

POLITECNICO DI BARI

Kinematic analysis of a McPherson suspension

FINAL PROJECT

IÑAKI URRUTIA JUAMPÉREZ

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Dr. Ing. Giuseppe Carbone.

1. INTRODUCTION

Automobile industry is currently one of the widest and biggest in the world; being the automobile an essential good for society.

All types of existing vehicles have been developed in order to transporting, which must be done in the best possible conditions for people as well as for cargo. Always minimizing vibrations and decreasing interferences with the ground and environment. This will reduce the fatigue of the travellers. Regarding the comfort, it is very important for the driver, because it will make possible to maintain concentration levels, and therefore safety.

The suspension system is on charge of keeping the best transporting conditions. Besides, it must maintain the tyre and the road in contact. This is the most important function, due to the fact that enables the facility to control the vehicle.

In this Project it is tried to explain in an easy and clear way the different types of suspension systems. It will also be explained how to perform a kinematic analysis with ADAMS.

With the currently computer simulation programs it is possible to obtain a high accuracy of the behaviour of a certain simulated system. Simulation can be defined as the imitation of some real object, state of affairs, or process. The act of simulate something generally entails representing certain key characteristics or behaviors of a selected physical or abstract system.

The main objective of this Final Project is to study the behavior of McPherson suspension. For that, it is going to be used a computerized model and study its response.

It is probably true to say that the average member of the vehicle owning public is unaware of the range of duties performed by a vehicle suspension. Certainly many would recognize the importance of the suspension for ride, but fewer would identify its importance in the handling of a vehicle.

In reality a vehicle suspension is required to perform effectively under a range of operating conditions including high levels of braking and accelerating, cornering at speed and traversing rough terrain; manoeuvres which are required to be done in comfort and with safety. These requirements present the chassis engineer with some challenging problems and introduce some unavoidable design compromises.

The aim here is to give a broad treatment of the main issues for the aspiring automotive designer.

The text begins by identifying the functions of suspensions and goes on to introduce the student to the mechanics of suspension systems. Suspension kinematics is analyzed and suspension test results and characteristics are discussed.

It concludes with a section outlining some of the developments in controllable suspensions.

2. THE ROLE OF A VEHICLE SUSPENSION

The principle requirements suspensions must have are:

- To provide good ride and handling performance – this requires the suspension to have vertical compliance providing chassis isolation and ensuring that the wheels follow the road profile with very little tyre load fluctuation.
- To ensure that steering control is maintained during maneuvering. This requires the wheels to be maintained in the proper positional attitude with respect to the road surface.
- To ensure that the vehicle responds favorably to control forces produced by the tires as a result of longitudinal braking and accelerating forces, lateral cornering forces and braking and accelerating torques, which requires the suspension geometry to be designed to resist squat, dive and roll of the vehicle body;
- To provide isolation from high frequency vibration arising from tire excitation. This requires appropriate isolation in the suspension joints to prevent the transmission of road noise to the vehicle body.

It will be seen that these requirements are virtually impossible to achieve simultaneously, leading to design compromises with less than ideal performance.

3. FACTORS AFFECTING DESIGN

Suspension design, like other forms of vehicle design are affected by the reduced development times dictated by market forces. This means that for new vehicles, refined suspensions need to be designed quickly with a minimum of rig and vehicle testing prior to launch.

Consequently considerable emphasis is placed on computer-aided design requiring the use of multi-body systems analysis software of which ADAMS is typical. This software enables many ‘what-if’ scenarios to be tested quickly without the need for a lot of development testing, but they do require sophisticated mathematical models to be developed for various components and sub-systems.

In addition to the functional constraints placed on a given design, suspensions are also required to meet certain performance targets which vary across the range of vehicles. There are also other limitations such as cost, weight, packaging space, requirements for robustness and reliability, together with manufacturing, assembly and maintenance constraints.

4. DEFINITIONS AND TERMINOLOGY

There is a lot of terminology associated with suspension design which may appear novel to the student meeting the subject for the first time. Most of this will be described as it arises, but it may prove useful to introduce the vehicle axis system and terminology associated with wheel position at this stage.

4.1 Vehicle axis system and terminology

In simple studies of whole vehicle braking, accelerating and turning analyses, it is appropriate to position the origin of the vehicle axes at the centre of gravity (CG) of the whole vehicle.

However, in suspension design we are concerned with the movement of the vehicle body (the sprung mass) relative to the other moving parts of the suspension and wheels (the unsprung masses). So in this case it is usual to place the vehicle axis system (a right-handed set of axes) at the CG of the sprung mass as shown in Figure 1.

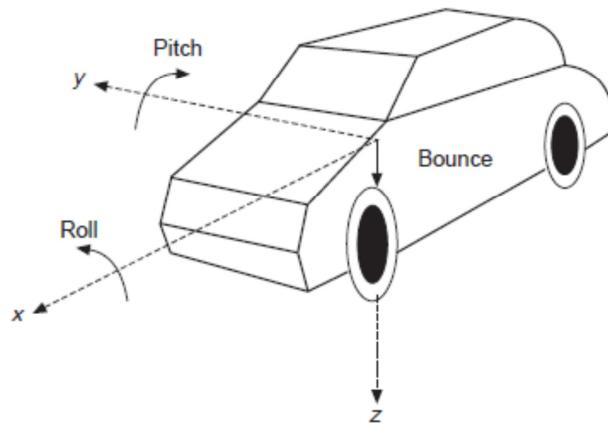


Figure 1. Sprung mass axes and displacements relevant to suspension analyses

Since the sprung mass is treated as a rigid body, it has six degrees of freedom (DOF) comprising three translations and three rotations. Only three of these are relevant for suspension studies, bounce, roll and pitch as shown in Figure 1.

4.2 Definitions for wheel orientation

One of the functions of a suspension system is to maintain the position of the wheels constant relative to the road throughout the motion of the suspension, so it is important to identify how the wheel position is defined. Figure 2 provides these definitions:

- Camber angle: is the angle between the wheel plane and the vertical – taken to be positive when the wheel leans outwards from the vehicle.
- Swivel pin (kingpin) inclination: is the angle between the swivel pin axis and the vertical. The swivel pin inclination has the effect of causing the vehicle to rise when the wheels are turned and produces a noticeable self-centring effect for swivel pin inclinations greater than 15°.
- Swivel pin (kingpin) offset: is the distance between the centre of the tyre contact patch and the intersection of the swivel pin axis and the ground plane, taken to be positive when the intersection point is at the inner side of the wheel. The swivel-pin offset reduces steering effort because the wheel tends to roll during turning. With zero offset, the kingpin axis intersects the centre of the tyre contact patch. If the wheel is steered under these conditions there is significant tyre scrub at the front and rear of the contact patch leading to a significant steering effort.

The disadvantage of offset is that longitudinal forces at the tyre contact patch due to braking, or striking a bump or pothole is transmitted through the steering mechanism to the steering wheel.

- Castor angle: is the inclination of the swivel pin axis projected into the fore-aft plane through the wheel centre, positive in the direction shown. Castor angle provides a selfaligning torque for non-driven wheels.
- Toe-in and Toe-out: is the difference between the front and rear distances separating the centre plane of a pair of wheels, quoted at static ride height, toe-in is when the wheel centre planes converge towards the front of the vehicle as shown in Figure 2(c).

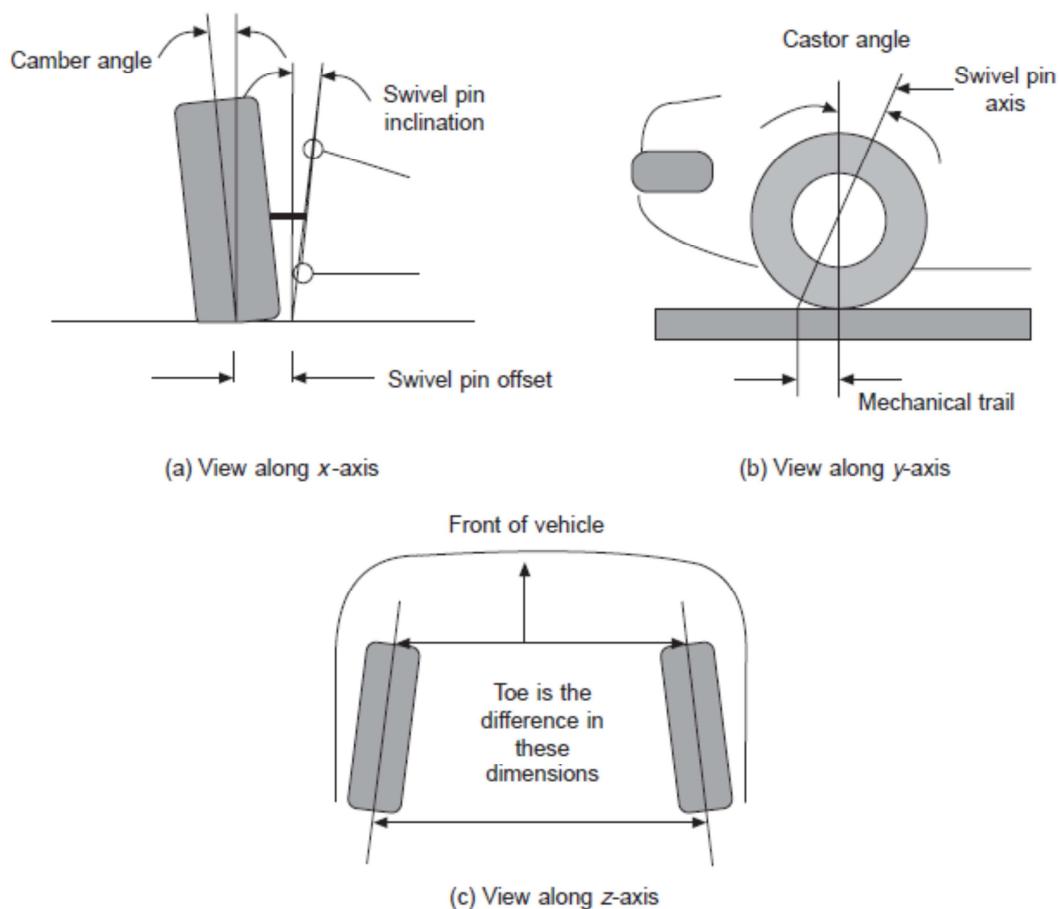


Figure 2. Wheel orientation

Suspension travel can result in changes in wheel orientation relative to the ground and consequently to steering effects unrelated to those initiated by the driver of the vehicle. When these arise from vertical travel of the unsprung mass they are referred to as bump steer effects.

Roll of the sprung mass can induce roll steer and flexibility in the suspension mechanism can also give rise to compliance steer.

5. MOBILITY OF SUSPENSION MECHANISMS

Suspension systems are in general three-dimensional mechanisms and as such are difficult to analyse fully without the aid of computer packages. Their analysis is complicated by the inclusion of many compliant bushes which effectively result in links having variable lengths.

Not with standing these complications it is possible to gain an appreciation of the capabilities and limitations of various mechanisms used in suspension design by neglecting bush compliances and concentrating on the basic motion of suspension mechanisms.

A fundamental requirement of a suspension mechanism is the need to guide the motion of each wheel along a (unique) vertical path relative to the vehicle body without significant change in camber. This requirement has been addressed by employing various Single Degree of Freedom (SDOF) mechanisms which have straight line motion throughout the deflection of the suspension.

Despite the apparent complexity of some suspension systems, a basic understanding of their kinematics can be derived from a two-dimensional analysis, i.e. by considering the motion in a vertical transverse plane through wheel centre. Fundamental to this analysis is an understanding of how the number of degrees of freedom (mobility in mechanisms parlance) of a mechanism are related to the number of links and the types of kinematic constraint imposed on them.

In general the aim is for a SDOF or a mobility of one. Mechanisms which have a mobility of zero are structures, i.e. not designed for motion, while those having two degrees of freedom require two prescribed inputs to position them uniquely. This is not desirable for suspensions.

Most of the kinematic connections between the members of a suspension mechanism can be reduced down to the kinematic pairs shown in Figure 3.

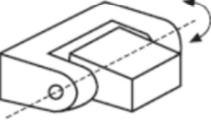
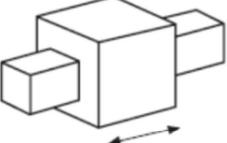
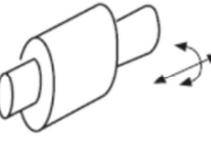
Name of pair	Geometric form	Degrees of freedom
Revolute R		1
Prism P		1
Cylinder C		2
Sphere S		3

Figure 3. Kinematic pairs

Each has an associated number of degrees of freedom and can be classified as lower pairs (connections having a SDOF) or higher pairs (more than one DOF). It has been shown that the mobility M , of a plane mechanism forming a closed kinematic chain, is related to the number of links n , the number of lower pairs j_l and the number of higher pairs j_h . According to the Kutzbach criterion:

$$M = 3(n-1) - j_h - 2j_l \quad (1)$$

For spatial mechanisms there is an equivalent equation.

The use of equation 1 can be illustrated with reference to McPherson strut suspensions whose kinematics are represented two dimensionally in Figure 4. This suspension can be seen to represent a single closed kinematic chain.

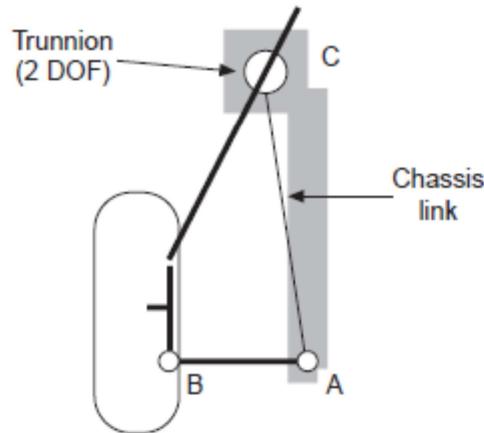


Figure 4. Two-dimensional kinematic of a common suspension mechanism

In the kinematically equivalent mechanism for the McPherson strut (Figure 3) the telescopic damper is replaced with an extension of the wheel attachment to pass through a trunnion (a 2DOF joint) at C. The mechanism thus has three links AB, BC and CA: $n = 3$.

There are two lower pairs (one at A and one at B) and one upper pair (at C). This gives $j_l = 2$ and $j_h = 1$. Giving $M = 3 \times (3 - 1) - 1 - (2 \times 2) = 1$.

While mobility analysis is useful for checking for the appropriate number of degrees of freedom, it does not help in developing the geometry of a mechanism to provide the desired motion. For suspension mechanisms this process is called position synthesis and requires the use of specialized graphical and analytical techniques aided by computer software. This departure from the well established suspension types is

only required when it is necessary to produce enhanced suspension characteristics, e.g. to produce changes in camber and toe under certain operating conditions to improve handling, which is out of this Project.

6. SUSPENSION TYPES

There is a range of generic suspensions which are commonly used. In this section I will describe some of them and discuss some of their important features. Some of the associated diagrams aim to convey only the essential kinematic principles.

Factors which primarily affect the choice of suspension type at the front or rear of a vehicle are engine location and whether the wheels are driven or guided. In general, suspensions can be classified as dependent or independent types.

With dependent suspensions the motion of a wheel on one side of the vehicle is dependent on the motion of its partner on the other side, that is when a wheel on one side of the vehicle strikes a pot-hole the effect of it is transmitted directly to its partner on the other side. This has a detrimental effect on the ride and handling of the vehicle.

With independent suspensions the motion of wheel pairs is independent, so that a disturbance at one wheel is not directly transmitted to its partner. This leads to better ride and handling capabilities.

6.1 Dependent systems

As a result of the upward trend in vehicle refinement, these are not so common on passenger cars. However, they are still commonly used on commercial and off-highway vehicles.

They have the advantages of being relatively simple in construction and almost completely eliminate camber change thereby reducing tyre wear.

They are occasionally used in conjunction with non-driven axles (dead axles) at the front of some commercial vehicles, but are more common at the rear of front-wheel drive light commercial and off-road vehicles.

This type of system is also used in conjunction with rear driven axles (live axles) on commercial and off-highway vehicles.

There are a number of ways of mounting a solid axle. The following two examples are very common.

The Hotchkiss rear suspension (Figure 5)

In this case the axle is mounted on longitudinal leaf springs, which are compliant vertically and stiff horizontally. The springs are pin-connected to the chassis at one end and to a pivoted link at the other. This enables the change of length of the spring to be accommodated due to loading.

Earlier problems with inter-leaf friction (which affected ride performance) have been overcome by replacing each of the multi-leaf springs with a single tapered leaf.

The requirements for good ride in passenger cars call for highly compliant leaf springs which lead to poor locational properties.

Such flexible springs are also unable to control high braking and accelerating torques (leading to axle tramp). These latter problems can be overcome to some extent by using Panhard rods to control lateral deflections and trailing arms to resist braking and accelerating torques. (To provide transverse stiffness without controlling suspension motion.) Despite these improvements this type of suspension is now rarely used on passenger cars.

However, for those vehicles where ride is not of primary importance they are still widely used. An example of this is the mid-range commercial van where load carrying capacity is important. In this case heavier or two-stage springing can be used to overcome the problems discussed above.

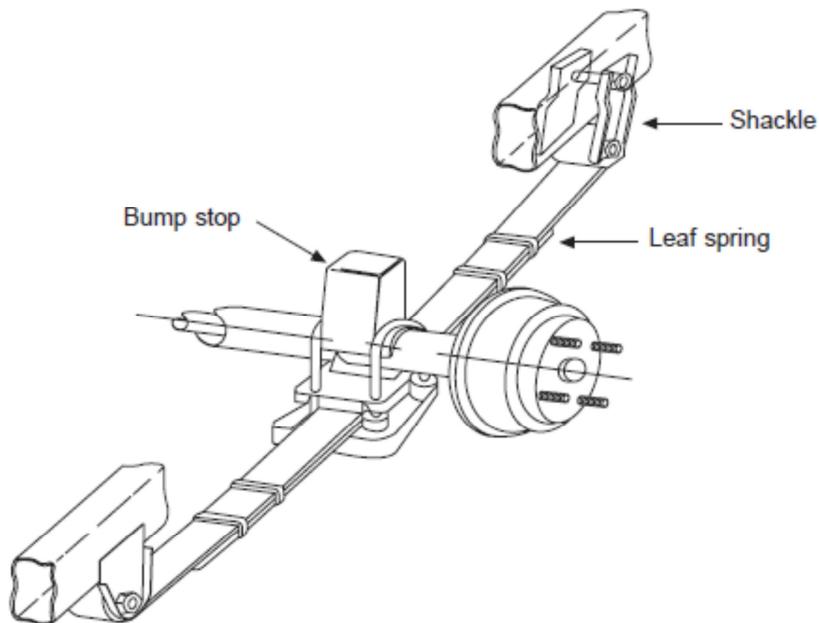


Figure 5. Hotchkiss rear suspension

Trailing arms (Figure 6)

Various configurations are possible, provided that they permit vertical and roll freedoms. Either coil or air springs can be used, the latter tending to give better ride performance. Lateral control can be provided by angling the upper links (as shown) or by using a Panhard rod. Compared to the Hotchkiss system, the four link design gives greater flexibility in the choice of roll-centre location, anti-squat and anti-dive geometry, and roll-steer.

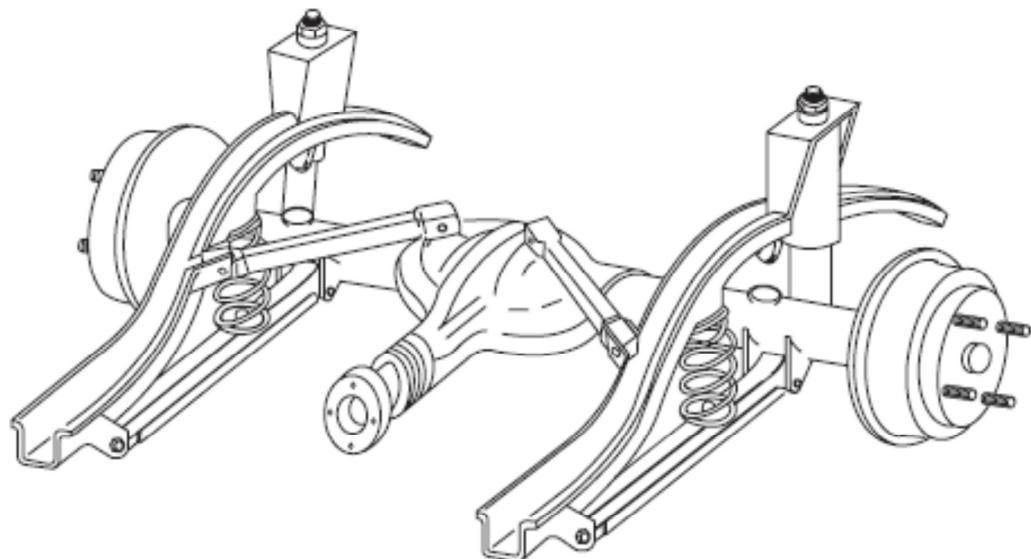


Figure 6. Trailing arm – rigid axle suspension

6.2 Semi-dependent systems

In this form of suspension, the rigid connection between pairs of wheels is replaced by a compliant link. This usually takes the form of a beam which can bend and flex providing both positional control of the wheels as well as compliance. Such systems tend to be very simple in construction but lack scope for design flexibility.

An example of this form of suspension system is the trailing twist axle design shown in Figure 7.

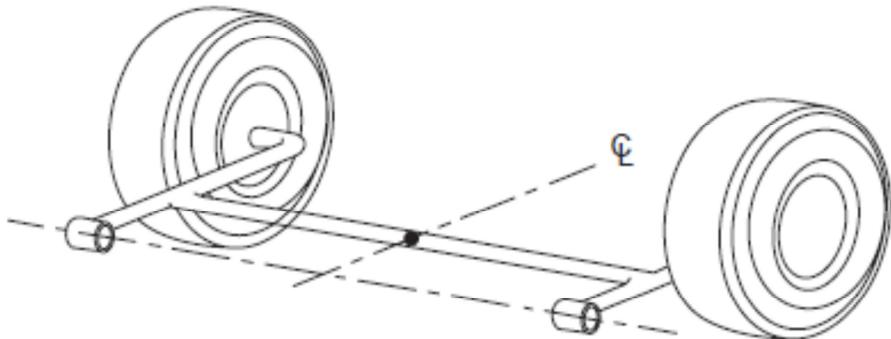


Figure 7 Trailing twist axle suspension

Additional compliance can be provided by rubber or hydro-elastic springs. Wheel camber is, in this case, the same as body roll.

6.3 Independent systems

This form of suspension has benefits in packaging and gives greater design freedom when compared to dependent systems. Some of the most common forms of front and rear designs will be considered. Both the McPherson strut, double wishbone

and multi-link systems are employed for front and rear wheel applications. The trailing arm, semi-trailing arm and swing axle systems tend to be used predominantly for rear wheel applications.

McPherson strut (Figure 8)

The vertical movement is constrained by the telescopic pivoted link (damper) and compliance is provided by a coil spring. Lateral constraint is provided by the lower transverse arm and longitudinal constraint is provided by the longitudinal link. Various options can be adopted for the constructional detail, these include replacing the lower transverse arm by an A-shaped member (A-arm) having the apex of the A connected to the knuckle and the base of the A connected to the chassis by two pin connections. This design obviates the need for the longitudinal member shown in the diagram since both longitudinal and transverse forces can now be reacted by the A-arm. Also the spring can be either co-axial or parallel with the damper. Simplicity is the main benefit in this case.

The disadvantages are:

- The installation height can be a problem when a low bonnet line is required by the vehicle stylist.
- The strut has to react against a moment imposed by wheel loading, but this problem can be lessened by angling the suspension spring.

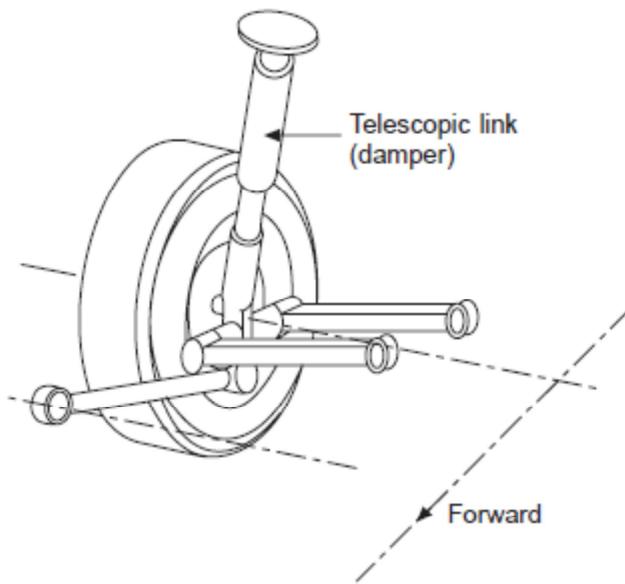


Figure 8 McPherson strut suspension

Double wishbones (Figure 9)

This design produces a classic four-bar mechanism when viewed from the front of the vehicle.

It has the knuckle located at the center of the coupler link and is therefore capable of providing straight-line motion to the knuckle. However, because of packaging constraints it is normal to make the upper wishbone shorter than the bottom. The double wishbones provide the constructional strength to react transverse and longitudinal loads.

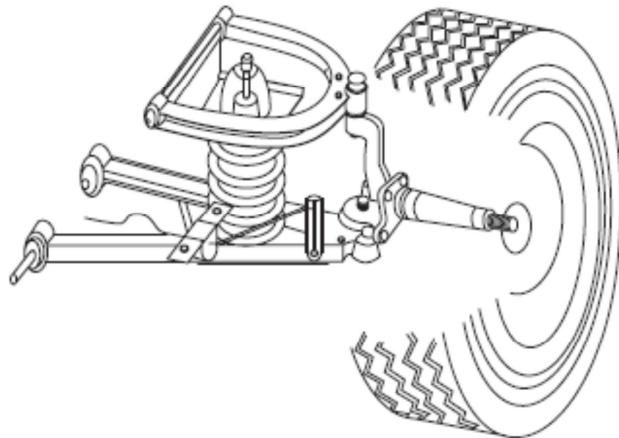


Figure 9 Double wishbone suspension

Trailing arms (Figure 10)

This is used in rear axle applications with either front or rear wheel drive. It can be used with a variety of springs including torsion bars, coil springs, rubber springs or hydro-elastic types.

It is a relatively low cost form of suspension, but offers little flexibility in terms of kinematic design options, but there is scope for anti-lift control by adjusting the pivot position.

Camber change is the same as body roll and caster change can be substantial. There is no toe change so roll steer is zero.

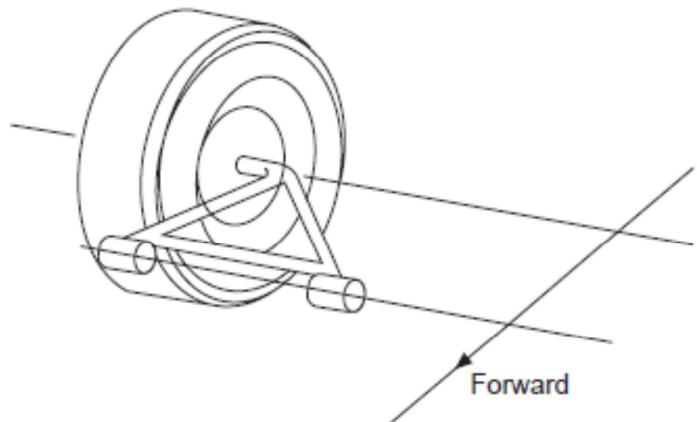


Figure 10 Trailing arm suspension

Swing axles (Figure 11)

This is a very simple form of suspension used with driven axles. With short swing axles the camber changes and tyre scrub (is related to the horizontal transverse movement of the tyre relative to the road surface) can be considerable for the range of suspension travel. This form of suspension is particularly prone to suspension jacking (a form of instability induced into the suspension system during cornering).

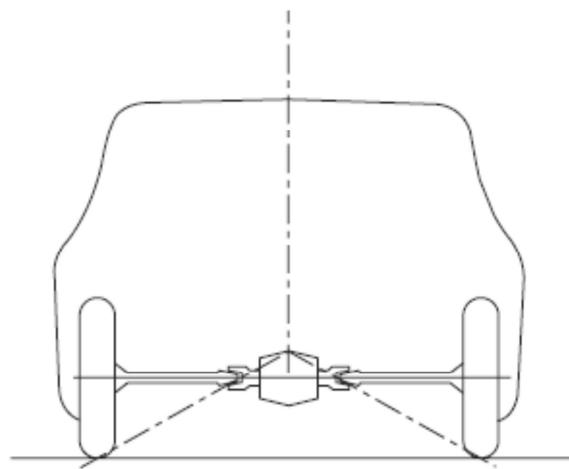


Figure 11 Swing axle suspension

Semi-trailing arms (Figure 12)

This type of suspension is a cross between the swing axle and pure trailing arm designs. It allows a compromise between the control of camber and jacking. It is essential that the geometry is carefully selected to limit the amount of steer induced by the trail angle. However, this is a feature which can be exploited to provide small amounts of rear wheel steer to improve handling performance.

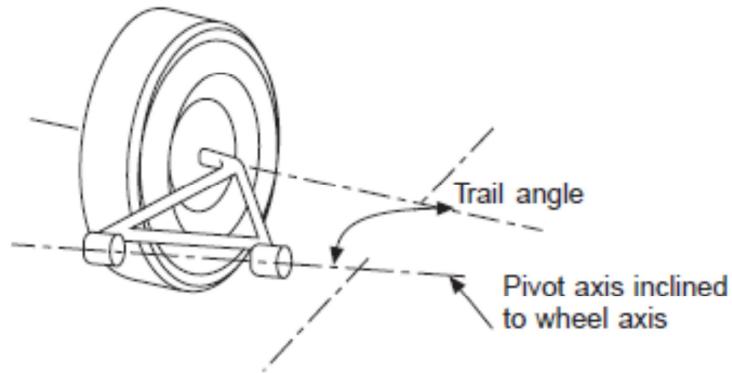


Figure 12 Semi-trailing arms

Multi-link suspensions (Figure 13)

There is a wide variety of multi-link designs. Figure 10.13 shows a five link design which is used to control the separate functions required of a suspension. The ends of the links are mounted in flexible bushes, which are necessary because the mechanism is over-constrained kinematically.

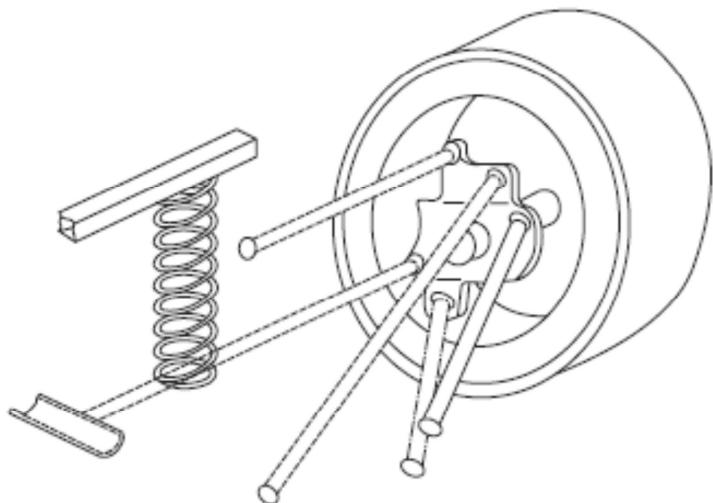


Figure 13 A multi-link suspension

7. KINEMATIC ANALYSIS

7.1 Creating template

One of the first stages of suspension design (once the type of suspension has been selected) is to size the mechanism and ensure it is capable of fitting into the packaging envelope. As part of this process it is necessary to check the geometry variations of the suspension over its operating range and ratios of spring and damper travel relative to wheel travel.

Comprehensive analysis of suspension motions requires the inclusion of joint compliances and the problem then becomes one of force-motion analysis. This requires the use of specialized computational software (ADAMS in this case) and associated modelling skills. If joint compliances are neglected, the problem is simplified

into a purely kinematic one and if, furthermore, the problem can be assumed to be 2-dimensional, an even more basic analysis can be carried out using graphical or computational methods. The latter can be aided with general purpose analysis software.

In this treatment of suspension design fundamentals, graphical and computational analysis will be restricted to two-dimensional examples.

In graphical analysis of suspension motion it should be recognized that relationships between the relative movement of parts of the mechanism can be determined from sets of velocity diagrams.

To cover the full range of suspension travel it is necessary to draw a number of diagrams corresponding to different positions of the mechanism. This has the advantage of providing a good sensation for what is happening, but the downside is the lack of accuracy and the tedium of drawing many diagrams..

If a computational approach is adopted, there can be significant effort required to formulate the problem, but once this has been done mathematical software can be used to solve equations and present the results numerically or graphically. The significant benefits of this approach are improved accuracy and the ability to try out ‘what-if’ scenarios.

ADAMS

Computer-aided design (CAD) is the use of computer technology for the process of design and design/documentation. CAD software, or environments, provides the user with tools for the purpose of designing processes; drafting, documentation, and manufacturing processes. It is often in the form of electronic files for print or machining operations. The development of CAD-based software is in direct correlation with the processes it seeks to economize; industry-based software (construction, manufacturing, etc.) typically uses vector-based (linear) environments whereas graphic-based software utilizes raster-based (pixelated) environments.

CAD environments often involve more than just shapes. As in the manual drafting of technical and engineering drawings, the output of CAD must convey information, such as materials, processes, dimensions, and tolerances, according to application-specific conventions.

It may be used to design curves and figures in two-dimensional (2D) space; or curves, surfaces, and solids in three-dimensional (3D) objects.

CAD is an important industrial art extensively used in many applications, including automotive, shipbuilding, aerospace industries, industrial and architectural design, prosthetics, and many more. CAD is also widely used to produce computer animation for special effects in movies, advertising and technical manuals. The modern ubiquity and power of computers means that even perfume bottles and shampoo dispensers are designed using techniques unheard of by engineers of the 1960s.

Because of its enormous economic importance, CAD has been a major driving force for research in computational geometry, computer graphics (both hardware and software), and discrete differential geometry.

MSC ADAMS is a complex CAD program formed of several working environments with a wide field of applications, which are:

Adams / Car

Adams / Chassis

Adams / Driveline

Adams / Insight

Adams / Flex

Adams / Engine

Adams / Postprocessor

Adams / Solver

Adams / View

Among all these options, the application which has been used in this Project is the first one, which is Adams Car, in order to work with the components and working conditions of a suspension. Adams/Car uses different templates to create subsystems for Steering, Suspension, brakes...

These subsystems are then assembled to create full vehicle assemblies, or as in this case, to create the suspension assembly.

Adams/Car is a powerful analysis tool for multi-body systems, such us rigid and flexible bodies, forces, links, and even 2D and 3D systems. Besides, it lets to visualize the results easily, due to the post processing tool, plotting and graphic analysis, all supported by exporting videos of the movements.

It all starts with the building of the template. A template is a pattern or overlay used in graphic arts (drawing, painting, etc) and sewing to replicate shapes or designs. Once the topology of the template has been built, the analysis starts.

STEPS AND PROCESS

✓ Creating a Template

It is needed to create a template in which to build suspension parts. Then, assign to the template a major role as a suspension template, because a major role defines the function the template serves for the vehicle.

To create a template:

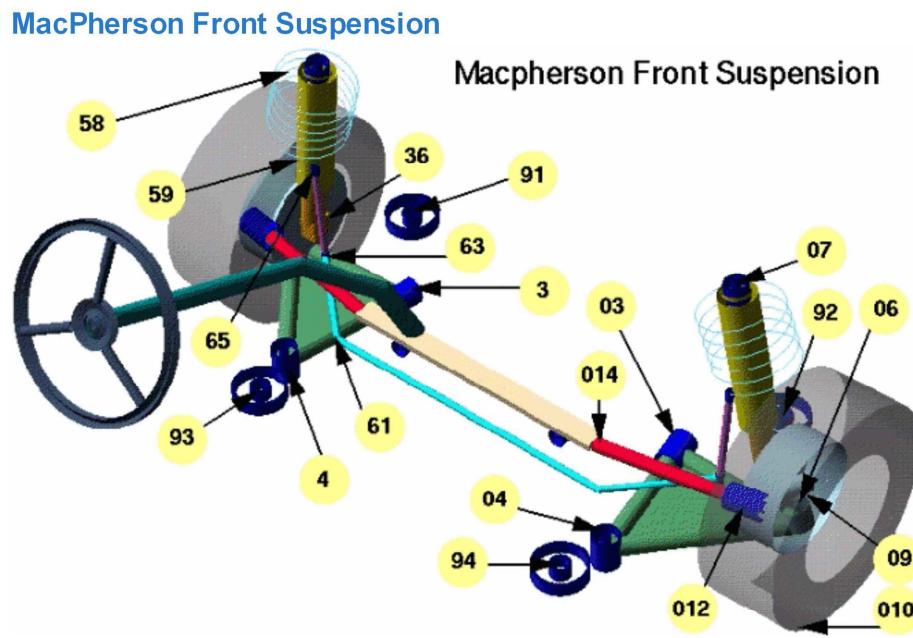
1.-Start Adams/Car Template Builder

2.-File, then select New.

3.-In the Name box, McPherson, and verify that Major Role is set. Ok.

✓ Building Suspension Parts

We create parts in ADAMS/Car through a three-step process. First, we need to create hardpoints that define key locations on the part. Then, we create the actual part. Finally we add geometry to our new part.



Geometry Hardpoints

3	lca_front	4	lca_rear
6	lower_ball_joint	7	top_mount
9	wheel_center	10	contact_patch
11	spindle_align	12	tierod_outer
14	tierod_inner	58	spring_seat_upper
59	spring_seat_lower	36	strut_knuckle
40	tensionstrut_lca	1	1
61	arb_bushing(left)	62	arb_bushing(right)
63	droplink_bar(left)	64	droplink_bar(right)
65	droplink_external(left)	66	droplink_external(right)
75	bumper_upper(left)	76	bumper_upper(right)
71	bumper_lower(left)	72	bumper_lower(right)
77	rebound_upper(left)	78	rebound_upper(right)
73	rebound_lower(left)	74	rebound_lower(right)
91	subframe_front(left)	92	subframe_front(right)
93	subframe_rear(left)	94	subframe_rear(right)

➤ Creating the Control Arm:

First, we have to define it by the hardpoints, and then we can create it. It is possible to modify later this hardpoints to see the effects in the suspension.

To build the hardpoints:

1. From the Build menu, Hardpoint, and then New.
2. In the Hardpoint Name text box, enter arm_outer.
3. Verify that Type is set to left.

All the hardpoints must be referred to the left. ADAMS/Car automatically creates a symmetrical pair about the central longitudinal axis.

4 In the Location text box, 0, -600, 0.

5 Then Apply.

6 We have to repeat steps from 2 to 5 to build these other hardpoints. (Figure 7.1)

Hardpoint Name:	Location:
arm_front	-150, -350, 0
arm_rear	150, -350, 0

Figure 7.1

Now we can create the control arm. From the Build menu, we go to Parts, General Parts and select New. We must fill in this dialog box. (Figure 7.2)

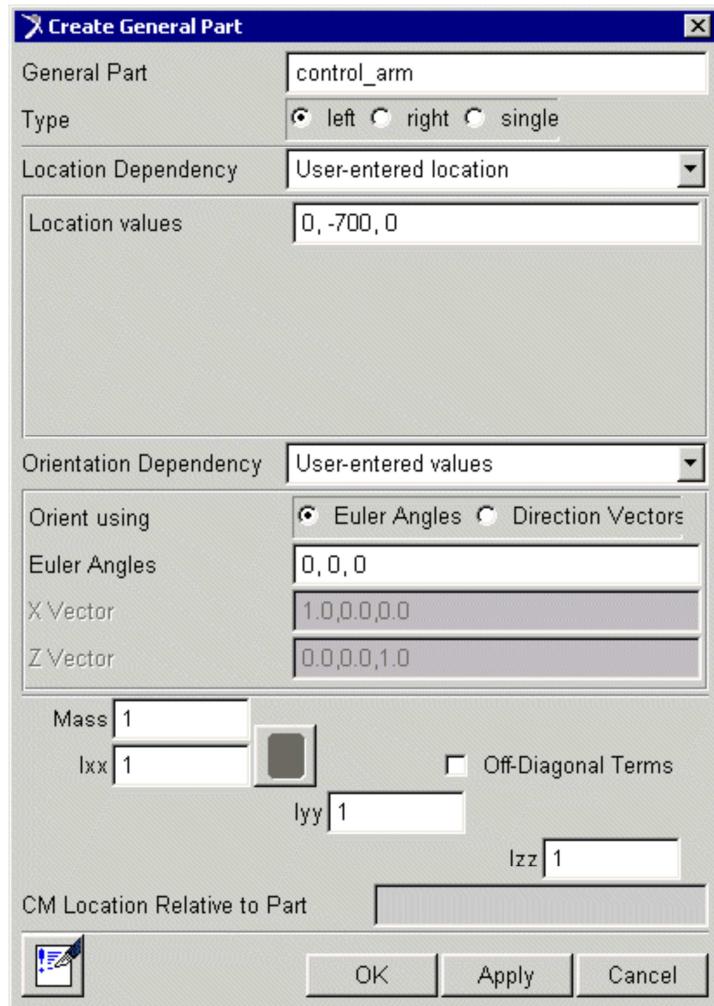


Figure 7.2

➤ To create the control arm geometry:

1 From the Build menu, Geometry, Arm, and then New.

2 Fill in the box of the control arm as follows:

❖ Arm Name: control_arm

❖ General Part: ._McPherson.gel_control_arm

❖ Coordinate Reference #1: ._McPherson.ground.hpl_arm_outer

❖ Coordinate Reference #2: ._McPherson.ground.hpl_arm_front

❖ Coordinate Reference #3: ._McPherson.ground.hpl_arm_rear

❖ Thickness: 10

3 Select Calculate Mass Properties of General Part.

4 Set Density to Material, the Ok.

➤ Creating the Wheel Carrier:

To create the wheel carrier, we must first create three hardpoints that define the location of the wheel carrier. Then we define the wheel carrier part using these hardpoint locations.

Next, we have to add link geometry to the wheel carrier.

To create the hardpoints:

1. Build, Hardpoint, New.

2. Fill in the table, pressing Apply after each hardpoint. (Figure 7.3)

Hardpoint Name:	Location:
wheel_center	0, -800, 100
strut_lower	0, -650, 250
tierod_outer	150, -650, 250

Figure 7.3

- To create the wheel carrier:

1. build, Parts, General Part and then, Wizzard.

- ❖ General Part Name: wheel_carrier
- ❖ Geometry Type: Arm
- ❖ Coordinate Reference #1: ._McPherson.ground.hpl_wheel_center
- ❖ Coordinate Reference #2: ._McPherson.ground.hpl_arm_outer
- ❖ Coordinate Reference #3: ._McPherson.ground.hpl_strut_lower
- ❖ Thickness: 10.

To finish, press Ok.

- To add the wheel carrier link geometry:

1. Build, Geometry, Link and New

2. Create the wheel carrier part as follows:

- ❖ Link Name: carrier_link
- ❖ General Part: ._McPherson.gel_wheel_carrier
- ❖ Coordinate Reference #1: ._McPherson.ground.hpl_strut_lower
- ❖ Coordinate Reference #2: ._McPherson.ground.hpl_tierod_outer
- ❖ Radius: 10

3. Select- Calculate Mass Properties of General Part. Then Ok.

➤ Creating the Strut:

In this section, we create the strut part for our suspension template. Just as done for the control arm, we enter the location, orientation, and mass properties for the strut part.

Because the strut geometry would not be visible from inside the damper, there is no need to give the strut any geometry.

To define the strut part:

1. Build, Parts, General Part, New.

2. Define the strut part as follows:

❖ General Part: strut

❖ Location values: 0, -600, 600

❖ Euler Angles: 0, 0, 0

❖ Mass/lxx/lyy/lzz: 1.

Then Ok.

➤ Creating the Tie Rod

First we must create a hardpoint that defines the tie in order to create it later. (Figure 7.4)

- ❖ Hardpoint Name: tierod_inner
- ❖ Location: 200, -350, 250

Figure 7.4

Then select Ok

Now, we create the tie rod part:

From the Build menu, we go to Parts, General Parts and Wizard.

- ❖ General Part Name: tierod
- ❖ Geometry Type: Link
- ❖ Coordinate Reference #1: ._McPherson.ground.hpl_tierod_outer
- ❖ Coordinate Reference #2: ._McPherson.ground.hpl_tierod_inner
- ❖ Radius: 10

Then Ok.

Besides, we have to create variables defining toe and camber angles, because they are commonly used for suspension analyses, and with ADAMS, it is possible to create both in one step.

From the Build menu, point to Suspension Parameters, Toe/Camber Values and then Set. In addition, we must fill in the box: (Figure 7.5)

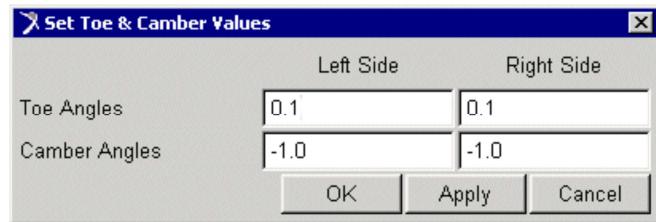


Figure 7.5

➤ Creating the Hub

We have to create the hub based on the construction frame, which is an element used when an entity requires a specified orientation in addition to a location.

To build the construction frame:

Build/Construction Frame/ New

- ❖ Construction Frame: hub_bearing
- ❖ Location Dependency: Delta location from coordinate
- ❖ Coordinate Reference: ._McPherson.ground.hpl_wheel_center
- ❖ Orientation Dependency: Toe/Camber
- ❖ Toe Parameter Variable: ._McPherson.pvl_toe_angle
- ❖ Camber Parameter Variable: ._McPherson.pvl_camber_angle

Ok

And now we can create the hub:

Build/Parts/General Part and New.

- ❖ General Part: hub
- ❖ Location Dependency: Delta location from coordinate
- ❖ Coordinate Reference: cfl_hub_bearing
- ❖ Location values: 0, 0, 0
- ❖ Orientation Dependency: Delta orientation from coordinate
- ❖ Construction Frame: cfl_hub_bearing
- ❖ Orientation: 0, 0, 0
- ❖ Mass/Ixx/Iyy/Izz: 1

Then Ok.

Next, we can create the cylinder geometry for the hub:

- ❖ Cylinder Name: hub
- ❖ General Part: ._McPherson.gel_hub
- ❖ Construction Frame: ._McPherson.ground.cfl_hub_bearing
- ❖ Radius: 30
- ❖ Length in Positive Z: 30
- ❖ Length in Negative Z: 0
- ❖ Color: magenta

We select Calculate Mass Properties of General Part, and then Ok.

The template now includes the hub.

The next step in the process is to create the damper and the spring. First we create a hardpoint which we will define the damper. Then we create the damper that is defined by force-velocity curve.

The hardpoints: (Figure 7.6)

Hardpoint Name: **strut_upper**
Location: **0, -600, 600**

From the Build menu, Forces, Damper and New:

- ❖ Damper Name: damper
- ❖ I Part: ._McPherson.gel_wheel_carrier
- ❖ J Part: ._McPherson.gel_strut
- ❖ I Coordinate Reference: ._McPherson.ground.hpl_strut_lower
- ❖ J Coordinate Reference: ._McPherson.ground.hpl_strut_upper

Then Ok.

Before we create the spring, we must create the hardpoint that defines the lower spring seat. Then, to define the spring, we must specify this:

- The two bodies between which we want the force to act.
- The location of this force on each body.

-The length of the spring, in order to derive the design preload on the spring.

ADAMS / Car calculates the force given to the spring by this equation: (Figure 7.7)

$$C = FL - IL + DM'(i, j)$$

Figure 7.7 Force = $-k(C - DM(i, j))$

-C is a constant.

-FL is the free length of the spring.

- IL is the defined installed length.

-DM(i,j) is the change in the displacement between the i and j coordinate reference points as the simulation progresses.

-DM'(i,j) is the initial displacement between the i and j coordinate reference points. If you enter a smaller value for DM(i,j), ADAMS/Car calculates an increased preload for the spring.

- “Force” represents the spring force.

- k is the nonlinear spring stiffness.

And now we can create the hardpoint of the spring: (Figure 7.8)

Hardpoint Name: **spring_lower**
Location: **0, -650, 300**

Then the actual spring:

-Build, forces, Spring and New. We must fill in the box. (Figure 7.9)

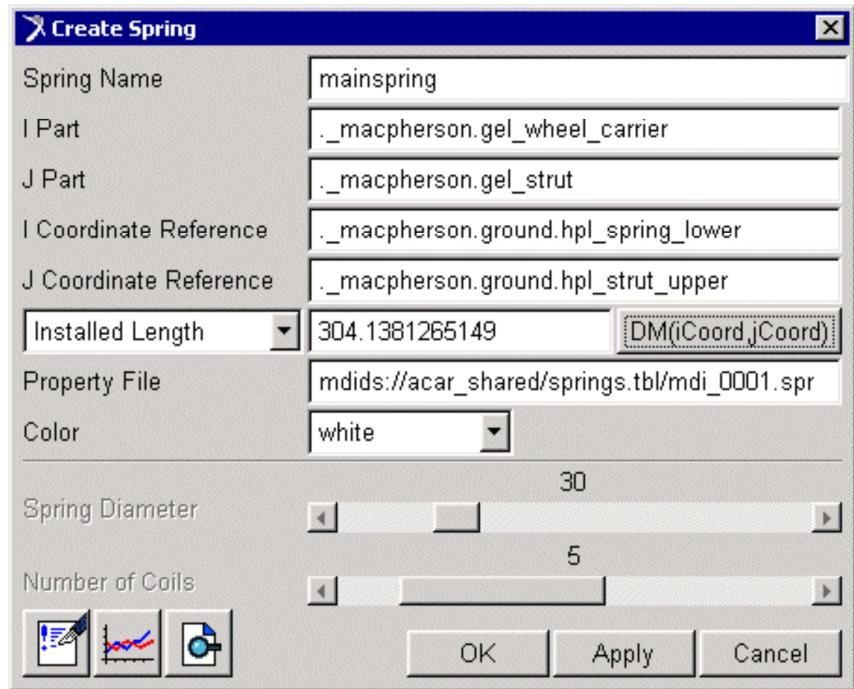


Figure 7.9

To complete the Installed Length blank, we must press the bottom next to it

DM(iCoord, jCoord) and then Fill Field. (Figure 7.10)

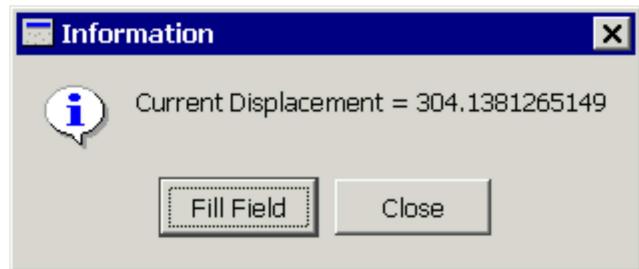


Figure 7.10

Now the template includes the damper and the spring.

Creating and Defining Attachments and Parameters

Now we have created all the parts, springs and dampers, we must create the attachments and parameters:

➤ Defining the Translational Joint

We first create a translational joint between the wheel carrier and the strut specifying that this joint is active regardless of the mode in which we simulate the model. As we are simulating a kinematic analysis, it uses constraints,(translational and revolute joints), to define attachments between the parts.

To define a translational joint: Build, Attachments, Joint and New. Then we create the joint as follows:

- ❖ Joint Name: strut_joint
- ❖ I Part: ._McPherson.gel_wheel_carrier
- ❖ J Part: ._McPherson.gel_strut
- ❖ Joint Type: translational
- ❖ Coordinate Reference: ._McPherson.ground.hpl_strut_upper
- ❖ Orientation Dependency: Orient axis along line
- ❖ Coordinate Reference #1: ._McPherson.ground.hpl_strut_lower
- ❖ Coordinate Reference #2: ._McPherson.ground.hpl_strut_upper

➤ Defining Control Arm Attachments

Before creating bushings and joints for the control arm, we must create the mount parts that act as placeholders for the attachments. A mount part is a massless part that is fixed to ground. The purpose of a mount part is to be replaced by another part (vehicle body, subframe, wheel, and so on).

When we create a mount part, ADAMS/Car automatically creates an input communicator for it of class mount. The input communicator requests the name of the part to which the mount part should connect. When ADAMS/Car finds a matching communicator during assembly, it replaces the mount part with the part that the output communicator indicates.

The replacement part is from another subsystem. If ADAMS/Car finds no matching output communicator, it replaces the mount part with the ground part.

To create a mount part, we specify a hardpoint and mount part name. If the hardpoint has a left or right symmetrical twin, ADAMS/Car creates left and right mount parts and input communicators. Otherwise, it creates a single mount part and a single input communicator.

After creating the mount parts, we create bushings for the control arm specifying the parameters, such as preload and offset, that define bushings.

We also set its inactive status to kinematic, so that during a kinematic analysis ADAMS/Car deactivates this bushing.

Finally, we create the control arm revolute joint. This time, you set its active status to kinematic. This tells ADAMS/Car that during a kinematic analysis it should use the revolute joint, but it should replace the joint with a bushing during a compliant analysis.

➤ Creating mount parts

-Build, Parts, Mount, New. Now, in the Mount Name, we have to enter "subframe_to_body".

In the Coordinate Reference, "._mcpherson.ground.hpl_arm_front", always verify that From Minor Role is set to Inherit, and then press Ok. With this, Adams creates fixed joints between the mount and ground.

➤ Now creating the front bushing for the control arm

Build, Menu, Attachments, Bushing and New, and complete the box. (Figure 7.11)

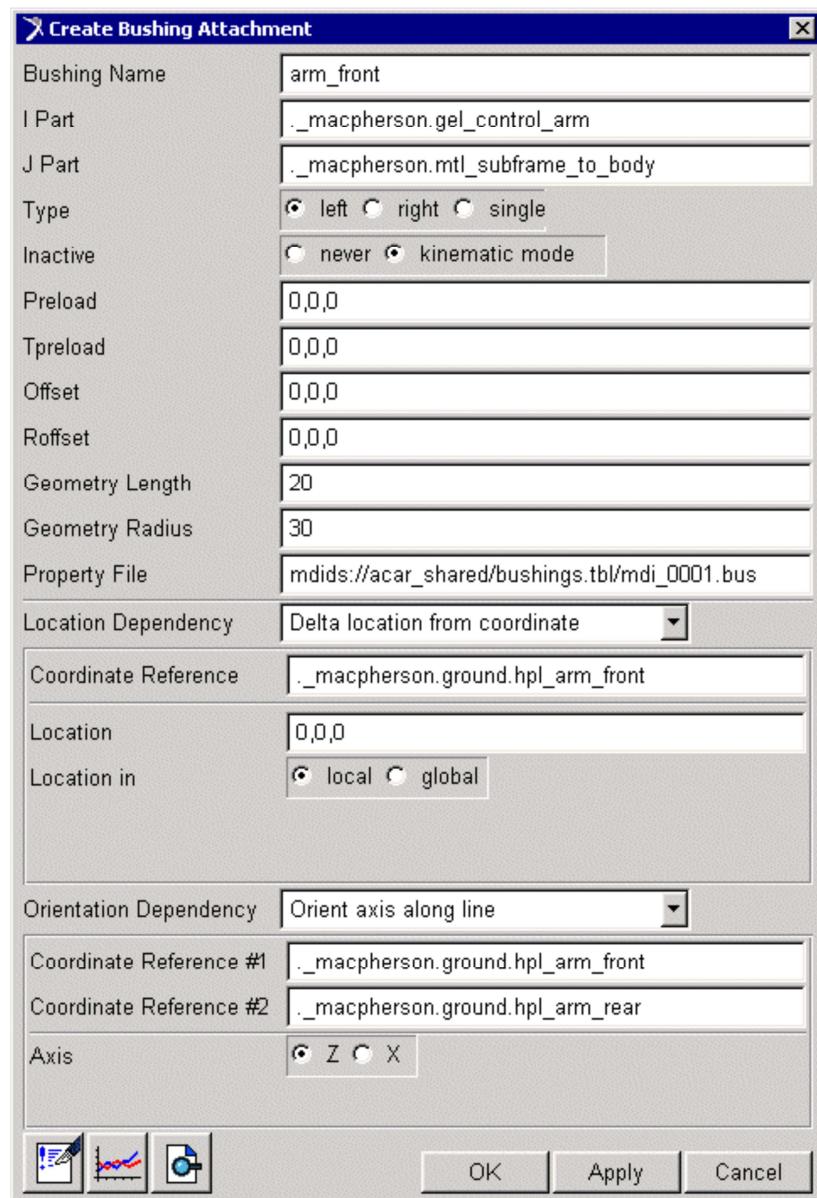


Figure 7.11

➤ Creating the rear bushing for the control arm

With the same box as before, once we have done the front one and pressed Apply, we enter “arm_rear” in the Bushing Name. In the Coordinate Reference blank, “._mcpherson.ground.hpl_arm_rear”, and then Ok.

- Creating the control arm revolute joint
 - ❖ Joint Name: arm_front
 - ❖ I Part: ._McPherson.gel_control_arm
 - ❖ J Part: ._McPherson.mtl_subframe_to_body
 - ❖ Joint Type: revolute
 - ❖ Active: kinematic mode
 - ❖ Coordinate Reference: ._McPherson.ground.hpl_arm_front
 - ❖ Orientation Dependency: Orient axis along line
 - ❖ Coordinate Reference #1: ._McPherson.ground.hpl_arm_front
 - ❖ Coordinate Reference #2: ._McPherson.ground.hpl_arm_rear

And then Apply.

Now we want to create an spherical joint, so with the same box:

- ❖ Joint Name: arm_outer
- ❖ I Part: ._McPherson.gel_wheel_carrier
- ❖ J Part: ._McPherson.gel_control_arm
- ❖ Joint Type: spherical
- ❖ Active: always

- ❖ Coordinate Reference: ._McPherson.ground.hpl_arm_outer

And now, pressing Ok.

➤ Defining the Strut Attachment

Before defining the strut attachment, we must define a mount part for the strut.

Then, we create a bushing for the strut, and next, we create a spherical joint to replace the strut mount bushing during kinematic analyses.

To define a mount part

- ❖ Mount Name: strut_to_body
- ❖ Coordinate Reference: ._McPherson.ground.hpl_strut_upper
- ❖ From Minor Role: inherit

And Ok again.

- To create the bushing for the strut: (Figure 7.12)

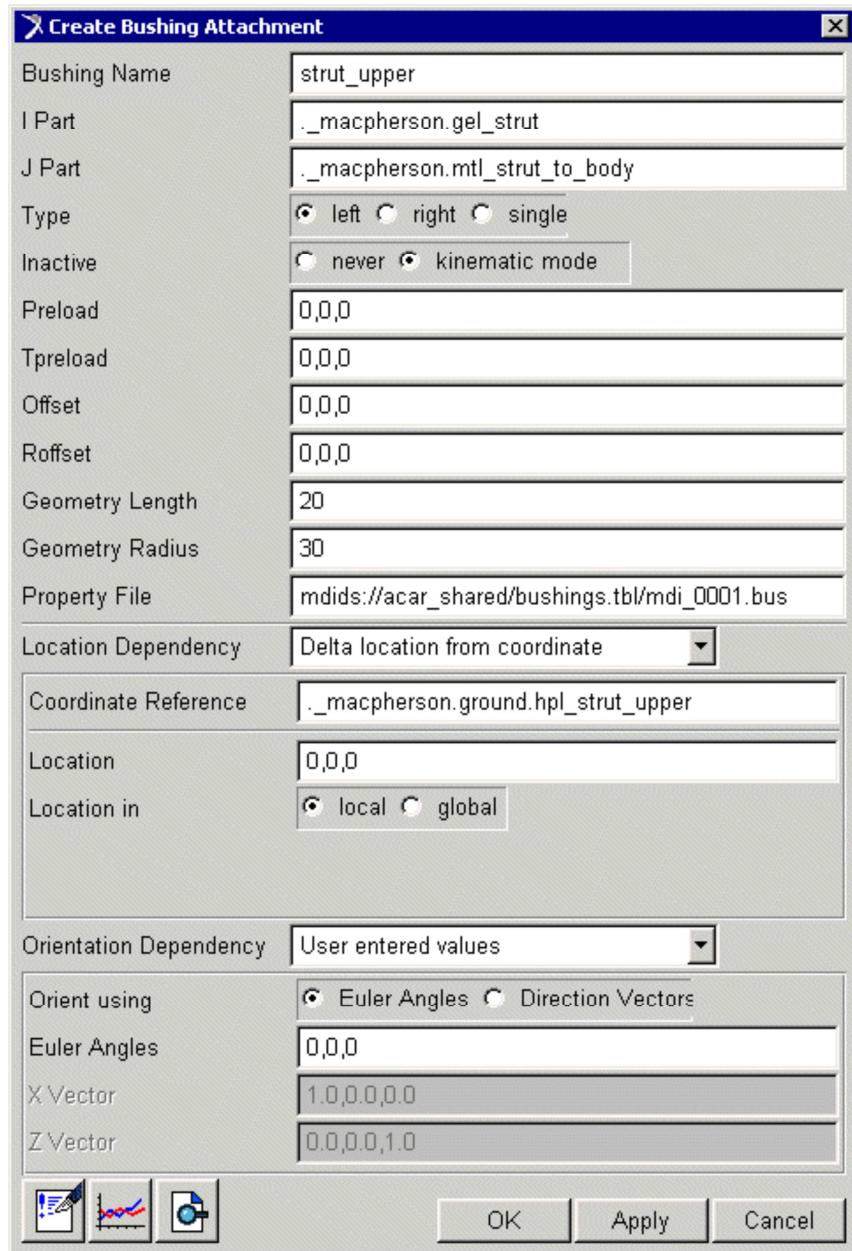


Figure 7.12

- Creating a spherical joint for the strut
 - ❖ Joint Name: strut_upper
 - ❖ I Part: ._McPherson.gel_strut
 - ❖ J Part: ._McPherson.mtl_strut_to_body
 - ❖ Joint Type: spherical
 - ❖ Active: kinematic mode
 - ❖ Coordinate Reference: ._McPherson.ground.hpl_strut_upper

And press Apply.

- Defining Wheel Carrier Attachments

Now, we define a spherical joint between the wheel carrier and the tie road. Then, we define the mount part that connects the suspension to the steering rack during assembly. To conclude, we create a hook joint between the tie road and the steering rack.

To create a spherical joint

- ❖ Joint Name: tierod_outer
- ❖ I Part: ._McPherson.gel_wheel_carrier
- ❖ J Part: ._McPherson.gel_tierod

- ❖ Joint Type: spherical
- ❖ Active: always
- ❖ Coordinate Reference: ._McPherson.ground.hpl_tierod_outer

And press Ok

➤ Creating a mount part for the hook joint

- ❖ Mount Name: tierod_to_steering
- ❖ Coordinate Reference: ._McPherson.ground.hpl_tierod_inner
- ❖ From Minor Role: inherit, and Ok

➤ To create a hook joint

- ❖ Joint Name: tierod_inner
- ❖ I Part: ._McPherson.gel_tierod
- ❖ J Part: ._McPherson.mtl_tierod_to_steering
- ❖ Joint Type: hooke
- ❖ Active: always
- ❖ Coordinate Reference: ._McPherson.ground.hpl_tierod_inner
- ❖ I-Part Axis: ._McPherson.ground.hpl_tierod_outer
- ❖ J-Part Axis: ._McPherson.ground.hpr_tierod_inner and apply.

➤ Defining hub attachment

We can define the hub bearing revolute joint between the wheel carrier and the hub.

- ❖ Joint Name: hub_bearing
- ❖ I Part: ._McPherson.gel_wheel_carrier
- ❖ J Part: ._McPherson.gel_hub
- ❖ Joint Type: revolute
- ❖ Active: always
- ❖ Coordinate Reference: ._McPherson.ground.hpl_wheel_center
- ❖ Orientation Dependency: Delta orientation from coordinate
- ❖ Construction Frame: ._McPherson.ground.cfl_hub_bearing

Then Ok

➤ Defining suspension parameters

We create a steering axis using the geometric method for calculating steer axes.

Adams calculates the steer axis by passing a line through two non-coincident hardpoints located on the steer axis.

Build, Suspension Parameters, Characteristics Array and Set.

We must complete the box: (Figure 7.13)

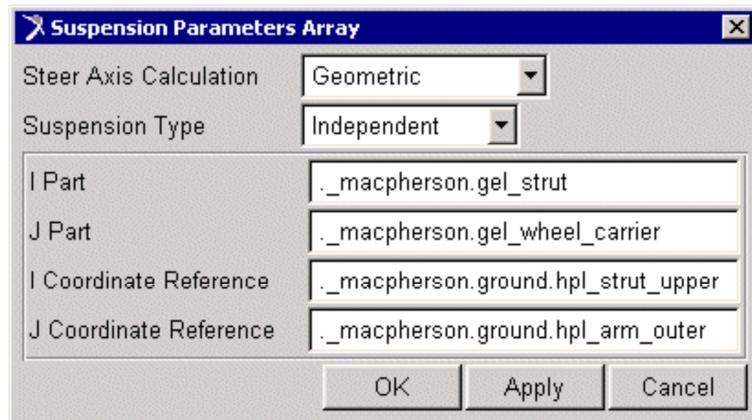


Figure 7.13

And press Ok.

➤ Assembling the Model for Analysis

In order to assemble correctly the model for analysis, Adams uses communicators.

Those are the elements that allow the subsystems and test rigs to exchange information about this:

- Topological data

- Array and parameters variables.

- locations, orientations...

Each subsystem in Adams has input communicators that need information from other subsystems to function properly in the model. Likewise, the corresponding subsystem needs a correctly named output communicator to send the information.

While the assembly is taking place, Adams scans the model for subsystems with minimal roles. It looks for any input communicator that needs information. Then it tries to find out an output communicator, of the same entity type.

➤ Defining communicators

First we must define how the suspension is in order to connect the suspension test rig. To do this, we define the communicators that attach the hub to the suspension test rig at the wheel center:

-Which part it needs to be connected to.

-Where the connection takes place.

-Which is the upright part.

When we run an analysis in static mode, we must lock the hub to the wheel; otherwise, our assembly will have a rotational degree of freedom that prevents the analysis from reaching a solution. Adams creates the actuator between the hub and the wheel carrier.

➤ Displaying information about the communicators

Build, Communicator, Info. Then we must clear the Model Names box. With the mouse right key, press the Model Names box, then Model, Guess and select “._mdi_suspension_testrig”. We choose Array, Location and mount. Finally, Ok.

There should appear a window. (Figure 7.14)

Information

Apply	Parent	Children	Modify	<input type="checkbox"/> Verbose	Clear	Read from File	Save to File	Close
Listing of input communicators in '_MDI_SUSPENSION_TESTRIG'								
Communicator Name:		Entity Class:	From Minor Role:	Matching Name:				
cis_suspension_parameters_ARRAY		array	any	suspension_parameters_s				
ci[lr]_diff_tripot		location	any	tripot_to_differential				
ci[lr]_wheel_center		location	any	wheel_center				
ci[lr]_jack_frame		mount	any	jack_frame				
ci[lr]_suspension_mount		mount	any	suspension_mount				
ci[lr]_suspension_upright		mount	any	suspension_upright				
cis_powertrain_to_body		mount	any	powertrain_to_body				
'_MDI_SUSPENSION_TESTRIG' contains:								
1 input communicator of type 'array'								
4 input communicators of type 'location'								
7 input communicators of type 'mount'								
Listing of output communicators in '_MDI_SUSPENSION_TESTRIG'								
Communicator Name:		Entity Class:	To Minor Role:	Matching Name:				
cos_characteristics_input_ARRAY		array	any	characteristics_input_s				
cos_leaf_adjustment_multiplier		array	any	leaf_adjustment_multipl				
cos_tire_forces_array_left		array	any	tire_forces_array_left				
cos_tire_forces_array_right		array	any	tire_forces_array_right				
co[lr]_tripot_to_differential		mount	any	tripot_to_differential				
'_MDI_SUSPENSION_TESTRIG' contains:								
4 output communicators of type 'array'								
0 output communicators of type 'location'								
2 output communicators of type 'mount'								

Figure 7.14

Those are the mount input communicators. Then we press Close.

- To create the output communicators

Build, Communicator, Output and New. Now, in the Output Communicator Name box, we have to enter “suspension_mount”, and we set Entity to Mount. Now, we check that Minor Role is set to Inherit, and in the Part Name box, “._mcpherson.gel_hub”; and finally, Apply.

Now, in the name box, we enter “suspension_upright”; in the Part Name, “._mcpherson.gel_wheel_carrier”, and Apply.

Again, Name, “wheel_center”; Entity to Location, Minor Role to Inherit, and in the Coordinate reference Name box, “._mcpherson.ground.hpl_wheel_center”, and to finish, Ok.

➤ Testing communicators

This is to verify that the input and output communicators are correctly specified, so we test them. It finds out whether to add or modify communicators to ensure that the template will assembly correctly:

Build, Communicator, Test. We complete the box as shown: (Figure 7.15)



Figure 7.15

And then Ok. The info window appears: (Figure 7.16)

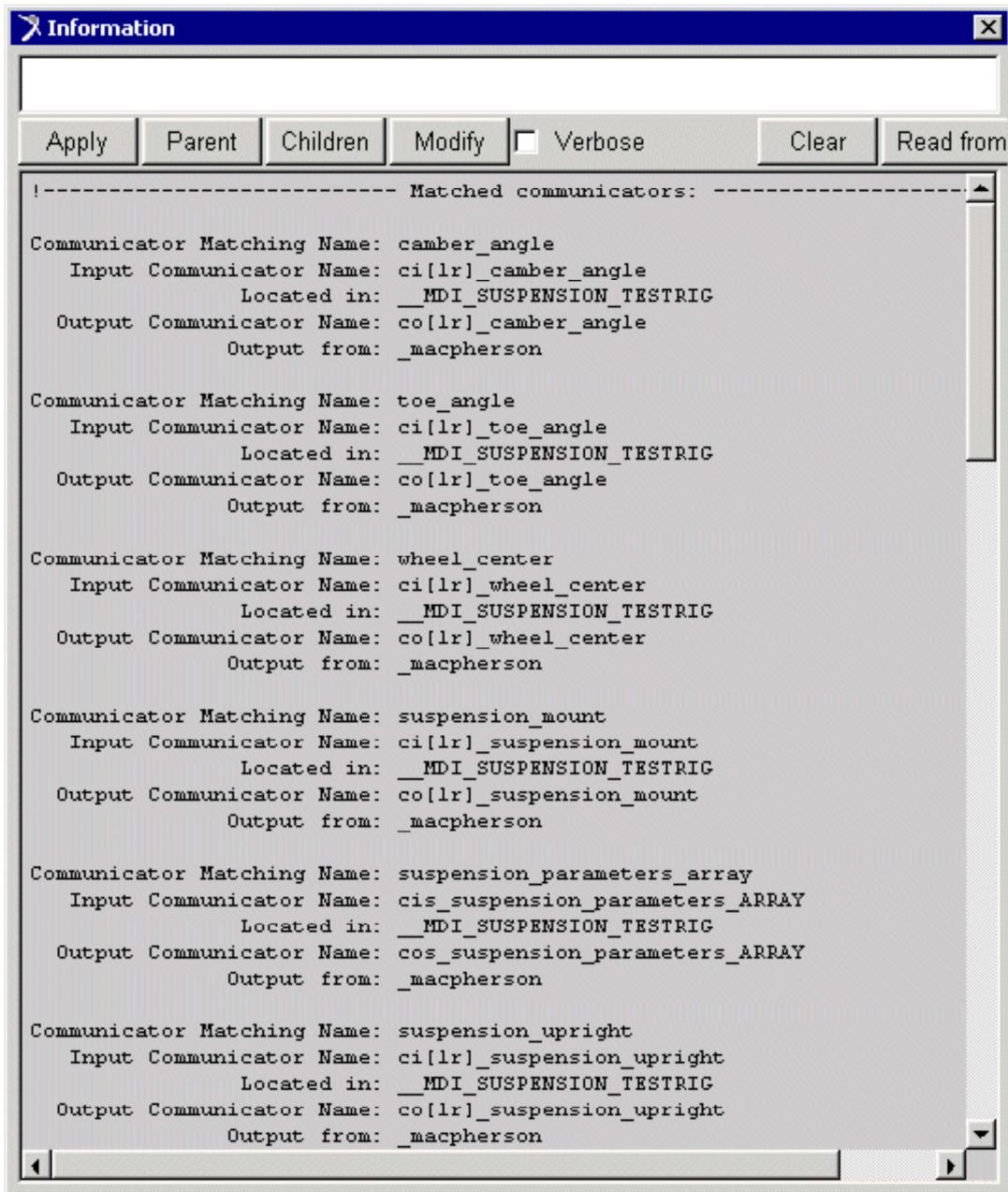


Figure 7.16

This window lists which communicators are matched and which are not. It shows the matched communicators that are correctly sending and receiving information.

The Information window also lists input and output communicators that are not matched correctly. Many communicators are unmatched. Many of these

communicators are related to items, such as the steering or chassis that we do not currently have opened. Finally, we Close.

➤ Concluding

The only step left to finish is saving the template, in order to be able to use it later in the creation of a suspension.

7.2 Creating suspension subsystem

Now this section explains how to create a suspension subsystem based on the template we previously have created.

To create the subsystem

File, New and Subsystem. Now we complete the box as this: (Figure 7.17)

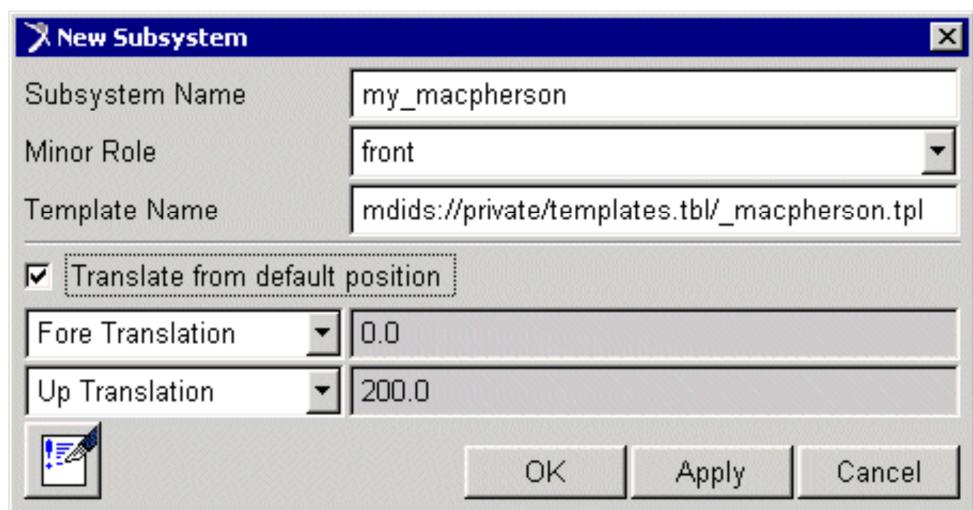


Figure 7.17

7.3 Analyzing an assembly with the template

Now we perform the principal analysis we were aimed to: kinematic analysis.

When performing a kinematic analysis we use the joints instead of the bushings, those we have defined when building the template.

First, we have to create a suspension assembly; after that, define a preload.

- To create a suspension assembly

File, New, Suspension Assembly.

In the Assembly Name text box, we have to enter susp_assy_1.

We have to check that the name of the subsystem is the same of ours, "my_mcpherson". And then Ok.

- To define a preload

We must click with the right bottom of the mouse, selecting Modify. Then, set Installed Length to 140, and Ok.

Now we must change the kinematic mode. From the Adjust menu, Kinematic Toggle. Then Current Mode must be set to Kinematic, and finally Ok.

- Performing a kinematic suspension analysis

Simulate, point to suspension Analysis and the Parallel Wheel Travel. This is to set that both wheel are going to move upside down in a parallel movement. Now we have to define the parameters of this movement: (Figure 7.18)

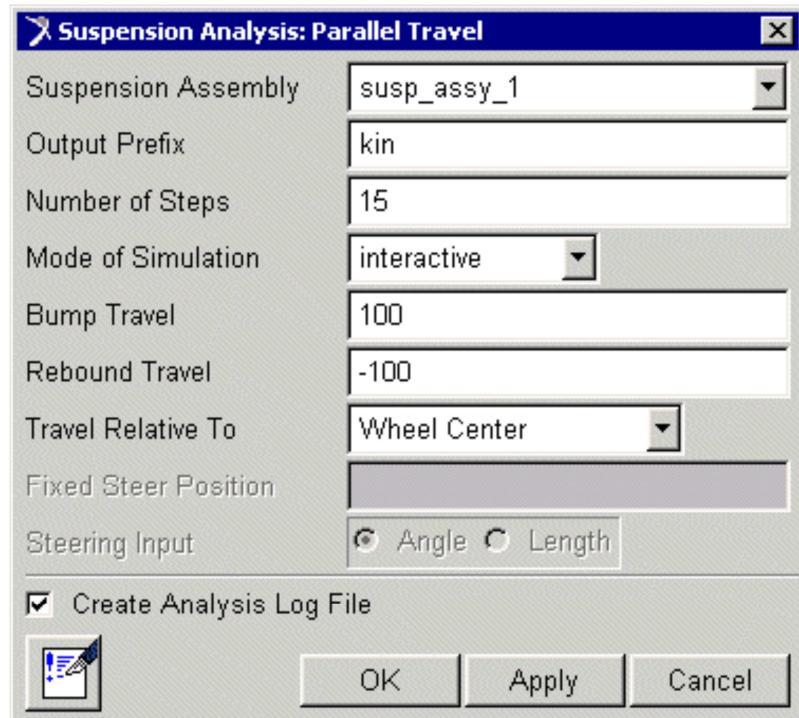


Figure 7.18

And finally, close.

Now, there is the possibility to see that movement and check that all parts of the suspension are moving properly. For that, we go to the Review menu, and click on Animation Controls, and then Play. The suspension animates through full jounce and rebound.

The last part of the analysis is to plot the results of the movement. Pressing F8 key, we enter in the Post Processor. Then from Plot menu, we go to Create Plots.

Then we must set up the plots as this:

❖ Plot Configuration File:

mdids://acar_shared/plot_configs.tbl mdi_suspension_parallel_travel.plt

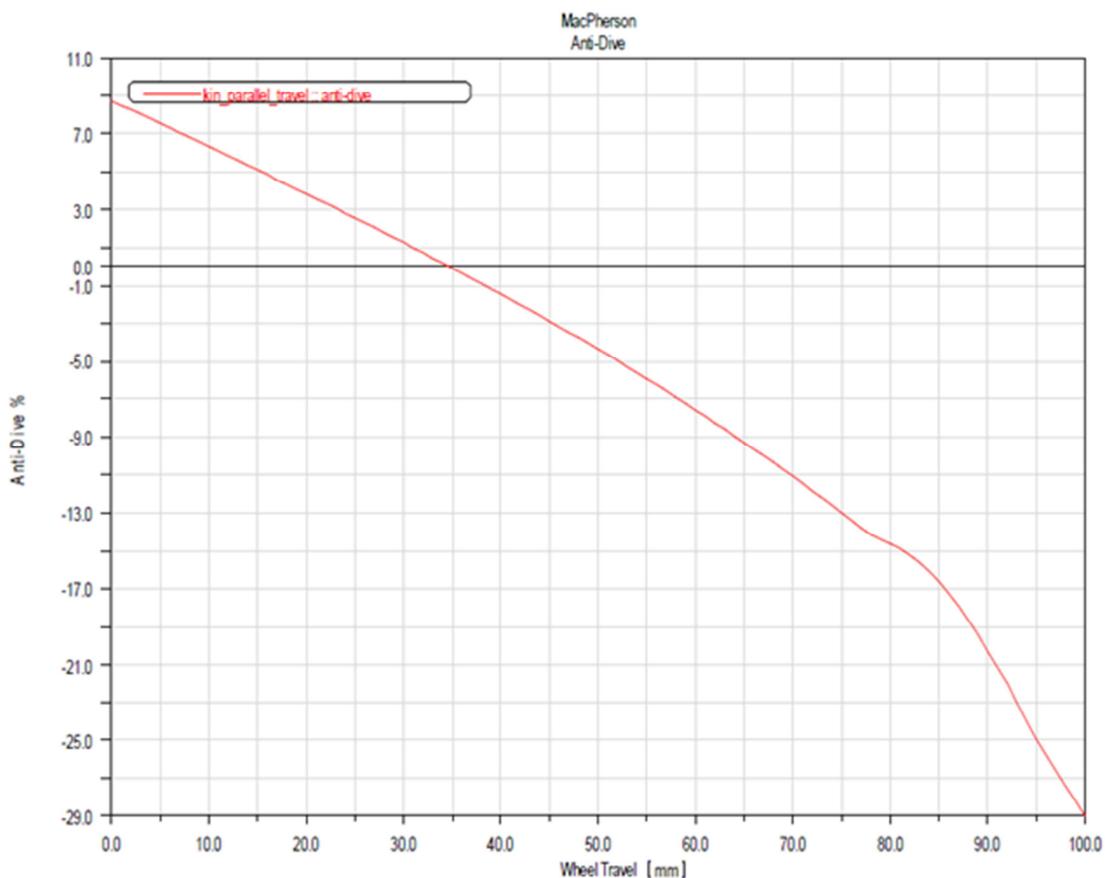
❖ Plot Title: McPherson

And we have to check that Cross Plotting is not selected; to finish, press Ok

7.4 Analyzing the results

Now, to conclude, the last part of this Project is to analyze and comment the results that ADAMS gives us regarding all the parameters and studies that have been taking place throughout it. For so, we must present the plots the program creates. There are several of them, and in order not to enlarge this Project, we are going to comment the ones I consider more important. All the plots shown in this part have been given by the program.

ANTI DIVE:



Anti-dive is a suspension parameter that affects the amount of suspension deflection when the brakes are applied (Figure 7.19).

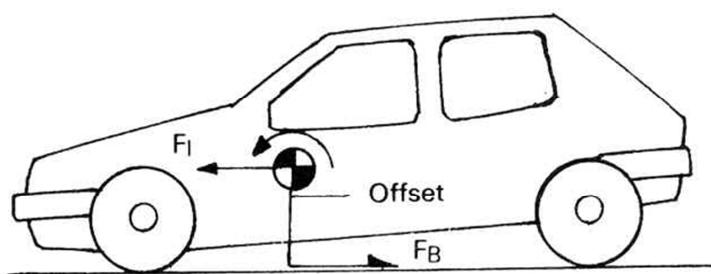


Figure 7.19

When a car is decelerating due to braking there is a load transfer from the rear to the front wheels proportional to the center of gravity height, the deceleration rate and inversely proportional to the wheelbase. If there is no anti-dive present, the vehicle suspension will deflect purely as a function of the wheel rate. This means only the spring rate is controlling this motion. As anti-dive is added, a portion of the load transfer is resisted by the suspension arms. The spring and the suspension arms are sharing the load in some proportion. If a point is reached called 100-percent anti-dive, all of the load transfer is resisted by the suspension arms and none is carried through the springs. When this happens there is no suspension deflection due to braking and no visible brake dive. There is still load transfer onto the wheels, but the chassis does not pitch nose down. (Figure 7.20)

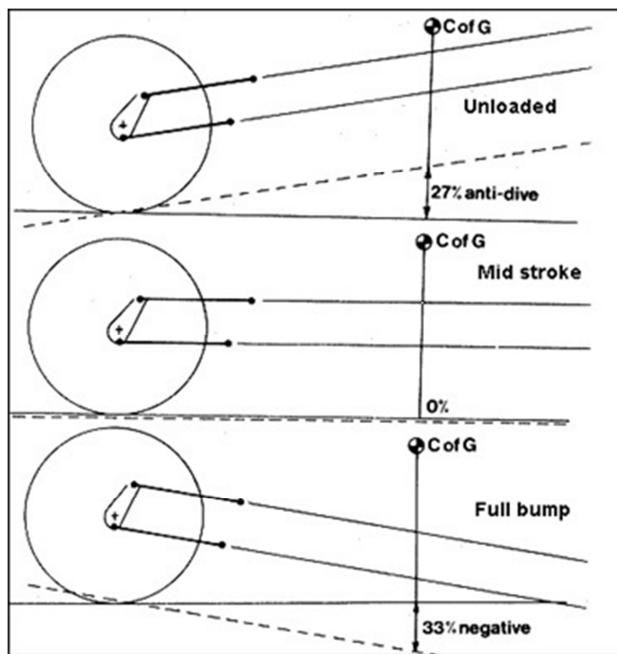


Figure 7.20

The method to achieve anti-dive is controlled by the upper and lower control-arm pivot points on the chassis. The exact determination is quite involved and is easily

established with a good three-dimensional geometry computer program. In all suspensions there is a factor called the "side view swing arm." This is a theoretical point of intersection of the arm planes projected into the fore-aft vertical plane through the wheel center. If this point is behind the wheel and above the ground, it will produce anti-dive. It will also provide anti-dive if it is below ground and ahead of the wheel. The other possible locations for this instant center are ahead and above ground as well as behind and below the ground. With these locations we have pro-dive. This means as the brakes are applied the suspension travels more than it would without any anti- features. The magnitude of the anti-feature, either pro or anti, is a function of how far the instant center is away from the ground. If it is on the ground, there is zero-percent anti-feature. (Figure 7.21)

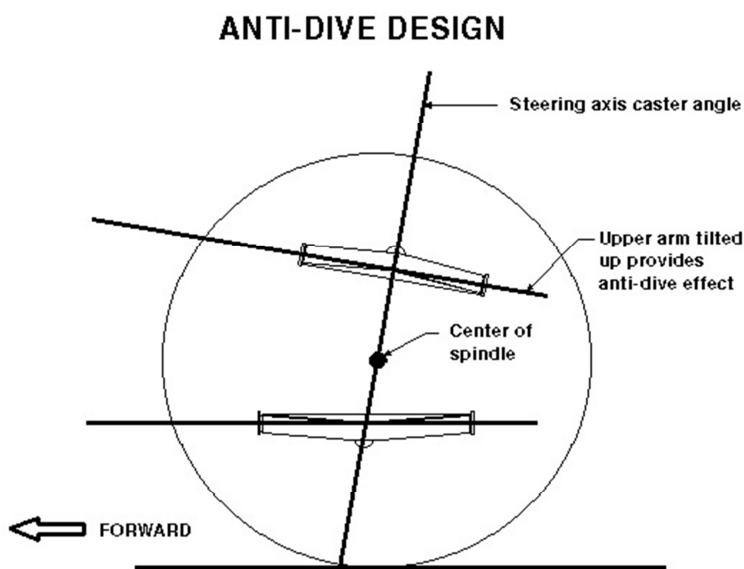


Figure 7.21

On passenger cars, anti-dive is added to make the pitch motions under braking more tolerable for the occupants because of the typical soft spring rates. The anti-dive

is made the same for both sides of the car because most people brake in a straight line, and there is no directional preference on the street or in road racing for that matter.

In the evolution of the circle-track chassis, they have come to a solution that provides pro-dive on the left-front and anti-dive on the right-front suspension. The reasons for this are varied. For one thing, as the brakes are applied the car will deflect more on the left front than on the right. This will give a feeling of the front leading into the corner and not rolling out right away. This combination also provides for more of the load transfer to the front to be carried by the right-front wheel than the left front. The total transfer is the same; just the distribution is affected by the anti-feature in the geometry. More load transfer to the right front on corner entry is like adding wedge with the brakes on and is a stabilizing effect.

Anti-dive has an effect when there is a torque trying to rotate the spindle relative to the chassis. The typical case for this is under braking, but there is another case that can alter the loads and deflections of the wheels. This is the drag component of cornering load. When the wheels are turned and the tires are cornering hard there is a longitudinal load relative to the chassis centerline trying to rotate the spindle.

Just like anti-dive in the front suspension, there can be anti-lift in the rear suspension that reduces rebound travel under braking. There is also anti-squat in the rear suspension under acceleration for rear-wheel-drive cars. (Figure 7.21)

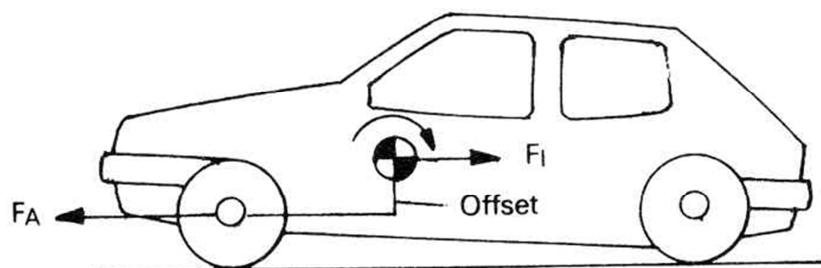
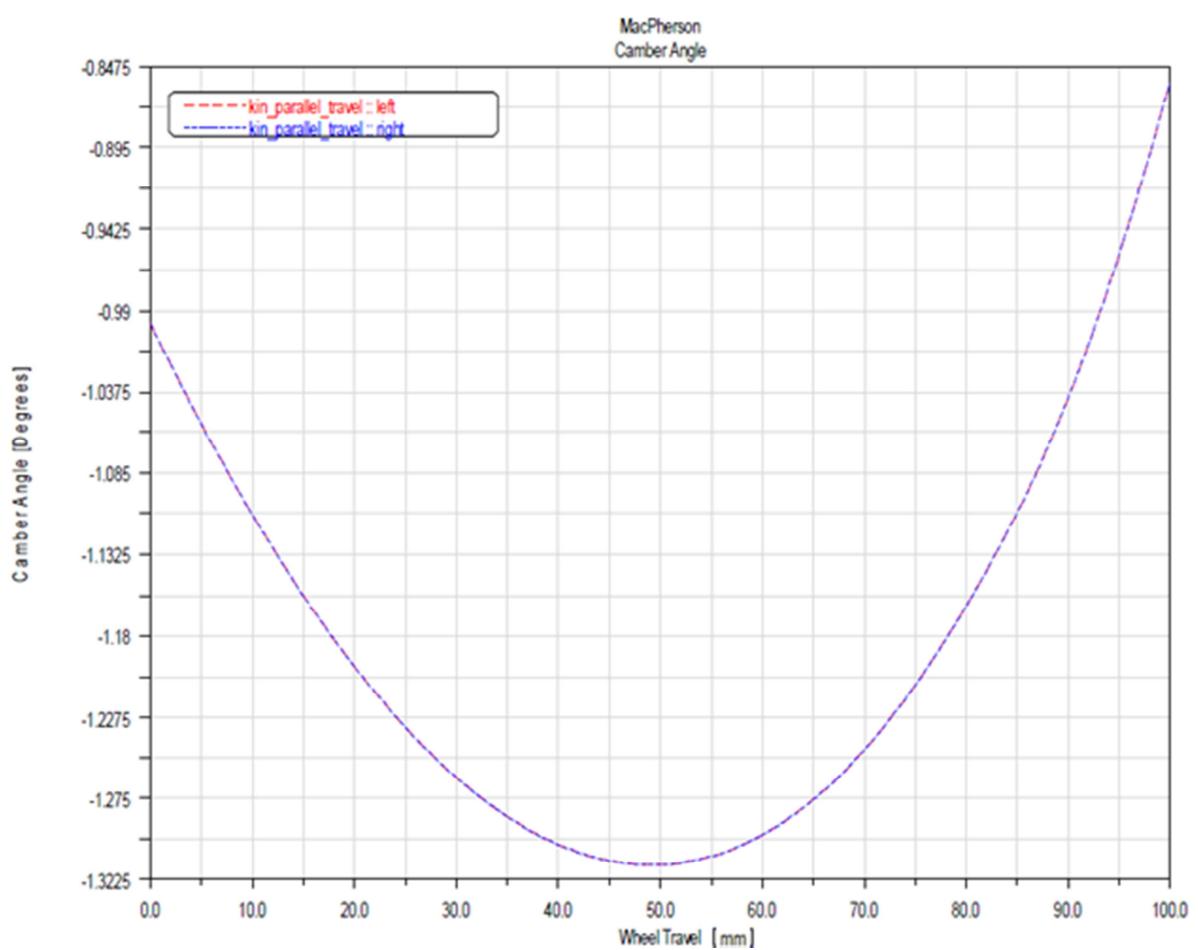


Figure 7.21

CAMBER ANGLE:



The principle of this angle is to distribute the cars weight evenly across the width of the tyre thus maximizing grip, but this angle demands the most complicated amount of maths of all the angles. (Figure 7.22)

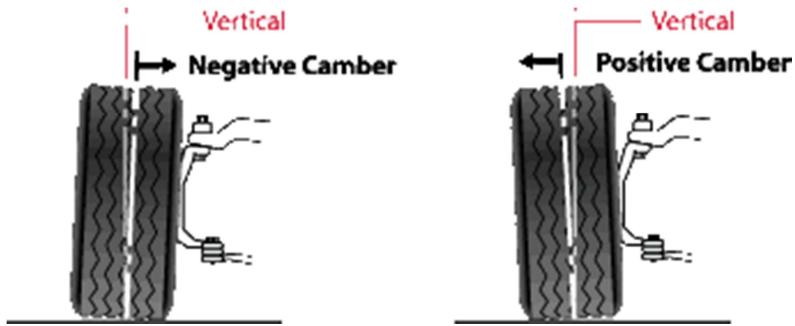


Figure 7.22

As the pictures suggest, if the camber is considered negative then it will lean in toward the cars centre line relative to the vertical, and the opposite will be found for positive, the main problem with camber is it has an active range which is expressed by the suspension, example being if the front camber setting is 30 min negative in a relaxed condition then it will progress to 60 min when hitting a bump (loaded) and drop to zero min off the bump (unloaded) this is fine but all those bumps and pot holes over the years may have moved, bent or compressed the cars camber beyond the relaxed 30 min, now loaded or unloaded the angle is dynamically out of range.

Camber is the most violent angle of them all as far as tyre wear is concerned, and has the ability to wear down to the wire in as little as 1000 miles, although not realizing the poor handling will need to be ignored

The way to recognize camber wear is by the damage, it is smooth and confined to a very small area normally about 10% of the total tyre width, often the remaining tread has little or no wear, the vast majority of camber problems lean toward the

negative and violently wears the tyre in an unseen area deep inside the heel arch, so it would take a conscious effort to realize a problem, usually it will be the tyre centre that informs you, sometimes the police, and the possibility of the centre reservation has been realized by many an unsuspecting motorist.

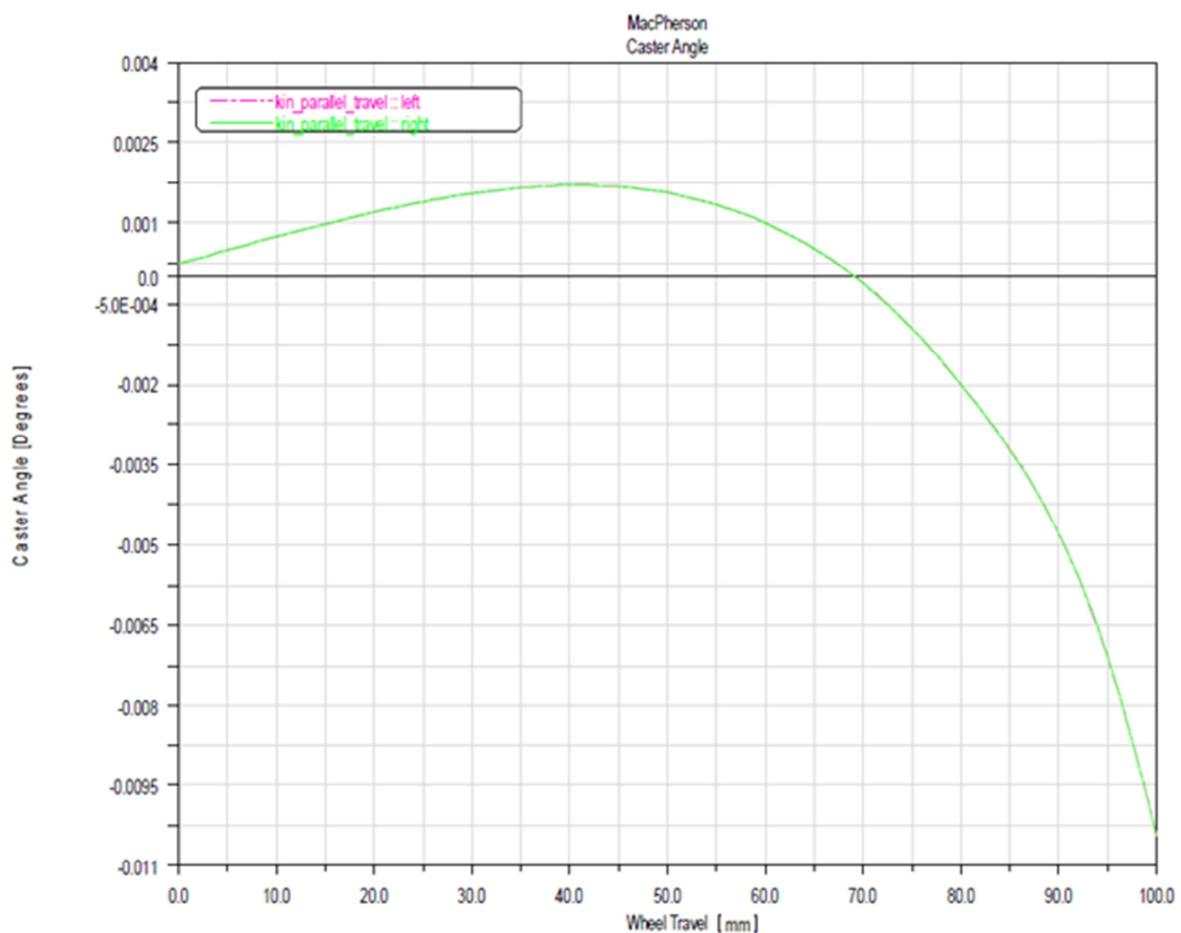
Another common reason for camber wear is changing from standard tyres and wheels, to a larger diameter rim and a smaller profile tyre, the geometry is based on the dynamic range of the suspension, and the on/off camber tolerances are calculated for a known tyre width, the wider tyre will move the weight distribution nearer to the (on,) or (loaded) state, so it can be seen that the active camber will be out of tolerance, and is now considered normal.

The intention to change to bigger, wider wheels and tyres must be approached with caution, the wheels camber must be re-set nearer to the off or (unloaded) setting, accommodating for the extra tyre width.

When any wheel alignment or geometry is performed there is made available conversions for the bigger than standard wheel, since it is the reference point that the machinery will gauge upon, there are two reasons for this, first, standard or linear toe readings will be amplified if there is a bigger wheel fitted so the reading and setting would be false, second the bigger wider tyre will address a different area in the geometry range so known corrections are made available for the operator to compensate.

The range of values is between 30 min and 1 degree, so in general terms, the suspension is correct. The outer values are result of having performed an extreme analysis, with a big wheel travel.

CASTER ANGLE:



It is the forward (negative) or rearward (positive) tilt of the steering axis as viewed from the side. Caster is usually measured in degrees, and only applies to the front wheels because they are the only ones that steer. (Figure 7.23)

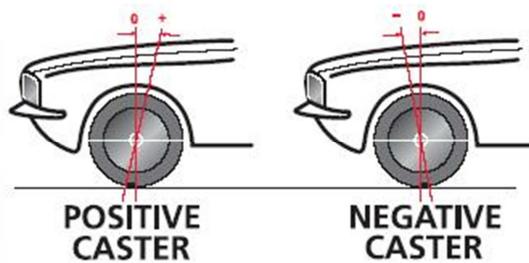


Figure 7.23

Caster is a weird angle because it does not affect tire wear directly. Its greatest effect is on steering stability, steering effort and steering return. So it is often the most ignored angle.

Most vehicles have a small amount of positive caster to provide quick steering return and high speed stability. This happens because caster forces the spindle to angle down slightly as the wheels turn. This lifts the chassis and brings more weight to bear on the wheels as they turns. The next effect is that caster helps keep the wheels aimed straight ahead for improved steering stability, and helps the wheels return to the straight ahead position after turning. Many high range autos have a lot of caster for this very reason because it provides a more stable feel at highway speeds. The downside is that it increases steering effort and steering feedback to the driver.

If there is too much difference in caster side-to-side, it can cause a vehicle to drift or lead to one side. Some alignment specifications call for a slight difference in caster to compensate for road crown. But as a rule, caster should usually be within half a degree in both sides.

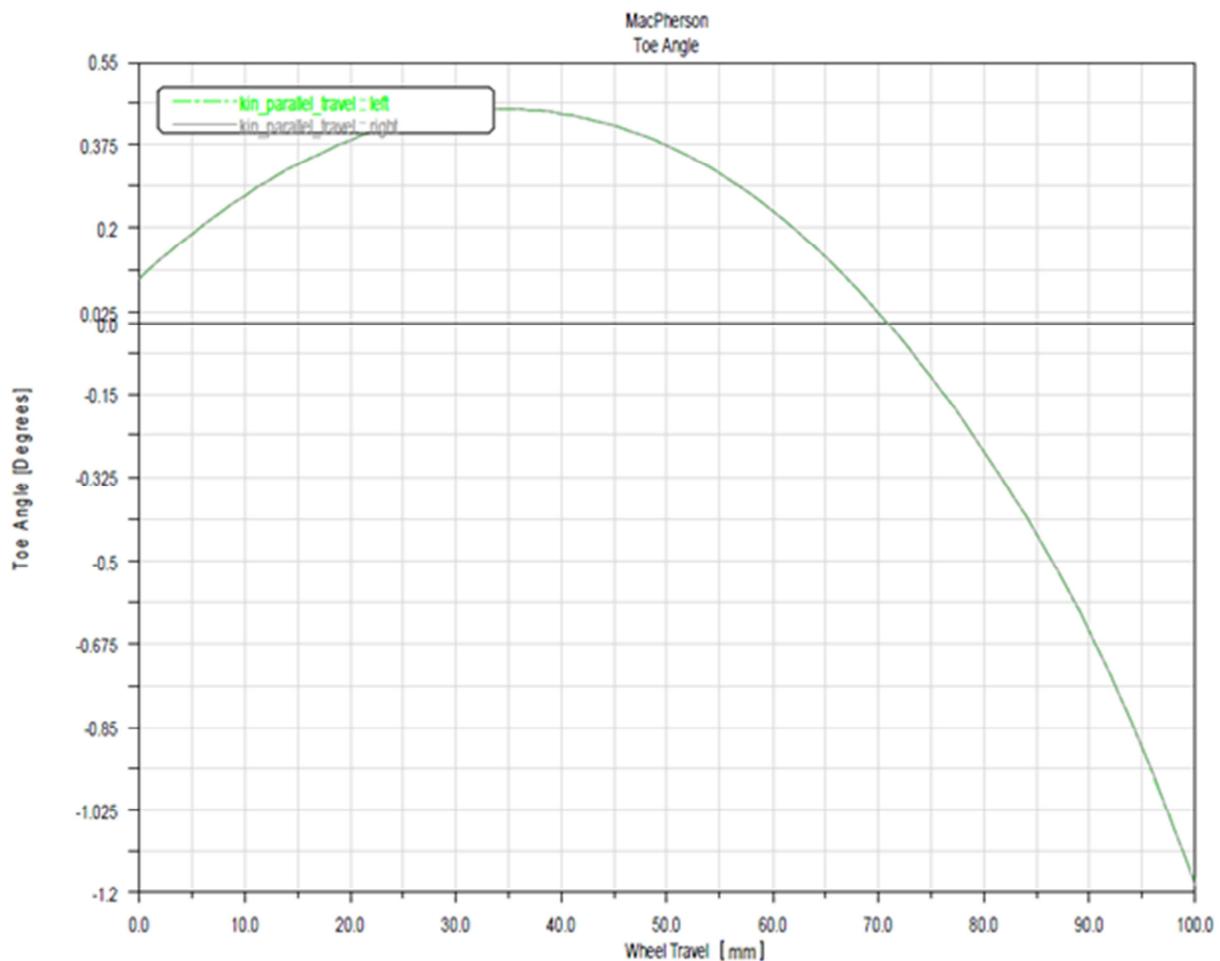
The same kind of problems that can cause camber misalignment can cause caster misalignment: a bent spindle, mislocated strut tower, bent strut, worn or collapsed control arm bushing, bent control arm or a weak or broken spring. So if any of these parts are replaced, caster should be checked and readjusted as necessary after the parts have been installed.

Ride height can also affect caster. Spring sag or overloading a vehicle can alter ride height up to several inches, which can change caster readings by up to a degree or more. This may contribute to steering instability or change steering effort. So checking

ride height is also an important element of aligning the wheels. If ride height is below specifications, weak springs should be replaced. Upgrade opportunities here include installing variable rate springs, air springs, overload shocks or air-assist shocks on a vehicle that is used for towing or hauling heavier than normal loads.

The values must be between 0 and 4 degrees, so we can observe the plot and confirm that the suspension is correct.

TOE ANGLE:



Toe-in and toe-out is the angle that the wheels deviate from the driving direction. (Figure 7.24)

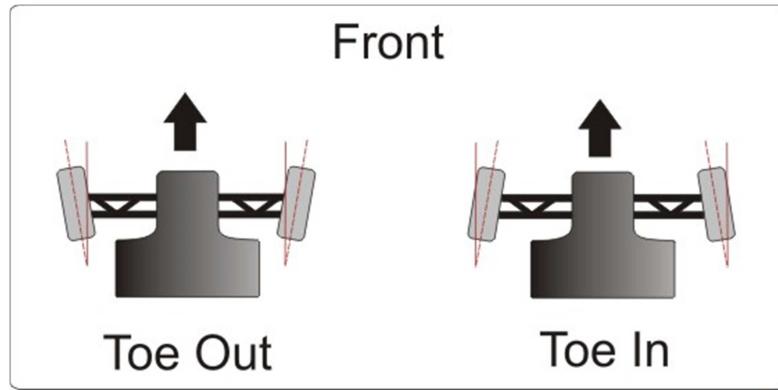
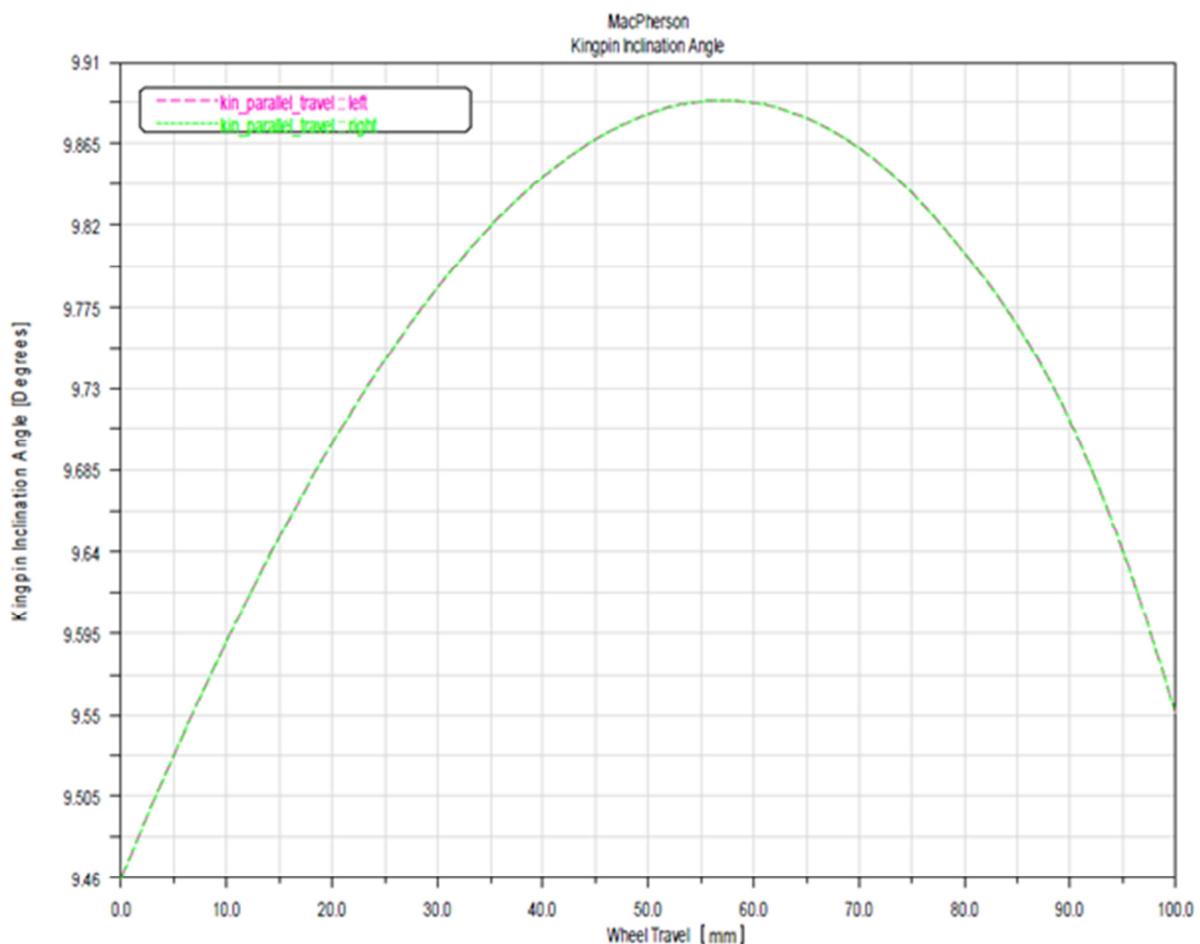


Figure 7.24

It is used because the driving force and road resistance tends to squeeze the rubbers in the suspension. By giving the car a toe-in or toe-out setting the wheels are in a straight line when driving. The rear wheels seldom have toe-in or toe-out. The front wheels of a rear wheel driven car mostly have toe-in. Toe-out is used in front wheel driven cars because the driving force tends to turn the wheel inwards. The kingpin inclination also plays a (minor) role in how much toe-in or toe-out is needed. The less kingpin inclination the more steering momentum is generated pulling the wheel inwards thus more toe-out is also needed to compensate.

The regular values usually are between 11 and 30 min, both positive and negative. In this case, we can see the values of the plot according with this interval, so the suspension is correct.

KINGPIN INCLINATION ANGLE:



To offer enough feel to the steering and to get a self returning action the suspension can be set up to achieve this by carefully choosing the caster angle and kingpin inclination angle. (Figure 7.25)

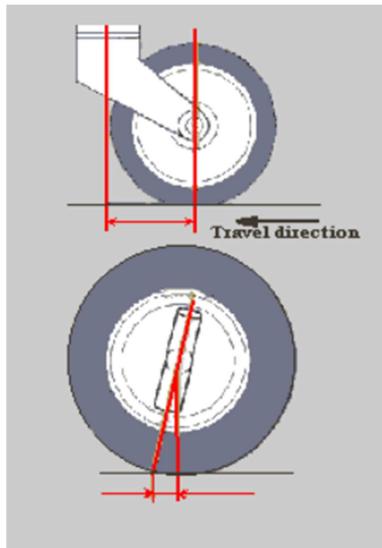


Figure 7.25

You can get a very strong self centring effect if the tyre footprint trails the steering axes. As shown here in the wheel of a handcart and the same principle as used in our car. The angle between the steering axis and a line from wheel center to the ground is called the caster angle. The bigger the angle is, the stronger the self centring action is. If the angle is negative the steering is very light and very nervous.

Another way to get a selfcentring action is by creating a steering offset. (3. In the figure 7.26). This can be created if the projected KPI angle does not align with the footprint center of the tyre. The wheelforces will try to pull the center of contact patch of the front wheels forward, thus the wheel will rotate about the point of the kingpin axle projected to the ground. The steer momentum is the product of distance 3 and the wheelforce. Increasing the inclination angle will decrease the self centring steering effect. Also notice that the steering effort here is changing with the amount of power supplied to the wheels. So for front wheel drives reducing steering effort by altering

the KPI is not a very elegant solution. It could be possible to increase the KPI so that distance 3. to zero, but with increasing KPI also the lateral forces on the cars increase thus making it more receptive to roll and instability. With McPherson an increase of KPI also means that the turrets protrude more deeply into the engine bay.

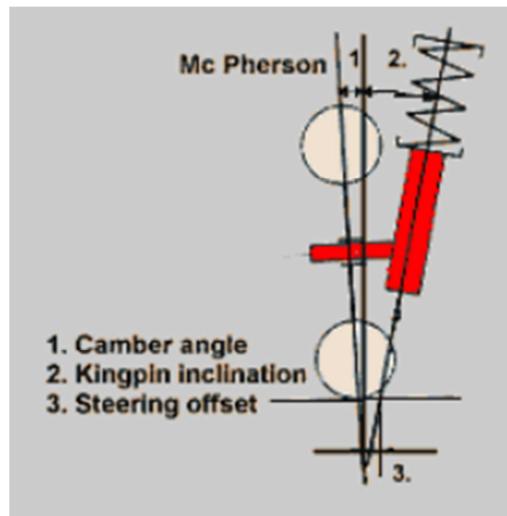


Figure 7.26

The values for this angle usually are between 8 and 10 degrees, so according to this, the suspension is correct.

8. CONCLUSIONS

What has been done in this Project is a kinematic analysis of a McPherson suspension. For that, the type of test chosen has been the parallel wheel travel, which consists of the movement at the same time of both wheels due to the compression of the buffers up and down, using for all that the support of the computer software ADAMS / Car. For this, the length of compression has been set to 100 mm.

While the study, I have focused my work on explaining all the types of suspensions, in order to have a general idea of what a suspension is and how can each one be distinguished and differentiate, with some introduction in all its components. The next part refers to building up the suspension, with the intermediate steps of creating the template needed with the subsequent analysis.

Finally, the last part is all about the results given by the program, deepening in those I consider more important, such us the anti-dive factor, the camber and caster angle, toe angle and kingpin inclination angle.

All the data used to create the suspension and perform the analysis, such us measures, sizes, weights, factors of the springs and bumpers have been taken from regular and standard values, in order to obtain more accurated and precise results so can be extrapolated to a real situation.

9. CONTENTS

1. INTRODUCTION	page 1
2. THE ROLE OF A VEHICLE SUSPENSION	page 3
3. FACTORS AFFECTING DESIGN	page 4
4. DEFINITIONS AND TERMINOLOGY	page 4
5. MOBILITY OF SUSPENSION MECHANISMS	page 8
6. SUSPENSION TYPES	page 12
7. KINEMATIC ANALYSIS	page 22
7.1 Creating a template	page 22
7.2 Creating suspension subsystem	page 54
7.3 Analyzing an assembly with the template	page 55
7.4 Analyzing the results	page 57
8. CONCLUSIONS	page 72