ESCUELA TÉCNICA SUPERIOR DE INGENIERÍA (ICAI)

GRADO EN INGENIERÍA ELECTROMECÁNICA

Especialidad Mecánica

ESTUDIO DE LA GEOMETRÍA DE LA SUSPENSIÓN FRONTAL DEL TROPHY TRUCK DE VILDOSOLA RACING

Autor: José Antonio López-Monís Yuste
Director: Dr. Matthew McGarry

Madrid
Julio 2018

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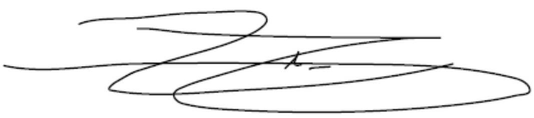

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ESTUDIO DE LA GEOMETRÍA DE LA SUSPENSIÓN FRONTAL DEL TROPHY TRUCK DE VILDOSOLA RACING

Autor: **López-Monís Yuste, José Antonio**

Director: McGarry, Matthew

Entidades colaboradoras:

- Vildosola Racing Team
- Shiley-Marcos School of Engineering , University of San Diego
- ICAI, Universidad Pontificia Comillas

Vildosola Racing Team es una empresa de carácter familiar fundada en 1968 que se dedica a la alta competición de vehículos todoterreno. Pese a ser un equipo compuesto en su gran mayoría por personal de origen mejicano, tiene su base en San Diego, posición estratégica que le aporta una gran ventaja a la hora de participar es una de las competiciones mas exigentes del mundo que sucede en su mayoría en Baja California: the SCORE international. Este equipo ha logrado ser el primero de origen mejicano en triunfar en las diferentes categorías de esta copetición, proclamandose campeones en varias ocasiones en las carreras Baja 1000, Baja 500 y Baja 250.

El objetivo de este proyecto consiste en realizar un estudio centrado en la geometría de la suspensión frontal del coche principal de la empresa, el Trophy Truck, y optimizar dicha suspensión con el fin de minimizar los problemas expuestos por Vildosola.

Hay que tener en cuenta que los resultados de este estudio se implementarán en el nuevo vehículo insignia de la marca, que se encuentra en periodo de diseño y podrá ser visto en la próxima temporada. El actual Trophy Truck resulta ser uno de los principales candidatos a la victoria en la competición debido al excelente trabajo del equipo de mecánicos que han conseguido reducir el peso total del vehículo en un 10%. Sin embargo, esta ventaja supone conlleva también un inconveniente puesto que la reducción de peso afecta a la tracción del vehículo, que se ve disminuida, incrementando así el desgaste de los neumáticos. En carreras de corta duración este hecho no supone un problema. Sin embrago, en competiciones como la SCORE international, en donde hay carreras de hasta 1000 millas (1600 km), un desgaste excesivo de los neumáticos conlleva aumentar el número de paradas y perder tiempo con respecto a los competidores, con lo que el desgaste de los neumáticos es un factor importante a tener en cuenta.

Sin embargo, el aspecto principal que la geometría de la suspensión frontal controla es el manejo del coche. Esto es la respuesta física del vehículo frente al aporte del piloto. Con la configuración actual de la suspensión frontal del Trophy Truck, el piloto, Tavo Vildosola, consigue realizar una curva fácilmente. Sin embargo, durante los tramos rectos el coche peca de una extrema sensibilidad ya que, a cualquier movimiento del volante, el vehículo tiende a girar. Este hecho unido a la ligereza del biplaza conlleva una excesiva inestabilidad del vehículo.

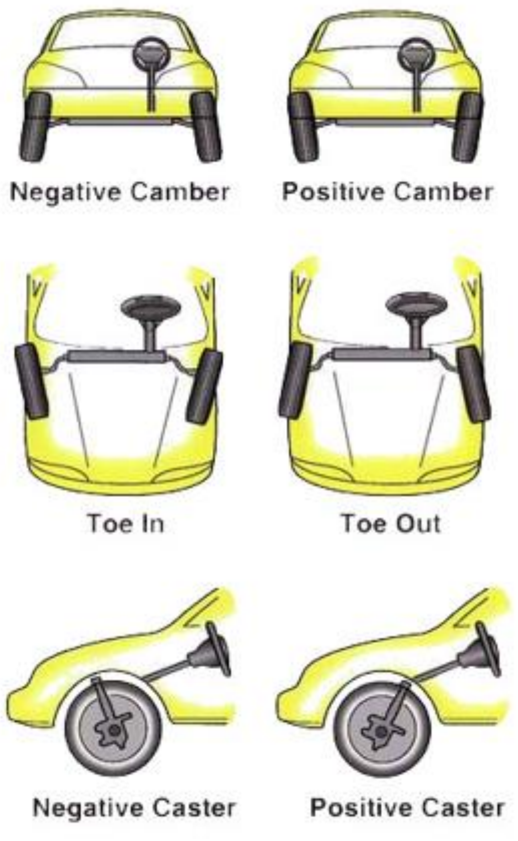
Este estudio analiza diferentes configuraciones de la suspensión frontal del vehículo expuesto con el fin de ayudar a Vildosola Racing a construir un nuevo Trophy Truck que minimice las debilidades del vehículo y se adapte en mayor medida a las exigencias del piloto facilitando así su labor.

El proyecto se centra en los siguientes aspectos de la suspensión frontal del vehículo:

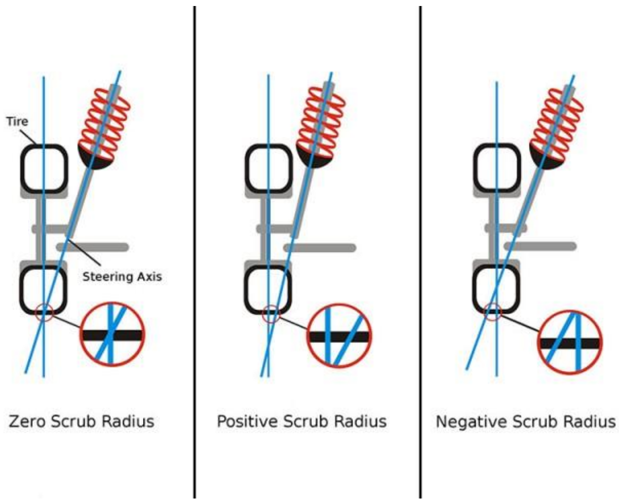
- Ángulo de caída (camber angle): ángulo que forma el eje de simetría de la rueda respecto a una vertical en la intersección entre el propio eje de simetría y el suelo.

- Ángulo de avance (caster angle): ángulo entre el eje que une la suspensión a la rueda y una vertical desde el centro de la rueda.

- Ángulo de convergencia (toe angle): ángulo entre la dirección de la rueda y la dirección del vehículo cuando éste marcha en línea recta.



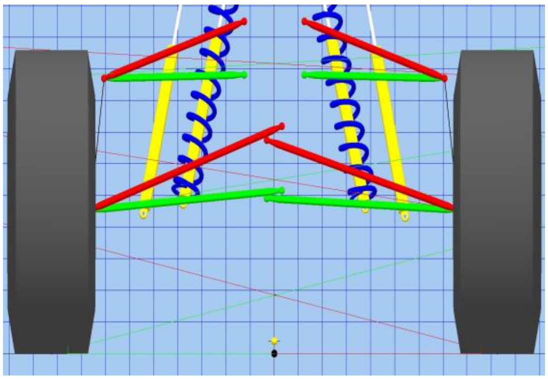
- Radio de pivotamiento (scrub radius): distancia entre la intersección de la línea imaginaria que prolonga el eje de pivotamiento y la intersección entre la línea de simetría de la rueda y el suelo.



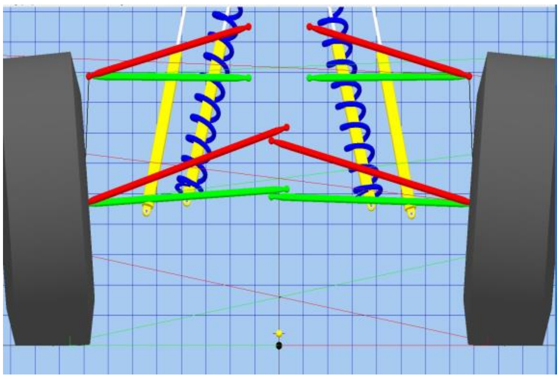
Con el fin de buscar aquellos ángulos óptimos se realizan en este estudio dos diferentes pruebas, una empleando un software de simulación y otra empleando un modelo real de una suspensión frontal.

Software de simulación

En este proyecto se emplea el software de simulación SusProg 3D que permite estudiar el comportamiento dinámico de la suspensión en función de los diferentes ángulos de la geometría. Así, se han estudiado varios casos en los cuales se introducen diferentes valores para los ángulos comentados previamente.



Ángulo de caída nulo constante
Ángulo de avance variable



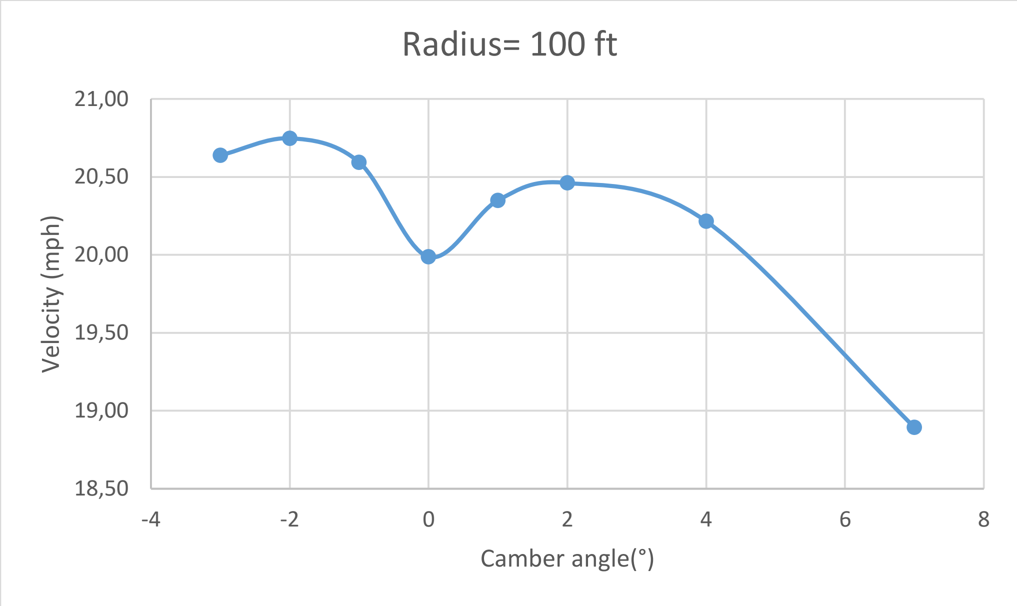
Ángulo de caída variable
Ángulo de avance nulo constante

Modelo real de una suspensión delantera

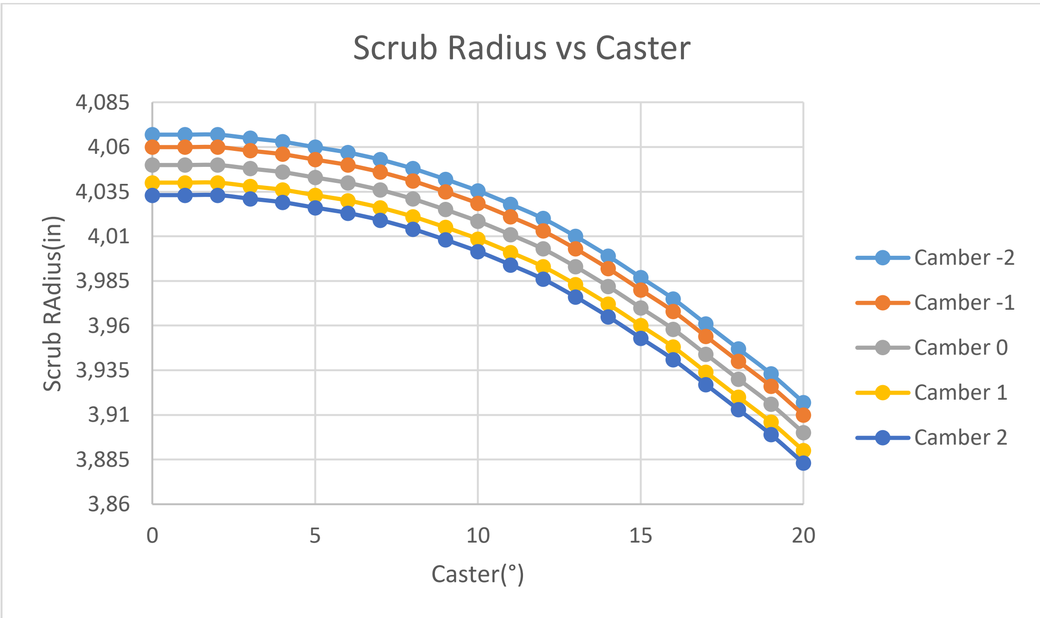
Mediante el uso de un modelo real de una suspensión frontal de un vehículo similar al Trophy Truck se ha estudiado el efecto del ángulo de caída en el comportamiento del coche durante las curvas. Así, para una curva de un radio determinado, se ha calculado la velocidad máxima de paso del vehículo en función del ángulo de caída, valor que depende de la fuerza lateral máxima que soporta la suspensión antes de patinar.

Resultados

En primer lugar, empleando el experimento llevado a cabo con el modelo de la suspensión se ha obtenido la velocidad máxima de paso por curva en función del ángulo de caída a partir de la fuerza lateral límite que soporta el sistema antes de deslizar. Así pues, cómo se puede observar en la siguiente gráfica, resulta favorable introducir al sistema un ángulo de pequeño valor preferiblemente negativo que maximiza la velocidad en cuerva, favoreciendo además la conservación de los neumáticos.



A partir de las diferentes simulaciones llevadas a cabo con SusProg 3D, se han obtenido diferentes gráficas de resultados en las cuales se observa el efecto que conlleva el cambio en el valor de alguno de los parámetros de la suspensión respecto a los demás. En ellas podemos observar que a medida que se aumenta el ángulo de avance, el radio d pivotamiento disminuye, favoreciendo así la estabilidad del vehículo. Sin embargo, valores muy altos del ángulo de avance se ven reflejados en una suspensión muy poco sensible, hecho que tampoco es deseable.



OPTIMIZATION OF THE GEOMETRY OF THE FRONT SUSPENSION OF THE TROPHY TRUCK OF VILDOSOLA RACING

Author: **López-Monís Yuste, José Antonio**

Director: McGarry, Matthew

Collaborating Entities:

- Vildosola Racing Team
- Shiley-Marcos School of Engineering , University of San Diego
- ICAI, Universidad Pontificia Comillas

Vildosola Racing Team is a family company founded in 1968 that is dedicated to the high competition of off-road vehicles. Despite being a team composed mostly of personnel of Mexican origin, is based in San Diego, a strategic position that provides a great advantage when participating is one of the most demanding competitions in the world that happens mostly in Baja California: the SCORE international. This team has managed to be the first of Mexican origin to triumph in the different categories of this match, proclaiming themselves champions in several occasions in the races Baja 1000, Baja 500 and Baja 250.

The objective of this project is to conduct a study focused on the geometry of the front suspension of the company's main car, the Trophy Truck, and optimize this suspension in order to minimize the problems exposed by Vildosola.

Keep in mind that the results of this study will be implemented in the new flagship vehicle of the brand, which is in design period and will be seen in the next season. The current Trophy Truck turns out to be one of the main candidates for victory in the competition due to the excellent work of the team of mechanics who have managed to reduce the total weight of the vehicle by 10%. However, this advantage also entails a disadvantage since the reduction of weight affects the traction of the vehicle, which is diminished, thus increasing the wear of the tires. In short races this fact is not a problem. However, in competitions such as SCORE international, where there are races of up to 1000 miles (1600 km), excessive tire wear leads to an increase in the number of stops and loss of time with respect to other participants, so it is necessary to consider the tires as an important factor.

However, the main aspect that the geometry of the front suspension controls is the handling of the car. This is the physical response of the vehicle against the contribution of the pilot. With the current configuration of the front suspension of the Trophy Truck, the driver, Tavo Vildosola, manages to make a curve easily. However, during the straight sections the car sins of an extreme sensitivity since, to any movement of the steering wheel, the vehicle tends to turn. This fact coupled with the lightness of the two-seater leads to excessive vehicle instability.

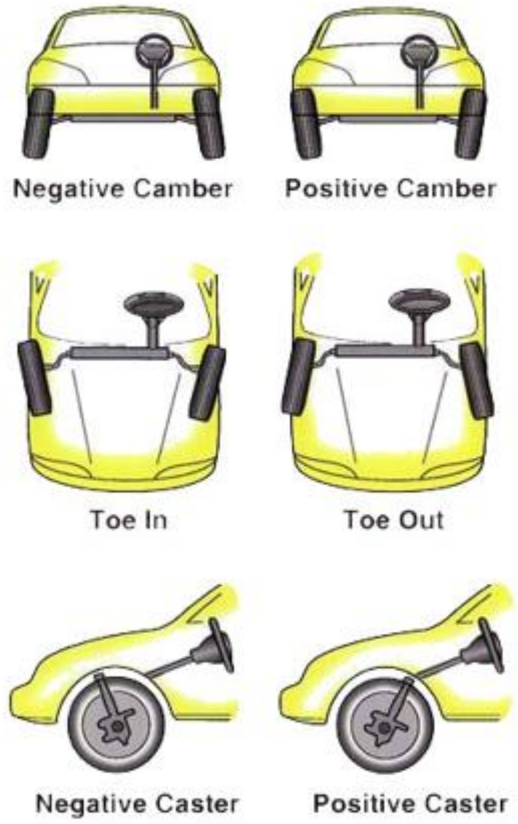
This study analyzes different configurations of the front suspension of the exposed vehicle in order to help Vildosola Racing to build a new Trophy Truck that minimizes the weaknesses of the vehicle and adapts it in order to solve the demands of the pilot.

The project focuses on the following aspects of the front suspension of the vehicle:

• Camber angle: An angle formed by the axis of symmetry of the wheel with respect to a vertical axis at the intersection between the axis of symmetry and the ground.

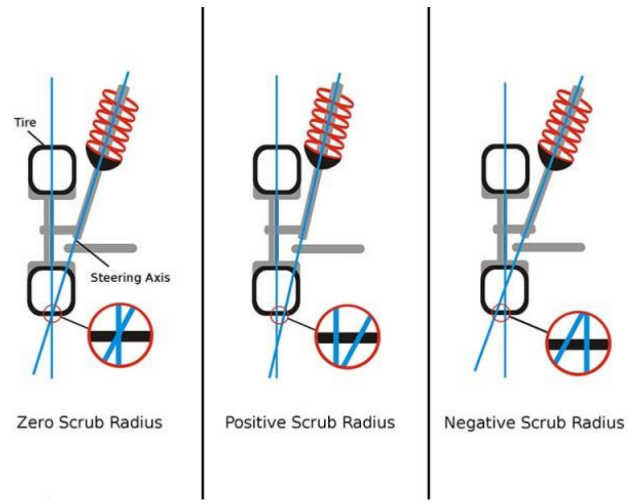
• Caster Angle: angle between the axis that joins the suspension to the wheel and a vertical from the center of the wheel.

• Toe angle: angle between the direction of the wheel and the direction of the vehicle when it is running in a straight line.



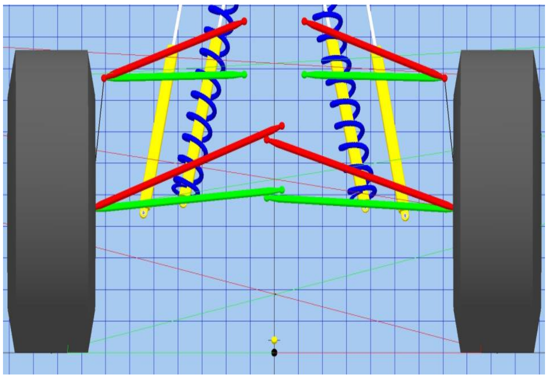
Scrub radius: distance between the intersection of the imaginary line that extends the direction axis and the intersection between the line of symmetry of the wheel and the ground.

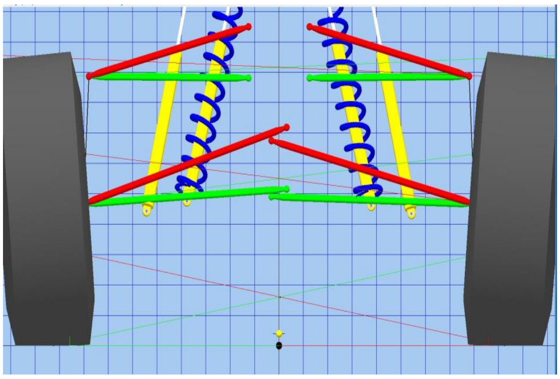
In order to find the optimal angles, two different tests are carried out in this study: one using simulation software and the other using a real model of a front suspension.



Simulation software

In this project, SusProg 3D simulation software is used to study the dynamic behavior of the suspension depending on the different angles of the geometry. Thus, several cases have been studied in which different values are introduced for the angles previously commented.





Angle of constant null fall
Angle of variable fall

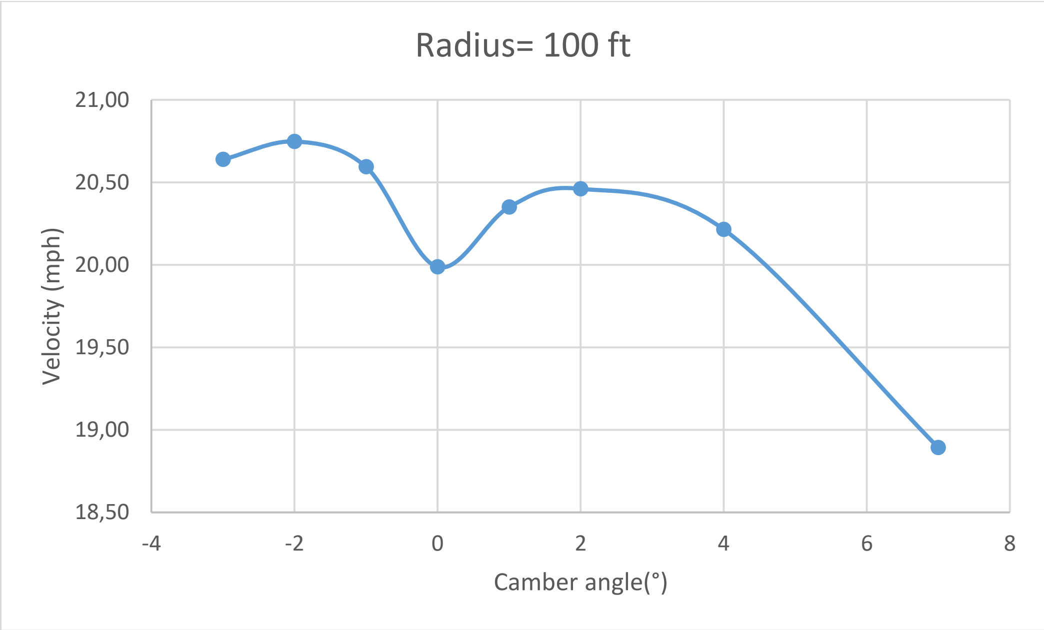
Variable advance angle
Constant null advance angle

Real model of a front suspension

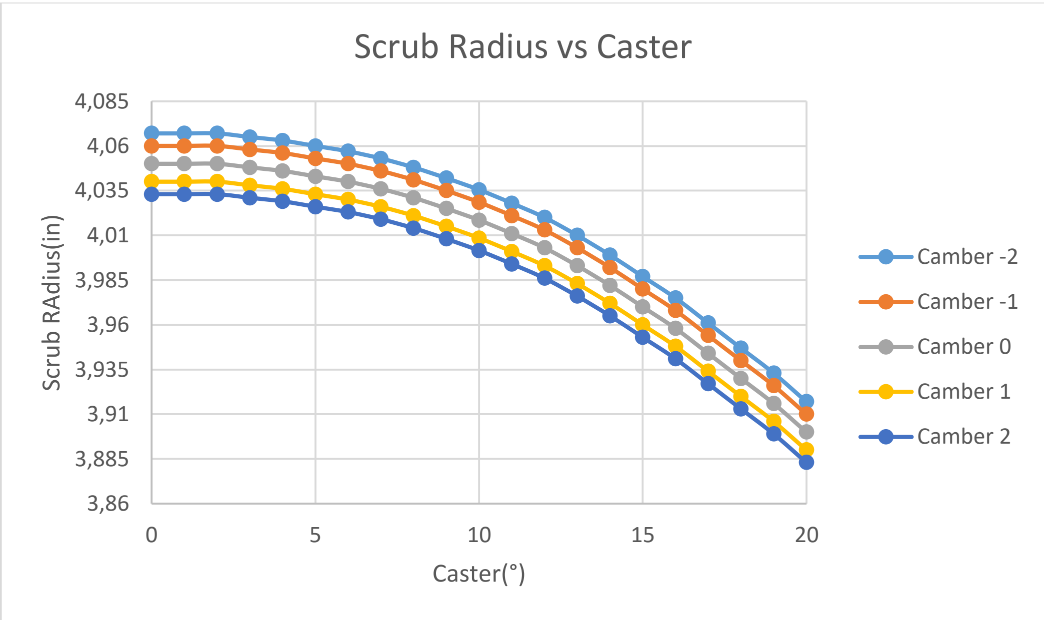
Through the use of a real model of a frontal suspension of a vehicle similar to the Trophy Truck, it has been studied the effect of the caber angle in the behavior of the car during the curves. Thus, for a corner of a certain radius, the maximum speed of the vehicle has been calculated as a function of the camber, a value that depends on the maximum lateral force that the suspension supports before skidding.

Results

In first place, using the experiment carried out with the suspension model, the maximum speed of passage through the curve has been obtained as a function of the camber from the limit lateral force that the system supports before sliding. Therefore, as can be seen in the following graph, it is favorable to introduce to the system an angle of small value preferably negative that maximizes the speed in the crow, also favoring the conservation of the tires.



From the different simulations carried out with SusProg 3D, different graphs of results have been obtained in which the effect of the change in the value of some of the parameters of the suspension with respect to the others is observed. On them, we can see that as the caster is increased, the scrub radius decreases, thus favoring the stability of the vehicle. However, very high values of the caster angle are reflected in a suspension that is not very sensitive, which is also not desirable.



Abstract

The Vildosola Racing Team seeks to gain a competitive advantage by improving the drivability and tire wear of their Baja Trophy Truck. Our team will determine the effect of front suspension geometries have on the overall vehicle handling and tire wear. Through various experiments of our suspension model, we can determine where improvements can be made, and suggest these designs to Vildosola.

We seek to model the Vildosola Trophy Truck, by using a front suspension kit model to analyze the effects of the camber and caster. This model will allow us to change the various suspension geometries and perform tests to determine the optimal configuration for most improved handling and minimal tire wear.

Our project also includes computer simulation data using the Solidworks model of the custom trophy truck suspension. Our work will build off of the continued efforts of the Vildosola team, as well as the conclusions of the previous two USD Senior Design teams.

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1 Context

1.1 Background

1.1.1 Background of need

The customers of this project are Vildosola Racing a professional SCORE Trophy Truck #21 off-road racing team based in San Diego. It is owned and managed by USD alumnus Gustavo Vildosola, Jr. ’06 and his family. Our project’s aim is to try to improve drivability; traction and tire wear on the company’s “trophy trucks,” which compete in various races in Baja California. The stakeholders are USD Advisors: Dr. McGarry and Dr. Codd, Vildosola Racing Owners, the mechanics, and advertisers.

The current truck specifications are:

1. Class: Trophy Truck
2. Make: Ford
3. Model: F-150
4. Chassis: Geiser Bros.
5. Weight: 5400 lbs
6. Engine: 454ci by Patton Racing Engines
7. Horsepower: 830 Horsepower
8. Torque: 750 ft/lb
9. Transmission: 3 Speed, TH-400 Rancho Drivetrain Engineering
10. Shocks: King 4.5” Bypass Shocks
11. Wheel Travel: 32’ Front / 36” Rear
12. Wheelbase: 122’
13. Width: 93” +/- 10%
14. Brakes: Brembo 6 Piston
15. Tires: BFGoodrich Baja T/A KR 39 x 13.5 R17
16. Wheels: 17” x 10” Method Race Wheels

The main objective of this project is to maximize the performance of the Vildosola Baja trophy truck by improving the weight ratio, suspension, and tire wear, while minimizing the possibility of breaking down during a race. For the users the goal is to upgrade the vehicle performance and handling/drivability. For the stakeholders the objective is to improve the vehicle dynamics, redesign the suspension, reduce weight, and decrease tire wear.

The user’s specific needs are:

- Improve front suspension in order to reduce weight and provide better performance; model relationships for future improvements.
- Increase traction of back tires and investigate option of AWD truck and reduce tire wear, especially for the rear tires.
- Optimize and model front to back weight ratio, taking into account weight shift during acceleration.
- Improve the drivability of the truck for ease of maneuverability in the course of the race.

We intend to further examine the following:

- Caster Angle:

A caster angle is built into the front suspension so the steering is more stable and will return to center. Camber is the angle of the tire to the road – negative camber is when the tire leans in at the top and in contrast, positive camber is when the tire to leans out at the top.

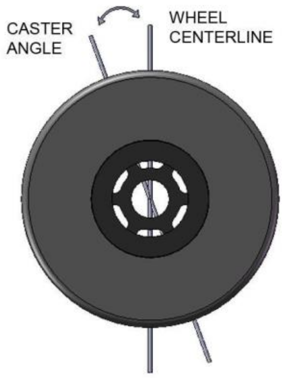


Figure 1: Caster Angle

- Camber Angle:

Camber is the angle of the tire to the road – negative camber is when the tire leans in at the top and in contrast, positive camber is when the tire to leans out at the top.

Figure 2:Camber angle

- Scrub Radius:

Scrub radius is the distance between the intersection of the imaginary line of the steering axis and the ground and the center of the contact footprint of the wheel.

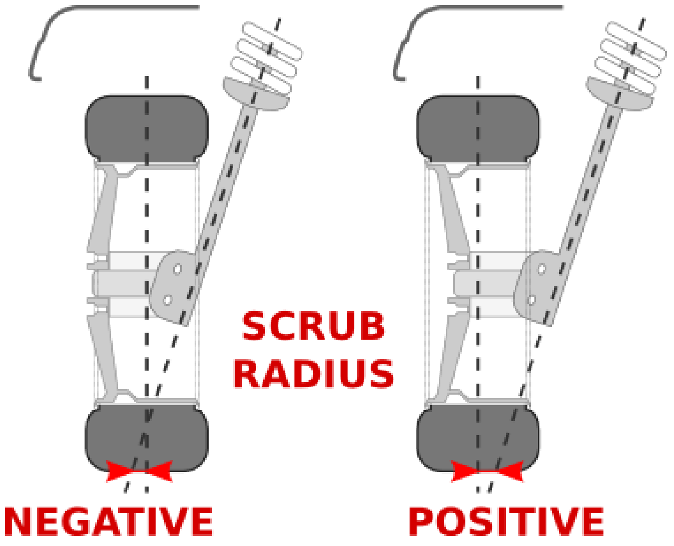


Figure 3: Scrub Radius

- Radius Arm Setup:

The radius arm design uses two arms that run parallel to the frame. They mount to a perch on the frame and solidly to the axle housing and allow the axle to pivot up and down. A track bar runs from the frame to the axle perpendicular to the radius arms to keep the axle centered on the frame. Since the radius arms are fixed at the axle end, the caster angle changes when the suspension cycles up and down, shown in the figure to the right.

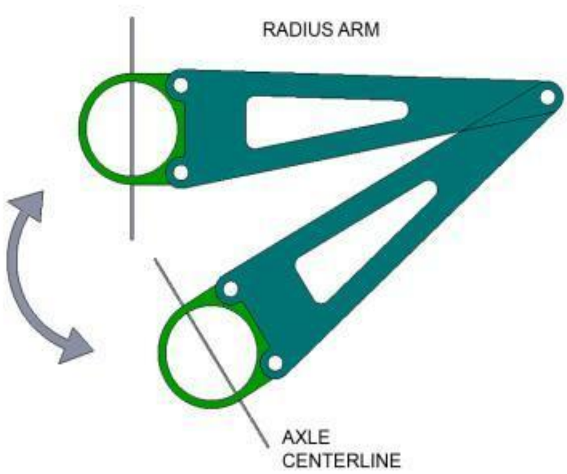


Figure 4: Radius Arms setup

- Parallel and Triangulated Four Link:

The manufacturers make kits that retrofit an existing radius arm suspension to a parallel four link design and use coil springs and a track bar to center the axle. Instead of a radius arm with a fixed mount on the axle, it uses an upper and lower link on each side with pivots on both ends. As the axle cycles up and down, the links allow it to maintain the same relationship with the ground and the caster angle remains constant.

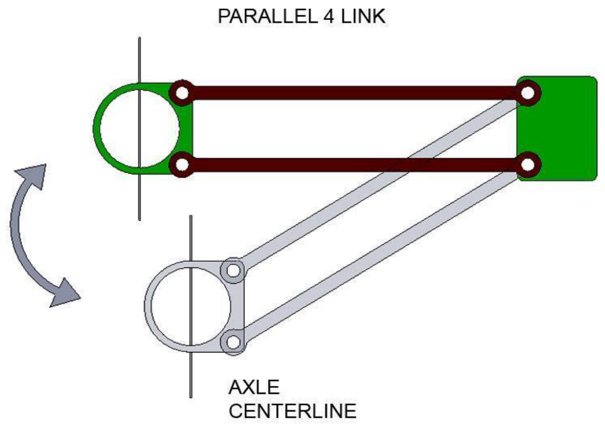


Figure 5: Parallel and Triangular Four Lin

1.1.2 Background of racing

Vildosola Racing Team is a family business and one of the most successful teams in off-road racing competition. It was founded in 1968 when Gustavo Vildosola, owner of the company, started racing in local races in Mexico.

After several years of experience in off-road racing he decided to face the Baja 1000, the most demanding truck racing in Baja California, Mexico, in 1968 with his cousins. In 1996 he built his own ProTruck and 5 years later he decided to move forward and he started his adventure on the first class, the Trophy Trucks.

It is important to understand that the SCORE Baja 1000 was dominated by North American teams until Vildosola Racing started participating. It consists on a race that's tarts in Ensenada and finishes either on La Paz or in the same place it starts.

There are also other categories of this competition as the Baja 500 or the Baja 250. This number represents the total miles the vehicles are supposed to do in the race. Since 2003, Vildosola Racing Team participates in the three races with vehicles in both the ProTruck and the Trophy truck categories.

Their first victory in a SCORE competitions was in the San Felipe 250 in 2003 with Gustavo Vildosola as a driver. In 2005 Gustavo's son, Tavo Vildosola, started his career as a driver for the company racing for the years in the ProTruck category. In 2008 he jumped into the Trophy Truck races and he achieved the first ever victory of a Mexican team in the Baja 1000 in the 43rd anniversary of the race, that took place in 2010.

Best result of Vildosola Racing team since 2010:

Year	Category	Place
2010	SCORE Baja 1000	1st
2011	SCORE Laughlin Desert Challenge	3rd
2012	SCORE Baja 1000	8th
2013	SCORE San Felipe 250	1st
2014	SCORE San Felipe 250	1st
2015	SCORE San Felipe 250	1st
2016	SCORE Baja 500	1st
2017	SCORE Baja 1000 Trophy Legend	1st
2018	SCORE Baja 500	1st

Table 1: Vildosola racing last best results

As it can be seen in the table, Vildosola Racing Teams finishes usually on the podium of at least one of the categories of the SCORE competitions. It is an honor for us as well as a big responsibility to work with such a successful team.

1.2 Customer Need Statement

The Vildosola Racing employees have presented a number of issues they are currently facing with their trophy truck including tire wear, drivability, and suspension. They would also like to locate the center of gravity of the truck so they can analyze the weight transfer during acceleration. Addressing these needs would help Vildosola Racing improve their trophy truck, and provide a competitive advantage over other competitors.

1.3 Literature Review

1.3.1 Prior Work

In previous years, senior design projects at USD have worked to improve the Vildosola trophy truck. The most recent of these projects focused on improving the aerodynamics of the truck through CFD analysis and modeling.

This year our project focuses on improving the suspension and tire wear, which have been issues Vildosola has been working on for quite some time. This problem is hard to solve because these trophy trucks are truly one of a kind. Most internet searches will result in suspension systems and off road tires for stock trucks. However, these trophy trucks are built by hand and therefore all the parts on them must be customized and unique. It would be helpful to find a technology to model the relationship between the front and rear suspension of the vehicle to clearly show the cause and effect minor changes will have, before building it. This will include a finite element analysis of the forces on each member of the truck.

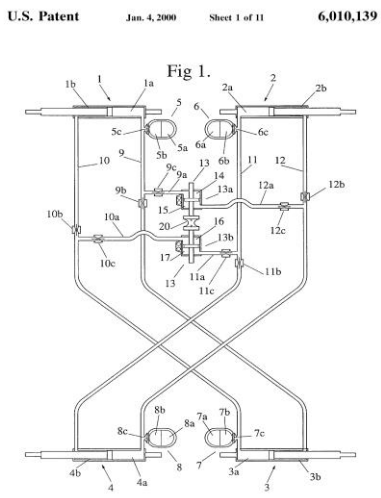
Our limitations will be apparent when we receive the Solidworks 3D model of the truck. Based on our visit to the facility, the mechanics often change things on the spot in the shop, without knowing the mathematical effect of their actions. It will be difficult to capture all these changes in the 3D model to make our calculations and analysis most accurate.

1.3.2 Patents

- **Vehicle suspension with independent pitch and roll control** (*1*)

A vehicle hydro pneumatic suspension comprising four double acting rams each between respective one of four spaced wheels at comers of the vehicle.

This invention relates to improvements in the suspension system or a vehicle, and is specifically related to controlling the disposition of the vehicle body relative to the ground when the vehicle is subject to variations in the contour of the surface being traversed.

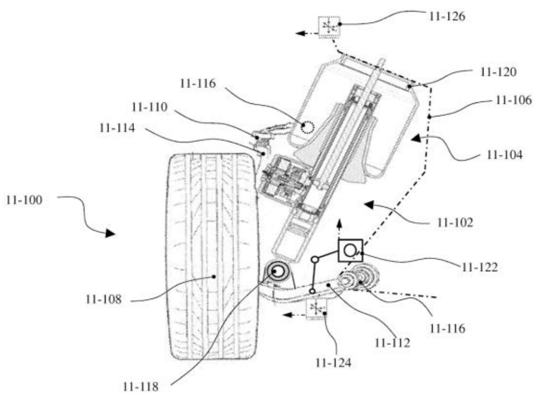


Patent number: US6010139 A
Patent holder: Christopher B. Heyring
Date of issue: 01-04-2000

Figure 6: Independent pitch and roll

- **Active vehicle suspension** *(2)*

A method of on-demand energy delivery to an active suspension system comprising an actuator body, hydraulic pump, electric motor, plurality of sensors, energy storage facility, and controller is provided. The method comprises disposing an active suspension system in a vehicle between a wheel mount and a vehicle body, detecting a wheel event requiring control of the active suspension; and sourcing energy from the energy storage facility and delivering it to the electric motor in response to the wheel event.

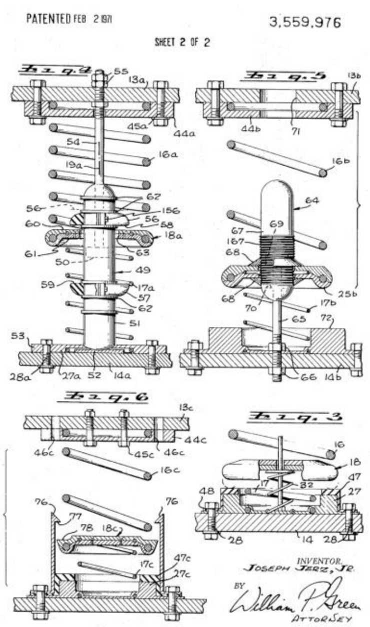


Patent holder: Zackary Martin Anderson
Patent number: US20150224845 A1
Date of issue: 08-13-2015

Figure 7: Active suspension

- **Variable stiffness suspension system** *(3)*

A vehicle suspension system including two springs connected in series, with one of the springs being stiffer than the other, and with the springs being so related that under normal load conditions the softer of the two springs is effective to provide a very gently cushioned ride, while upon the imposition of heavier load forces, the vehicle is supported more stiffly and primarily by the stronger spring. The conversion between these two conditions may be effected automatically, by engagement under heavy load conditions of a pair of stop shoulders acting to limit compression of the light spring. Similarly, upon excessive extension of the springs, an additional set of stop shoulders may automatically become effective to limit the amount of extension of the softer spring and cause the stiffer spring to resist further extension.



Patent holder: Jerz Joseph
Patent number: US3559976 A
Date of issue: 02-02-1971

Figure 8: Variable stiffness suspension

- **Tire tread wear sensor system** *(4)*

This patent can indicted a warn for user about the wear problem of tire or belt.

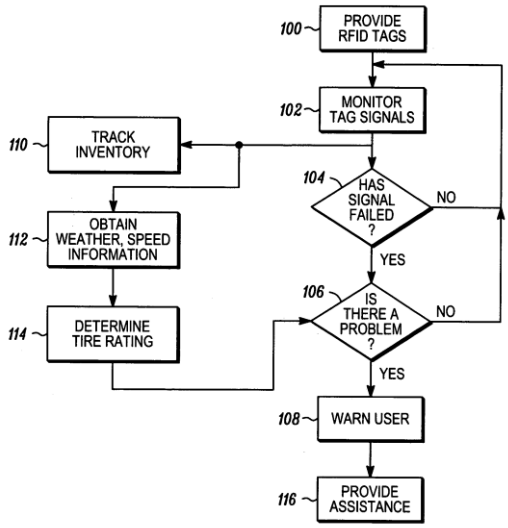


FIG. 3

The publish date: 3-11-2005
Patent number: US7180409B2
The inventor: Thomas A. Brey

Figure 9: Tread wear sensor system

- **Electromagnetic suspension system for vehicle** *(5)*

Electromagnetic suspension system, a changeover between the active control and the passive control is made. Accordingly, for example, in case that a vibration input is applied from tires during the active control for accomplishing an attitude control of the vehicle, this vibration input is also tackled by the active control to accomplish a vibration control. As a result, both the attitude control and the vibration control are required to be simultaneously accomplished under a motor control in an active manner.

FIG.3

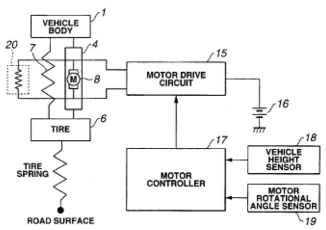


FIG.4

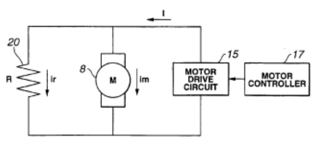
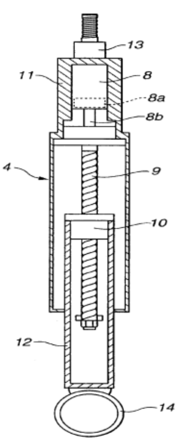


FIG.2

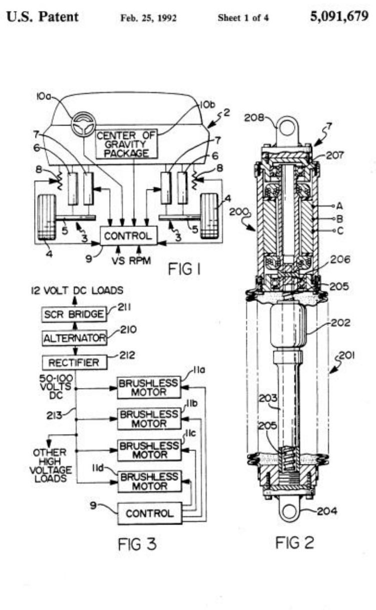


Patent number: US7005816 B2
The inventor: Koji Hio, Masaharu Sato, Takaaki Uno
The issue date: 2-2-2004

Figure 10: Electromagnetic suspension

- **Active vehicle suspension with brushless dynamoelectric actuator** *(6)*

This invention relates to a fully active suspension and ride control for a motor vehicle.



Patent number: US5091679A
Patent holder: BWI Co Ltd SA
Date of issue: 1992-02-2

Figure 11: Active vehicle suspension

2 Customer Requirements

2.1 Functional Requirements

These general functional requirements must apply to front suspension improvements:

1. Fundamental purpose of product: Improved traction and Drivability (H)
2. Performance: Modifying camber & caster angles to improve traction, which will result in a more drivable vehicle. (H)
3. Reliability
4. Serviceability (M)
5. Safety (M)
6. Reduce weight by structural analysis (M)
7. Analysis of suspension (H)
8. Tires should last for at least 80 miles (H)
9. Absorb road bumps equivalent to best-in-class competitors (H)
10. Improve the probability of the truck to have tire wear issues to 20% (H)

2.2 Physical Requirements

The following physical requirements must apply to the front suspension improvement:

Angle variations:

Caster $\pm 8\%$
Camber$\pm 3\%$
Toe $\pm 3\%$⅛ and 1/16 inches

1- the modification of the Caster $\pm 8\%$, Camber$\pm 3\%$, Toe $\pm 3\%$, and Scrub 4 %(H)
2- improve the tire wear by 10% (H)
3- visually appealing (L)

2.3 Assumptions

1. Small suspension modifications will not affect the cabin security.
2. Vildosola´s mechanicals will follow our recommendations.
3. Improving the suspension will improve tire wear.
4. The truck specifications will not change during the development period.
5. The project will be working with the initial data pf the truck unless Vildosola requires to update the truck specifications if they change.
6. The truck is based on the model we have been given.
7. Client will be trainable in Solidworks and CREO software after purchasing the license.
8. The team will have access to the trophy truck for modeling purposes.
9. Model will be viewed on a computer that supports the software used.
10. The car will be used for the BAJA competition, in a similar environment to previous years, and will be facing similar conditions.
11. The materials used will behave as determined by their specifications.
12. Trophy truck will exist in a simple 3-dimensional state such that it may be modeled.
13. The number of right and left turns will be similar in all the races.

2.4 Constraints

1. Project must be finished within 7 months.
2. Limited testing time as applying changes to the truck and taking it to a testing facility takes time.
3. Budget must be affordable and under $7000.
4. The front suspension geometry design should be less or equal to the current weight.
5. Limited options of the physical testing.
6. Limited knowledge of the team in this field.

3 Concept Development

3.1 Synthesis and Analysis of Overall Concepts

3.1.1 Types of off road truck's front suspensions

3.1.1.1 Solid axle

The solid axle front suspension is the type of suspension which most manufacturers have used for decades. It is bases on a simple axle that crosses the truck from one front wheel to the other one.

The principle advantage this suspension has is that the configuration allows enough space for leaf springs, which have a double task. First, it absorbs the different inputs from the ground without a big dependence on the weight of the truck due to the own configuration of the spring. Although most of the springs have a linear spring rate, the design of this device allows an increasing spring rate during its compression. The other main job of this design is that the spring fix the position of the axle.

Figure 12: Solid axle suspension truck

3.1.1.2 Ford Twin Traction Beam

The Twin Traction Beam of TTB is a Ford design that uses both the solid and part independent suspension configurations. The main difference between this suspension and the solid axil one is that this device carries a joint in the center of the truck that allows the relative movement between both axles and wheels.

The joint in the middle creates a pivoting movement of the wheels that causes a change on the camber angle, what could damage the performance of the vehicle. However, this design increases the strength of the suspension due to the length of the beams. The forces are spread out through them and it reaches a higher performance than A-Arms.

Figure 13: Ford twin Traction Beam truck

3.1.1.3 Fully independent Suspension

Although rear independent suspension is not commonly used, front independent suspension is one of the favorite configurations for competition off road vehicles. It is based on a system of struts, coils and torsion bars that suspend the chassis of the car.

The main advantage of this configurations is its small volume. However, this could also be the biggest downside due to the friction between all the parts of the suspension who try to occupy the same space, what could dramatically increase the wear of the pieces.

It is usually used a configuration in which the length of the A-Arms is different, where the upper A-Arm is shorter than the bottom Arm in order to maintain the camber angle almost stable in the desired position. The caster angle is fully fixed by the A-Arm configuration.

Figure 14: Fully independent suspension truck

3.1.1.4 Double wishbone suspension

The double wishbone suspension is an independent design built with two arms that locate the wheel. This configuration can also be named as “Double A-Arm suspension”. Each A-Arm has its own joint with the chassis.

The main advantage of this type of suspension is the possibility of carefully controlling all the aspects of the dynamics of the own suspension by letting the mechanics modify the different parameters of the suspension (caster, camber, toe angle…).

The current Trophy Truck of Vildosola uses this type of front suspension configuration due to the simple design, which reduces the cost, combined with the possibility of adapting the angles to a wide range of values.

Figure 15: Double wishbone suspension truck

3.1.2 Functional Decomposition

Our functional decomposition diagram categorizes the various solutions to our problem statement

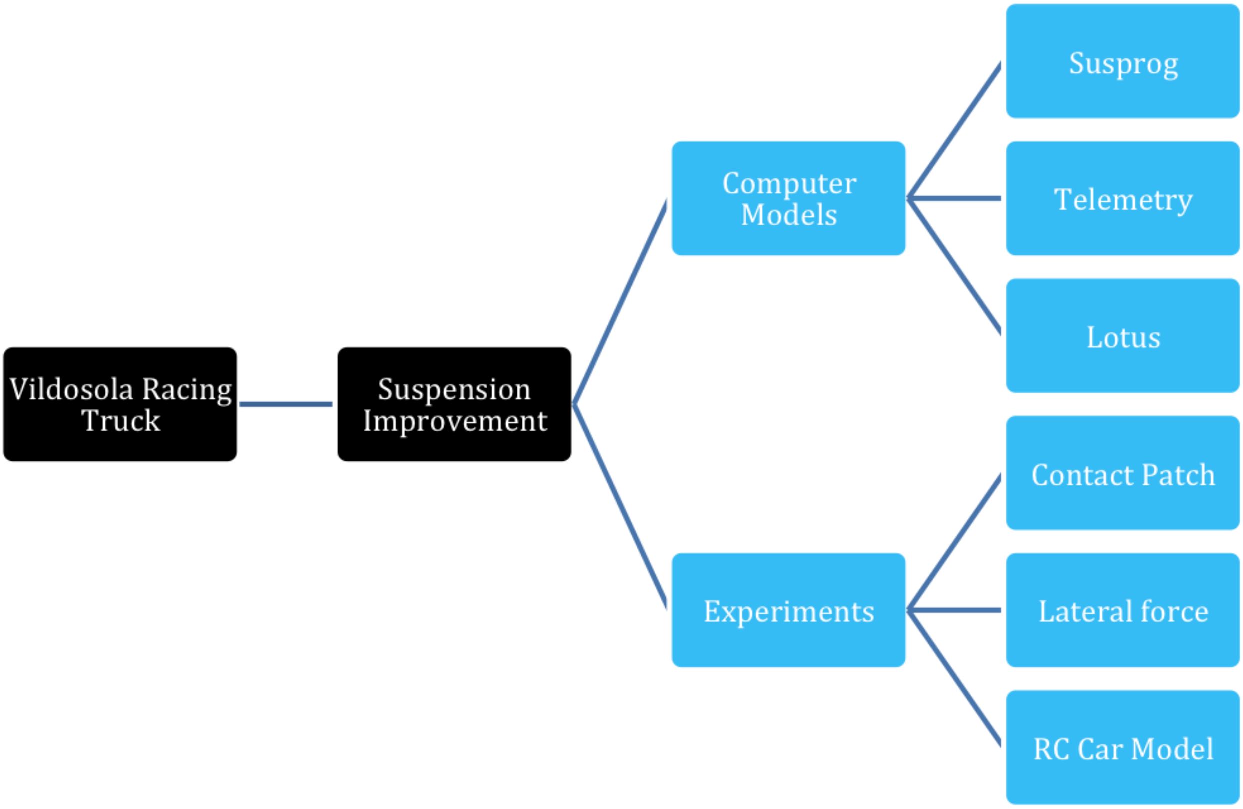


Table 2: Functional decomposition

3.1.3 Computer modeling

3.1.3.1 SusProg3D

SusProg3D software analysis that the premier kinematic Suspension by Design is a complete software package enabling the design, evaluation and visualization of race and road car suspension characteristics.

- Camber and caster adjustment location on top, bottom or both links.
- Store and recall suspension design data.
- Display of camber, roll centre, castor and wheel scrub in roll and bump.
- Weight distribution

So, it obtained the variation of camber and caster. when the car is rolling to the right, when the shocks are being compressed and when the shocks are being decompressed.

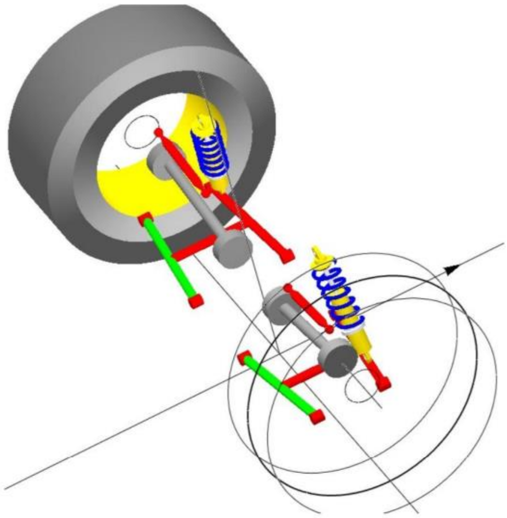


Figure 16: SusProg 3d picture

3.1.3.2 Telemetry Software MoTec i2 Pro

Figure 17: MoTec i2 Pro picture

Using sophisticated analysis software and data collected from a variety of sensors, the behavior of a vehicle can be comprehensively investigated, recorded and analyzed. This would include information about temperature, speed, acceleration, strain and movement. This software lets users gain valuable insights into performance and reliability, resulting in more efficient testing and tuning and greater predictability on race.

3.1.3.3 Lotus Software

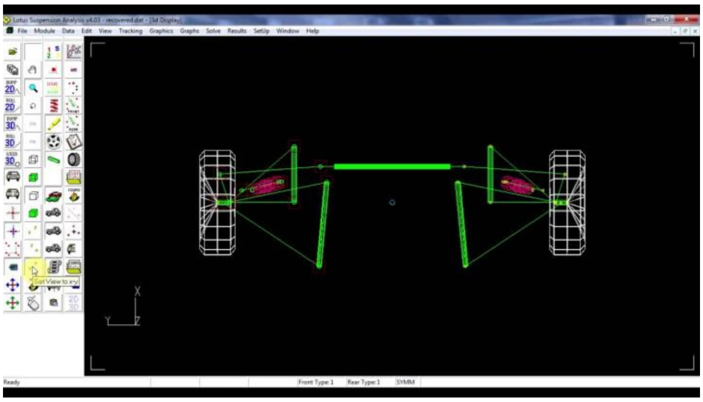


Figure 18: Lotus software picture

Lotus is used by both designers and analysts alike for the layout of the suspension hard point positions, in order that the required kinematic behavior is achieved. Any number of results can be displayed graphically, (e.g. Camber angle, Toe angle), against bump motion, roll motion or steering motion. These results are updated in ‘real time’ as the suspension hard points are moved. The inclusion of compliant bushes to the kinematic model allows the tuning of bush properties to be carried out, to achieve required compliant response for items such as lateral force steer.

3.1.4 Physical experiments

3.1.4.1 Contact patch

A contact patch is used for measuring the pressure between the ground and the tire. There are two different possibilities for this experiment:

- Straight path: we can see how the different suspension geometry affects the tire wear.
- Corners: tires do not behave in the same way in straight and turning lines. During corners, the pressure increases on the external surface of the tire generating a non-symmetric wear over it.

This experiment consists on studying the different pressures between the tire and the road during both straight and corner truck´s behavior. The expected outcome will help us to decrease the tire wear by modifying the front suspension geometry.

Figure 19: Contact surface tyre-ground

3.1.4.2 Lateral force

Lateral or corner force counters inertial forces during turnings. This means that the capacity of turning is directly related with this type of force. The experiment proposed consists on a partial front suspension assembly loaded with a normal force which represents the inertial force of the truck and a load that represents the truck mass. The measurement of the force will be made by strain gauges located between both surfaces.

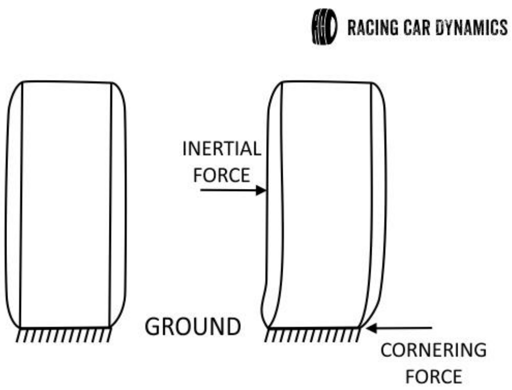


Figure 20: Lateral force over the tyre

3.2 Governing Principles

- Solid mechanics / FEA
- Kinematics / Dynamics
- Mechanical behaviour
- Factor of safety
- Machine Design
- Tribology (bearings and lubrication)

3.3 Analysis

3.3.1 Front Suspension Geometry

The Following analysis illustrate the front suspension Geometry calculation by dividing into front view and side view

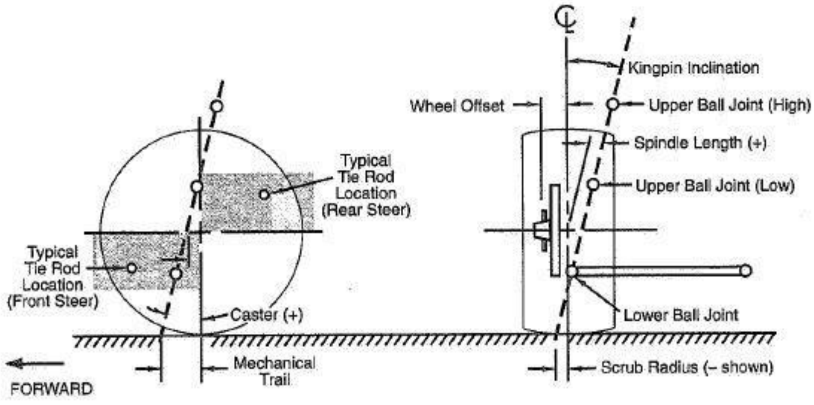


Figure 21: Front suspensión geometry

a. Front View Swing Arm Geometry (FVSA)

The front view arm instant center controls the roll center height (RCH), the camber change rate, and tire lateral scrub.

The Instant Center can be located ground level or below ground. The location is up to designer performance requirement

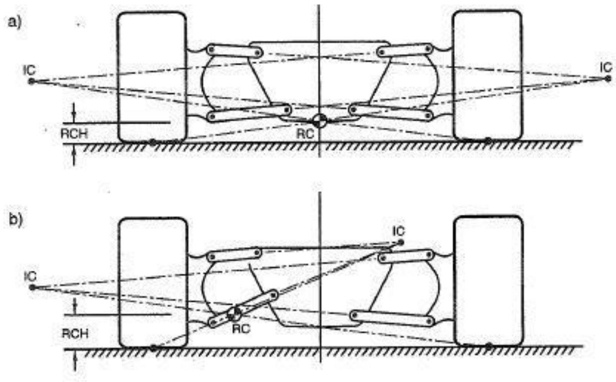


Figure 22: Front view of the suspension

Roll Center Height

The roll center establishes the force coupling point between the unsprung and sprung weights

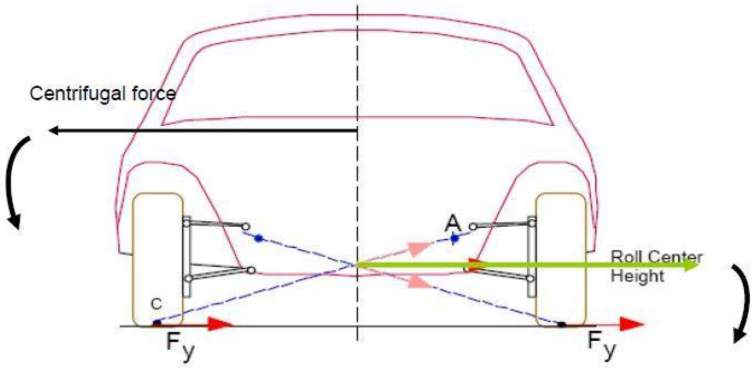


Figure 23: Roll center height

Jacking Effect

If roll center height is high, the lateral force acting at the tire generates a moment about the instant center. This moment pushes the sprung weight up and wheel down-jacking effect. The reverse happens if the roll center is below the ground

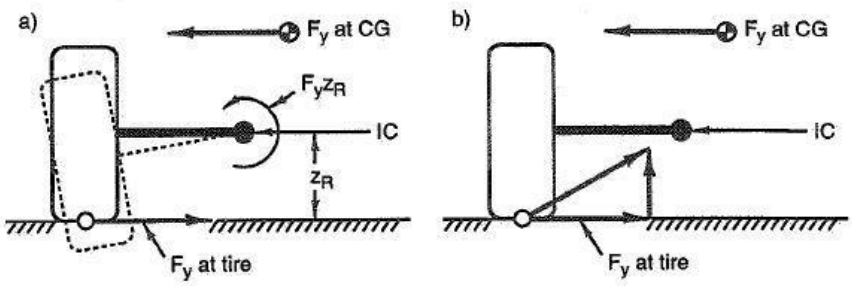


Figure 24Jacking effect

Camber change rate

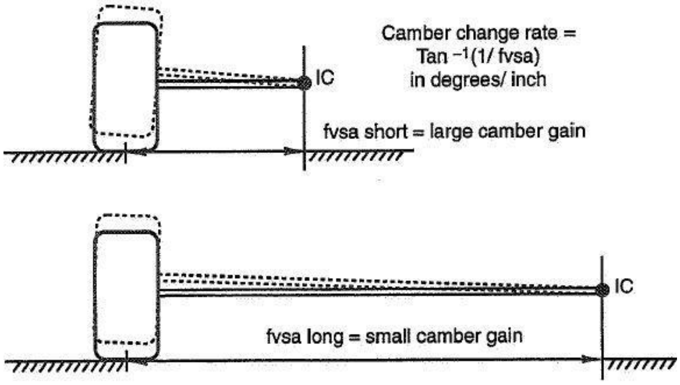


Figure 25: Camber change rate

Scrub Radius

This is the lateral motion relative to the ground that results from vertical motion of the wheel. Scrub occurs in every suspension system.

The amount of scrub is function of absolute and relative lengths of the control arms and the position of the front suspension view instant center relative to ground.

If the front view instant center is at any position other than the ground level scrub radius is increased

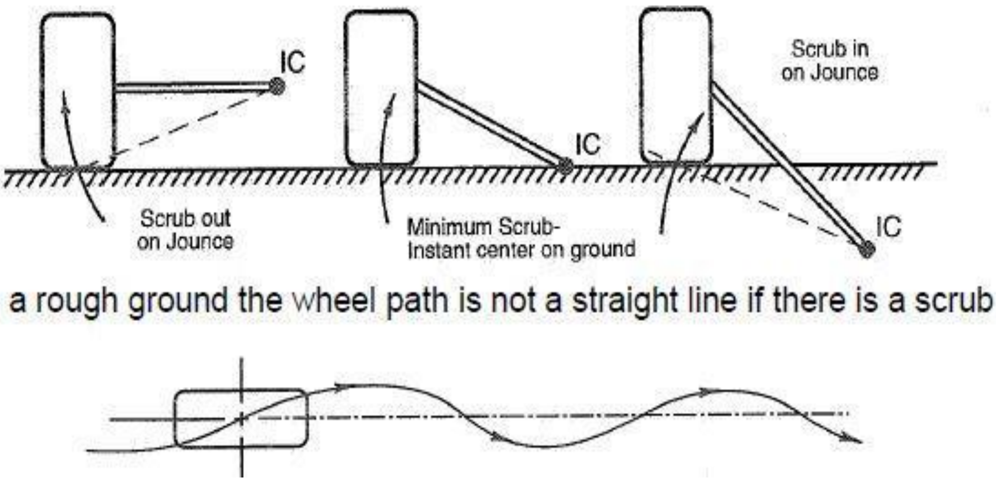


Figure 26: Scrub radius

b. Side View Arm Geometry (SVSA)

the instant center is behind and above the wheel center on front and it is ahead and on most rear suspensions. Also, define the wheel forth and near path, anti and anti-dive/ squat information, and caster change rate.

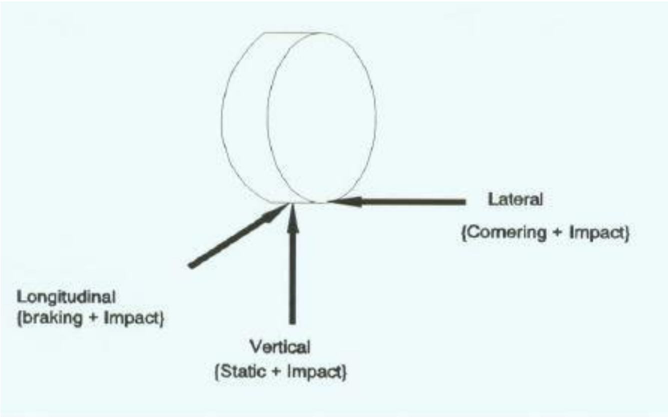
Caster change

Just like camber in front view, caster changes in side view as function of the length of the side swing arm. The result of caster change with suspension travel is that bump-steer curve is more difficult to make linear throughout the total range of travel

Camber is the tilting angle of the wheel, measured in degrees, when viewed from the front of the vehicle. When the wheel's tilt outward at the top, the camber is positive, and when the wheel tilts inward at the top, the camber is negative. The amount of tilt is measured in degrees from the vertical. These camber settings are known to influence the truck's drivability and tire wear.

On the other hand, caster is the tilting of the uppermost point of the steering axis (either forward or backward when viewed from the side of the vehicle). If the top of the pivot is leaning toward the rear of the car, then the caster is positive; if it is leaning toward the front, it is negative. If the caster is out of adjustment, it can cause problems in straight line tracking. If the caster is different from side to side, the vehicle will pull to the side with the less positive caster. If the caster is equal but too negative, the steering will be too light and the vehicle will wander and be difficult to keep in a straight line. If the caster is equal but too positive, the steering will be heavy and the steering wheel may kick when the truck hits a bump. The caster angle has little effect on tire wear, however adjustments made will affect drivability.

c. Suspension Weight Distribution



μ= tire-road coefficient of friction
W = total vehicle weight
bcg = Cg-to-rear axle distance (LR)
l = wheelbase length
a = longitudinal acceleration (deceleration)
m = vehicle mass
hcg = Cg height
g = acceleration of gravity

Figure 27: Forces over the wheel

Front Axle Braking per wheel:

$$FB = \frac{\mu}{2}\left[Static \quad + dynamic\ load\right] = \frac{\mu}{2}[W\frac{b_{cg}}{l} + m\bar{a}\frac{h_{cg}}{l}]$$

$$FB = \frac{\mu}{2}W[\frac{b_{cg}}{l} + \frac{\bar{a}}{g}\frac{h_{cg}}{l}]$$

<u>Vertical</u>: (commonly considered as a 3 g load)

$$V = \frac{3}{2}\left[W\frac{b_{cg}}{l} + m\bar{a}\frac{h_{cg}}{l}\right] = \frac{3}{2}W[\frac{b_{cg}g + \bar{a}h_{cg}}{l}]$$

<u>Lateral</u>: (commonly considered as a 2 g load)

$$F_L = W[\frac{b_{cg}g + \bar{a}h_{cg}}{gl}]$$

3.4 Evaluation

3.4.1 Evaluation of Computer Modeling Software:

Customer Requirements	Weight factor	SusProg 3d	MoTec I2	Lotus
Manufacturability	4	8	7	8
Derivability	9	8	8	8
Weight	7	8	5	3
Suspension travel	5	7	6	7
Cost	7	8	8	4
Sum Score	32	39	34	30
Average	-	0.82	0.94	1.07

Table 3: Computer software evaluation

3.4.2 Evaluation of Physical Experiments:

Customer Requirements	Weight factor	Contact Patch	Lateral Force	RC car
Manufacturability	4	4	6	4
Derivability	9	6	8	9
Weight	7	6	8	4
Suspension travel	5	7	8	6
Cost	7	5	7	8
Sum Score	32	28	37	31
Average	-	1.14	0.86	1.03

Table 4: Physical experiments evaluation

3.5 Refinement and Selection

Vildosola Racing Team seeks to improve the suspension system in order to increase the drivability of the car as well as to decrease tire wear during the competition.

The current Trophy Truck uses a camber angle of 1° and a caster angles of 3°. According to the car's pilot, Tavo Vildosla, the current configuration of the front suspension geometry creates a small response on the vehicle in spite of a big input from the driver.

Vildosola Racing has achieved to reduce the total weight of the truck in almost 300 lb. This fact makes them one of the fastest teams in the SCORE competitions. However, it also affects the traction of the car because of a smaller normal force and this increases the tire wear of the truck.

Considering all the facts above, we decided to reject the idea of the RC car with adjustable suspension geometry because it would not give us what we need, so we decided to test the latera force on the tires. This experiment will consist on a partial front suspension assembly loaded with a normal force which represents the inertial force of the truck and a load that represents the truck mass that will measure the force that will be made by strain gauges located between both surfaces.

We can also use some computer simulation that can show us the complete behavior of the car when a change is produced in the geometry of the front suspension. The software we will use is SusProg 3D.

. By testing and finding the optimal angle settings the trophy truck will achieve better performance, reliability, and drivability. We chose the above experiments since they will be the most useful compared to the others in terms of the road condition vs off road terrain conditions and time constraint that we have.

4 Design Specifications

4.1 Design Overview

4.1.1 Description

Our project contains two main deliverables: a computer simulation and a physical experiment.

The computer simulation would provide the calculation of adjusting varies angles on the tire, which will help us calculate the lateral force.

The second deliverable of two experiments which would be the experiment on altering the angles on the tires will help us choose the best angle for the racing truck.

Both deliverables will be used to determine the alterations on the truck to decrease tire wear and improve drivability. After experimenting and delivering the optimal choices to the Vildosola Racing team, they will have the option to choose if they would like to follow our team's data and apply it to their race truck. We have divided the project into two subsystems: the computer model Susprog3D, and the scale model lateral force experiment.

4.1.2 Design Schematics

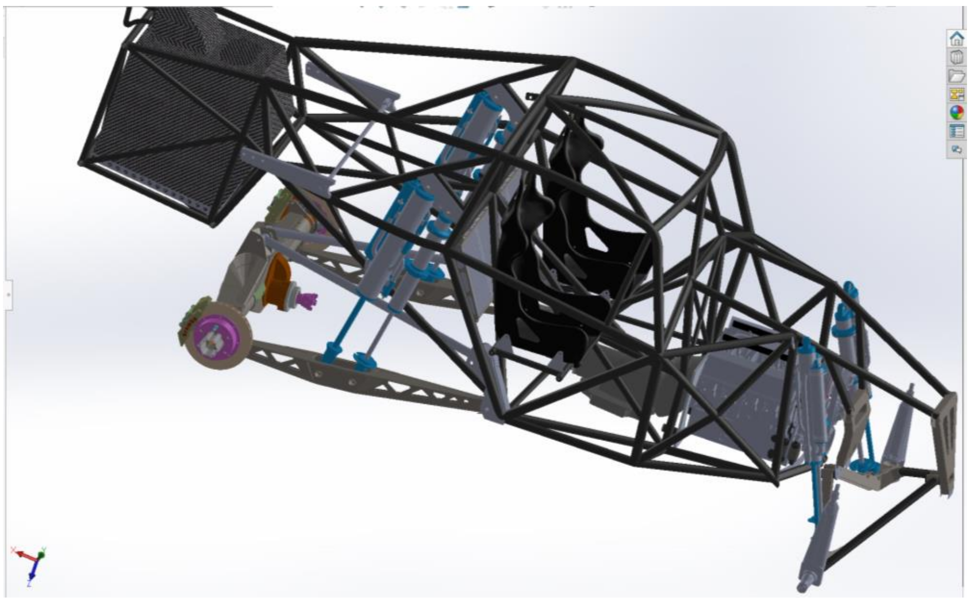


Figure 28:Process Schematic for Front Suspension

4.2 Functional Specifications

- Compatibility with Vildosola’s requirements for modelling and viewing software.
- Compatibility with Vildosola’s manufacturing abilities.
- Multiple revised and analyzed models to investigate and choose the optimal angles of the front suspension.

4.3 Physical Specifications

There are different aspects in which the front suspension will be modified. Three angles will be tackled.

- Camber: $\Delta\% < 3$
- Caster: $\Delta\% < 8$
- Toe: $\Delta\% < 3$

An optimization of these three angles combined would result in improving drivability which will be measured with the lateral external force test.

4.4 Subsystems

4.4.1 Susprog3D Simulation

First subsystem is Susprog 3D, a software that study and analysis the suspension behavior of the race car. It demonstrated the variation of camber and caster angles. when the car is rolling to the right, when the shocks are being compressed and when the shocks are being decompressed.

Build front suspension geometry is the same as the trophy truck specification as shown in figure 28.

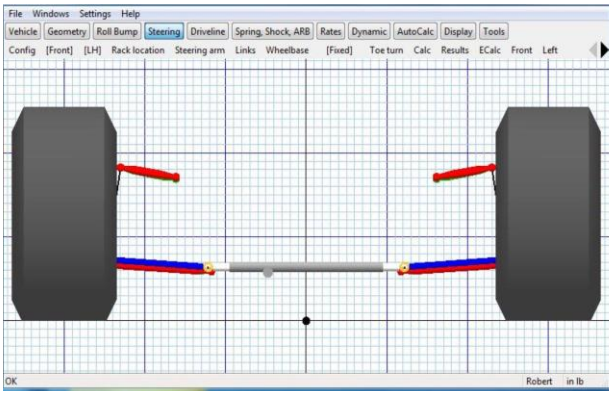


Figure 29: Example of SuspProg 3d

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	1.000	0.218	8.654	4.223	0.000	0.000	0.000	0.000	-0.245	123.894
0.50 roll	-0.389	0.960	0.209	9.043	4.227	0.000	-0.004	0.000	-1.662	-0.241	148.525
1.00 roll	-0.798	0.919	0.199	9.452	4.232	0.000	-0.008	0.000	-3.364	-0.228	185.991
1.50 roll	-1.228	0.878	0.190	9.881	4.237	-0.001	-0.013	0.000	-5.159	-0.207	250.384
2.00 roll	-1.678	0.837	0.180	10.332	4.243	-0.001	-0.017	0.000	-7.125	-0.178	388.397
2.50 roll	-2.151	0.795	0.170	10.804	4.250	-0.001	-0.021	0.000	-9.417	-0.141	903.018
3.00 roll	-2.646	0.753	0.160	11.299	4.257	-0.002	-0.025	0.000	-12.412	-0.098	-2370.410
3.50 roll	-3.164	0.710	0.150	11.817	4.265	-0.002	-0.030	0.000	-17.514	-0.048	-492.637
4.00 roll	-3.707	0.666	0.140	12.360	4.274	-0.002	-0.034	0.000	-39.690	0.017	-267.569

RH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	1.000	0.218	8.654	4.223	0.000	0.000	0.000	0.000	-0.245	123.894
0.50 roll	0.369	1.040	0.228	8.285	4.218	0.000	0.004	0.000	-1.662	-0.241	106.386
1.00 roll	0.718	1.079	0.237	7.936	4.215	0.000	0.008	0.000	-3.364	-0.228	93.243
1.50 roll	1.047	1.118	0.246	7.607	4.211	-0.001	0.012	0.000	-5.159	-0.207	82.970
2.00 roll	1.357	1.157	0.255	7.298	4.208	-0.001	0.016	0.000	-7.125	-0.178	74.688
2.50 roll	1.646	1.195	0.264	7.009	4.206	-0.001	0.020	0.000	-9.417	-0.141	67.846
3.00 roll	1.915	1.232	0.273	6.740	4.203	-0.002	0.024	0.000	-12.412	-0.098	62.080
3.50 roll	2.163	1.270	0.282	6.492	4.201	-0.002	0.028	0.000	-17.514	-0.048	57.141
4.00 roll	2.390	1.306	0.291	6.265	4.199	-0.003	0.031	0.000	-39.690	0.017	52.851

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	rc offset	roll centre height chassis	roll centre height ground	f\
4.000 bump	-4.300	1.591	0.333	12.956	4.283	-0.260	0.057	0.000	0.000	0.906	-3.094	32.
3.500 bump	-3.470	1.522	0.322	12.125	4.269	-0.210	0.051	0.000	0.000	0.562	-2.938	36.
3.000 bump	-2.731	1.451	0.309	11.387	4.258	-0.162	0.044	0.000	0.000	0.288	-2.712	41.
2.500 bump	-2.080	1.379	0.295	10.735	4.249	-0.119	0.037	0.000	0.000	0.078	-2.422	47.
2.000 bump	-1.511	1.306	0.281	10.166	4.241	-0.081	0.030	0.000	0.000	-0.076	-2.076	54.
1.500 bump	-1.022	1.231	0.266	9.676	4.235	-0.050	0.023	0.000	0.000	-0.178	-1.678	63.
1.000 bump	-0.608	1.155	0.251	9.262	4.230	-0.025	0.015	0.000	0.000	-0.237	-1.237	75.
0.500 bump	-0.268	1.078	0.235	8.922	4.226	-0.008	0.008	0.000	0.000	-0.257	-0.757	94.

Table 5: Example of SusProg 3D output

4.4.2 Front Suspension Kit

Static lateral force experiment: In order to measure drivability with a certain level of accuracy we will simulate the truck turning at a certain slip angle.

With a stationary wheel, a lateral force is applied on its axis, acting perpendicular to the direction of the wheel. This external force stimulates the lateral force a tire experiences when turning, which is equal to the inertial force that the vehicle suffers during a corner. This lateral or cornering force appears with different in each one of the 4 wheels of the car.

Although the lateral force in each wheel is different, due to the equipment of the experiment the measured force includes both the external and the internal forces suffered by the wheels.

In order to keep the tire in contact with the ground, another external force is applied downwards. The team can vary the angles to tackle, while measuring the lateral force needed for the tire to slip. The greater the lateral force, the better the drivability.

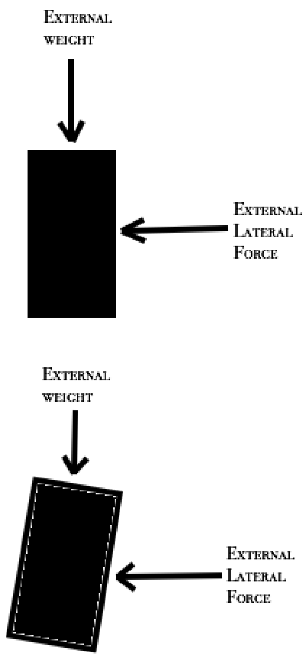


Figure 30: Static lateral force experiment

By purchasing a front suspension kit similar to the trophy truck's suspension we ensure to have a small highly detailed model of Vildosola's truck on which to experiment our changes of the angles and study the outcomes of them on the truck's deliverables. Although caster angle is determined, it is possible to accurately change both camber and toe in order to test the suspension.

The main objective of this experiment is to study the geometry angles and will help us decide the optimal angle numbers for the truck to help minimize the tire wear and increase the handling of the truck.

Figure 31: Helix Suspension Kit

5 Design Analysis

5.1 SusProg3D Simulation

For this part, we used SusProg 3D suspension modeler to analyze the front suspension effects of camber and caster on the scrub radius of the front suspension. We used Vildosola's trophy truck CAD model and varied the camber between 0 - 2 degrees, and the caster from 0 - 20 degrees. For each combination, we measured the scrub radius. With this data we are able to predict the truck's drivability characteristics.

The scrub radius can be altered by varying the camber, caster, and suspension design. This will affect how the tire contact patch interacts with the road during cornering, braking, and acceleration, and can cause unique toe characteristics depending on the design. A positive scrub radius tends to be beneficial while braking into a turn. A negative scrub radius tends to better for braking stability in the event of brake failure. Typically, the scrub radius is configured to be as small as possible to minimize the effects under braking, but squirm (from zero scrub radius) is also undesirable.

For a front-drive truck, when scrub radius is negative the front wheels will try to toe out as they pull the car forward while accelerating. The length of the scrub radius influences how much toe-out force is generated. If one front tire has more traction than the other, it will feel like the steering wheel is being pulled out of the driver's hands. Since traction on the course varies constantly, this situation would not be desirable for application. Similarly, when the truck is braking, the front wheels try to toe in. In this case, the car's weight and large scrub radius amplifies the toe-in affect, which reduces stopping distance, but only if he driver can hold the steering wheel straight.

Materials Used:

- SusProg 3D software
- Vildosola provided CAD model of Trophy Truck

Procedure:

Once we obtained a complete CAD model of the Vildosola Trophy Truck, we converted the SolidWorks file into SusProg 3D. We ran a series of tests, using the specifications of the truck and assumptions outlined below. Keeping all suspension geometries constant except camber and caster, we record all the calculated data under the roll and dump. Also, we study the behavior of the camber and caster angle of the front suspension when steering turns in maximum angle of 30 degrees. Finally, we graphed the results in Excel.

Effect of caster change:

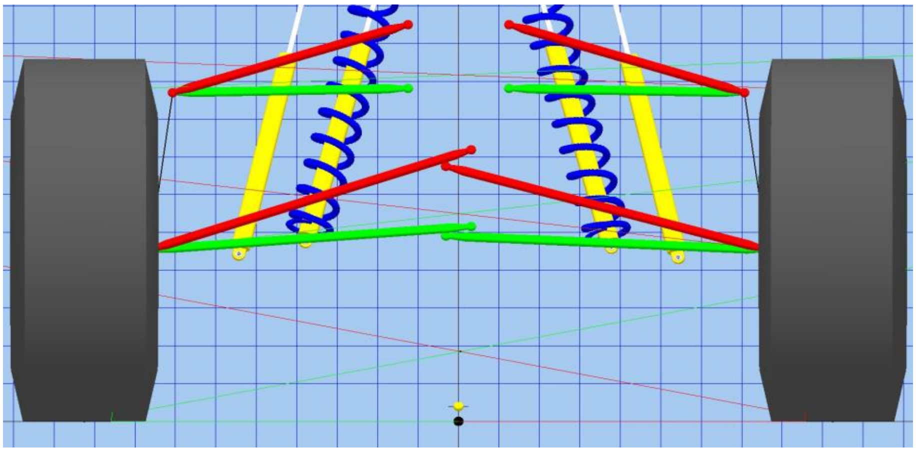


Figure 32: Camber constant(zero) and caster alternates

Effect of camber change:

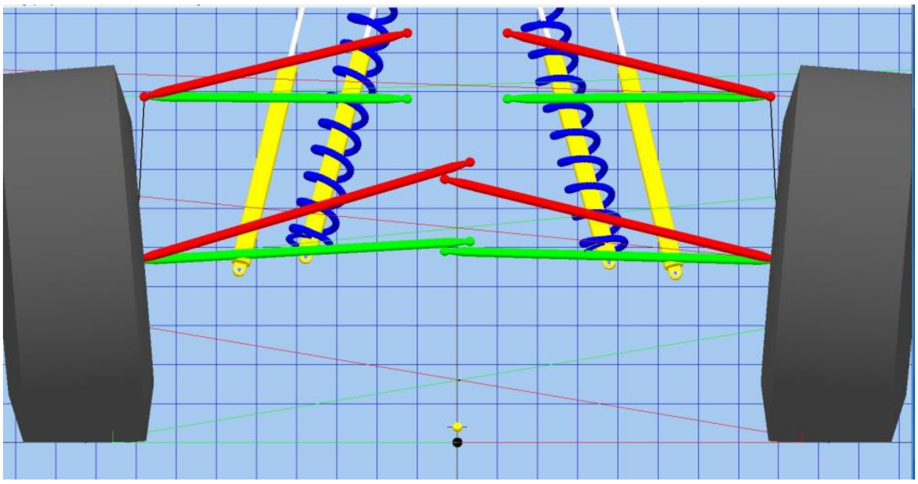


Figure 33: Caster constant(zero) and camber alternates

5.2 Front Suspension Kit

We used the suspension kit to test out the maximum velocity the truck can turn without slipping which will be determined by calculating the slip angle. The main objective of this experiment would be useful for the drivability aspect of our approach on modifying the truck's suspension. The second objective of this experiment is to study the geometry angles and will help us decide the optimal angle numbers for the truck to help minimize the tire wear and increase the handling of the truck.

Materials Used:

- Suspension Kit
- 660 lb. crane scale
- Wheels
- Drill press
- Mill machine
- Welding Equipment
- Hydraulic press

Procedure:

We ordered the suspension kit from Helix Suspension which came out unassembled, so we had to assemble it ourselves. We started with welding the suspension arms and the steering rack mounts and then we assembled all the other parts. We also had to press the bearings into the brake discs and drill the holes for them to be mounted onto the suspension, we then attached the wheels on to the kit.

Then in order to measure drivability with a certain level of accuracy we simulated the truck turning at a fixed angle and calculated the slip angle. With a stationary wheel, using the crane scale to apply a lateral force is applied on its axis, acting perpendicular to the direction of the wheel. This external force stimulates the lateral force a tire experiences when turning, which is in reality caused by friction between the tire and the road. To keep the tire in contact with the ground, another external force is applied downwards.

The team can vary the angles to tackle, while measuring the lateral force needed for the tire to slip. The greater the lateral force, the better the drivability. As well as to experiment the changes of the angles and study the outcomes of them on the truck's deliverables.

Figures 34 and 35 shows the configuration of the test.

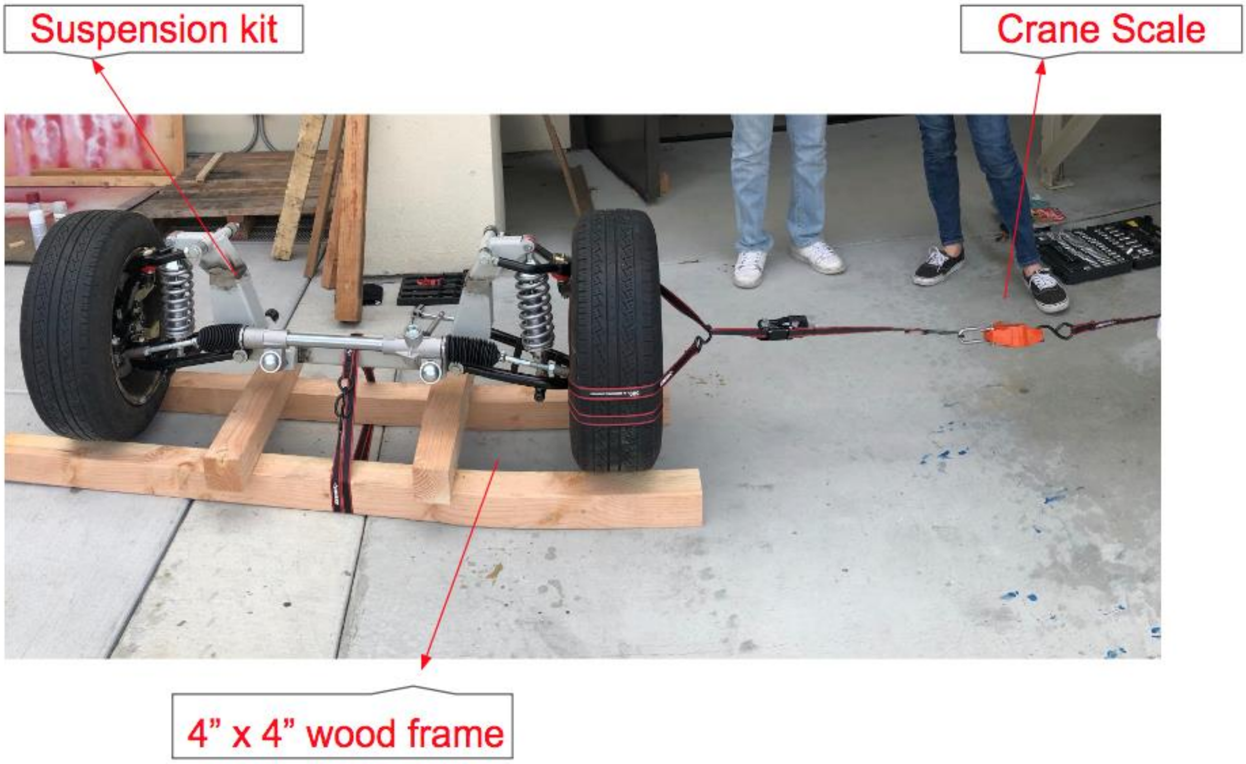


Figure 34: Physical experiment

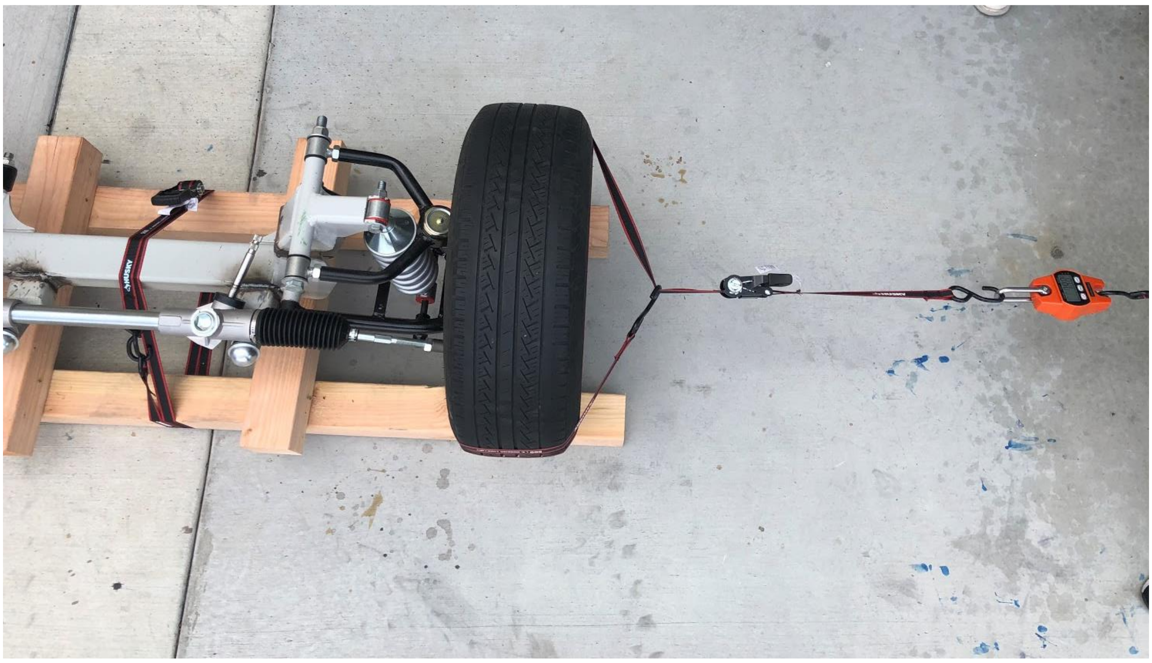


Figure 35:Top view of physical experiment

6 Results and conclusions

6.1 Physical experiment

In this graph are recorder all the data obtained from the test that was carried out with the front suspension kit with the procedure described in section 5.2:

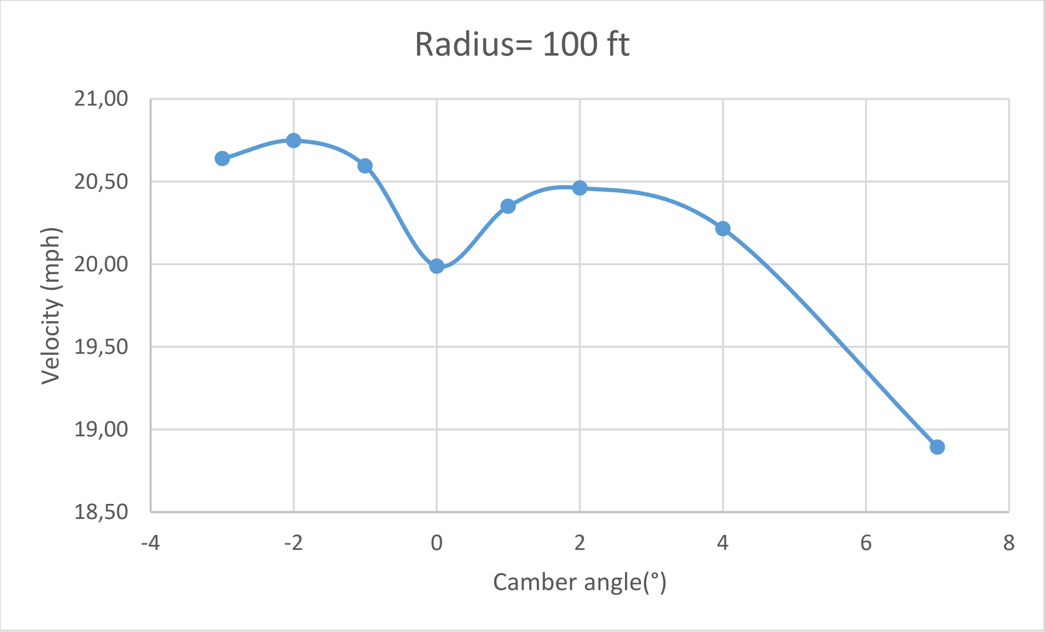


Figure 36: Camber-turning speed

Due to limitations in the suspension kit, the minimum caster angle testable was -3°. However, it can be seen that the behavior of the suspension referring to lateral force is almost symmetric respect the 0 angle, so the maximum lateral force decreases exponentially with lower values than -3° and higher values than 3°. This fact limits the possible range of optimal camber angle to the angles between -3° and 3°.

In spite of the similar behavior between positive and negative values of the angle due to the symmetric configuration of the suspension, there exists an effect that makes both negative and positive camber different. In order to understand this, we need to study the dynamic behavior of the vehicle during corners:

- The inertial force is a fictitious force that appears when a mass is accelerated.
- In a curved path, this force is equal to the centripetal force in value and opposite to the centripetal acceleration in direction.

$$F = ma_c$$

- This force pushes the weight of the vehicle towards the outside heel, creating a torque over the instant center of rotation, which is the point of contact between the tire and the ground. This effect makes the outside wheel loose this tilt due to camber angle.

It is essential to understand that the wheels which support the majority of the weight of the vehicle during a turn are the outside wheels, while the inner ones support less force. Figure 38 shows how this effect happens.

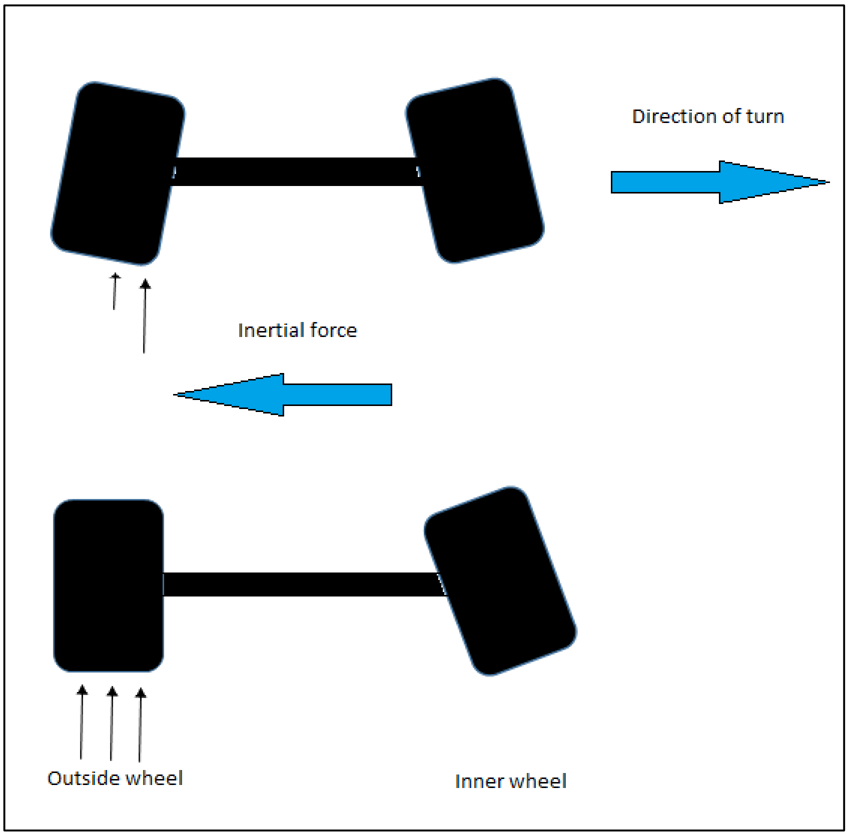


Figure 37: Negative camber under inertial force

On the other hand, if we focus on tire wear, increasing the angle over positive or negative 3° could dramatically magnify the damage on the external or internal surface of the tire depending on the type of camber applied when the vehicle runs in straight parts of the circuit. As the angle increases, the surface area which suffer the weight of the vehicle decreases, raising then the wear.

In a turn is also better to use a negative camber configuration for the suspension. In this case, the outside wheel suffers more than the inner one, so it is recommendable that the surface of contact between this wheel and the ground is as higher as possible.

If we focus on the Figure 38, a negative camber angle will distribute the efforts during the corner through a bigger surface of the tire than with positive or zero camber configurations, as shown in Figure

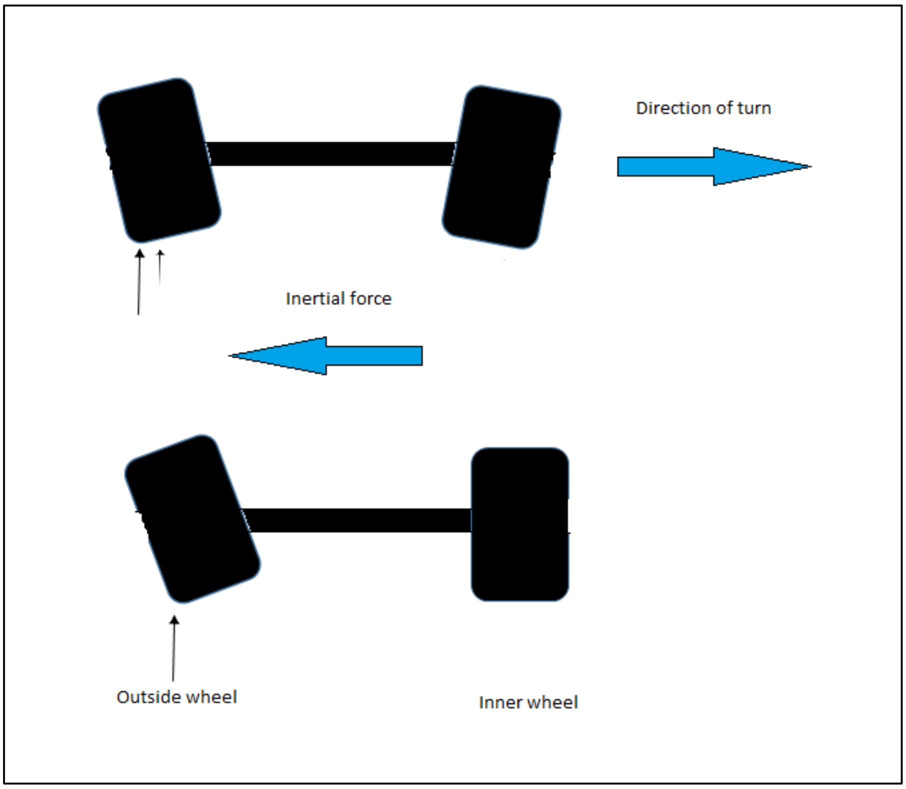


Figure 38: Positive camber under inertial force

On terms of tire wear, and assuming that the number of right and left turns during a competition are similar, we will focus on the outside wheel in order to obtain the optimal camber angle. If the wear of the outside wheel is minimized, then the inner tire will be minimized as well because this one suffers less stress than the outside one.

In conclusion, from the physical experiment it can be deduced that the optimal value for the camber angle in terms of drivability thru tire wear is between -1° and -2°, depending on what fact do you prioritize. If your main problem is drivability, then it is recommendable to increase the negative value of camber to -2°. However, if you want to reduce the tire wear as maximum, use small negative values for camber, but not zero camber, because as demonstrated before, a zero camber will damage the tires more than with a small negative value. In both cases there is an increment of about a 4% on the maximum speed thru a turn.

These improvements in the suspension increase the possibilities of Vildosola of winning the SCORE international races because they imply a reduction on the number of stops for tire changing as well as an increment on the average speed of the Trophy Truck in a race, that is the same as a time gain.

6.2 Software simulation

The scrub radius is the distance between the center of the contact path between the tire and the road and the intersection between the imaginary prolongation line of the direction axis and the ground. If the value of this distance is different to zero, there would be external forces and torques over the direction axis because the wheel wants to turn over a pivot which is not located in the instant center of the wheel (the contact point between the tire and the road). Because of that, the ideal and desired scrub radius is zero. However, due to an undesired instability in the direction produced by a zero value of this distance, this scrub radius is unfeasible.

Caster is the inclination angle of the direction axis. Most of the front suspension systems have a positive caster angle because the engine propulsion creates an instability on the front wheels that has to be countered. This effect is canceled by tilting the direction axis, which creates a new point B (Figure 40), which is the intersection between this axis and the ground, giving then stability to the front suspension.

When the pilot changes the directions to make a turn, the wheel pivots over point B (Figure 40), which makes point A (Figure 40) to move towards A' (Figure 40), creating a torque that tries to move back the wheel to its straight line position where torque is zero because d is zero.

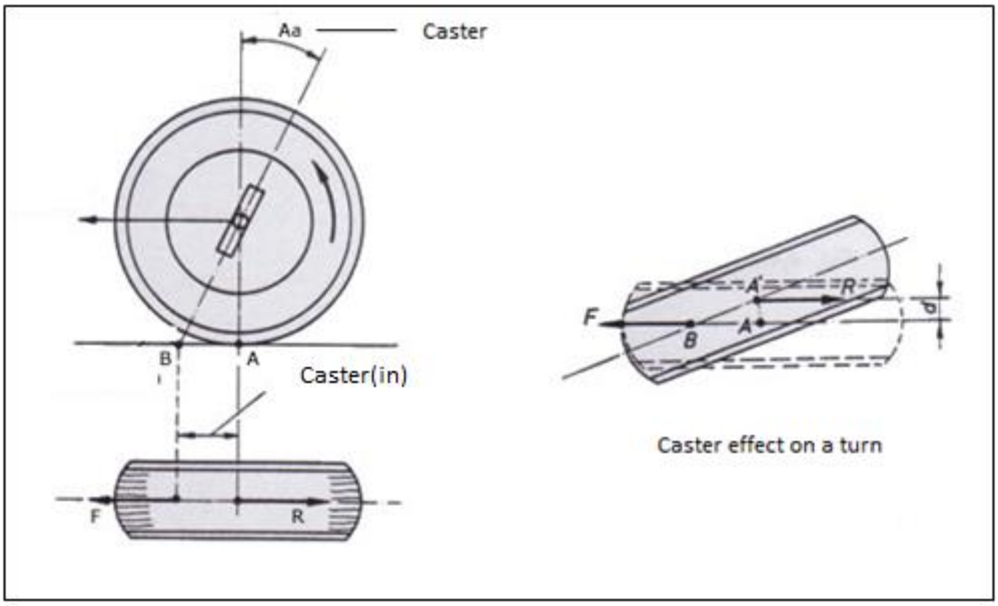


Figure 39: Caster effects

This effect provides stiffness and stability to the suspension, since the different disturbances of the road are faced by this torque that moves back the direction into the straight line position. Caster needs to have enough value to fulfill its task without damaging another component of the suspension. If this angle is too large, then the torque is large as well, forcing the wheels to make violently turns. However, if this angle is insufficient, then the instability of the vehicle's direction will not disappear. Typical values for caster angle are in a rage of values between 4° and 12°.

The following graphs shows the behavior of the suspension when changes over the configuration are made:

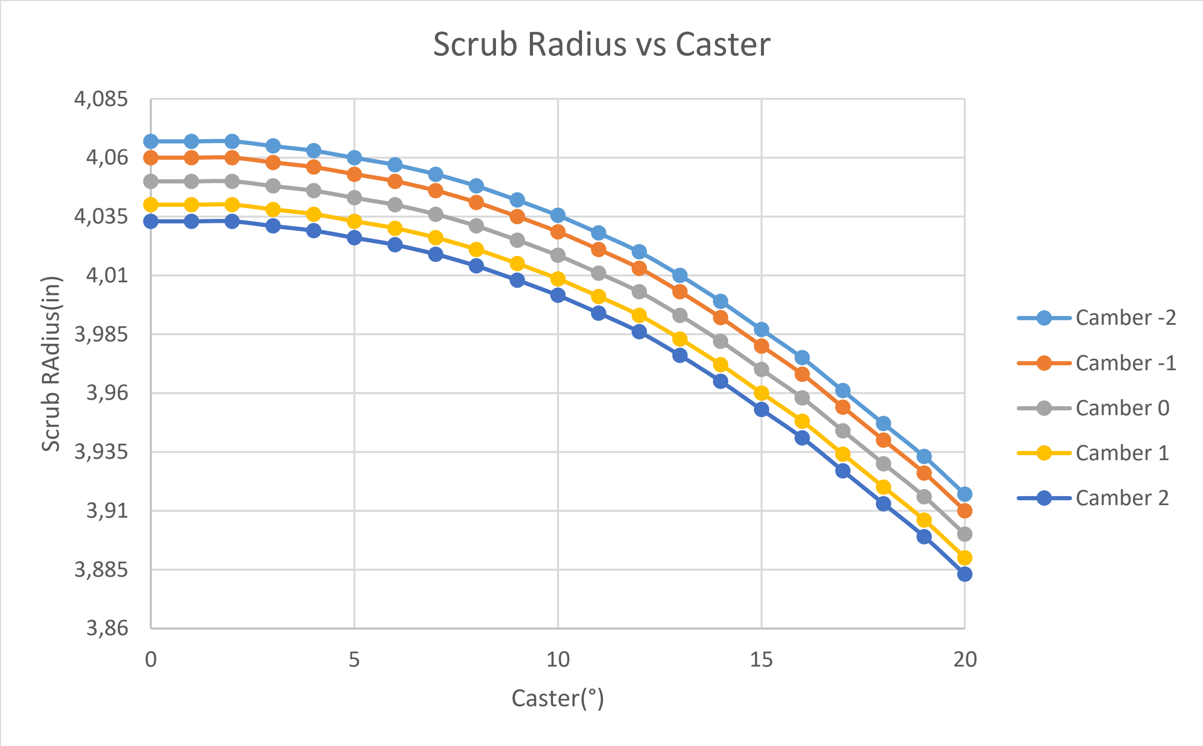


Figure 40: Scrub radius- Caste

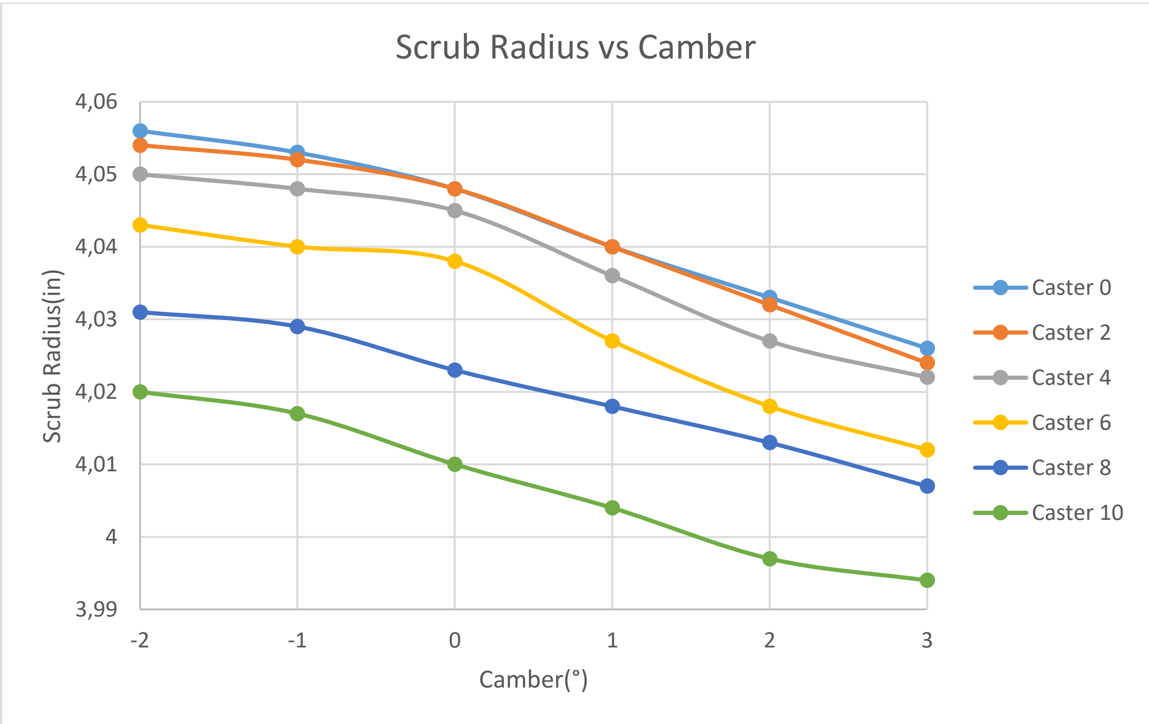


Figure 41: Camber vs Scrub Radius

The data obtained from the different software simulations carried out (ANEX 8.3.2) is focused on the range of values shown in section 5.3.1 where the optimal camber angle can be found.

All the data that appear on the different graphs were took from simulations where the configuration was updated with the current Trophy Truck in order to obtain real data on how do affect the changes on the real suspension geometry. With this graphs, Vildosola can obtain useful information about the dynamics of the suspension, what will help them on the design of the new Trophy Truck.

As both figures 40 and 41 show, the scrub radius of the Trophy Truck's suspension increases as the camber angle decreases, so, as said before, we would try to minimize the scrub radius by implementing big values of camber angle. However, this would affect the general performance of the vehicle by decreasing the maximum speed thru corners and deteriorating more the tires. In this case, it is recommended to slightly sacrifice scrub radius and stability in order to maximize the performance of the Truck.

On the other hand, Caster affects scrub radius by decreasing its value as this angle increases, so it is preferable to set the suspension with the biggest possible caster angle, without forgetting to consider that excessive values of caster would damage the behavior of the vehicle, reducing then the sensibility of the direction.

7 Project Plan

7.1 Construction

For the front suspension kit, we will need to modify the suspension geometry. The kit will be purchased from Johnny Law Motors and assembled in the Loma Hall Machine Shop.

All constructions and technical changes will be made in Loma Hall.

Subsystem	Percent Completion	Remaining Work	Completion Date
Material Procurement and Inventory	100%	N/A	4-April
Suspension Assembly	100%	N/A	14-March
Testing & Data-Gathering	40%	Complete suspension kit experiments for varying camber and caster angles	23-April
Computer Simulation	75%	Finish modeling front suspension and print graphs to show results clearly	16-April
Analysis	25%	Analyze data gathered from the suspension kit experiments and compare with calculations from computer simulations	30-April

Table 6: Construction Plan

7.2 Testing

Customer Requirement/ Constraint	Test Procedure
Improve tire wear	Optimizing the camber, caster, and toe angles will decrease the probability of tire wear by more than 10%
Drivability / Traction	Adjusting the toe by ⅛ or 1/16 inches will increase the responsiveness of turning during the race and also will improve the ride quality.

Table 7: Testing Procedure

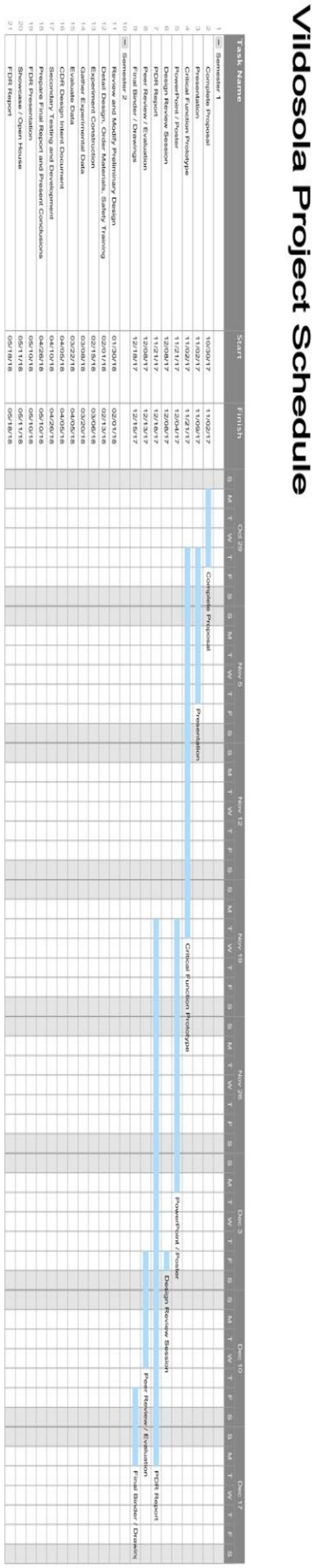
7.3 Safety

Safety glasses and proper tools have been used when assembling the suspension kit. Thus far, there have been no safety concerns. If the team is not certain on a direction or how to use a tool, several faculty moderators are present to answer questions. As a team, we are mindful of others and our surroundings to avoid minor accidents.

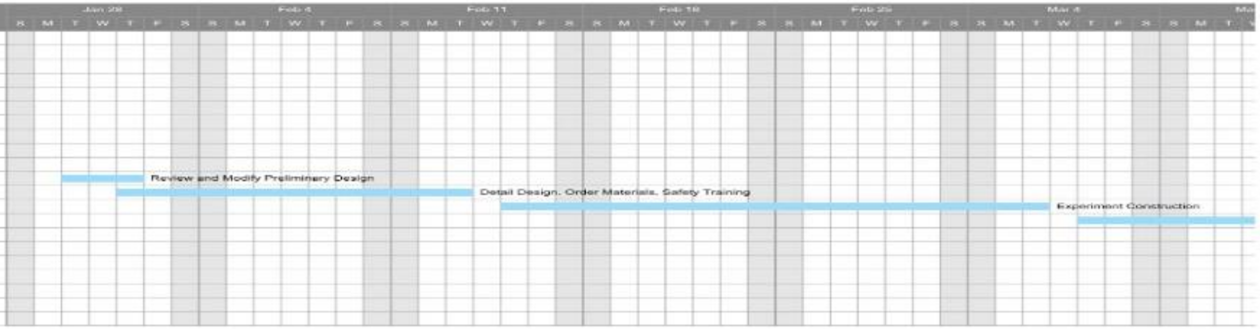
7.4 Project Deliverables

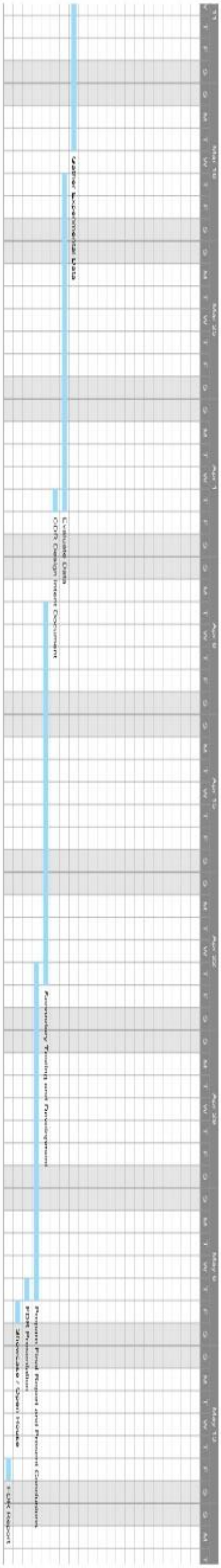
1. First Semester:
    - Preliminary Design Report with associated CAD modeling and analyses
    - Bill of Materials and cost estimates
    - Order materials including Suspension kit
2. Second Semester:
    - Final Design Report including test results
    - Susprog Simulation Graphs and Analysis
    - Lateral force experiment using suspension kit

7.5 Schedule


Vildosola Project Schedule
Task Name | Start | Finish
1 Semester 1
2 Complete Proposal 10/30/17 11/02/17
3 Presentation 11/02/17 11/09/17
4 Critical Function Prototype 11/02/17 11/21/17
5 PowerPoint / Poster 11/21/17 12/04/17
6 Design Review Session 12/08/17 12/08/17
7 PDR Report 11/21/17 12/18/17
8 Peer Review / Evaluation 12/08/17 12/13/17
9 Final Binder / Drawings 12/18/17 12/15/17
10 Semester 2
11 Review and Modify Preliminary Design 01/30/18 02/01/18
12 Detail Design, Order Materials, Safety Training 02/01/18 02/13/18
13 Experiment Construction 02/15/18 03/06/18
14 Gather Experimental Data 03/08/18 03/20/18
15 Evaluate Data 03/22/18 04/05/18
16 CDR Design Intent Document 04/05/18 04/05/18
17 Secondary Testing and Development 04/10/18 04/26/18
18 Prepare Final Report and Present Conclusions 04/26/18 05/10/18
19 FDR Presentation 05/10/18 05/10/18
20 Showcase / Open House 05/11/18 05/11/18
21 FDR Report 05/18/18 05/18/18
Oct 29
Nov 5
Nov 12
Nov 19
Nov 26
Dec 3
Dec 10
Dec 17

	Task Name	Start	Finish
1	Semester 1		
2	Complete Proposal	10/30/17	11/02/17
3	Presentation	11/02/17	11/09/17
4	Critical Function Prototype	11/02/17	11/21/17
5	PowerPoint / Poster	11/21/17	12/04/17
6	Design Review Session	12/08/17	12/08/17
7	PDR Report	11/21/17	12/18/17
8	Peer Review / Evaluation	12/08/17	12/13/17
9	Final Binder / Drawings	12/10/17	12/15/17
10	Semester 2		
11	Review and Modify Preliminary Design	01/30/18	02/01/18
12	Detail Design, Order Materials, Safety Training	02/01/18	02/13/18
13	Experiment Construction	02/15/18	03/06/18
14	Gather Experimental Data	03/08/18	03/20/18
15	Evaluate Data	03/22/18	04/05/18
16	CDR Design Intent Document	04/05/18	04/05/18
17	Secondary Testing and Development	04/10/18	04/26/18
18	Prepare Final Report and Present Conclusions	04/26/18	05/10/18
19	FDR Presentation	05/10/18	05/10/18
20	Showcase / Open House	05/11/18	05/11/18
21	FDR Report	05/18/18	05/18/18




Mar 18
Mar 25
Apr 1
Apr 8
Apr 15
Apr 22
Apr 29
May 6
May 13
Gather Experimental Data
Evaluate Data
CDR Design Intent Document
Secondary Testing and Development
Prepare Final Report and Present Conclusions
FDR Presentation
Showcase / Open House
FDR Report

7.6 Budget

Column1	Part/Material	Supplier	Cost	Quantity	Subtotal
Computer simulation	SusProg 3D software	SusProg.com	$ -	1	$ -
Physical experiment	Helix Corner Killer IFC Suspension Kit	Johnny Law Motors	$2,645.00	1	$2,645.00
	Crane Scale	Modern Step	$48.00	1	$48.00
	Tires	Local Tire Shop	$110.00	2	$110.00
	Miscellanous Lumber	The Home Depot	$36.00	1	$36.00
Total Cost					$2,949.00
Budget					$7000.00
					$4,051.00

Table 8: Budget

8 References

***[1]: https://patents.google.com/patent/US6010139A/en?oq=US6010139+A*+**

[2]:https://patents.google.com/patent/US20150224845A1/en?oq=US2015022484 5+A1

[3]: https://patents.google.com/patent/US7180409B2/en?oq=US7180409B2

[4]: https://patents.google.com/patent/US7005816B2/en?oq=US7005816+B2

[5]: https://patents.google.com/patent/US5091679A/en?oq=US5091679A

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http://www.aficionadosalamecanica.net/direccion-geometria.htm

http://bevenyoung.com.au/

9 Appendices

9.1 CV

José Antonio López-Monís Yuste

Personal data

Birth date: 09/02/1996

Location of birth: Navalmoral de la Mata, Cáceres, Spain

Addres: Cerro del Espino nº7 3ºA, Majadahonda, Comunidad de Madrid, 92109

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Highly motivated student with excellent influential and communication skills. Currently studying fourth year of industrial engineering at US. Specialized in mechanical engineering. Fluent English speaker with experience living abroad in San Diego, US.

Education

- 2014-May 2018 University of Comillas-ICAI
- 2017-May 2018 University of San Diego
    - Exchange student
    - Engaged in an industrial project for Vildosola Racing Team.

Languages

- Spanish: native language
- English: advanced level C1(99/120)

Experience

- Member of the QC/QA department for Cymi in the project "El niño".
    - Client: International Paper
    - Project: transform a recycled paper factory in a recycled paperboard factory.

Skills

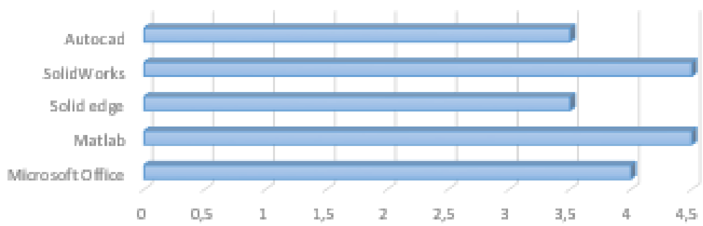


Figure 42: CV

9.2 Front Suspension Kit

9.2.1 Suspension assembly process

Figure 43: Unassembled suspension kit

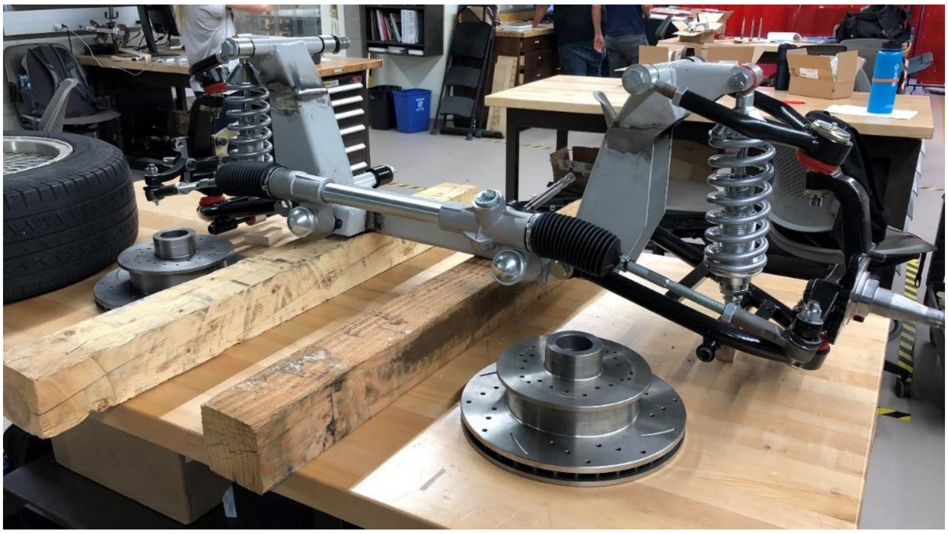


Figure 44: Assembled suspension kit

Calculus justification:

In order to obtain the maximum speed for a determined radius of a turn it is necessary to use the Newtonian mechanics in a circular path and considering he whole vehicle as a single solid of mass m:

$$F = ma_c \qquad (1)$$

$$a_c = \frac{|v|^2}{R} \qquad (2)$$

Where:

- m [lb]= mass
- F [lbs] = inertial force/ lateral force
- a_c $[ft/s^2]$= centripetal acceleration
- v [ft/s] = linear speed
- R [ft] = turn radius

Combining both (1) and (2) we can obtain the maximum velocity on a circular turn.

$$F = m\frac{|v|^2}{R} \qquad (3)$$

$$v = \sqrt{\frac{FR}{m}} \qquad (4)$$

In this case is not necessary to consider an extra coefficient due to the difference of mass between the kit and the real Truck because the lateral force suffered is equal to the friction force, which engages the normal force:

$$F = \mu N = \mu mg \qquad (5)$$

$$ma = \mu mg \qquad (1 + 5)$$

Where:

- g [ft/s^2] = gravity force = 32.17
- µ [-]= friction coefficient

The information contained in the following table is all the data recovered from the physical experiment of the suspension kit.

Weight(Fixed)	Lateral Force (lbf)	Radius (ft)	Velocity (ft/s)	Velocity (mi/hr)
150	296.4	100	14.06	20.62
150	294.5	100	14.01	20.55
150	300.2	100	14.15	20.75
150	302.1	100	14.19	20.81
150	294.5	100	14.01	20.55
150	304	100	14.24	20.88
150	300.2	100	14.15	20.75
150	290.7	100	13.92	20.42
150	296.4	100	14.06	20.62
150	267.9	100	13.36	19.60
150	290.7	100	13.92	20.42
150	277.4	100	13.60	19.95
150	283.1	100	13.74	20.15
150	294.5	100	14.01	20.55
150	288.8	100	13.88	20.35
150	285	100	13.78	20.22
150	294.5	100	14.01	20.55
150	296.4	100	14.06	20.62
150	292.6	100	13.97	20.48
150	285	100	13.78	20.22
150	277.4	100	13.60	19.95
150	245.1	100	12.78	18.75
150	252.7	100	12.98	19.04
150	248.9	100	12.88	18.89

Table 9: Suspension Kit Experiment Data

The average of the previous data that's was used to create the conclusion graphs is:

Camber	Lateral Force(lbf)	Velocity (ft/s)	Velocity (mph)
-3	297.03	14.07	20.64
-2	300.20	14.15	20.75
-1	295.77	14.04	20.59
0	278.67	13.63	19.99
1	288.80	13.88	20.35
2	291.97	13.95	20.46
4	285.00	13.78	20.22
7	248.90	12.88	18.89

Table 10: Average of Suspension Kit Experiment

9.3 Software simulation data

9.3.1 Vehicle data used in SusProg 3D

1. Vehicle:

- Datum location zero for the longitudinal datum reference point x-axis is zero and the y-axis is zero.
- Wheelbase: 0.040.
- Ride height, single point front LH, single point rear LH, front ride height LH location (chassis datum) x: -1.250 Y: 27 Z: -13.5, static (from ground): 27.00
- Mass: corner weights (unsprung): front LH: 94.5 RH: 94.5 rear LH 0.00 RH: 0.00, corner weights (vehicle): front LH: 1240 RH: 1240 rear LH 0.00 RH: 0.00
- Wheel and Tire: Wheel, Rim Diameter 16.350 in Rim width 9.5 in Rim mounting offset -0.750 in
- Tire: tire tread width 12 in tire section width 14.5 in tire rolling radius 19 in tire diameter 38.7 in tire spring rate 2000 lb/in toe reference length 30 in, (185/70 VR 15)

2. Geometry:

- Chassis [front LH]: Top A-arm mounting, front X: 5.00 Y: 42.00 Z: -13.750, Bottom Top A-arm mounting X: -1.250 Y:27.00 Z: -13.500, Top A-arm mounting, Rear X: 5.00 Y: 35.250 Z: 3.625 Bottom A-arm mounting, Rear X: -1.250 Y: 19.625 Z: 7.00 Steering rack ball joint X: 3.00 Y: 26.750 Z: -4.250
- Upright [front LH]: Top mounting ball joint, top A-arm mounting X: 9.5 Y: 15.250 Z: 4.400, Bottom mounting Ball Joint X: 7.250 Y: -0.850 Z: 0.000, Steering arm ball joint X: 6.5 Y: 5.560 Z: -5.00

3. Steering:

- Toe Turn maximum turn angle 30 degree, increment 5 degree

9.3.2 SusProg 3D output

In the Software Susprog 3D, the specification of the trophy truck has been used such as chassis and tire diameter. The calculations below determine the different angles of the camber and the caster and show how that impact into the tires deformation during the rolling and dumping.

The different tables shown in section 8.3.2.1 and 8.3.2.2 gave us the necessary data to create the graphs shown in section 5.3. and take final conclusions.

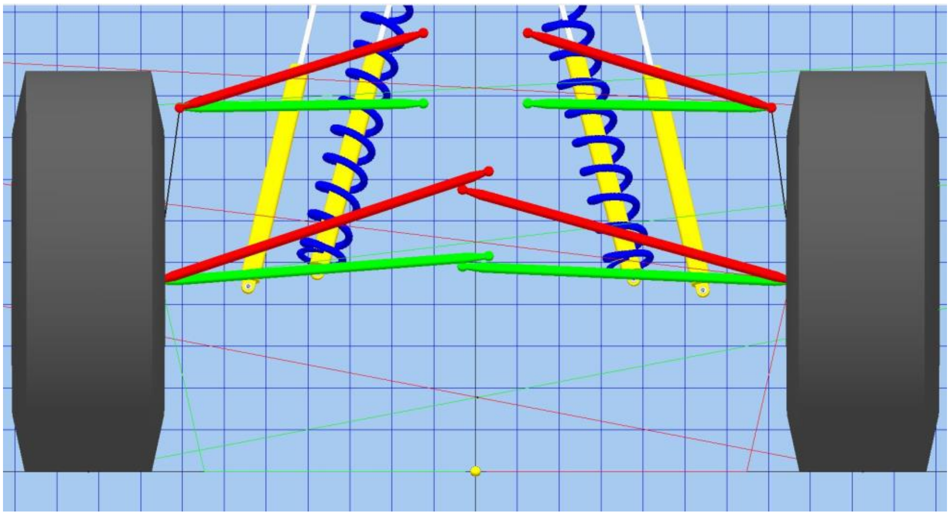


Figure 45:front suspension interface in the simulation program.

9.3.2.1 Caster changes and camber equal to zero

Roll and bump [Front]

SusProg3D Vildofrontsusp.s3d Front Roll and bump

Chassis roll values calculated every 1.00 degrees. Roll left.
Semi dynamic roll centre, inclined roll axis. Roll starts at Static.
Toe variation has NOT been calculated.

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	4.750	1.354	7.704	4.043	0.000	0.000	0.189	0.175	7.386	225.586
1.50 roll	0.131	4.951	1.422	7.575	4.041	-0.043	-0.789	0.189	1.735	7.352	227.145
3.00 roll	0.267	5.149	1.490	7.441	4.039	-0.108	-1.577	0.189	3.300	7.281	228.851
4.00 roll	0.409	5.344	1.556	7.302	4.037	-0.195	-2.364	0.189	4.870	7.173	230.707

RH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	4.750	1.354	7.704	4.043	0.000	0.000	0.189	0.175	7.386	228.763
1.50 roll	-0.126	4.546	1.285	7.827	4.045	0.021	0.788	0.189	1.735	7.352	228.357
3.00 roll	-0.246	4.339	1.215	7.946	4.047	0.019	1.576	0.189	3.300	7.281	228.070
4.00 roll	-0.361	4.129	1.144	8.059	4.048	-0.005	2.362	0.189	4.870	7.173	227.902

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	rc offset	roll centre height chassis	roll centre height ground	fvsax
2.796 bump	-0.813	4.970	1.419	8.519	4.050	0.459	0.973	0.189	0.383	8.066	5.270	155.744
Static	0.000	4.750	1.354	7.704	4.043	0.000	0.000	0.189	0.175	7.386	7.386	225.586
3.084 droop	0.531	4.484	1.270	7.170	4.039	-0.708	-1.091	0.189	0.048	6.664	9.748	493.716

RH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	rc offset	roll centre height chassis	roll centre height ground	fvsax
2.796 bump	-0.802	4.846	1.378	8.507	4.051	0.452	1.009	0.189	0.383	8.066	5.270	157.388
Static	0.000	4.750	1.354	7.704	4.043	0.000	0.000	0.189	0.175	7.386	7.386	228.763
3.084 droop	0.524	4.623	1.316	7.178	4.038	-0.705	-1.131	0.189	0.048	6.664	9.748	506.338

Equivalent suspension travel due to chassis roll

	RH	LH
0.00 roll	0.000	0.000
1.50 roll	-0.005	0.005
3.00 roll	-0.011	0.011
4.00 roll	-0.016	0.016

Side view swing axle and instant centre LH

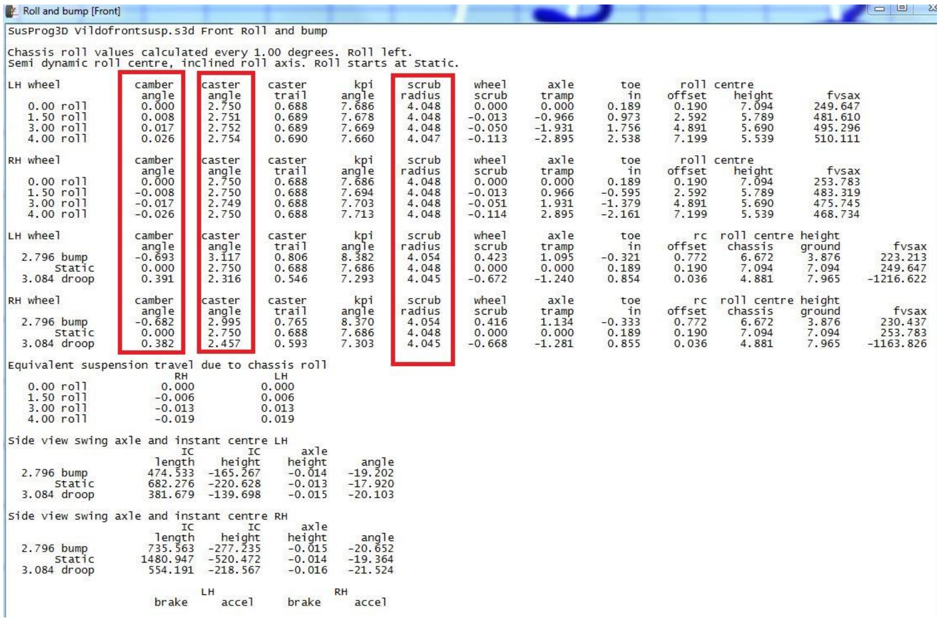
	IC length	IC height	axle height	angle
2.796 bump	768.422	-246.188	-0.013	-17.765
Static	703.437	-227.219	-0.013	-17.901
3.084 droop	640.724	-209.023	-0.013	-18.068

Side view swing axle and instant centre RH

	IC length	IC height	axle height	angle
2.796 bump	1862.492	-647.687	-0.014	-19.175
Static	1557.199	-546.309	-0.014	-19.332
3.084 droop	1304.057	-462.338	-0.014	-19.521

LH		RH	
brake	accel	brake	accel

Figure 46: caster 4.75 degree and camber zero



Roll and bump [Front]

SusProg3D Vildofrontsusp.s3d Front Roll and bump

Chassis roll values calculated every 1.00 degrees. Roll left.
Semi dynamic roll centre, inclined roll axis. Roll starts at Static.

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	2.750	0.688	7.686	4.048	0.000	0.000	0.189	0.190	7.094	249.647
1.50 roll	0.008	2.751	0.689	7.678	4.048	-0.013	-0.966	0.973	2.592	5.789	481.610
3.00 roll	0.017	2.752	0.689	7.669	4.048	-0.050	-1.931	1.756	4.891	5.690	495.296
4.00 roll	0.026	2.754	0.690	7.660	4.047	-0.113	-2.895	2.538	7.199	5.539	510.111

RH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	2.750	0.688	7.686	4.048	0.000	0.000	0.189	0.190	7.094	253.783
1.50 roll	-0.008	2.750	0.688	7.694	4.048	-0.013	0.966	-0.595	2.592	5.789	483.319
3.00 roll	-0.017	2.749	0.688	7.703	4.048	-0.051	1.931	-1.379	4.891	5.690	475.745
4.00 roll	-0.026	2.750	0.688	7.713	4.048	-0.114	2.895	-2.161	7.199	5.539	468.734

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	rc offset	roll centre height chassis	height ground	fvsax
2.796 bump	-0.693	3.117	0.806	8.382	4.054	0.423	1.095	-0.321	0.772	6.672	3.876	223.213
Static	0.000	2.750	0.688	7.686	4.048	0.000	0.000	0.189	0.190	7.094	7.094	249.647
3.084 droop	0.391	2.316	0.546	7.293	4.045	-0.672	-1.240	0.854	0.036	4.881	7.965	-1216.622

RH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	rc offset	roll centre height chassis	height ground	fvsax
2.796 bump	-0.682	2.995	0.765	8.370	4.054	0.416	1.134	-0.333	0.772	6.672	3.876	230.437
Static	0.000	2.750	0.688	7.686	4.048	0.000	0.000	0.189	0.190	7.094	7.094	253.783
3.084 droop	0.382	2.457	0.593	7.303	4.045	-0.668	-1.281	0.855	0.036	4.881	7.965	-1163.826

Equivalent suspension travel due to chassis roll

	RH	LH
0.00 roll	0.000	0.000
1.50 roll	-0.006	0.006
3.00 roll	-0.013	0.013
4.00 roll	-0.019	0.019

Side view swing axle and instant centre LH

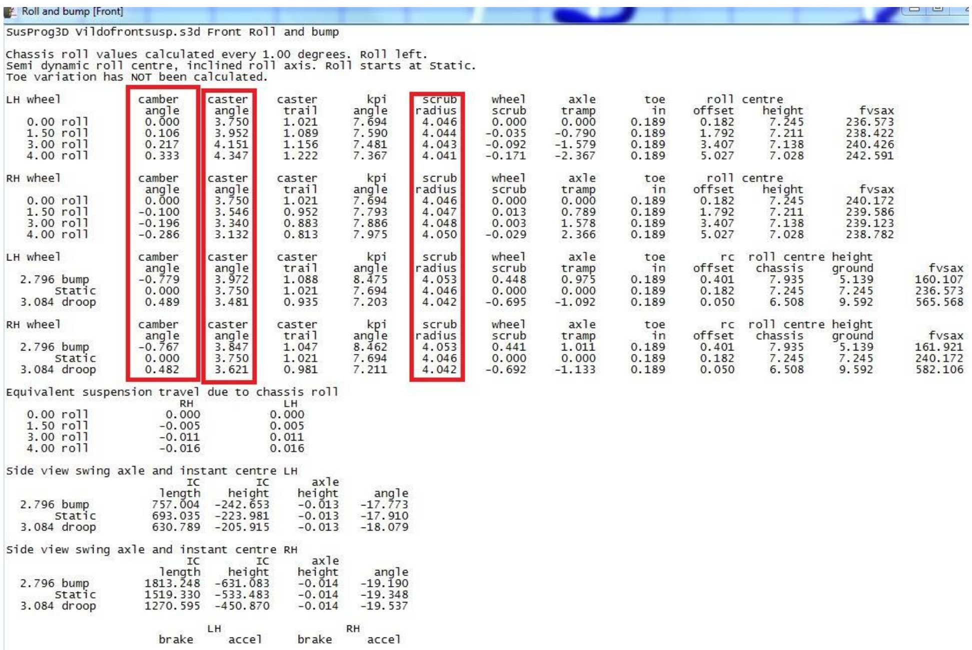
	IC length	IC height	axle height	angle
2.796 bump	474.533	-165.267	-0.014	-19.202
Static	682.276	-220.628	-0.013	-17.920
3.084 droop	381.679	-139.698	-0.015	-20.103

Side view swing axle and instant centre RH

	IC length	IC height	axle height	angle
2.796 bump	735.563	-277.235	-0.015	-20.652
Static	1480.947	-520.472	-0.014	-19.364
3.084 droop	554.191	-218.567	-0.016	-21.524

LH brake	LH accel	RH brake	RH accel

Figure 47: caster 2.75 degree and camber zero



Roll and bump [Front]

SusProg3D Vildofrontsusp.s3d Front Roll and bump

Chassis roll values calculated every 1.00 degrees. Roll left.
Semi dynamic roll centre, inclined roll axis. Roll starts at Static.
Toe variation has NOT been calculated.

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	3.750	1.021	7.694	4.046	0.000	0.000	0.189	0.182	7.245	236.573
1.50 roll	0.106	3.952	1.089	7.590	4.044	-0.035	-0.790	0.189	1.792	7.211	238.422
3.00 roll	0.217	4.151	1.156	7.481	4.043	-0.092	-1.579	0.189	3.407	7.138	240.426
4.00 roll	0.333	4.347	1.222	7.367	4.041	-0.171	-2.367	0.189	5.027	7.028	242.591

RH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	roll centre offset	roll centre height	fvsax
0.00 roll	0.000	3.750	1.021	7.694	4.046	0.000	0.000	0.189	0.182	7.245	240.172
1.50 roll	-0.100	3.546	0.952	7.793	4.047	0.013	0.789	0.189	1.792	7.211	239.586
3.00 roll	-0.196	3.340	0.883	7.886	4.048	0.003	1.578	0.189	3.407	7.138	239.123
4.00 roll	-0.286	3.132	0.813	7.975	4.050	-0.029	2.366	0.189	5.027	7.028	238.782

LH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	rc offset	roll centre height chassis	height ground	fvsax
2.796 bump	-0.779	3.972	1.088	8.475	4.053	0.448	0.975	0.189	0.401	7.935	5.139	160.107
Static	0.000	3.750	1.021	7.694	4.046	0.000	0.000	0.189	0.182	7.245	7.245	236.573
3.084 droop	0.489	3.481	0.935	7.203	4.042	-0.695	-1.092	0.189	0.050	6.508	9.592	565.568

RH wheel	camber angle	caster angle	caster trail	kpi angle	scrub radius	wheel scrub	axle tramp	toe in	rc offset	roll centre height chassis	height ground	fvsax
2.796 bump	-0.767	3.847	1.047	8.462	4.053	0.441	1.011	0.189	0.401	7.935	5.139	161.921
Static	0.000	3.750	1.021	7.694	4.046	0.000	0.000	0.189	0.182	7.245	7.245	240.172
3.084 droop	0.482	3.621	0.981	7.211	4.042	-0.692	-1.133	0.189	0.050	6.508	9.592	582.106

Equivalent suspension travel due to chassis roll

	RH	LH
0.00 roll	0.000	0.000
1.50 roll	-0.005	0.005
3.00 roll	-0.011	0.011
4.00 roll	-0.016	0.016

Side view swing axle and instant centre LH

	IC length	IC height	axle height	angle
2.796 bump	757.004	-242.653	-0.013	-17.773
Static	693.035	-223.981	-0.013	-17.910
3.084 droop	630.789	-205.915	-0.013	-18.079

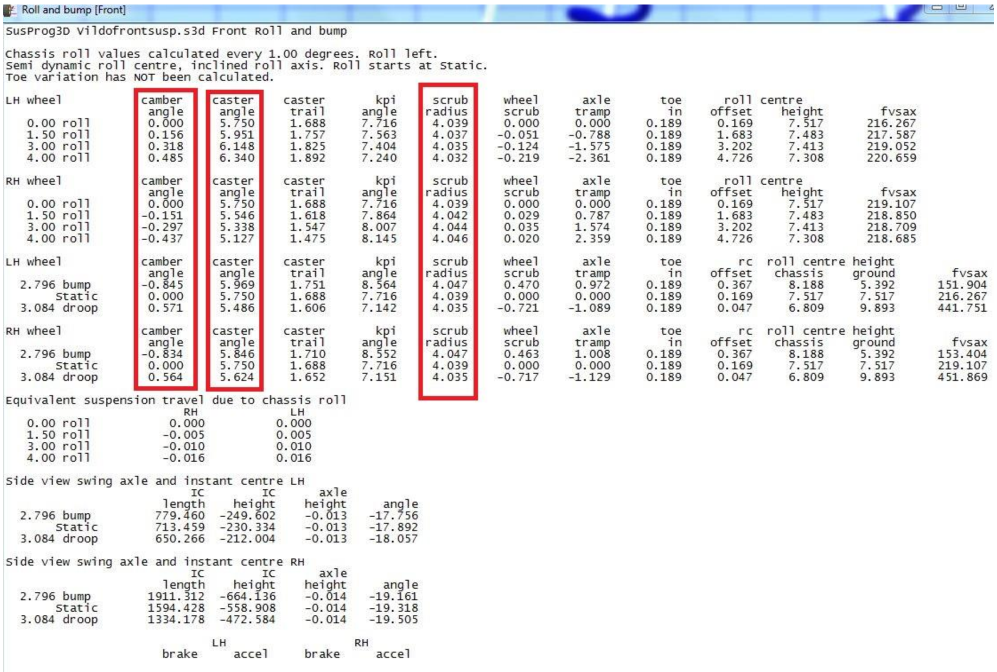
Side view swing axle and instant centre RH

	IC length	IC height	axle height	angle
2.796 bump	1813.248	-631.083	-0.014	-19.190
Static	1519.330	-533.483	-0.014	-19.348
3.084 droop	1270.595	-450.870	-0.014	-19.537

LH brake	LH accel	RH brake	RH accel

Figure 48: caster 3.75 degree and camber zero

9.3.2.2 Camber changes and caster equal to zero



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Roll and bump [Front]
SusProg3D Vildofrontsusp.s3d Front Roll and bump

Chassis roll values calculated every 1.00 degrees. Roll left.
Semi dynamic roll centre, inclined roll axis. Roll starts at Static.
Toe variation has NOT been calculated.

LH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe   roll centre        fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  height
  0.00 roll    0.000   5.750   1.688   7.716   4.039   0.000   0.000   0.189   0.169   7.517    216.267
  1.50 roll    0.156   5.951   1.757   7.563   4.037  -0.051  -0.788   0.189   1.683   7.483    217.587
  3.00 roll    0.318   6.148   1.825   7.404   4.035  -0.124  -1.575   0.189   3.202   7.413    219.052
  4.00 roll    0.485   6.340   1.892   7.240   4.032  -0.219  -2.361   0.189   4.726   7.308    220.659

RH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe   roll centre        fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  height
  0.00 roll    0.000   5.750   1.688   7.716   4.039   0.000   0.000   0.189   0.169   7.517    219.107
  1.50 roll   -0.151   5.546   1.618   7.864   4.042   0.029   0.787   0.189   1.683   7.483    218.850
  3.00 roll   -0.297   5.338   1.547   8.007   4.044   0.035   1.574   0.189   3.202   7.413    218.709
  4.00 roll   -0.437   5.127   1.475   8.145   4.046   0.020   2.359   0.189   4.726   7.308    218.685

LH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe      rc roll centre height       fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  chassis   ground
 2.796 bump   -0.845   5.969   1.751   8.564   4.047   0.470   0.972   0.189   0.367   8.188     5.392     151.904
     Static    0.000   5.750   1.688   7.716   4.039   0.000   0.000   0.189   0.169   7.517     7.517     216.267
 3.084 droop   0.571   5.486   1.606   7.142   4.035  -0.721  -1.089   0.189   0.047   6.809     9.893     441.751

RH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe      rc roll centre height       fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  chassis   ground
 2.796 bump   -0.834   5.846   1.710   8.552   4.047   0.463   1.008   0.189   0.367   8.188     5.392     153.404
     Static    0.000   5.750   1.688   7.716   4.039   0.000   0.000   0.189   0.169   7.517     7.517     219.107
 3.084 droop   0.564   5.624   1.652   7.151   4.035  -0.717  -1.129   0.189   0.047   6.809     9.893     451.869

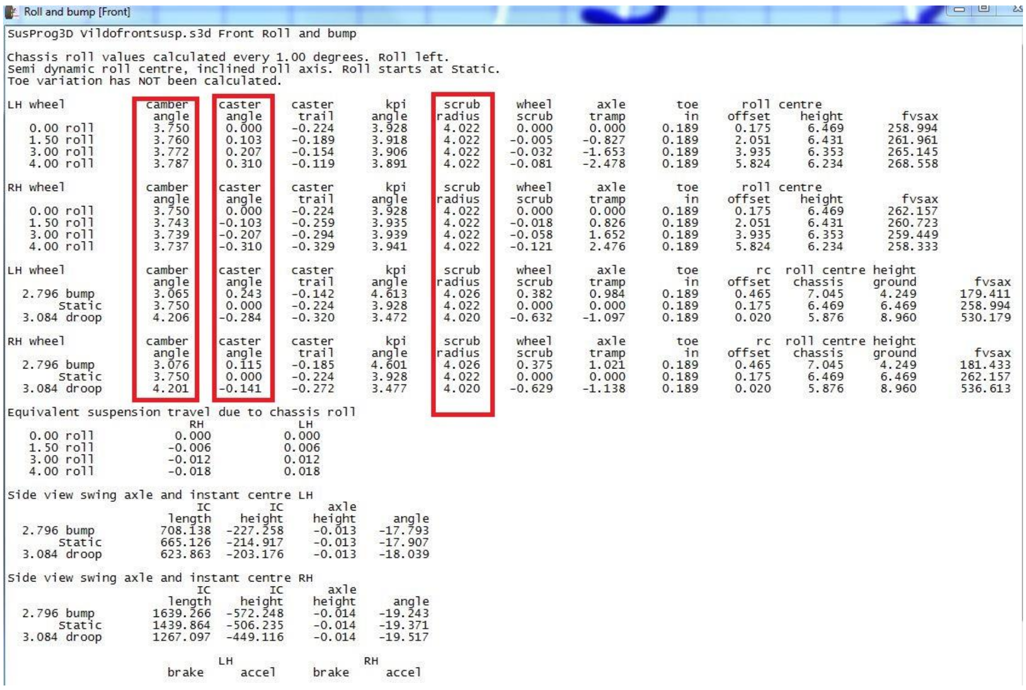
Equivalent suspension travel due to chassis roll
                    RH        LH
  0.00 roll       0.000     0.000
  1.50 roll      -0.005     0.005
  3.00 roll      -0.010     0.010
  4.00 roll      -0.016     0.016

Side view swing axle and instant centre LH
                    IC        IC      axle
                length    height    height     angle
 2.796 bump    779.460  -249.602    -0.013   -17.756
     Static    713.459  -230.334    -0.013   -17.892
 3.084 droop   650.266  -212.004    -0.013   -18.057

Side view swing axle and instant centre RH
                    IC        IC      axle
                length    height    height     angle
 2.796 bump   1911.312  -664.136    -0.014   -19.161
     Static   1594.428  -558.908    -0.014   -19.318
 3.084 droop  1334.178  -472.584    -0.014   -19.505

                       LH                  RH
                brake     accel     brake     accel
```

Figure 49: caster 5.75 degree and camber zero



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Roll and bump [Front]
SusProg3D Vildofrontsusp.s3d Front Roll and bump

Chassis roll values calculated every 1.00 degrees. Roll left.
Semi dynamic roll centre, inclined roll axis. Roll starts at Static.
Toe variation has NOT been calculated.

LH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe   roll centre        fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  height
  0.00 roll    3.750   0.000  -0.224   3.928   4.022   0.000   0.000   0.189   0.175   6.469    258.994
  1.50 roll    3.760   0.103  -0.189   3.918   4.022  -0.005  -0.827   0.189   2.051   6.431    261.961
  3.00 roll    3.772   0.207  -0.154   3.906   4.022  -0.032  -1.653   0.189   3.935   6.353    265.145
  4.00 roll    3.787   0.310  -0.119   3.891   4.022  -0.081  -2.478   0.189   5.824   6.234    268.558

RH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe   roll centre        fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  height
  0.00 roll    3.750   0.000  -0.224   3.928   4.022   0.000   0.000   0.189   0.175   6.469    262.157
  1.50 roll    3.743  -0.103  -0.259   3.935   4.022  -0.018   0.826   0.189   2.051   6.431    260.723
  3.00 roll    3.739  -0.207  -0.294   3.939   4.022  -0.058   1.652   0.189   3.935   6.353    259.449
  4.00 roll    3.737  -0.310  -0.329   3.941   4.022  -0.121   2.476   0.189   5.824   6.234    258.333

LH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe      rc roll centre height       fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  chassis   ground
 2.796 bump    3.065   0.243  -0.142   4.613   4.026   0.382   0.984   0.189   0.465   7.045     4.249     179.411
     Static    3.750   0.000  -0.224   3.928   4.022   0.000   0.000   0.189   0.175   6.469     6.469     258.994
 3.084 droop   4.206  -0.284  -0.320   3.472   4.020  -0.632  -1.097   0.189   0.020   5.876     8.960     530.179

RH wheel      camber  caster  caster     kpi   scrub   wheel    axle     toe      rc roll centre height       fvsax
               angle   angle   trail   angle  radius   scrub   tramp      in  offset  chassis   ground
 2.796 bump    3.076   0.115  -0.185   4.601   4.026   0.375   1.021   0.189   0.465   7.045     4.249     181.433
     Static    3.750   0.000  -0.224   3.928   4.022   0.000   0.000   0.189   0.175   6.469     6.469     262.157
 3.084 droop   4.201  -0.141  -0.272   3.477   4.020  -0.629  -1.138   0.189   0.020   5.876     8.960     536.613

Equivalent suspension travel due to chassis roll
                    RH        LH
  0.00 roll       0.000     0.000
  1.50 roll      -0.006     0.006
  3.00 roll      -0.012     0.012
  4.00 roll      -0.018     0.018

Side view swing axle and instant centre LH
                    IC        IC      axle
                length    height    height     angle
 2.796 bump    708.138  -227.258    -0.013   -17.793
     Static    665.126  -214.917    -0.013   -17.907
 3.084 droop   623.863  -203.176    -0.013   -18.039

Side view swing axle and instant centre RH
                    IC        IC      axle
                length    height    height     angle
 2.796 bump   1639.266  -572.248    -0.014   -19.243
     Static   1439.864  -506.235    -0.014   -19.371
 3.084 droop  1267.097  -449.116    -0.014   -19.517

                       LH                  RH
                brake     accel     brake     accel
```
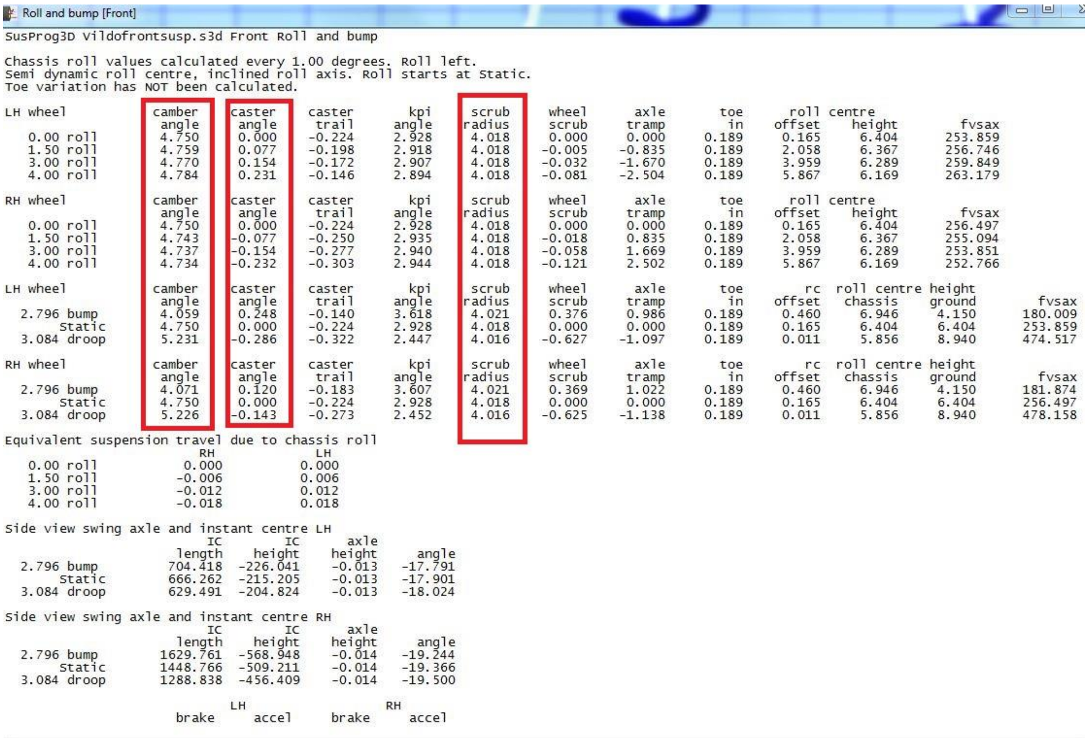

Figure 50: caster zero degree and camber 3.75 degree



```
Roll and bump [Front]
SusProg3D Vildofrontsusp.s3d Front Roll and bump

Chassis roll values calculated every 1.00 degrees. Roll left.
Semi dynamic roll centre, inclined roll axis. Roll starts at Static.
Toe variation has NOT been calculated.

LH wheel        camber  caster  caster    kpi   scrub   wheel    axle    toe   roll centre
                 angle   angle   trail  angle  radius   scrub   tramp     in  offset  height     fvsax
   0.00 roll     4.750   0.000  -0.224  2.928   4.018   0.000   0.000  0.189   0.165   6.404   253.859
   1.50 roll     4.759   0.077  -0.198  2.918   4.018  -0.005  -0.835  0.189   2.058   6.367   256.746
   3.00 roll     4.770   0.154  -0.172  2.907   4.018  -0.032  -1.670  0.189   3.959   6.289   259.849
   4.00 roll     4.784   0.231  -0.146  2.894   4.018  -0.081  -2.504  0.189   5.867   6.169   263.179

RH wheel        camber  caster  caster    kpi   scrub   wheel    axle    toe   roll centre
                 angle   angle   trail  angle  radius   scrub   tramp     in  offset  height     fvsax
   0.00 roll     4.750   0.000  -0.224  2.928   4.018   0.000   0.000  0.189   0.165   6.404   256.497
   1.50 roll     4.743  -0.077  -0.250  2.935   4.018  -0.018   0.835  0.189   2.058   6.367   255.094
   3.00 roll     4.737  -0.154  -0.277  2.940   4.018  -0.058   1.669  0.189   3.959   6.289   253.851
   4.00 roll     4.734  -0.232  -0.303  2.944   4.018  -0.121   2.502  0.189   5.867   6.169   252.766

LH wheel        camber  caster  caster    kpi   scrub   wheel    axle    toe      rc  roll centre height
                 angle   angle   trail  angle  radius   scrub   tramp     in  offset  chassis  ground     fvsax
  2.796 bump     4.059   0.248  -0.140  3.618   4.021   0.376   0.986  0.189   0.460    6.946   4.150   180.009
      Static     4.750   0.000  -0.224  2.928   4.018   0.000   0.000  0.189   0.165    6.404   6.404   253.859
  3.084 droop    5.231  -0.286  -0.322  2.447   4.016  -0.627  -1.097  0.189   0.011    5.856   8.940   474.517

RH wheel        camber  caster  caster    kpi   scrub   wheel    axle    toe      rc  roll centre height
                 angle   angle   trail  angle  radius   scrub   tramp     in  offset  chassis  ground     fvsax
  2.796 bump     4.071   0.120  -0.183  3.607   4.021   0.369   1.022  0.189   0.460    6.946   4.150   181.874
      Static     4.750   0.000  -0.224  2.928   4.018   0.000   0.000  0.189   0.165    6.404   6.404   256.497
  3.084 droop    5.226  -0.143  -0.273  2.452   4.016  -0.625  -1.138  0.189   0.011    5.856   8.940   478.158

Equivalent suspension travel due to chassis roll
                      RH            LH
   0.00 roll       0.000         0.000
   1.50 roll      -0.006         0.006
   3.00 roll      -0.012         0.012
   4.00 roll      -0.018         0.018

Side view swing axle and instant centre LH
                    IC        IC      axle
                length    height    height     angle
  2.796 bump   704.418  -226.041    -0.013   -17.791
      Static   666.262  -215.205    -0.013   -17.901
  3.084 droop  629.491  -204.824    -0.013   -18.024

Side view swing axle and instant centre RH
                    IC        IC      axle
                length    height    height     angle
  2.796 bump  1629.761  -568.948    -0.014   -19.244
      Static  1448.766  -509.211    -0.014   -19.366
  3.084 droop 1288.838  -456.409    -0.014   -19.500

                          LH                  RH
                 brake       accel     brake      accel
```

Figure 51: caster zero degree and camber 4.75

9.3.2.3 Graphs' Tables

Related with figure 41.

Camber	Scrub radius					
	Caster 0	Caster 2	Caster 4	Caster 6	Caster 8	Caster 10
-2	4.056	4.054	4.05	4.043	4.031	4.02
-1	4.053	4.052	4.048	4.04	4.029	4.017
0	4.048	4.048	4.045	4.038	4.023	4.01
1	4.04	4.04	4.036	4.027	4.018	4.004
2	4.033	4.032	4.027	4.018	4.013	3.997
3	4.026	4.024	4.022	4.012	4.007	3.994

Table 11: Camber/Caster vs Scrub Raidus

Related with Figure 40.

Caster	Scrub Radius				
	Camber -2	Camber -1	Camber 0	Camber 1	Camber 2
0	4.067	4.060	4.050	4.040	4.033
1	4.067	4.060	4.050	4.040	4.033
2	4.067	4.060	4.050	4.040	4.033
3	4.065	4.058	4.048	4.038	4.031
4	4.063	4.056	4.046	4.036	4.029
5	4.060	4.053	4.043	4.033	4.026
6	4.057	4.050	4.040	4.030	4.023
7	4.053	4.046	4.036	4.026	4.019
8	4.048	4.041	4.031	4.021	4.014
9	4.042	4.035	4.025	4.015	4.008
10	4.036	4.029	4.019	4.009	4.002
11	4.028	4.021	4.011	4.001	3.994
12	4.020	4.013	4.003	3.993	3.986
13	4.010	4.003	3.993	3.983	3.976
14	3.999	3.992	3.982	3.972	3.965
15	3.987	3.980	3.970	3.960	3.953
16	3.975	3.968	3.958	3.948	3.941
17	3.961	3.954	3.944	3.934	3.927
18	3.947	3.940	3.930	3.920	3.913
19	3.933	3.926	3.916	3.906	3.899
20	3.917	3.910	3.900	3.890	3.883

Table 12: Caster/Camber vs Scrub Radius