;
SCALE.Fz0=1;
SCALE.Ky=1;
SCALE.Hy=1;
SCALE.Vy=1;
%
% Aligning Moment Scaling Factors
SCALE.Gammaz=1;
SCALE.t=1;
SCALE.r=1;
%
% Overturning Moment Scaling Factors
SCALE.Mx=1;
SCALE.VMx=1;
%
% Longitudinal Combined Force Scaling Factors
SCALE.xSA=1.1;
%
% Lateral Combined Force Scaling Factors
SCALE.ySR=1.1;
SCALE.VySR=1;
% Aligning Moment Combined Scaling Factors
SCALE.s=1;
% Degressive Friction Factor
SCALE.AMew=10;

VariableName=[TireName '_SCALE'];
SaveName=[VariableName '.mat'];

eval(['VariableName '.Gammax=SCALE.Gammax;']);
eval(['VariableName '.Cx=SCALE.Cx;']);
eval(['VariableName '.Mewx=SCALE.Mewx;']);
eval(['VariableName '.Ex=SCALE.Ex;']);
eval(['VariableName '.KxKappa=SCALE.KxKappa;']);
eval(['VariableName '.Hx=SCALE.Hx;']);
eval(['VariableName '.Vy=SCALE.Vy;']);
eval(['VariableName '.Vx=SCALE.Vx;']);
eval(['VariableName '.Gammay=SCALE.Gammay;']);
eval(['VariableName '.Cy=SCALE.Cy;']);
eval(['VariableName '.Mewy=SCALE.Mewy;']);
eval(['VariableName '.Ey=SCALE.Ey;']);
eval(['VariableName '.Fx0=SCALE.Fx0;']);
eval(['VariableName '.Ky=SCALE.Ky;']);
eval(['VariableName '.Hy=SCALE.Hy;']);
eval(['VariableName '.Vy=SCALE.Vy;']);
eval(['VariableName '.Gammaz=SCALE.Gammaz;']);
eval(['VariableName '.t=SCALE.t;']);
eval(['VariableName '.r=SCALE.r;']);
eval(['VariableName '.Mx=SCALE.Mx;']);
eval(['VariableName '.VMx=SCALE.VMx;']);
eval(['VariableName '.xSA=SCALE.xSA;']);
eval(['VariableName '.ySR=SCALE.ySR;']);
eval(['VariableName '.VySR=SCALE.VySR;']);
eval(['VariableName '.s=SCALE.s;']);
eval(['VariableName '.AMew=SCALE.AMew;']);

eval(['save ' SaveName ' ' VariableName '']);
clear all;

D.4. PAC2002 Expansion Function

function [FX, FY, MX, MY, MZ, Fx0, Fy0, Mz0] = pac2002(COEFF, SCALE, SA, SR, FZ, IA)
% PAC2002 Pacejka 2002 ADAMS Model Expansion
% [FX, FY, MX, MY, MZ, Fx0, Fy0, Mz0] = pac2002(COEFF, SCALE, SA, SR, FZ, IA)
% Expansion routine for PAC2002 Magic-Formula by Pacejka. The coefficients
% are generated utilizing the ADAMS/Tire version of the equations. As such
% the outputs and inputs must follow the ISO tire coordinate system in the
% wheel axis system.
% The coefficient file utilized is a data structure generated by Fred
% Jabs's Pacejka fitting script or other defining procedure.
%
The scaling structure defines the scaling factors utilized. An argument of 'unity' will apply a scaling structure equal to one.

The units of the supplied arguments of the tire's running state are respectively: Radians, Dimensionless (%), Newtons and Radians. The units of the outputs are Newtons and Newton*Meters.

The supplied arguments SA and SR must be equal in size; each set corresponding to one running condition of the tire. They may be singular or a vector of length m.

FZ and IA may be singular or a vector. In the case of a vector of length n and p respectively the returned results will be of dimension n*p*m; a full test array of all combinations of FZ and IA will be calculated.

Alternate fitting methodology for D_t for the trail function in the pure side slip aligning moment definition. The alternate method eliminates a power of FZ to achieve a better fit of the data.

Defining the Scaling Matrix

In the case that 'unity' is supplied:

```matlab
if strcmp(SCALE,'unity')
    clear SCALE
    % Longitudinal Pure Force Scaling Factors
    SCALE.Gammax=1;
    SCALE.Cx=1;
    SCALE.Mewx=1;
    SCALE.Ex=1;
    SCALE.KxKappa=1;
    SCALE.Hx=1;
    SCALE.Vx=1;
    % Lateral Pure Force Scaling Factors
    SCALE.Gammay=1;
    SCALE.Cy=1;
    SCALE.Mewy=1;
    SCALE.Ey=1;
    SCALE.Fz0=1;
    SCALE.Ky=1;
    SCALE.Hy=1;
    SCALE.Vy=1;
    % Aligning Moment Scaling Factors
    SCALE.Gammaz=1;
    SCALE.t=1;
    SCALE.r=1;
    % Overturning Moment Scaling Factors
    SCALE.Mx=1;
    SCALE.VMx=1;
    % Longitudinal Combined Force Scaling Factors
    SCALE.xSA=1;
    % Lateral Combined Force Scaling Factors
    SCALE.ySR=1;
    SCALE.VySR=1;
    % Aligning Moment Combined Scaling Factors
    SCALE.s=1;
    % Degressive Friction Factor
    SCALE.AMew=10;
```

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end

SCALE.MewyPrime=SCALE.AMew*SCALE.Mewy/(1+(SCALE.AMew-1)*SCALE.Mewy);
SCALE.MewxPrime=SCALE.AMew*SCALE.Mewx/(1+(SCALE.AMew-1)*SCALE.Mewx);

%% Defining Matrices for Computation
if size(SA)~=size(SR)
disp('SA and SR differ in size');
    return;
elseif length(FZ)~=1 || length(IA)~=1
    MATSIZE=[length(FZ) length(IA) length(SA)];
    FZm=zeros(MATSIZE); IAm=zeros(MATSIZE);
    SAm=zeros(MATSIZE); SRm=zeros(MATSIZE);
    for K=1:MATSIZE(1)
        for L=1:MATSIZE(2)
            SAm(K,L,:)=SA; SRm(K,L,:)=SR;
            IAm(K,L,:)=IA(L); FZm(K,L,:)=FZ(K);
        end
    end
elseif length(FZ)==1 && length(IA)==1 && max(size(SR))>1
    SAm=SA; SRm=SR;
    MATSIZE=size(SR);
    IAm=ones(MATSIZE).*IA;
    FZm=ones(MATSIZE).*FZ;
else
    disp('FZ or IA appear to be in error. Check function inputs.');
    return
end

%% Setup Calculations
Fz0=COEFF.Fz0.*SCALE.Fz0;
dFz=(FZm-Fz0)./Fz0;

%% Lateral Force (Pure Side Slip) Computation
IAFy=IAm.*SCALE.Gammay;
Cy=COEFF.pCy1.*SCALE.Cy;
Dy=FZm.*(COEFF.pDy1+COEFF.pDy2.*dFz).*(1-COEFF.pDy3.*IAFy.^2).*SCALE.Mewy;
Ky=COEFF.pKy1.*Fz0.*sin(2.*atan(FZm./(COEFF.pKy2.*Fz0))).*(1-COEFF.pKy3.*abs(IAFy)).*SCALE.Ky;
By=Ky./(Cy.*Dy);
SHy=(COEFF.phy1+COEFF.phy2.*dFz).*SCALE.Hy+COEFF.phy3.*IAFy;
SVy=FZm.*(COEFF.pVy1+COEFF.pVy2.*dFz).*SCALE.Vy+...
    (COEFF.pVy3+COEFF.pVy4.*dFz).*IAFy).*SCALE.MewyPrime;
SAFy=SAm+SHy;
Ey=(COEFF.pey1+COEFF.pey2.*dFz).*(1-(COEFF.pey3+...
    COEFF.pey4.*IAFy).*SAFy./abs(SAFy)).*SCALE.Ey;
Fy0=Dy.*sin(Cy.*atan(By.*SAFy-Ey.*(By.*SAFy-atan(By.*SAFy))))+SVy;

%% Longitudinal Force (Pure Longitudinal Slip) Computation
IAFx=IAm.*SCALE.Gammax;
Cx=COEFF.pCx1.*SCALE.Cx;
Dx=FZm.*(COEFF.pDx1+COEFF.pDx2.*dFz).*(1-COEFF.pDx3.*IAFx.^2).*SCALE.Mewx;
Kx=FZm.*(COEFF.pkx1+COEFF.pkx2.*dFz).*exp(COEFF.pkx3.*dFz).*SCALE.KxKappa;
Bx=Kx./(Cx.*Dx);
SHx = (COEFF.pHx1 + COEFF.pHx2 .* dFz) .* SCALE.Hx;
SVx = Fz.* (COEFF.pVx1 + COEFF.pVx2 .* dFz) .* SCALE.Vx .* SCALE.MewxPrime;
SRFx = SRm + SHx;
Ex = (COEFF.pEx1 + COEFF.pEx2 .* dFz + COEFF.pEx3 .* dFz.^2) .* (1 - COEFF.pEx4.* ...
SRFx ./ abs(SRFx)). * SCALE.Ex;
Fx0 = Dx.* sin(Cx.* atan(Bx.* SRFx - Ex.* (Bx.* SRFx - atan(Bx.* SRFx))) ) + SVx;

%% Aligning Moment (Pure Side Slip) Computation
IAMz = IAm .* SCALE.Gammaz;
Bmzt = (COEFF.qBz1 + COEFF.qBz2 .* dFz + COEFF.qBz3 .* dFz.^2) .* (1 + COEFF.qBz4.* ...
IAMz + COEFF.qBz5 .* abs(IAMz)) .* SCALE.Ky ./ SCALE.Mewy;

if isfield(COEFF,'qDz1') && isfield(COEFF,'qDz2')
    DMzt = Fz.* (COEFF.qDz1 + COEFF.qDz2 .* dFz).* (1 + COEFF.qDz3 .* IAMz + ...
    COEFF.qDz4 .* IAMz.^2) .* COEFF.R0 ./ COEFF.Fz0 .* SCALE.t;
elseif isfield(COEFF,'qDz1m') && isfield(COEFF,'qDz2m')
    DMzt = (COEFF.qDz1m + COEFF.qDz2m .* dFz).* (1 + COEFF.qDz3 .* IAMz + ...
    COEFF.qDz4 .* IAMz.^2) .* COEFF.R0 ./ COEFF.Fz0 .* SCALE.t;
end
SHmzt = COEFF.qHz1 + COEFF.qHz2 .* dFz + (COEFF.qHz3 + COEFF.qHz4 .* dFz) .* IAMz;
SAMzt = SAM + SHmzt;
Emzt = (COEFF.qEz1 + COEFF.qEz2 .* dFz + COEFF.qEz3 .* dFz.^2).* ...
(1 + (COEFF.qEz4 + COEFF.qEz5 .* IAMz) .* (2/pi) .* atan(Bmzt .* CMzt .* SAMzt));
SHF = SHy + SVy./ Ky;
SAMzr = SAM + SHF;

%% Lateral Force (Combined Slip) Computation
Byk = COEFF.rBy1 .* cos(atan(COEFF.rBy2 .* (SAM - COEFF.rBy3))) .* SCALE.ySR;
Cyk = COEFF.rCy1;
Eyk = COEFF.rEy1 + COEFF.rEy2 .* dFz;
SHyk = COEFF.rHy1 + COEFF.rHy2 .* dFz;
DVyk = Dy.* (COEFF.rVy1 + COEFF.rVy2 .* dFz + COEFF.rVy3 .* IAM).* ...
    cos(atan(COFF.rVy4).* SAM));
SVyk = DVyk .* sin(atan(COFF.rVy5 .* atan(COFF.rVy6 .* SRm))).* SCALE.VySR;
SRs = SRm + SHyk;
GyK = cos(Cyk.* atan(Byk.* SRs-Eyk.* (Byk.* SRs-atan(Byk.* SRs))));/...
    cos(Cyk.* atan(Byk.* SHyk-Eyk.* (Byk.* SHyk-atan(Byk.* SHyk))));
Fy = Fy0.* Gyk + SVyk;

%% Longitudinal Force (Combined Slip) Computation
Bxa = COEFF.rBx1 .* cos(atan(COFF.rBx2 .* SRm)) .* SCALE.xSA;
Cxa = COEFF.rCx1;
Exa = COEFF.rEx1 + COEFF.rEx2 .* dFz;
SHxa = COEFF.rHx1;
SAM = SAM + SHxa;
Gxa = cos(Cxa.* atan(Bxa.* SAM-Exa.* (Bxa.* SAM-atan(Bxa.* SAM))));/...
    cos(Cxa.* atan(Bxa.* SHxa-Exa.* (Bxa.* SHxa-atan(Bxa.* SHxa))));
Fx = Fx0.* Gxa;
%% Aligning Moment (Combined Slip) Computation
SAteq=(SAmzt.^2+(Kx./Ky).^2.*SRm.^2).^.5.*SAmzt./abs(SAmzt);
SAreq=(SAmzr.^2+(Kx./Ky).^2.*SRm.^2).^.5.*SAmzr./abs(SAmzr);
Mzreq=Dmzr.*cos(atan(Bmzr.*SAreq)).*cos(SAm);
Mzteq=-Dmzt.*cos(Cmzt.*atan(Bmzt.*SAteq-
(Emzt.*
(Bmzt.*SAteq-
atan(Bmzt.*SAteq))))).*cos(SAm).*((Fy-SVyk);
Mzseq=(COEFF.R0.*((COEFF.sSz1+COEFF.sSz2.*
(Fy./Fz0)+(COEFF.sSz3+COEFF.sSz4.*dFz).*IAm).*SCALE.s).*Fx;
Mz=Mzreq+Mzteq+Mzseq;

%% Overturning Moment (Combined Slip) Computation
Mx=COEFF.R0.*FZm.*(COEFF.qSx1.*SCALE.VMx-COEFF.qSx2.*
IAm+COEFF.qSx3.*(Fy./Fz0)).*SCALE.Mx;

%% Rolling Resistance Computation
% Simple Rolling Resistance, Normal Load Sensitive, Assumed Constant .01
% My = R0*Fz*My-Coeff (.01)
% My should be negative in the ISO system.
My=COEFF.R0.*FZm.*-.01;

%% Finish function
FX=Fx; FY=Fy; MZ=Mz; MX=Mx; MY=My;

D.5. PAC2002 Expansion Examples

D.5.1. Combined Tire Force Plotting

% Demonstration of utilization of the pac2002 function to generate data for
% tire force and moment graphs.
% Fitted data from Round 4 as well as the fitted variables entered in mat
% files from the Stackpole Engineering Services are listed below.
clear all
close all

% Slip Angle Range in Degrees, Fitted over -12:12
SASpacing=1; % Slip Angle spacing in degrees
SA=-12:SASpacing:12;

% Slip Ratio Range in %, Fitted over -.20:.15
SRSpacing=.01; % Spacing of slip ratios
SR=-.12:SRSpacing:.12;

% FZ in Newtons, Fitted over 222:1557
FZ=[200 900 1600];

% IA in Degrees, Fitted over 0:4
IAP=[0 2 4];

% Axis extent
AxisExtent=3;
XYAxis=[-AxisExtent AxisExtent -AxisExtent AxisExtent 0 2.5];
AxisSetup='axis(XYAxis);'
% Grid setup
GridSetup='grid on;';

% Unity scaling? 1 or 0
Unity=1;

% Scaling Structure Name
ScaleName='Hoosier20x6_13_6inch_SCALE';

% Select Inflation Pressure {Pa55 Pa69 Pa83 Pa97 Pa83COLD}
InflationPressure='Pa83';

% Modified aligning moment (MZ) fitment?
ModMZ=0;

% Uncomment wanted tire file, and variable declarations.
% Round 4 Data
TireName='GoodyearD2696_7inch';
TireName='GoodyearD2696_8inch';
TireName='Hoosier20x6_13_6inch';
TireName='Hoosier20x6_13_7inch';
TireName='Hoosier20x7_13_6inch';
TireName='Hoosier20x7_13_7inch';
TireName='Hoosier20x75_13_6inch';
TireName='Hoosier20x75_13_7inch';
TireName='Dunlop_7inch';
TireName='Dunlop_8inch';
TireName='Michelin_7inch';
TireName='Michelin_8inch';

if ModMZ
    OutputName=[TireName '_Pac2002mod'];
else
    OutputName=[TireName '_Pac2002'];
end

eval(['load ' OutputName '.mat;']);
eval(['GraphVar=' OutputName '_' InflationPressure ';']);

% Round 3 Data - Stackpole Engineering Services - Here for reference

% load SESPAC_Goodyear20x70_13_D2692_6inch_Pac2002_Pa83.mat;
% GraphVar=Goodyear20x70_13_D2692_6inch_Pac2002;
% OutputName='Goodyear20x70_13_D2692_6inch_Pac2002';
% load SESPAC_Goodyear20x70_13_D2692_7inch_Pac2002_Pa83.mat;
% GraphVar=Goodyear20x70_13_D2692_7inch_Pac2002;
% OutputName='Goodyear20x70_13_D2692_7inch_Pac2002';
% load SESPAC_Goodyear20x70_13_D2692_8inch_Pac2002_Pa83.mat;
% GraphVar=Goodyear20x70_13_D2692_8inch_Pac2002;
% OutputName='Goodyear20x70_13_D2692_8inch_Pac2002';
% load SESPAC_Hoosier20x75_13_43161_7inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier20x75_13_43161_7inch_Pac2002;
% OutputName='Hoosier20x75_13_43161_7inch_Pac2002';
% load SESPAC_Hoosier20x75_13_43161_8inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier20x75_13_43161_8inch_Pac2002;
% OutputName='Hoosier20x75_13_43161_8inch_Pac2002';
% load SESPAC_Hoosier205x70_13_43129_6inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier205x70_13_43129_6inch_Pac2002;
OutputName='Hoosier205x70_13_43129_6inch_Pac2002';
% load SESPAC_Hoosier205x70_13_43129_7inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier205x70_13_43129_7inch_Pac2002;
OutputName='Hoosier205x70_13_43129_7inch_Pac2002';
% load SESPAC_Hoosier205x70_13_43129_8inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier205x70_13_43129_8inch_Pac2002;
OutputName='Hoosier205x70_13_43129_8inch_Pac2002';
% load SESPAC_Michelin16_25_13_F146045_7inch_Pac2002_Pa83.mat;
% GraphVar=Michelin16_25_13_F146045_7inch_Pac2002;
OutputName='Michelin16_25_13_F146045_7inch_Pac2002';
% load SESPAC_Michelin16_25_13_F146045_8inch_Pac2002_Pa83.mat;
% GraphVar=Michelin16_25_13_F146045_8inch_Pac2002;
OutputName='Michelin16_25_13_F146045_8inch_Pac2002';
% Round 2 Data - Stackpole Engineering Services - Reference Only
% load SESPAC_Avon20x62_13_FITO9241_6inch_Pac2002_Pa83.mat;
% GraphVar=Avon20x62_13_FITO9241_6inch_Pac2002;
% load SESPAC_Avon20x72_13_HDTO9760_6inch_Pac2002_Pa83.mat;
% GraphVar=Avon20x72_13_HDTO9760_6inch_Pac2002;
% load SESPAC_Goodyear18x65_10_D1383_6inch_Pac2002_Pa83.mat;
% GraphVar=Goodyear18x65_10_D1383_6inch_Pac2002;
% load SESPAC_Hoosier205x6_13_43128_6inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier205x6_13_43128_6inch_Pac2002;
% load SESPAC_Hoosier205x70_13_43129_6inchR2_Pac2002_Pa83.mat;
% GraphVar=Hoosier205x70_13_43129_6inchR2;
% Round 1 Data - Stackpole Engineering Services - Reference Only
% load SESPAC_Goodyear20x7_13_D1385_6inch_Pac2002_Pa83.mat;
% GraphVar=Goodyear20x7_13_D1385_6inch_Pac2002;
OutputName='Goodyear20x7_13_D1385_6inch_Pac2002';
% load SESPAC_Goodyear20x7_13_D2509_6inch_Pac2002_Pa83.mat;
% GraphVar=Goodyear20x7_13_D2509_6inch_Pac2002;
OutputName='Goodyear20x7_13_D2509_6inch_Pac2002';
% load SESPAC_Hoosier18x6_10_43101_6inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier18x6_10_43101_6inch_Pac2002;
OutputName='Hoosier18x6_10_43101_6inch_Pac2002';
% load SESPAC_Hoosier20x6_13_43131_6inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier20x6_13_43131_6inch_Pac2002;
OutputName='Hoosier20x6_13_43131_6inch_Pac2002';
% load SESPAC_Hoosier20x7_13_43157_6inch_Pac2002_Pa83.mat;
% GraphVar=Hoosier20x7_13_43157_6inch_Pac2002;
OutputName='Hoosier20x7_13_43157_6inch_Pac2002';
if Unity
  ScaleVar='unity';
else
  % Non-unity scaling file
  eval(['load ' ScaleName '.mat;']);
  eval(['ScaleVar=' ScaleName '']);
end
% Radian conversion
SA=SA*pi/180;
IA=IAP*pi/180;

MATSIZE=[length(SR) length(SA)];
SRM=zeros(MATSIZE);
SAM=zeros(MATSIZE);

for k=1:MATSIZE(1)
    SAM(k,:)=SA;
end

for k=1:MATSIZE(2)
    SRM(:,k)=SR;
end

[non Xind]=min(abs(SR));
[non Yind]=min(abs(SA));

for p=1:length(IA)
    for q=1:length(FZ)
        [FX FY MX MY MZ Fx0 Fy0 Mz0]=
        pac2002(GraphVar,'unity',SAM,SRM,FZ(q),IA(p));

        n=1;
        GraphTitle=strcat('-','GraphVar.TirePressure','-','...
                       GraphVar.TireID,'-IA',int2str(IAP(p)),'-FZ',int2str(FZ(q)));

        % IA's arranged vertically
        % SubCallOut=strcat('subplot(',int2str(length(IAP)),',',int2str...
        % (length(FZ)),',',int2str(length(FZ)*(p-1)+q),');');

        % IA's arranged horizontally
        SubCallOut=strcat('subplot(',int2str(length(FZ)),',',int2str...
                        (length(IAP)),',',int2str(length(IAP)*(q-1)+p),');');

        % Longitudinal Force Mesh
        figure(n); eval(SubCallOut); n=n+1; hold on;
title(['Fx' GraphTitle]); xlabel('Slip Angle (deg)');
ylabel('Slip Ratio (%)'); zlabel('Force (Newtons)'); colorbar;
mesh(SAM*180/pi,SRM*100,FX);
%

% Normalized Longitudinal Force Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['Fx/Fz' GraphTitle]); xlabel('Slip Angle (deg)');
ylabel('Slip Ratio (N/N)'); zlabel('Force (N/N)'); colorbar;
mesh(SAM*180/pi,SRM*100,FX./FZ(q));
%

% Lateral Force Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['Fy' GraphTitle]); xlabel('Slip Angle (deg)');
ylabel('Slip Ratio (%)'); zlabel('Force (Newtons)'); colorbar;
mesh(SAM*180/pi,SRM*100,FY);
%

% Normalized Lateral Force Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['Fy/Fz' GraphTitle]); xlabel('Slip Angle (deg)');
ylabel('Slip Ratio (%)'); zlabel('Force (N/N)'); colorbar;
mesh(SAM*180/pi,SRM*100,FY./FZ(q));

% Overturing Moment Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['Mx' GraphTitle]); xlabel('Slip Angle (deg)');
ylabel('Slip Ratio (%)'); zlabel('Moment (N*m)'); colorbar;
mesh(SAM*180/pi,SRM*100,MX);

% Aligning Moment Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['Mz' GraphTitle]); xlabel('Slip Angle (deg)');
ylabel('Slip Ratio (%)'); zlabel('Moment (N*m)'); colorbar;
mesh(SAM*180/pi,SRM*100,MZ);

% Absolute Force Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['F abs' GraphTitle]); xlabel('Slip Angle (deg)');
ylabel('Slip Ratio (%)'); zlabel('Force (Newtons)'); colorbar;
mesh(SAM*180/pi,SRM*100,(FX.^2+FY.^2).^.5);

% Absolute Force Elipse Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['F abs' GraphTitle]); xlabel('FX (N/N) SR Spacing - num2str(SRSpacing*100) %');
ylabel('FY (N/N) SA Spacing - num2str(SASpacing) deg');
zlabel('Abs Force (Newtons)'); colorbar;
mesh(FX./FZ(q),FY./FZ(q),(FX.^2+FY.^2).^.5);
plot3(FX(Xind,Yind)./FZ(q),FY(Xind,Yind)./FZ(q),abs(SAM.*180/pi),...'
'MarkerSize',8,'MarkerFaceColor','g','LineWidth',2);
hidden off; axis(XYAxis(1:4)); eval(GridSetup);

% Normalized Absolute Force Elipse Mesh
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['F abs' GraphTitle]); xlabel('FX (N/N) SR Spacing - num2str(SRSpacing*100) %');
ylabel('FY (N/N) SA Spacing - num2str(SASpacing) deg');
zlabel('Abs Force (Newtons)'); colorbar;
mesh(FX./FZ(q),FY./FZ(q),(FX.^2+FY.^2).^.5./FZ(q));
plot3(FX(Xind,Yind)./FZ(q),FY(Xind,Yind)./FZ(q),abs(SAM.*180/pi),...'
'MarkerSize',8,'MarkerFaceColor','g','LineWidth',2);
hidden off; axis(XYAxis(1:4)); eval(GridSetup); axis equal;

figure(n); eval(SubCallOut); n=n+1; hold on;
title(['F vs Abs Slip Angle' GraphTitle]);
xlabel('FX (N/N) SR Spacing - num2str(SRSpacing*100) %');
ylabel('FY (N/N) SA Spacing - num2str(SASpacing) deg');
zlabel('Abs Slip Angle (Degrees)'); colorbar;
mesh(FX./FZ(q),FY./FZ(q),abs(SAM.*180/pi));
plot3(FX(Xind,Yind)./FZ(q),FY(Xind,Yind)./FZ(q),0,'ro',...'
'MarkerSize',8,'MarkerFaceColor','g','LineWidth',2);
hidden off; eval(GridSetup); axis(XYAxis(1:4));
title(['F vs Slip Angle' GraphTitle]);
xlabel(['FX (N/N) SR Spacing - ' num2str(SRSpacing*100) ' %']);
ylabel(['FY (N/N) SA Spacing - ' num2str(SASpacing) ' deg']);
ylabel('Slip Angle (Degrees)'); colorbar;
mesh(FX./FZ(q),FY./FZ(q),SAM.*180/pi);
plot3(FX(Xind,Yind)./FZ(q),FY(Xind,Yind)./FZ(q),0,'ro',...
'MarkerSize',8,'MarkerFaceColor','g','LineWidth',2);
hidden off; eval(GridSetup); axis(XYAxis(1:4));
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['F vs Abs Slip Ratio' GraphTitle]);
xlabel(['FX (N/N) SR Spacing - ' num2str(SRSpacing*100) ' %']);
ylabel(['FY (N/N) SA Spacing - ' num2str(SASpacing) ' deg']);
ylabel('Abs Slip Ratio (%)'); colorbar;
mesh(FX./FZ(q),FY./FZ(q),abs(SRM.*100));
plot3(FX(Xind,Yind)./FZ(q),FY(Xind,Yind)./FZ(q),0,'ro',...
'MarkerSize',8,'MarkerFaceColor','g','LineWidth',2);
hidden off; eval(GridSetup); axis(XYAxis(1:4));
figure(n); eval(SubCallOut); n=n+1; hold on;
title(['F vs Slip Ratio GraphTitle']);
xlabel(['FX (N/N) SR Spacing - ' num2str(SRSpacing*100) ' %']);
ylabel(['FY (N/N) SA Spacing - ' num2str(SASpacing) ' deg']);
ylabel('Abs SR+SA (radian+%'); colorbar;
title(['F vs SR+SA GraphTitle']);
mesh(FX./FZ(q),FY./FZ(q),abs(SRM)+abs(SAM));
plot3(FX(Xind,Yind)./FZ(q),FY(Xind,Yind)./FZ(q),0,'ro',...
'MarkerSize',8,'MarkerFaceColor','g','LineWidth',2);
hidden off; eval(GridSetup); axis(XYAxis(1:4));
figure(n); eval(SubCallOut); n=n+1; hold on;
xlabel(['FX (N/N) SR Spacing - ' num2str(SRSpacing*100) ' %']);
ylabel(['FY (N/N) SA Spacing - ' num2str(SASpacing) ' deg']);
ylabel('Abs SR+SA (radian+%'); colorbar;
GraphFile=strcat('Plot_',OutputName,'_',GraphVar.TirePressure,...
' _IA',int2str(IAP),'_FZ',int2str(FZ),'.fig');
title(['F vs SR+SA GraphTitle']);
mesh(FX./FZ(q),FY./FZ(q),abs(SRM)+abs(SAM));
plot3(FX(Xind,Yind)./FZ(q),FY(Xind,Yind)./FZ(q),0,'ro',...
'MarkerSize',8,'MarkerFaceColor','g','LineWidth',2);
hidden off; axis equal; eval(AxisSetup); eval(GridSetup);
end
end

% Save graphs for comparison
% saveas(gcf,GraphFile);
D.5.2. Inflation Pressure and Rim Width Comparison

% Plotting utility to compare different tires. Specifically the intent is
% to allow rim width and inflation pressure effect comparisons.

clear all;
close all;

Colors=get(gca,'ColorOrder');

% Comparison Graph Loop
for m=1:2
    switch m
        case 1
            ColorCode=1; LineCall='--';
            % Tires with a full set of runs (no debeading)
            % BaseName='Hoosier20x7_13_7inch_Pac2002';
            % BaseName='Hoosier20x7_13_6inch_Pac2002';
            BaseName='Hoosier20x6_13_6inch_Pac2002';
            % BaseName='Hoosier20x6_13_7inch_Pac2002';
            % BaseName='GoodyearD2696_7inch_Pac2002';
            % BaseName='Dunlop_7inch_Pac2002';
            % BaseName='Michelin_7inch_Pac2002';
            % BaseName='Michelin_8inch_Pac2002';
            Pressure=[55 69 83 97]; ColorOffset=0;
            % % Tires with debeading at low pressure (55 Pascals)
            % BaseName='GoodyearD2696_8inch_Pac2002';
            % BaseName='Dunlop_8inch_Pac2002';
            % BaseName='Hoosier20x75_13_7inch_Pac2002';
            % BaseName='Hoosier20x75_13_8inch_Pac2002';
            % Pressure=[69 83 97]; ColorOffset=1;
        case 2
            ColorCode=2; LineCall='-.';
            % Tires with a full set of runs (no debeading)
            % BaseName='Hoosier20x7_13_7inch_Pac2002';
            % BaseName='Hoosier20x7_13_6inch_Pac2002';
            % BaseName='Hoosier20x6_13_6inch_Pac2002';
            BaseName='Hoosier20x6_13_7inch_Pac2002';
            % BaseName='Hoosier20x6_13_7inch_Pac2002';
            % BaseName='GoodyearD2696_7inch_Pac2002';
            % BaseName='Dunlop_7inch_Pac2002';
            % BaseName='Michelin_7inch_Pac2002';
            % BaseName='Michelin_8inch_Pac2002';
            Pressure=[55 69 83 97]; ColorOffset=0;
            % % Tires with debeading at low pressure (55 Pascals)
            % BaseName='GoodyearD2696_8inch_Pac2002';
            % BaseName='Dunlop_8inch_Pac2002';
            % BaseName='Hoosier20x75_13_7inch_Pac2002';
            % BaseName='Hoosier20x75_13_8inch_Pac2002';
            % Pressure=[69 83 97]; ColorOffset=1;
    end

LoadName=[BaseName '.mat'];
eval(['load ' LoadName ' ;']);
for n=1:length(Pressure)
    eval(['Pa' int2str(Pressure(n)) '=' BaseName '_Pa'...
        int2str(Pressure(n)) ';']);
eval(['TireID=' BaseName '_Pa83.TireID;']);

Var1='pCy1';
Var2='pDy1';
Var3='pDy2';
Var4='pKy1';
Var5='pKy2';
Var6='pCx1';
Var7='pDx1';
Var8='pDx2';
Var9='pKx1';
Var10='pKx2';
VarCount=10;

clear GraphVar GraphY

for n=1:VarCount
eval(['GraphVar=Var' int2str(n) ';']);
for p=1:length(Pressure)
eval(['GraphY(' int2str(p) ')=Pa' int2str(Pressure(p))... '
' GraphVar ';']);
end
figure(n); hold on;
plot(Pressure,GraphY,'Color',Colors(ColorCode,:));
xlabel('Tire Pressure (Pa)');
GraphTitle=[TireID ' - ' GraphVar ' vs. Inflation Pressure'];
title(GraphTitle);
end

FZ=25:1:1800;
IA=0*pi/180;
clear GraphVar
for n=1:length(Pressure)
eval(['GraphVar=Pa' int2str(Pressure(n)) ';']);
DGraphY=(GraphVar.pDy1+GraphVar.pDy2.*((FZ(GraphVar.pKy2.*GraphVar.Fz0))).*(1-GraphVar.pDx3.*IA.^2);
KGraphY=GraphVar.pKy1.*GraphVar.Fz0.*sin(2.*atan(FZ./...
(GraphVar.pKy2.*GraphVar.Fz0))).*(1-GraphVar.pKx3.*abs(IA));
DGraphX=(GraphVar.pDx1+GraphVar.pDx2.*((FZ-GraphVar.Fz0))).*(1-GraphVar.pDx3.*IA.^2);
KGraphX=FZ.*(GraphVar.pKx1+GraphVar.pKx2.*)((FZ-GraphVar.Fz0)).*exp(GraphVar.pKx3.*((FZ-GraphVar.Fz0)).
(GraphVar.Fz0));
figure(VarCount+1); hold on;
plot(FZ,DGraphY,'Color',Colors(n+ColorOffset,:),...
'LineStyle',LineCall);
figure(VarCount+2); hold on;
plot(FZ,KGraphY,'Color',Colors(n+ColorOffset,:),...
'LineStyle',LineCall);
figure(VarCount+3); hold on;
plot(FZ,DGraphX,'Color',Colors(n+ColorOffset,:),...
'LineStyle',LineCall);
figure(VarCount+4); hold on;
plot(FZ,KGraphX,'Color',Colors(n+ColorOffset,:),...
end

figure(VarCount+1); hold on;
GraphTitle=[TireID...  
' - Inflation Pressure Effect on F_y_0 Peak Mu_y vs F_z'];
title(GraphTitle);
xlabel('F_z (Newtons)'); ylabel('Peak Mu_y (Newtons/Newtons)');  
switch length(Pressure)
  case 3
    legend('69 Pascals','83 Pascals','97 Pascals');
  case 4
    legend('55 Pascals','69 Pascals','83 Pascals','97 Pascals');
end
figure(VarCount+2); hold on;
GraphTitle=[TireID...  
' - Inflation Pressure Effect on Cornering Stiffness K_y vs F_z'];
title(GraphTitle);
xlabel('F_z (Newtons)'); ylabel('Cornering Stiffness K_y (Newtons/rad)');  
switch length(Pressure)
  case 3
    legend('69 Pascals','83 Pascals','97 Pascals');
  case 4
    legend('55 Pascals','69 Pascals','83 Pascals','97 Pascals');
end
figure(VarCount+3); hold on;
GraphTitle=[TireID...  
' - Inflation Pressure Effect on Fx0 Peak Mu_x vs F_z'];
title(GraphTitle);
xlabel('F_z (Newtons)'); ylabel('Peak Mu_x (Newtons/Newtons)');
switch length(Pressure)
  case 3
    legend('69 Pascals','83 Pascals','97 Pascals');
  case 4
    legend('55 Pascals','69 Pascals','83 Pascals','97 Pascals');
end
figure(VarCount+4); hold on;
GraphTitle=[TireID...  
' - Inflation Pressure Effect on Longitudinal Stiffness K_x vs F_z'];
title(GraphTitle);
xlabel('F_z (Newtons)'); ylabel('Longitudinal Stiffness K_x (Newtons/%)');
switch length(Pressure)
  case 3
    legend('69 Pascals','83 Pascals','97 Pascals');
  case 4
    legend('55 Pascals','69 Pascals','83 Pascals','97 Pascals');
end

D.6. Tire Vertical Spring Rate Fitting Script

% Tire Rate and Unloaded Radius Processing Script

% Utilizing Round 4 data pre-processed via the Data Wrangler tire rate and
% unloaded radius parameters for different inclination angles are computed.
% The available data files that are the standard outputs of the data
% wrangler ISO version scripts are listed below.

clear all;
close all;

%% Tire data file selection
% Select ONE and only ONE!
% pFileName='GoodyearD2696_7inch';
% pFileName='GoodyearD2696_8inch';
% pFileName='Hoosier20x6_13_6inch';
% pFileName='Hoosier20x6_13_7inch';
% pFileName='Hoosier20x7_13_6inch';
% pFileName='Hoosier20x7_13_7inch';
% pFileName='Hoosier20x75_13_6inch';
% pFileName='Hoosier20x75_13_7inch';
% pFileName='Dunlop_7inch';
% pFileName='Dunlop_8inch';
% pFileName='Michelin_7inch';
pFileName='Michelin_8inch';

%% Stiffness and Unloaded Radius Calculation
eval(['load ' pFileName '.mat;']);

vFileName=['SpringRate_' pFileName];

% Source number of Spring Rate Tests to be fitted.
eval(['SpringIndex=' pFileName '.SpringRate.SpringIndex;']);

% Fitted IA
IAvector=[0 2 4];
eval([vFileName '.IAVector=IAvector;']);

% Fitted P
Pvector=[55 69 83 97];
eval([vFileName '.PVector=Pvector;']);

% Static Counter
Scounter=ones(length(Pvector),length(IAvector));

% Dynamic Counter
Dcounter=ones(length(Pvector),length(IAvector));

% Static Output
eval([vFileName '.Static.Rate=nan(length(Pvector),length(IAvector),10);']);
eval([vFileName '.Static.Radius=nan(length(Pvector),length(IAvector),10);']);
eval([vFileName '.StaticRate=nan(length(Pvector),length(IAvector));']);
eval([vFileName '.StaticRadius=nan(length(Pvector),length(IAvector));']);

% Dynamic Output
eval([vFileName '.Dynamic.Rate=nan(length(Pvector),length(IAvector),10);']);
eval([vFileName '.Dynamic.Radius=nan(length(Pvector),length(IAvector),10);']);
eval([vFileName '.DynamicRate=nan(length(Pvector),length(IAvector));']);
eval([vFileName '.DynamicRadius=nan(length(Pvector),length(IAvector));']);
for k=1:SpringIndex
    clear FZN RLN IAN VN TIREN TESTN FZ RL IA V tireid testid TireK
    FZN=['pFileName ' '.SpringRate.Test' int2str(k) '.FZ'];
    RLN=['pFileName ' '.SpringRate.Test' int2str(k) '.RL'];
    IAN=['pFileName ' '.SpringRate.Test' int2str(k) '.IAAVG'];
    PN=['pFileName ' '.SpringRate.Test' int2str(k) '.PAVG'];
    PVN=['pFileName ' '.SpringRate.Test' int2str(k) '.PAVG'];
    VN=['pFileName ' '.SpringRate.Test' int2str(k) '.VAVG'];
    TIREN=['pFileName ' '.SpringRate.Test' int2str(k) '.tireid'];
    TESTN=['pFileName ' '.SpringRate.Test' int2str(k) '.testid'];
    clear FZ RL IA P PV V tireid testid
    eval(['FZ=' FZN ';']);
    eval(['RL=' RLN ';']);
    eval(['IA=' IAN ';']);
    eval(['P=' PN ';']);
    eval(['PV=' PVN ';']);
    eval(['V=' VN ';']);
    eval(['tireid=' TIREN ';']);
    eval(['testid=' TESTN ';']);
    % Convert RL from centimeters to meters
    RL=RL/100;
    % First order polyfit to ascertain tire rate and initial diameter.
    [TireK]=polyfit(RL,FZ,1);
    [UnloadedR]=polyfit(FZ,RL,1);
    PVavg=mean(PV);
    [Non m]=min(abs(Pvector-P));
    [Non n]=min(abs(IAvector-IA));
    % Use velocity and inclination angle information to classify data.
    switch V
    case 0
        eval(['vFileName ' '.Static.Rate(m,n,Scounter(m,n))=TireK(1);']);
        eval(['vFileName ' '.Static.Radius(m,n,Scounter(m,n))=UnloadedR(2);']);
        Scounter(m,n)=Scounter(m,n)+1;
    otherwise
        eval(['vFileName ' '.Dynamic.Rate(m,n,Dcounter(m,n))=TireK(1);']);
        eval(['vFileName ' '.Dynamic.Radius(m,n,Dcounter(m,n))=UnloadedR(2);']);
        Dcounter(m,n)=Dcounter(m,n)+1;
    end
end
% Store average values to value in higher part of data structure.
for m=1:length(Pvector)
    for n=1:length(IAvector)
        if Scounter(m,n)>1
            eval(['vFileName ' '.StaticRate(m,n)=mean(abs(' vFileName ' '.Static.Rate(m,n,1:Scounter(m,n)-1)));']);
        end
    end
end
eval(['vFileName' '.StaticRadius(m,n)=mean(abs(' vFileName...
  '.Static.Radius(m,n,1:Scounter(m,n)-1)));']);
end
if Dcounter(m,n)>1
  eval(['vFileName' '.DynamicRate(m,n)=mean(abs(' vFileName...
    '.Dynamic.Rate(m,n,1:Dcounter(m,n)-1)));']);
eval(['vFileName' '.DynamicRadius(m,n)=mean(abs(' vFileName...
    '.Dynamic.Radius(m,n,1:Dcounter(m,n)-1)));']);
end
end

% Units information.
UnitMsg='Newtons,Meter';
eval(['vFileName' '.UNITS=UnitMsg;']);

% Data Information.
DateRun=datestr(now);
eval(['vFileName' '.DateRun=DateRun;']);

%% Store Information and Terminate Script
% Save out Spring Rate Structure to mat file.
eval(['save ' vFileName ' StaticRate;']);
eval(['save ' vFileName ' StaticRadius;']);
eval(['save ' vFileName ' DynamicRate;']);
eval(['save ' vFileName ' DynamicRadius;']);

% Some Graphs
figure(1);
eval(['mesh(Pvector,IAvector,' vFileName ' DynamicRate);']);
title('Tire Rate'); xlabel('Inflation Pressure (Pa)');
ylabel('Inclination Angle (deg)'); zlabel('Tire Rate (Newton/Meter)');
figure(2);
eval(['mesh(Pvector,IAvector,' vFileName ' DynamicRadius);']);
title('Tire Unloaded Radius'); xlabel('Inflation Pressure (Pa)');
ylabel('Inclination Angle (deg)'); zlabel('Tire Radius (Meter)');

% Clean up.
clear all

D.7. Thermal Effects Analysis Script

%% Tire Thermal Effects Test
% During the conditioning and warmup of the tires during round 4, a Cold to
% Hot test was performed to explore the thermal effects on performance. In
% the documentation of the test plan the test was summarized as:
% Cold to Hot test: ±12 deg SA @ 8 deg/sec, 0 IA, 250 lb, 12 psi --repeat 8
% sweeps
% Each sweep takes 6 seconds, thus a total test time of 48 seconds.

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These graphs are a demonstration of that information only. There are many more ways to analyze and view the data. Of special note is the effect of temperature on the disparity in peak values of force during the two cross over points in the slip angle sweep. Every tire test has this information that could be analyzed as well.

clear all;
close all;

%% User Input Selections
% Adjust the smoothing factor for vectors going through the numerical derivation process.
DerivS=15;

%% Tire Data File Selection - Listed are the standard outputs of the data wrangler scripts for ISO formatted data. The script is made to function with this specific input format.

pFileName='GoodyearD2696_7inch'; % Test3 available
pFileName='GoodyearD2696_8inch';
pFileName='Hoosier20x6_13_6inch';
pFileName='Hoosier20x6_13_7inch';
pFileName='Hoosier20x7_13_6inch';
pFileName='Hoosier20x7_13_7inch'; % Test3 available
pFileName='Hoosier20x75_13_6inch';
pFileName='Hoosier20x75_13_8inch';
pFileName='Dunlop_7inch'; % Test3 available
pFileName='Dunlop_8inch';
pFileName='Michelin_7inch'; % Test3 available
pFileName='Michelin_8inch';

% Tests at a later time were done on a select number of tires, resulting in a third testing. The first test was done during the cornering warm up, while the second was done during the brake drive warmup. Test3 is only available for the select runs listed above.
% Test='Test1';
% Test='Test2';
Test='Test3';

%% Data File Loading and Variable Assignment
eval(['load ' pFileName '.mat;'])

% pressure run for fitting
% Possibilities are [Pa55 Pa69 Pa83 Pa97 Pa83COLD]
pressure='Pa83';

ETN=[pFileName '.Cold2Warm.' Test '.ET'];
FZN=[pFileName '.Cold2Warm.' Test '.FZ'];
SAN=[pFileName '.Cold2Warm.' Test '.SA'];
FYN=[pFileName '.Cold2Warm.' Test '.FY'];
MZN=[pFileName '.Cold2Warm.' Test '.MZ'];
MXN=[pFileName '.Cold2Warm.' Test '.MX'];
REN=[pFileName '.Cold2Warm.' Test '.RE'];
VN=[pFileName '.Cold2Warm.' Test '.V'];
TSTCN=[pFileName '.Cold2Warm.' Test '.TSTC'];
TSTIN=[pFileName '.Cold2Warm.' Test '.TSTI'];
TSTON=[pFileName '.Cold2Warm.' Test '.TSTO'];
TIREN=[pFileName '.Cold2Warm.' Test '.tireid'];

eval(['ET=' ETN '];
 eval(['FZ=' FZN ']);
 eval(['SA=' SAN ']);
 eval(['FY=' FYN ']);
 eval(['MZ=' MZN ']);
 eval(['MX=' MXN ']);
 eval(['RE=' REN ']);
 eval(['V=' VN ']);
 eval(['TSTC=' TSTCN ']);
 eval(['TSTI=' TSTIN ']);
 eval(['TSTO=' TSTON ']);
 eval(['tireid=' TIREN ']);

FYN4s=[pFileName '.Cornering.Pa83.IA0.FZ1112.FY'];
FZN4s=[pFileName '.Cornering.Pa83.IA0.FZ1112.FZ'];
SAN4s=[pFileName '.Cornering.Pa83.IA0.FZ1112.SA'];

eval(['FY4s=' FYN4s ']);
eval(['FZ4s=' FZN4s ']);
eval(['SA4s=' SAN4s ']);

% Fahrenheit Conversion
% TSTC=TSTC.*(9/5)+32;
% TSTI=TSTI.*(9/5)+32;
% TSTO=TSTO.*(9/5)+32;
clear ETN FYN FZN MXN MZN SAN TIREN TSTCN TSTIN TSTON

%% Calculation and Plotting

c=1; % Plot Counter Initiation

% Phase Temperature data to match force measurement
% Temperature sensors observe temperatures 3/4 of a revolution after they
% leave the roadway.
SensorPhase=.75*2*pi;
PhaseOffset=round(((SensorPhase*mean(RE)/100)/(mean(V)*1000/3600))/(ET(2)-ET(1)));
ET=ET(1:length(TSTO)-PhaseOffset+1);
FZ=FZ(1:length(TSTO)-PhaseOffset+1);
SA=SA(1:length(TSTO)-PhaseOffset+1);
FY=FY(1:length(TSTO)-PhaseOffset+1);
TSTI=TSTI(PhaseOffset:length(TSTO));
TSTC=TSTC(PhaseOffset:length(TSTO));
TSTO=TSTO(PhaseOffset:length(TSTO));

ET=ET-min(ET); % Zero out time vector for distinct test

% Tire Temps vs. Elapsed Time
figure(c); subplot(1,2,1); hold on;
plot(ET,TSTI,'b--',ET,TSTC,'r-',ET,TSTO,'c--');
title([tireid ' - Tire Temps vs. Elapsed Time']);
xlabel('Elapsed Time (sec)');
ylabel('Tire Temp (deg Celcius)');
legend('Inner Tread', 'Center Tread', 'Outer Tread');

% Compute the sampling period.
DeltaT = (ET(10) - ET(1))/9;

% Tire Temps vs Slip Angle
figure(c); subplot(1,2,2); c=c+1; hold on;
plot(SA, TSTI, 'b--', SA, TSTC, 'r-', SA, TSTO, 'c--');
title(['tireid - Tire Temps vs. Slip Angle']);
xlabel('Slip Angle (deg)');
ylabel('Tire Temp (deg Celcius)');
legend('Inner Tread', 'Center Tread', 'Outer Tread');

% Lateral Force vs Slip Angle
figure(c); c=c+1;
plot(SA, smooth(FY, 25), 'b-', SA4s, smooth(FY4s, 25), 'r-');
title(['tireid - Lateral Force vs. Slip Angle']);
xlabel('Slip Angle (deg)');
ylabel('Lateral Force (Newtons)');
legend('Temperature Study 8^o/sec SV', 'SA1 Run 4^o/sec SV');
figure(c); c=c+1;
plot(SA, FY);
title(['tireid - Raw Lateral Force vs. Slip Angle']);
xlabel('Slip Angle (deg)');
ylabel('Lateral Force (Newtons)');
figure(c); c=c+1;
plot(SA, smooth(FY./FZ, 25));
title(['tireid - Normalized Lateral Force vs. Slip Angle']);
xlabel('Slip Angle (deg)');
ylabel('FY/FZ (mu)');
figure(c); c=c+1;
plot3(ET, SA, smooth(FY./FZ, 25));
title(['tireid - Normalized Lateral Force vs. Slip Angle vs Time']);
xlabel('Elapsed Time (sec)');
ylabel('FY/FZ (mu)');
figure(c); c=c+1;
plot(ET, SA);
title(['tireid - Slip Angle vs Elapsed Time']);
xlabel('Elapsed Time (sec)');
ylabel('Slip Angle (deg)');
figure(c); c=c+1;
plot(ET, smooth(FY./FZ, 25));
title(['tireid - Normalized Lateral Force vs. Elapsed Time']);
xlabel('Elapsed Time (sec)');
ylabel('FY/FZ (mu)');

% Intensive Lateral Force vs Slip Angle Study
FYStudy = smooth(FY, 25);
LinearLength = 3;
StiffCount = 1;
DeLineate = 100;
PeakCount = [1 1 1 1];
LinearCount = [1 1 1 1];
for k = 100:length(FYStudy)
    if (FYStudy(k)>0 && FYStudy(k-1)<=0)

if k<=LinearLength+1
    LinearBound=[1 k+LinearLength];
else
    LinearBound=[k-LineareLength k+LinearLength];
end
LinearFit=polyfit(SA(LinearBound(1):LinearBound(2)),...
    FYStudy(LinearBound(1):LinearBound(2)),1);
CornerStiffness(StiffCount)=LinearFit(1);
StiffCount=StiffCount+1;
if k>=150
    for m=1:150
        if abs(FYStudy(k+1-m)-(LinearFit(1)*SA(k+1-m)+...
            LinearFit(2)))>=DeLineate
            Linear1(LinearCount(1))=k+1-m;
            LinearCount(1)=LinearCount(1)+1;
            break;
        end
    end
    for m=1:150
        if abs(FYStudy(k+1-m))>=abs(FYStudy(k-m)) &&...
            abs(FYStudy(k+1-m))>=abs(FYStudy(k+2-m))
            Peak1(PeakCount(1))=k+1-m;
            PeakCount(1)=PeakCount(1)+1;
            break;
        end
    end
end
if k<=length(SA)-150
    for m=1:150
        if abs(FYStudy(k+1+m)-(LinearFit(1)*SA(k+1+m)+...
            LinearFit(2)))>=DeLineate
            Linear2(LinearCount(2))=k+1-m;
            LinearCount(2)=LinearCount(2)+1;
            break;
        end
    end
    for m=1:150
        if abs(FYStudy(k+1+m))>=abs(FYStudy(k+m)) &&...
            abs(FYStudy(k+1+m))>=abs(FYStudy(k-2+m))
            Peak2(PeakCount(2))=k+1-m;
            PeakCount(2)=PeakCount(2)+1;
            break;
        end
    end
end
if (FYStudy(k)<0 && FYStudy(k-1)>=0)
    if k<=LinearLength+1
        LinearBound=[1 k+LinearLength];
    else
        LinearBound=[k-LineareLength k+LinearLength];
    end
    LinearFit=polyfit(SA(LinearBound(1):LinearBound(2)),...
        FYStudy(LinearBound(1):LinearBound(2)),1);
    CornerStiffness(StiffCount)=LinearFit(1);
    StiffCount=StiffCount+1;
if k>=150

for m=1:150
    if abs(FYStudy(k+1-m)-(LinearFit(1)*SA(k+1-m)+...)
        LinearFit(2)))>=DeLineate
        Linear3(LinearCount(1))=k+1-m;
        LinearCount(3)=LinearCount(3)+1;
        break;
    end
end
for m=1:150
    if abs(FYStudy(k+1-m))>=abs(FYStudy(k-m)) &&...
        abs(FYStudy(k+1-m))>=abs(FYStudy(k+2-m))
        Peak3(PeakCount(3))=k+1-m;
        PeakCount(3)=PeakCount(3)+1;
        break;
    end
end
if k<=length(SA)-150
    for m=1:150
        if abs(FYStudy(k-1+m)-(LinearFit(1)*SA(k-1+m)+...)
            LinearFit(2)))>=DeLineate
            Linear4(LinearCount(4))=k-1+m;
            LinearCount(4)=LinearCount(4)+1;
            break;
        end
    end
end
end
end
end
for m=1:4
    eval(['Peak=Peak' int2str(m) ';']);
    eval(['Linear=Linear' int2str(m) ';']);
    for n=1:length(Peak)
        eval(['Peak' int2str(m) 'SA(n)=SA(' int2str(Peak(n)) ')';']);
        eval(['Peak' int2str(m) 'FY(n)=FYStudy(' int2str(Peak(n)) ')';']);
        eval(['Peak' int2str(m) 'ET(n)=ET(' int2str(Peak(n)) ')';']);
    end
    for n=1:length(Linear)
        eval(['Linear' int2str(m) 'SA(n)=SA(' int2str(Linear(n)) ')';']);
        eval(['Linear' int2str(m) 'FY(n)=FYStudy(' int2str(Linear(n)) ')';']);
        eval(['Linear' int2str(m) 'ET(n)=ET(' int2str(Linear(n)) ')';']);
    end
end
figure(c); hold on; c=c+1;
for m=1:3
    ...
subplot(2,2,m);
plot3(ET,SA,FYStudy,'b-',...  
    Peak1ET,Peak1SA,Peak1FY,'go-',Peak2ET,Peak2SA,Peak2FY,'ro-',...  
    Peak3ET,Peak3SA,Peak3FY,'mo-',Peak4ET,Peak4SA,Peak4FY,'co-',...  
    Linear1ET,Linear1SA,Linear1FY,'gs-',...  
    Linear2ET,Linear2SA,Linear2FY,'rs-',...  
    Linear3ET,Linear3SA,Linear3FY,'ms-',...  
    Linear4ET,Linear4SA,Linear4FY,'cs-');
title([tireid ' - Lateral Force vs. Slip Angle']);
xlabel('Elapsed Time (sec)');
ylabel('Slip Angle (deg)');
zlabel('Lateral Force (Newtons)');
end
 subplot(2,2,1);
legend('FY vs SA vs ET','Peak Variation: +SA,-SV',...  
    'Peak Variation: -SA,-SV','Peak Variation: -SA,+SV',...  
    'Peak Variation: +SA,+SV','Delineation: +SA,-SV',...  
    'Delineation: -SA,-SV','Delineation: -SA,+SV',...  
    'Delineation: +SA,+SV','Location','NorthEast');
 subplot(2,2,2); grid on; AXIS=axis; AXIS(3)=-10; AXIS(4)=10; axis(AXIS);
 subplot(2,2,3); grid on;
 subplot(2,2,4);
 plot(CornerStiffness);
title([tireid ' - Cornering Stiffness Variation']);
ylabel('C_F_y (Newtons/Degree)');
xlabel('Zero Crossing Count');

% Tire Temps vs FY/FZ vs Slip Angle (3-d plot)
 figure(c); c=c+1;
 plot3(SA,smooth(FY./FZ,25),TSTC,'r-');
title([tireid ' - Center Temperature vs. FY/FZ vs. Slip Angle']);
xlabel('Slip Angle (deg)');
ylabel('FY/FZ (mu)');
zlabel('Center Tread Temp (deg Celcius)');
figure(c); c=c+1;
 plot3(SA,smooth(FY./FZ,25),TSTI,'b-');
title([tireid ' - Inner Temperature vs. FY/FZ vs. Slip Angle']);
xlabel('Slip Angle (deg)');
ylabel('FY/FZ (mu)');
zlabel('Inner Tread Temp (deg Celcius)');
figure(c); c=c+1;
 plot3(SA,smooth(FY./FZ,25),TSTO,'c-');
title([tireid ' - Outer Temperature vs. FY/FZ vs. Slip Angle']);
xlabel('Slip Angle (deg)');
ylabel('FY/FZ (mu)');
zlabel('Outer Tread Temp (deg Celcius)');

% Tire Temps vs FY/FZ and Inverse
 figure(c); c=c+1;
 plot(smooth(FY./FZ,25),TSTC,'r-')
title([tireid ' - Center Tread Temp vs. FY/FZ']);
ylabel('Center Temperature (deg Celcius)');
xlabel('FY/FZ (mu)');
figure(c); c=c+1;
plot(smooth(FY./FZ,25),TSTI,'b-')
title({'tireid ' - 'Inner Tread Temp vs. FY/FZ'});
ylabel('Inner Temperature (deg Celcius)');
xlabel('FY/FZ (mu)');
figure(c); c=c+1;
plot(smooth(FY./FZ,25),TSTO,'c-')
title({'tireid ' - 'Outer Tread Temp vs. FY/FZ'});
ylabel('Outer Temperature (deg Celcius)');
xlabel('FY/FZ (mu)');
figure(c); subplot(1,3,2); hold on;
plot(smooth(FY./FZ,25),TSTC,'r-')
title({'tireid ' - 'Tread Temp vs. FY/FZ'});
ylabel('Center Temperature (deg Celcius)');
xlabel('FY/FZ (mu)');
axis(AxLim);
figure(c); subplot(1,3,1); hold on;
plot(smooth(FY./FZ,25),TSTI,'b-')
figure(c); subplot(1,3,3); hold on; c=c+1;
plot(smooth(FY./FZ,25),TSTO,'c-')
figure(c); c=c+1;
plot(TSTC,smooth(FY./max(abs(FY)),7),'r-')
title({'tireid ' - 'Normalized FY vs Center Tread Temp'});
ylabel('FY/max(FY)');
xlabel('Center Temperature (deg Celcius)');
figure(c); c=c+1;
plot(TSTI,smooth(FY./max(abs(FY)),7),'b-')
title({'tireid ' - 'FY/max(FY) vs Inner Tread Temp'});
ylabel('FY/max(FY)');
xlabel('Inner Temperature (deg Celcius)');
figure(c); c=c+1;
plot(TSTO,smooth(FY./max(abs(FY)),7),'c-')
title({'tireid ' - 'FY/max(FY) vs Outer Tread Temp'});
ylabel('FY/max(FY)');
xlabel('Outer Temperature (deg Celcius)');
figure(c); c=c+1;
plot(FY,TSTC,'r-')
title({'tireid ' - 'Center Tread Temp vs. Raw FY'});
ylabel('Center Temperature (deg Celcius)');
xlabel('FY (Newtons)');
figure(c); c=c+1;
plot(smooth(FY./FZ,25),TSTC,'r-',smooth(FY./FZ,25),TSTI,'b-',...
    smooth(FY./FZ,25),TSTO,'c-')
title({'tireid ' - 'Tread Temp vs. FY/FZ'});
ylabel('Tread Temperature (deg Celcius)');
xlabel('FY/FZ (mu)');
legend('Center Tread','Inner Tread','Outer Tread');
% Numerical Derivative Preparation
FYs=smooth(FY./FZ,DerivS);

MATSIZE=size(TSTC);
dTSTC=zeros(MATSIZE);
dTSTI=zeros(MATSIZE);
dTSTO=zeros(MATSIZE);
dFYs=zeros(MATSIZE);
DFYs=zeros(MATSIZE);

% Numerical Derivative Loop
for k=2:length(FY)
    dTSTC(k)=(TSTC(k)-TSTC(k-1))/DeltaT;
    dTSTI(k)=(TSTI(k)-TSTI(k-1))/DeltaT;
    dTSTO(k)=(TSTO(k)-TSTO(k-1))/DeltaT;
    dFYs(k)=(FYs(k)-FYs(k-1))/DeltaT;
    DFYs(k)=(FYs(k)-FYs(k-1))/(SA(k)-SA(k-1));
    if k==2
        dTSTC(k-1)=(TSTC(k)-TSTC(k-1))/DeltaT;
        dTSTI(k-1)=(TSTI(k)-TSTI(k-1))/DeltaT;
        dTSTO(k-1)=(TSTO(k)-TSTO(k-1))/DeltaT;
        dFYs(k-1)=(FYs(k)-FYs(k-1))/DeltaT;
        DFYs(k-1)=DFYs(k);
    end
end

% Postprocess smoothing of time derivative of FY
sdFYs=smooth(dFYs,25);

% Center Temperature Derivative Graphs
figure(1); c=c+1;
subplot(1,2,1);
plot(smooth(FY./FZ,25),smooth(dTSTC,7),'r-');
title([tireid ' - Center Tread dT/dt vs. FY/FZ']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('FY/FZ (mu)');

subplot(1,2,2);
plot(SA,smooth(dTSTC,7),'r-');
title([tireid ' - Center Tread dT/dt vs. SA']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('Slip Angle (deg)');
figure(c); c=c+1;
plot3(smooth(FY./FZ,25),ET,smooth(dTSTC,7),'r-');
title([tireid ' - Center Tread dT/dt vs. ET vs. FY/FZ']);
zlabel('Temperature Delta (deg C/sec)');
ylabel('Elapsed Time (sec)');
xlabel('FY/FZ (mu)');

figure(c); c=c+1;
plot(smooth(MZ./FY,25),smooth(dTSTC,7),'r-');
figure(c); c=c+1;
plot(smooth(MX./FZ,25),smooth(dTSTC,7),'r-');
figure(c); c=c+1;
plot(sdFYs,smooth(dTSTC,7),'r-');
title([tireid ' - Center Tread dT/dt vs. dFY/dt']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('\(dFY/dt\) (Newton\(s\)/Sec)');

figure(c); c=c+1;
plot3(sdFYs,smooth(FY./FZ,25),smooth(dTSTC,7),'r-');
title([\text{\texttt{tireid}} ' - Center Tread dT/dt vs. FY/FZ vs dFY/dt']);
zlabel('Temperature Delta (deg C/sec)');
xlabel('dFY/dt (Newton\(s\)/Sec)');
ylabel('FY/FZ (\(\mu\))');

figure(c); c=c+1;
plot(FY./FZ,smooth(FY/FZ,25),'r-');
title([\text{\texttt{tireid}} ' - FY/FZ vs dFY/dt']);

xlabel('dFY/dt (Newton\(s\)/Sec)');
ylabel('FY/FZ (\(\mu\))');

figure(c); c=c+1;
plot3(abs(sdFYs),smooth(FY./FZ,25),smooth(TSTC,7),'r-');
title([\text{\texttt{tireid}} ' - Center Tread Temp vs. FY/FZ vs abs(dFY/dt)']);
zlabel('Temperature (deg Celcius)');
xlabel('dFY/dt (Newton\(s\)/Sec)');
ylabel('FY/FZ (\(\mu\))');

Sample=1:1:length(ET);

figure(c); c=c+1;
plot3(sdFYs,smooth(FY./FZ,25),Sample,'r-');
title([\text{\texttt{tireid}} ' - Center Tread Temp vs. FY/FZ vs Index']);
zlabel('Matrix Index');
xlabel('dFY/dt (Newton\(s\)/Sec)');
ylabel('FY/FZ (\(\mu\))');


\% Inner Temperature Derivative Graphs
figure(c); c=c+1;
plot(smooth(FY./FZ,25),smooth(dTSTI,7),'b-');
title([\text{\texttt{tireid}} ' - Inner Tread dT/dt vs. FY/FZ']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('FY/FZ (\(\mu\))');

figure(c); c=c+1;
plot3(smooth(FY./FZ,25),ET,smooth(dTSTI,7),'b-');
title([\text{\texttt{tireid}} ' - Inner Tread dT/dt vs. ET vs. FY/FZ']);
zlabel('Temperature Delta (deg Celcius)');
xlabel('FY/FZ (\(\mu\))');
ylabel('Elapsed Time (sec)');

figure(c); c=c+1;
plot(SA,smooth(dTSTI,7),'b-');
title([\text{\texttt{tireid}} ' - Inner Tread dT/dt vs. SA']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('SA (deg)');

\% figure(c); c=c+1;
% plot(smooth(MZ./FY,25),smooth(dTSTI,7),'b-');
% figure(c); c=c+1;
% plot(smooth(MX./FZ,25),smooth(dTSTI,7),'b-');
% figure(c); c=c+1;
plot(sdFYs,smooth(dTSTI,7),'b-');
title([\text{\texttt{tireid}} ' - Inner Tread dT/dt vs. dFY/dt']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('dFY/dt (Newton/Sec)');
plot3(sdFYS, smooth(FY./FZ, 25), smooth(dTSTI, 7), 'b-');
title(['tireid ' - Inner Tread dT/dt vs. FY/FZ vs dFY/dt']);
zlabel('Temperature Delta (deg C/sec)');
ylabel('FY/FZ (mu)');
figure(c); c=c+1;
plot3(sdFYS, smooth(FY./FZ, 25), smooth(TSTI, 7), 'b-');
title(['tireid ' - Inner Tread Temp vs. FY/FZ vs dFY/dt']);
zlabel('Temperature (deg Celcius)');
ylabel('FY/FZ (mu)');

% Inner Temperature Derivative Graphs
figure(c); c=c+1;
plot(smooth(FY./FZ, 25), smooth(dTSTO, 7), 'c-');
title(['tireid ' - Outer Tread dT/dt vs. FY/FZ']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('FY/FZ (mu)');
figure(c); c=c+1;
plot3(smooth(FY./FZ, 25), ET, smooth(dTSTO, 7), 'c-');
title(['tireid ' - Outer Tread dT/dt vs. ET vs. FY/FZ']);
zlabel('Temperature Delta (deg C/sec)');
ylabel('Elapsed Time (sec)');
xlabel('FY/FZ (mu)');
figure(c); c=c+1;
plot(SA, smooth(dTSTO, 7), 'c-');
title(['tireid ' - Outer Tread dT/dt vs. SA']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('SA (deg)');
figure(c); c=c+1;
% figure(c); c=c+1;
% plot(smooth(MZ./FY, 25), smooth(dTSTO, 7), 'c-');
% figure(c); c=c+1;
% plot(smooth(MX./FZ, 25), smooth(dTSTO, 7), 'c-');
% figure(c); c=c+1;
plot(sdFYS, smooth(dTSTO, 7), 'c-');
title(['tireid ' - Outer Tread dT/dt vs. dFY/dt']);
ylabel('Temperature Delta (deg C/sec)');
xlabel('dFY/dt (Newton/Sec)');
plot3(sdFYS, smooth(FY./FZ, 25), smooth(dTSTO, 7), 'c-');
title(['tireid ' - Outer Tread dT/dt vs. FY/FZ vs dFY/dt']);
zlabel('Temperature Delta (deg C/sec)');
ylabel('FY/FZ (mu)');
figure(c); c=c+1;
plot3(sdFYS, smooth(FY./FZ, 25), smooth(TSTO, 7), 'c-');
title(['tireid ' - Outer Tread Temp vs. FY/FZ vs dFY/dt']);
zlabel('Temperature (deg Celcius)');
xlabel('dFY/dt (Newton/Sec)');
ylabel('FY/FZ (mu)');

% Compute effect of temperature on Peak Factor D's major component, Mu Y.
k=1;
m=1;
n=1;
sFYFZ=smooth(FY./FZ, 25);
sTSTC=smooth(TSTC, 7);
sTST0=smooth(TST0,7);
sTSTI=smooth(TSTI,7);

% Compute test lengths to do apply fitting methods to.
for k=1:length(TST0)
    if m==1 && abs(sFYFZ(k))/sFYFZ(k)>=1 && abs(sdFYs(k))/sdFYs(k)>=1
        SegmentVec(n)=k;
        m=0;
        n=n+1;
    elseif m==0 && abs(sFYFZ(k))/sFYFZ(k)<=-1 && abs(sdFYs(k))/sdFYs(k)<=-1
        SegmentVec(n)=k;
        m=1;
        n=n+1;
    end
end

% dFY/dET will be zero at peak magnitude for both increasing and decreasing
% SA magnitude at both positive and negative slip angles. Relaxation
% effects result in non-linearities when trying to fit static curve data to
% a dynamic test. Thus the peaks will be plot with other peaks of like
% conditions. Curves should be similar in shape, but possess offsets due
% to uncompensated transient effects. As such, the data will be seperated
% into 4 sections based on slip angle sign and the sign of the derivative
% of the absolute of the slip angle.

ind1=1;ind2=1;ind3=1;ind4=1;
for k=1:length(SegmentVec)-1
    Start=SegmentVec(k); Finish=SegmentVec(k+1);
    clear mxind mnind
    nomin=1;
    for m=Finish:-1:Start
        if abs(sdFYs(m))/sdFYs(m)-abs(sdFYs(m+1))/sdFYs(m+1)~=0
            mxind=-m;
            break
        end
    end
    for m=Start:mxind-10
        if abs(sdFYs(m))/sdFYs(m)-abs(sdFYs(m+1))/sdFYs(m+1)~=0
            mnind=m;
            nomin=0;
            break
        end
    end
    if round(k/2)==k/2 % quadrant 1 & 2 data
        PeakInd2(ind2)=mxind;
        ind2=ind2+1;
        if nomin==0
            PeakInd1(ind1)=mnind;
            ind1=ind1+1;
        end
    else
        PeakInd4(ind4)=mxind;
        ind4=ind4+1;
        if nomin==0
            PeakInd3(ind3)=mnind;
            ind3=ind3+1;
        end
    end
% For the indices found, the data is pulled from the respective vectors.
for k=1:4
  eval(['PeakL=length(PeakInd' int2str(k) ');']);
  for m=1:PeakL
    eval(['PeakTSTC_' int2str(k) '(' int2str(m) ')=sTSTC(PeakInd'...
        int2str(k) ');']);
    eval(['PeakTSTI_' int2str(k) '(' int2str(m) ')=sTSTI(PeakInd'...
        int2str(k) ');']);
    eval(['PeakTSTO_' int2str(k) '(' int2str(m) ')=sTSTO(PeakInd'...
        int2str(k) ');']);
    eval(['PeakFYFZ_' int2str(k) '(' int2str(m) ')=sFYFZ(PeakInd'...
        int2str(k) ');']);
  end
end

% Mu Y vs Center Temperature for the 4 Slip Angle conditions. The most
% useful and intuitive of the 3 temperatures.
figure(c); subplot(2,1,1); hold on;
plot(PeakTSTC_3,abs(PeakFYFZ_3),',',PeakTSTC_4,abs(PeakFYFZ_4),',',...
    PeakTSTC_1,abs(PeakFYFZ_1),',',PeakTSTC_2,abs(PeakFYFZ_2),');
title([tireid ' - Peak FY/FZ Variation with Center Temperature']);
xlabel('Center Temperature (deg Celcius)');
ylabel('Peak Factor D - FY/FZ (mu)');
legend('Increasing abs(SA), Negative SA',...
    'Decreasing abs(SA), Negative SA',
    'Increasing abs(SA), Positive SA',
    'Decreasing abs(SA), Positive SA',
    'Location','SouthEast');
figure(c); subplot(2,1,2); hold on; c=c+1;
plot(PeakFYFZ_3,PeakTSTC_3,  
    PeakFYFZ_4,PeakTSTC_4, 
    PeakFYFZ_1,PeakTSTC_1,  
    PeakFYFZ_2,PeakTSTC_2,  
    smooth(FY./FZ,25),TSTC,'m:');
title([tireid ' - Center Temperature vs FY/FZ with Peak Force Overlay']);
xlabel('Center Temperature (deg Celcius)');
ylabel('FY/FZ (mu)');
figure(c); hold on; c=c+1;
plot(PeakTSTC_3,abs(PeakFYFZ_3)./max(abs(PeakFYFZ_3)),',',PeakTSTC_4,  
    abs(PeakFYFZ_4)./max(abs(PeakFYFZ_4)),',',PeakTSTC_1,  
    abs(PeakFYFZ_1)./max(abs(PeakFYFZ_1)),',',PeakTSTC_2,  
    abs(PeakFYFZ_2)./max(abs(PeakFYFZ_2)),');
title([tireid ' - Normalized Peak FY/FZ Variation with Center Temperature']);
xlabel('Center Temperature (deg Celcius)');
ylabel('Peak Factor D - FY/FZ (mu)');
legend('Increasing abs(SA), Negative SA',...
    'Decreasing abs(SA), Negative SA',
    'Increasing abs(SA), Positive SA',
    'Decreasing abs(SA), Positive SA',
    'Location','SouthEast');

% Mu Y vs Inside Temperature for the 4 Slip Angle conditions.
figure(c); c=c+1;
plot(PeakTSTI_3,abs(PeakFYFZ_3),',',PeakTSTI_4,abs(PeakFYFZ_4),',',
    PeakTSTI_1,abs(PeakFYFZ_1),',',PeakTSTI_2,abs(PeakFYFZ_2),');
title([tireid ' - Peak FY/FZ Variation with Inner Temperature']);
xlabel('Inner Temperature (deg Celcius)');
ylabel('Peak Factor D - FY/FZ (mu)');
legend('Increasing abs(SA), Negative SA',...
'Decreasing abs(SA), Negative SA', 'Increasing abs(SA), Positive SA', ...
'Decreasing abs(SA), Positive SA', 'Location', 'SouthEast');

% Mu Y vs Outside Temperature for the 4 Slip Angle conditions.
figure(c); c=c+1;
plot(PeakTSTO_3,abs(PeakFYFZ_3),'', PeakTSTO_4,abs(PeakFYFZ_4),'', ...
PeakTSTO_1,abs(PeakFYFZ_1),'', PeakTSTO_2,abs(PeakFYFZ_2),'');
title([ingredients ' - Peak FY/FZ Variation with Outside Temperature']);
xlabel('Outside Temperature (deg Celcius)');
ylabel('Peak Factor D - FY/FZ (mu)');
legend('Increasing abs(SA), Negative SA', ...
'Decreasing abs(SA), Negative SA', 'Increasing abs(SA), Positive SA', ...
'Decreasing abs(SA), Positive SA', 'Location', 'SouthEast');

D.8. Transient Effects Analysis – Relaxation Length

%% Relaxation Length Fitting Utility - Lateral Edition
% Tire test run and pressure selection for curve fitment.
% Pacejka fitting routine is Pac2002 MSC.Adams version.
% DATA MUST BE IN ISO TIRE SYSTEM!
% Calspan data is in SAE tire system natively.
clear all;
close all;

% Pacejka 2002 Adams Curve Fitting Utility
% Round 4 FSAE TTC data sectored utilizing data wrangler.

% pFileName='GoodyearD2696_7inch'; D=20.0;
% pFileName='GoodyearD2696_8inch'; D=20.0; % No Pa55
% pFileName='Hoosier20x6_13_6inch'; D=20.5;
% pFileName='Hoosier20x6_13_7inch'; D=20.5;
% pFileName='Hoosier20x7_13_6inch'; D=20.5;
% pFileName='Hoosier20x7_13_7Inch'; D=20.5;
% pFileName='Hoosier20x75_13_7inch'; D=20.0; % No Pa55
% pFileName='Hoosier20x75_13_8inch'; D=20.0; % No Pa55
% pFileName='Dunlop_7inch'; D=(2*175*.505+13*25.4)/1000*2/.0254;
% pFileName='Dunlop_8inch'; D=(2*175*.505+13*25.4)/1000*2/.0254;
% pFileName='Michelin_7inch'; D=(2*160*.53+13*25.4)/1000*2/.0254;
% pFileName='Michelin_8inch'; D=(2*160*.53+13*25.4)/1000*2/.0254;

eval(['load ' pFileName '.mat;'])
load Hoosier20x6_13_6inch_SCALE.mat
SCALE=Hoosier20x6_13_6inch_SCALE;

% pressure run for fitting
% Possibilities are [Pa55 Pa69 Pa83 Pa97 Pa83COLD]
pressure='Pa83';
eval(['load ' pFileName '_Pac2002_' pressure '.mat;'])
eval(['COEFF=' pFileName '_Pac2002;']);

% Lateral Force - Pure Side Slip - Initial Coefficient Definition
% Inclination Angle Runs
IAP=[0 1 2 3 4];
% Normal Load Runs
FZP=[222 445 667 1112 1557];

% IA Index
n=1;
% FZ Index
m=4;

% Cornering load averaged from the test given will be used.
% The middle loading of 667 N (150 lbf) will be utilized as pFz0.
FZN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n)) '.FZ' ...
int2str(FZP(m)) '.FZ'];
SAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n)) '.FZ' ...
int2str(FZP(m)) '.SA'];
FYN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n)) '.FZ' ...
int2str(FZP(m)) '.FY'];
ETN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n)) '.FZ' ...
int2str(FZP(m)) '.ET'];
IAN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n)) '.FZ' ...
int2str(FZP(m)) '.IA'];
tireidN=[pFileName '.Cornering.' pressure '.IA' int2str(IAP(n)) '.FZ' ...
int2str(FZP(m)) '.tireid'];

eval(['FZ=' FZN ' ;']);
eval(['SA=' SAN '.*pi./180;' ]); 
eval(['FY=' FYN ' ;']);
eval(['ET=' ETN ' ;']);
eval(['IA=' IAN ' ;']);
eval(['tireid=' tireidN ' ;']);

TimeConstant=.080;
Integrator=tf(1,[TimeConstant/2 1]);
TimeDelay=inv(tf(1,[TimeConstant 1])+.00000001);
FYn=lsim(TimeDelay,smooth(FY,15),ET-ET(1));
figure(1); hold
plot(SA,smooth(FY,15), '' );
hold on;
axis([-0.02 .02 -500 500])
plot(SA,FYn,'R-');

% pFileName='Hoosier20x7_13_7inch'; D=20.5; Pa55
% Results=[.045 .045 .045 .050 .045; ...
% .060 .060 .060 .055 .055; ...  
% .075 .075 .075 .070 .070; ... 
% .085 .085 .080 .080 .080; ... 
% .080 .080 .080 .080 .080];

% pFileName='Hoosier20x7_13_7inch'; D=20.5; Pa83
% Results=[.035; .050; .060; .075; .085] 8 deg/sec = .165

% pFileName='GoodyearD2696_7inch'; D=20.0; Pa55
% Results=[.045; .060; .075; .080; .075] 8 deg/sec = .165/2
\% pFileName='GoodyearD2696_8inch'; D=20.0; Pa83
\% Results=[.030; .040; .050; .065; .070;]

\% pFileName='Dunlop_7inch'; D=20.0; Pa83
\% Results=[.035; .045; .050; .060; .060;]

\% pFileName='Dunlop_7inch'; D=20.0; Pa55
\% Results=[.040; .050; .055; .060; .065;]

t=-.5*pi:pi/10000:3.5*pi;
Input=sawtooth(t,.5);
SAp=Input*12*pi/180;
t=t/pi*3;
SRp=0;
FZp=mean(FZ);
IAP=IAP(n);
\[FXp,FYp,Mxp,My,p,Mzp,Fx0p,Fy0p,Mz0p\] = pac2002(COEFF,'unity', ...
SAp,SRp,FZp,IAp);
FYm=lsim(Integrator,FYp,t-t(1));
figure();
plot(SAp,FYm, '' ,SAp,FYp, '' ,SA-.002,FY, '' );
legend('Filtered Pacejka Model','Steady State Pacejka Model','Raw Data');
xlabel('Slip Angle (rad)');
ylabel('FY (Newton)');
title([tireid ' - Application of Filter Method']);

\% Thermal test data sourced - slip velocity = 8.0 deg/sec
Test='Test3';
clear FZN SAN FYN ETN IAN
FZN=[pFileName '.Cold2Warm.' Test '.FZ'];
SAN=[pFileName '.Cold2Warm.' Test '.SA'];
FYN=[pFileName '.Cold2Warm.' Test '.FY'];
ETN=[pFileName '.Cold2Warm.' Test '.ET'];
IAN=[pFileName '.Cold2Warm.' Test '.IA'];
eval(['FZt=' FZN ' ;']);
eval(['SAt=' SAN '.*pi./180;']);
eval(['FYt=' FYN ' ;']);
eval(['ETt=' ETN ' ;']);
eval(['IAt=' IAN ' ;']);

TimeConstant8=.080;
Integrator8=tf(1,[TimeConstant8/2 1]);
TimeDelay8=inv(tf(1,[TimeConstant8 1])+.00000001);
FYtn=lsim(TimeDelay8,smooth(FYt,20),ETt-ETt(1));
figure(3); hold off;
plot(SAt,smooth(FYt,20), '');
figure(4); hold off;
plot(SAt,smooth(FYt,20), '');
figure(3); hold on;
axis([-0.02 .02 -500 500])
plot(SAt,FYtn, 'R-');
figure(4); hold on;
axis(axis);
plot(SAt,FYtn, 'R-');

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FZpt=mean(FZt);
IAPt=IAP(n);
% COEFF.pKy1=COEFF.pKy1*1.4;

[FXpt,FYpt,MXpt,MYpt,MZpt,Fx0pt,Fy0pt,Mz0pt] = pac2002(COEFF,'unity',...  
SAp,SRp,FZpt,IAPt);
t=-.5*pi:pi/10000:3.5*pi;
t=t/pi*1.5;
FYmt=lsim(Integrator8,FYpt,t(1));
figure(5);
plot(SAp,FYmt,
SAp,FYpt,

%% Relaxation Length Fitting Utility - Longitudinal Edition
% Tire test run and pressure selection for curve fitment.
% Pacejka fitting routine is Pac2002 MSC.Adams version.
% DATA MUST BE IN ISO TIRE SYSTEM!
% Calspan data is in SAE tire system natively.

clear all;
close all;

% Pacejka 2002 Adams Curve Fitting Utility
% Round 4 FSAE TTC data sectored utilizing data wrangler.

pFileName='GoodyearD2696_7inch'; D=20.0;
pFileName='GoodyearD2696_8inch'; D=20.0; % No Pa55
pFileName='Hoosier20x6_13_6inch'; D=20.5;
pFileName='Hoosier20x6_13_7inch'; D=20.5;
pFileName='Hoosier20x7_13_6inch'; D=20.5;
pFileName='Hoosier20x7_13_7inch'; D=20.5;
pFileName='Hoosier20x7_13_8inch'; D=20.5;
pFileName='Hoosier20x75_13_7inch'; D=20.0; % No Pa55
pFileName='Hoosier20x75_13_8inch'; D=20.0; % No Pa55
pFileName='Dunlop_7inch'; D=(2*175*.505+13*25.4)/1000*2/.0254;
pFileName='Dunlop_8inch'; D=(2*175*.505+13*25.4)/1000*2/.0254;
pFileName='Michelin_7inch'; D=(2*160*.53+13*25.4)/1000*2/.0254;
pFileName='Michelin_8inch'; D=(2*160*.53+13*25.4)/1000*2/.0254;

eval(['load ' pFileName '.mat;'])
load Hoosier20x6_13_6inch_SCALE.mat  
SCALE=Hoosier20x6_13_6inch_SCALE;

% pressure run for fitting
% Possibilities are [Pa55 Pa69 Pa83 Pa97 Pa83COLD]
pressure='Pa83';

eval(['load ' pFileName '_Pac2002_' pressure '.mat;'])
eval(['COEFF=' pFileName '_Pac2002;'])

% Lateral Force - Pure Side Slip - Initial Coefficient Definition

% Inclination Angle Runs
IAP=[0 2 4];
% Normal Load Runs
FZP=[222 667 1112 1557];
% IA Index
n=1;
% FZ Index
m=3;

% Cornering load averaged from the test given will be used.
% The middle loading of 667 N (150 lbf) will be utilized as pFz0.
FZN=['pFileName '.BrakeDrive0SA.' pressure '.IA int2str(IAP(n)) '.FZ'
int2str(FZP(m)) '.FZ'];
SLN=['pFileName '.BrakeDrive0SA.' pressure '.IA int2str(IAP(n)) '.FZ'
int2str(FZP(m)) '.SL'];
FXN=['pFileName '.BrakeDrive0SA.' pressure '.IA int2str(IAP(n)) '.FZ'
int2str(FZP(m)) '.FX'];
ETN=['pFileName '.BrakeDrive0SA.' pressure '.IA int2str(IAP(n)) '.FZ'
int2str(FZP(m)) '.ET'];
IAN=['pFileName '.BrakeDrive0SA.' pressure '.IA int2str(IAP(n)) '.FZ'
int2str(FZP(m)) '.IA'];
tireidN=['pFileName '.BrakeDrive0SA.' pressure '.IA int2str(IAP(n)) '.FZ'
int2str(FZP(m)) '.tireid'];

eval(['FZ=' FZN ';']);
eval(['SL=' SLN ';']);
eval(['FX=' FXN ';']);
eval(['ET=' ETN ';']);
eval(['IA=' IAN ';']);
eval(['tireid=' tireidN ';']);

TimeConstant=.095;
Integrator=tf(1,[TimeConstant/2 1]);
TimeDelay=inv(tf(1,[TimeConstant 1])+.00000001);
Time1=0:.02:max(ET-ET(1)+.5);
Time=Time1(1:length(ET));
FXn=lsim(TimeDelay,smooth(FX,25),Time);
figure(1); hold off;
plot(SL,FX,"'");
AXIS=axis;
hold on;
% AXIS=axis;
% AXIS(1)=-.05; AXIS(2)=-AXIS(1);
% axis(AXIS);
plot(SL,FXn,"R-'");
axis(AXIS);

% pFileName='Hoosier20x7_13_7inch'; D=20.5; Pa83
% Results=[.025; .065; .095; .125] IA=0

SAp=0;
SLp=smooth(SL,15);
FZp=mean(FZ);
IAp=IAP(n);
[FXp,FYp,MXp,MYp,MZp,Fx0p,Fy0p,Mz0p] = pac2002(COEFF,'unity',...
    SAp,SLp,FZp,IAp);
FXm=lsim(Integrator,FXp,Time);
figure();
plot(SLp,FXm,"'",SLp,FXp,"',SL-.001,FX,'');
legend('Filtered Pacejka Model','Steady State Pacejka Model','Raw Data');
xlabel('Slip Angle (rad)');
ylabel('FY (Newtons)');
title(['tireid ' - Application of Filter Method']);