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Compliant Multi-Link Vehicle Suspensions

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COMPLIANT MULTI-LINK VEHICLE SUSPENSIONS

A Thesis
Presented to
the Graduate School of
Clemson University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science
Mechanical Engineering

by
Souharda Raghavendra
August 2008

Accepted by:
Dr. Beshah Ayalew, Committee Chair
Dr. John C. Ziegert
Dr. Mohammed Daqaq

ABSTRACT

One of the goals of automotive lightweight engineering is to achieve reduction in mass, cost, and complexity of vehicle components, subsystems and systems without sacrificing functionality and expected performance. This thesis addresses functionally integrated suspension systems that could lead to reduction in part count and mass and save packaging space. It deals with the analysis of multi-link suspensions that combine the function of energy storage and the mechanism of wheel location and guidance within individual compliant links and members.

To explore possibilities, a generic kinematic model of an independent five-link suspension was built in the MSC.ADAMS multi-body dynamics simulation environment. Models of the compliant energy storage and kinematic guidance members were created using a finite element analysis package and interfaced with the MSC.ADAMS environment. Then, the main spring, and individual and multiple rigid links of the reference suspension were replaced with compliant members, and subsequently, the resulting kinematic characteristics of the compliant multi-link suspension were compared against those of the reference rigid multi-link suspension. Under certain achievable assumptions and a suitable choice of the dimensions of the compliant links, it was found that similar kinematic characteristics as the reference suspension could be achieved by variants of the compliant multi-link suspension consisting of compliant links.

The analysis was also applied to the development of a compliant suspension concept for an existing high performance vehicle. Model validation data were obtained from actual tests conducted on a kinematic and compliance test rig. Evaluation of

possible compliant variants of the rear suspension for this vehicle led to the replacement of the upper control arm of the original suspension with a ternary-link compliant member. The kinematic and compliance characteristics of this modified suspension were thoroughly analyzed through simulations and some of the characteristics were validated with tests conducted using a test-fixture employing many parts of the actual suspension and an aftermarket composite member for the compliant ternary-link.

The compliant suspension concepts evaluated in both phases use fewer parts, and therefore exhibit reduced mass and complexity. Further research and development is required to comprehensively optimize the design of the compliant links for certain desired response attributes, such as better toe control.

DEDICATION

This thesis is dedicated to my parents, Mr. Raghavendra Ramaiah Muniyappa, Mrs. Dakshayini Raghavendra and my elder brother, Mr. Santhosh Raghavendra without whose unconditional love support this thesis would not be complete.

ACKNOWLEDGMENTS

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CHAPTER 1

INTRODUCTION

Vehicle suspensions can be collectively described as an arrangement of kinematic linkages and force elements (springs, dampers and bushings) connecting the chassis and the wheels to ensure constant contact of the wheels with the road. The functions of kinematic links, springs and dampers is, respectively, to provide wheel location, energy storage and energy dissipation and to isolate the vibrations excited by road irregularities and thereby enhance driving pleasure. In doing so, suspensions have a direct effect on the ride and handling comfort of the vehicle.

The motive behind this thesis is to research the possibility of effectively integrating energy storage and wheel guidance capabilities of the suspension system using compliant links that replace the original rigid kinematic links within the suspension, with little or no compromise in system performance. This integration has advantages such as reduction in weight, part count, and cost, and thereby offers enhanced reliability compared to traditional suspension systems. The latter occupy as much as 12 % of the overall mass of the vehicle (1).

The thesis draws ideas from the use of compliant mechanisms. Compliant mechanisms are defined as mechanisms that provide motion transfer while at the same time act as means for storing energy within the mechanism. They are capable of integrating energy storage function with that of wheel location. Compliant mechanisms are nowadays being used in many other application areas such as medical field, consumer

goods and commercial applications. Compliance is typically incorporated in suspension designs in the form of bushings or springs (coil and leaf). The very basic and the elementary type of compliant suspension which was used and still being put to use in some vehicles is the leaf spring. Figure 1 shows the traditional leaf springs used for storing energy and wheel guidance in early Mercedes vehicles.

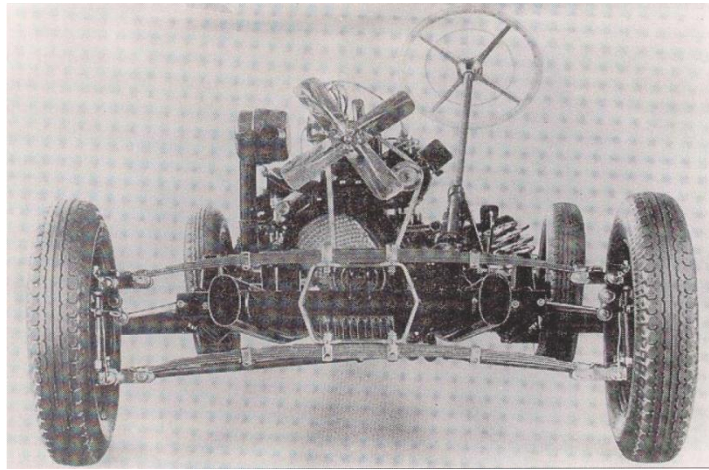


Figure 1 Traditional leaf spring used on Mercedes 170V in 1935 (2)

The wheel guiding mechanism in a leaf spring is obtained from the flexing of the leaf spring in bounce and rebound motion. Although leaf springs had certain advantages, there were disadvantages which hindered the performance of a vehicle like fatigue failure and reduced control over the vertical motion of the wheel. These led to the design of many other complicated kinematic models which enhanced the performance characteristic of the suspension. Since all these designs included solely rigid kinematic links, the need to combine both the performance characteristics of the kinematic suspension designs and the compliance to the system, arise which would still deliver the

same or better performance characteristics with advantages like reduced weight, reduced cost and reduced number of parts in the suspension assembly.

Since the rigid multi-link independent suspension is used as a benchmark for various current suspension designs, an attempt is made to make the multi-link suspension compliant. This includes research and design of new compliant suspension concepts that are focused on improving ride quality, comfort, and handling and at the same time; minimize weight and manufacturing cost of the system.

1.1 Thesis Objective

The objectives of this thesis are given below.

- Address the concept of integrating the functionalities of energy storage and wheel guidance mechanism within the suspension.
- Develop compliant suspension concepts which achieve similar or better performance characteristics compared to existing suspensions.
- Achieve reduction in mass, cost, and complexity, with reduced packaging space as an additional advantage.
- Carry out feasibility analysis on compliant suspension concepts designed to verify the use of compliant members in suspensions.

1.2 Thesis Outline

The organization of each chapter and their contents are briefly explained in this section.

Chapter – 2 begins with the fundamentals of suspension systems and a brief introduction on the types of suspensions. Fundamentals of compliant mechanisms are first explained with reference to non-automotive applications. Different forms of existing compliance in suspensions and other compliant suspension concepts developed are also explained. Background on important performance characteristics for rear suspensions is presented.

Chapter – 3 discusses a generic multi-link suspension model chosen as the reference for the design of compliant multi-link suspension. Detailed insight on the development of the compliant multi-link suspension concept along with constraints and simulation experiments conducted are explained. Comparison of performance characteristics results of the compliant multi-link suspension against the reference multi-link suspension is presented.

Chapter – 4 focuses on an existing suspension as a reference for the development of compliant link suspension. Potential compliant suspension design and the compliant element used for the development of compliant suspension for the reference vehicle are explained.

Chapter – 5 discusses the summary of the results obtained from the compliant link suspension concepts for both the generic multi-link and existing reference suspensions. This chapter also outlines items that need to be addressed in future work to further improve certain performance characteristics of compliant link suspensions.

CHAPTER 2

BACKGROUND AND LITERATURE REVIEW

2.1 Suspension Systems

Suspension systems form one key subsystem of an automobile that are used to isolate the occupants from shocks and vibrations induced due to road surface irregularities. It is also used as a wheel locating and guiding mechanism when the vehicle is in motion. Some of the basic functions of a suspension system are (3)

1. Provide vertical compliance by isolating the chassis from irregularities of the road.
2. Control wheel/axle in proper location and orientation in relation to the road surface.
3. React to longitudinal forces (acceleration and braking), lateral forces (cornering) and vertical forces.
4. Support vehicle weight and provide minimal load variations during transient maneuvers.
5. Resist roll of the chassis.

Some of the basic elements that constitute the suspension system are

1. Springs – Stores energy and provides flexibility by compression and expansion as the wheel traverses along the irregular road surface.
2. Dampers – Dissipates energy and dampens out the vibrations induced in the suspension system.

3. Kinematics links – Helps in guiding and locating the wheel or axle.
4. Bushings – Replaces kinematic joints by inducing compliance in the system by allowing extra degrees of freedom during bounce and rebound motion. They also isolate the chassis from shocks and vibrations during suspension movement.

2.1.1 Front Suspensions

Front suspensions are classified as dependent and independent suspensions. The most common dependent front suspension is the beam axle, which is used less and less in recent vehicles because of numerous disadvantages like large unsprung mass, packaging space, and considerable caster change. However some off-road application vehicles tend to still use the beam axle dependent front suspension as they offer high articulation and high ground clearance.

The most common types of front independent suspensions are the double wishbone suspension and the Macpherson strut. The double wishbone suspension also known as the double A-arm suspension has parallel lower and upper lateral control arms. The main advantage of the double wishbone is that the camber can be adjusted easily by varying the length of the lateral upper control arm such that it has a negative camber in jounce. The MacPherson strut type suspension consists of a single lower wishbone arm which controls the lateral and longitudinal location of the wheel. The coil spring and the shock absorber are combined into a single unit extending vertically making it more compact. A few disadvantages of this type of suspension are that it requires sufficient vertical space and a strong top mount.

2.1.2 Rear Suspensions

Similar to the front suspensions, rear suspensions too are of dependent and independent suspension types. Some of the commonly used dependent rear suspensions are the twist beam, leaf springs, live and dead axles. The main advantage of a twist beam is that it is inexpensive, compact and is suitable for small cars where package space is limited. Live rear axles use longitudinal leaf spring to attach the axle to the vehicle chassis. Live rear axles are not used in small cars due to their high unsprung mass and are used mainly only on pickup trucks and SUV's.

Some of the independent type rear suspensions are the swing axles, semi trailing arms, wishbones, multi-link suspensions. Wishbones suspensions are similar to front wishbone suspensions. Multi-link suspension is the most commonly used type of rear independent type suspension. Multi-link suspension has 3 or more lateral arms arranged in space. They have the greatest flexibility in modifying any suspension parameter to suit the required vehicle application.

2.2 General Compliant Mechanisms

Compliant mechanisms provide motion transfer or transformation while at the same time providing means for energy storage within the mechanism. Compliant members derive their mobility from flexibility or deflection of members rather than movable kinematic joints only. There are several advantages of using a compliant mechanism for inducing flexibility in the system. Some of the advantages given by (4) are:

1. Reduced maintenance
2. Weight reduction
3. Reduced wear
4. Increased precision and reliability
5. Cost reduction
6. Reduction in number of parts
7. Simplified manufacturing process
8. Reduced noise and vibration due to the absence of friction

A very trivial example of a compliant member is the bow and arrow system or the diving board in a swimming pool shown in figure 2. When the bow is bent, energy is stored in the form of strain energy. This strain energy can be used later to transform into the form of kinetic energy to the arrow. Similar is the case with diving board used in a swimming pool. Thus unlike trivial energy storage mechanisms used with any mechanism, compliant members stores and release energy within the mechanism.

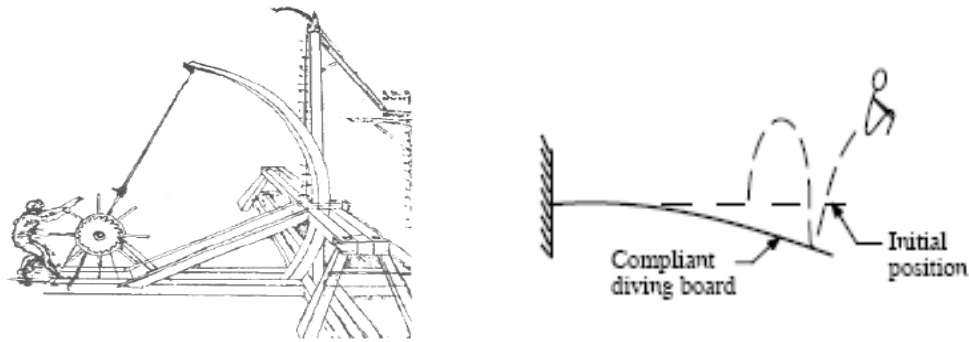


Figure 2 Examples of general compliant mechanisms (4)

Compliant mechanisms are found in many areas where flexibility is present. A brief outline of compliant mechanisms used in different fields is included below.

Compliant mechanisms are used for prosthetics in the medical field. One such evident example is the artificial foot for the physically challenged persons (5). Figure 3 shown below depicts a prosthetic feet. This system stores and releases energy constantly while walking. It is light yet extremely strong and can handle a load of 365 lb.

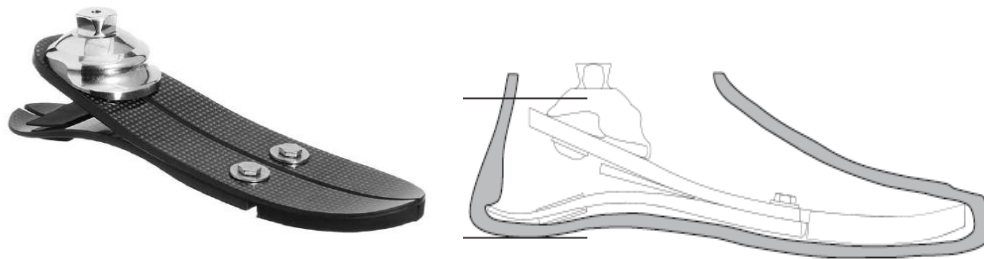


Figure 3 Prosthetic leg as a compliant mechanism (5)

Consumer goods also make significant use of compliant mechanisms. Some of the basic compliant mechanisms used are the ice cream scoop, lens cap of a camera and the paper clip (4). Moving on to commercial mechanisms, a bicycle brake is one example for compliant mechanism (6). The initial rigid version of the bicycle brakes had a redundant 4-bar mechanism with coil return spring. The use of compliant mechanism in the brakes

has eliminated the use of coil return spring and 2 fewer pin joints. Figure 4 shows some of the compliant mechanisms used as consumer goods and commercial mechanisms.

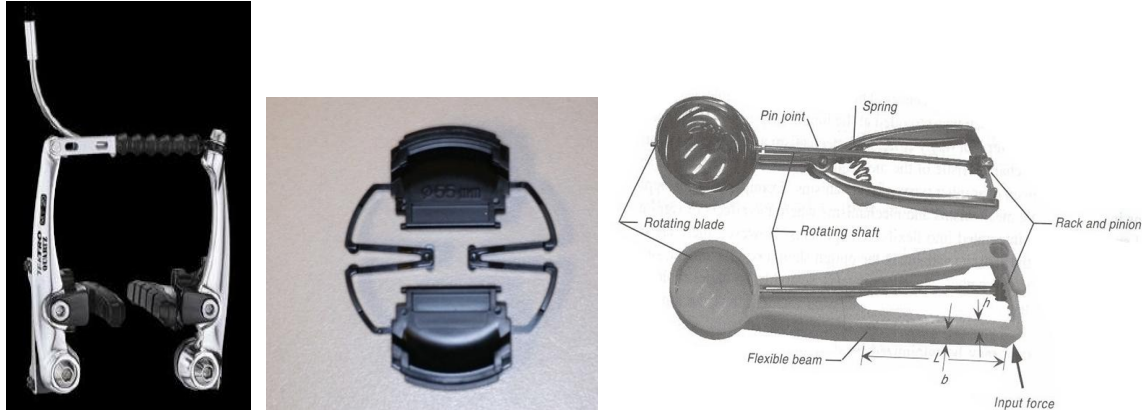


Figure 4 Commercially available compliant mechanisms (4) (6)

The field of Micro Electrical Mechanical Systems (MEMS), also has found the use of compliant mechanisms. Compliant mechanisms are used to produce linear motion (7) with MEMS. The folded beam linear motion mechanism shown in Figure 5 is one the most commonly used compliant mechanism in MEMS. It is shown in the initial and deflected positions. This mechanism is a planar mechanism with only one degree of freedom, translating in the vertical direction without translating or rotating in other directions. Additional support in other directions can be obtained by placing an additional mechanism oriented orthogonally.

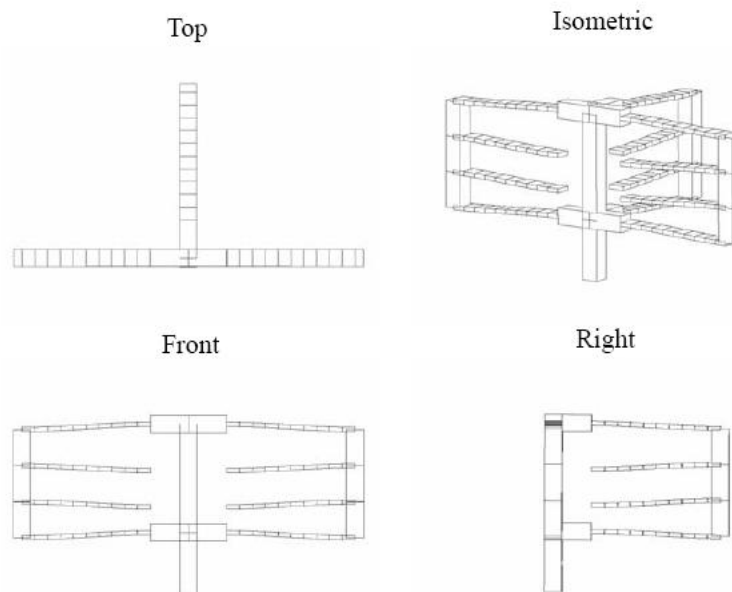


Figure 5 Compliant mechanism used in MEMS (2)

2.3 Suspension Concepts With Compliant Members

This section discusses in detail the compliant suspension concepts adapted to existing suspensions.

2.3.1 Leaf Spring

One of the earliest forms of compliance used in suspensions is the leaf spring. They are also called semi-elliptic leaf springs as they represent one half of an ellipse. They can also be referred to as the simplest type of suspension where the wheel location and energy storage are taken care by the leaf springs. Leaf springs can either be placed longitudinally or laterally depending on the type of vehicle. Longitudinal leaf springs usually have progressively shorter leaves stacked with shackles at the rear end of the arc.

The centre of the leaf spring provides wheel guidance, while the shackle allows for the elongation of the leaf springs during bounce and rebound motions. The advantages of leaf springs are that they are simple, cheap and offer reduction in weight. Longitudinal leaf springs were mostly used for live or dead axles as shown in Figure 6.



Figure 6 Longitudinal leaf spring used in Jeep (8)

Figure 7 shows different combinations of both lateral and longitudinal leaf spring (2). Since no kinematic links are used, leaf springs have limited control over the handling of the vehicle. Longitudinal forces cause the wheel to windup due to the effect of moment caused at the wheel centre.

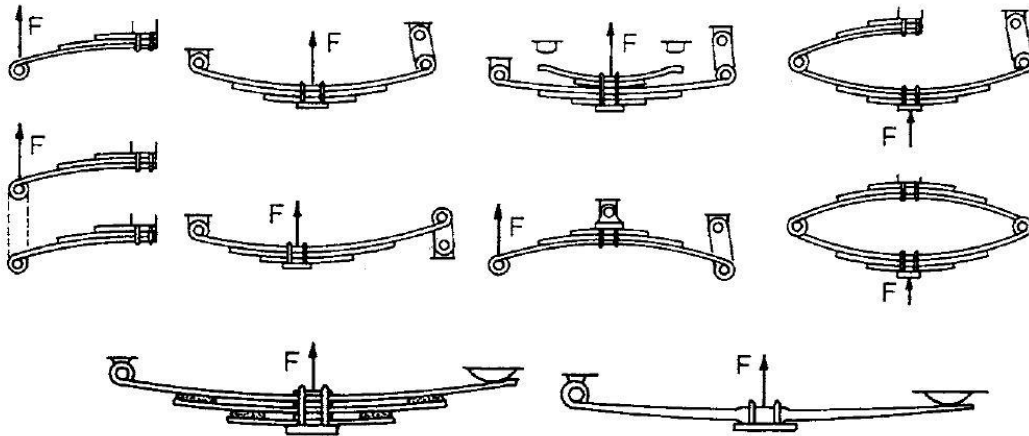


Figure 7 Different configurations of longitudinal and transverse leaf springs (2)

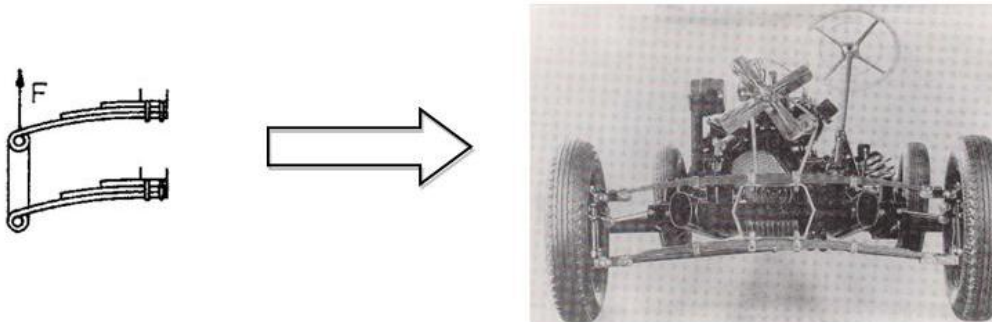


Figure 8 Example of known compliant suspension similar to the configuration shown -1

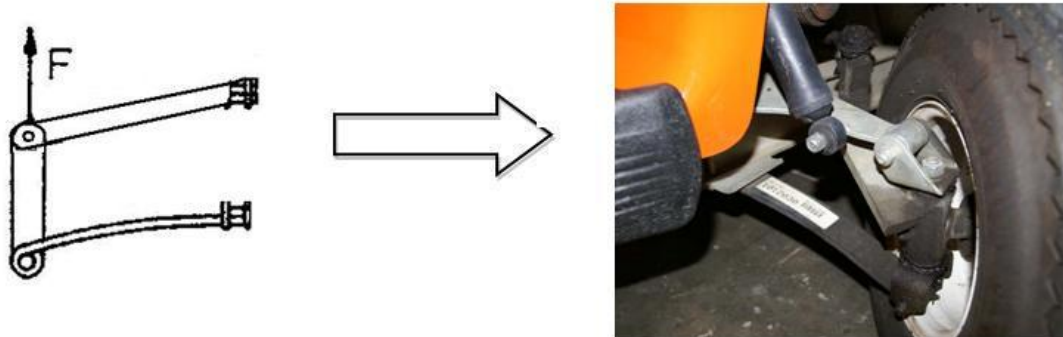


Figure 9 Example of existing compliant suspension with leaf springs - 2

Figure 10 shows examples of transverse leaf springs used in GM Corvette and Ford Escort production vehicle respectively (9) (10).

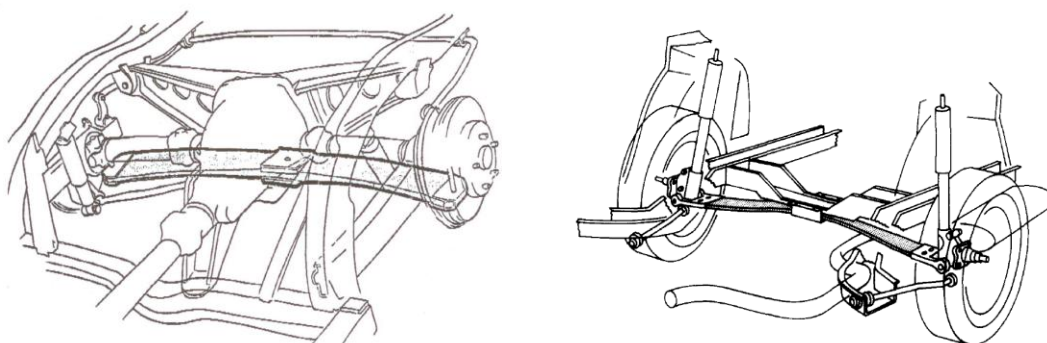


Figure 10 Examples of transverse leaf springs used in Corvette (left) and Escort (right) production vehicle

Figure 11 shows examples of longitudinal leaf springs used in a hybrid truck vehicles (9) and a Bugatti type 57SC (8).

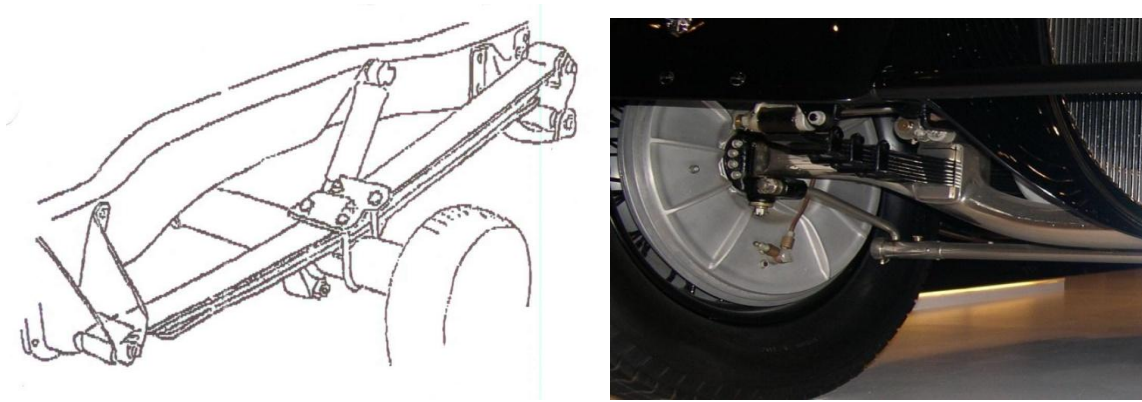


Figure 11 Examples of longitudinal leaf spring used in hybrid trucks and Bugatti

2.3.2 Double A-arm Suspension

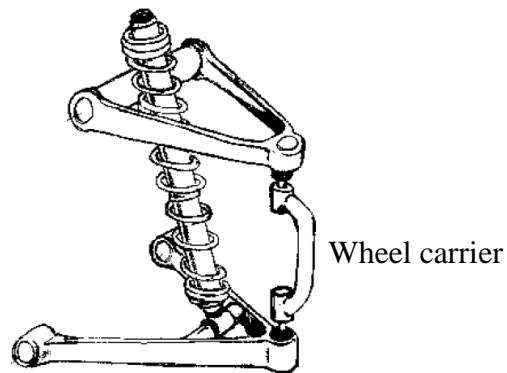


Figure 12 Double A-arm suspension (7)

Figure 12 shows schematic representation of a typical double A-arm suspension. The double A-arm or the wishbone suspension can be simplified as a 4 bar planar mechanism.

Flexural pivots have also been used to modify the front upper A-arm of Ferrari F1 suspensions. These flexural pivots act as a joint with some torsional stiffness replacing mechanical joints in race cars (11).

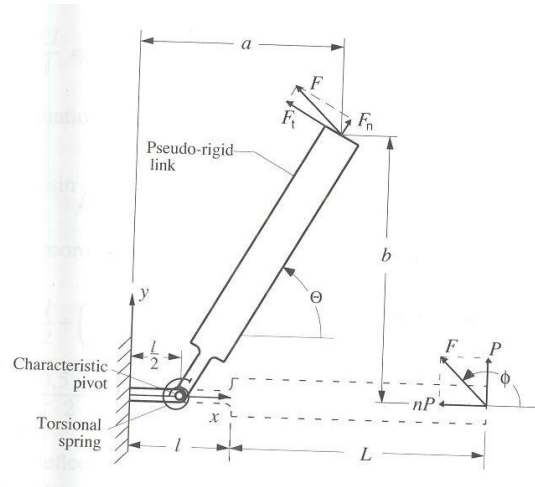
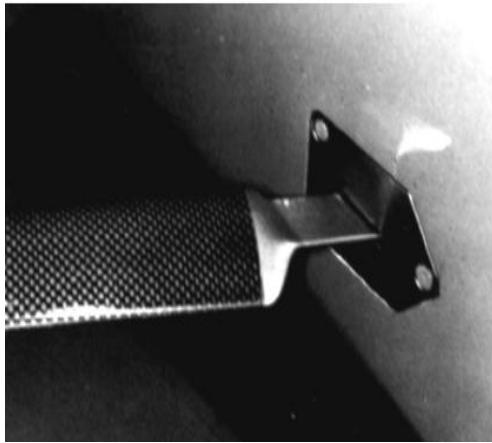


Figure 13 Double A-arm suspension using flexural pivots in F1 race cars (11) (4)

2.3.3 McPherson Strut

McPherson strut suspension basically consists of a lower control arm housing the spring and the damper unit. The best possible way to introduce compliance into the system is by replacing the lower control arm with a transverse leaf spring. Figure 14 shows a schematic representation of a typical McPherson strut and its compliant counterpart with the lower control arm and the spring replaced by a transverse leaf spring. (2) (10).

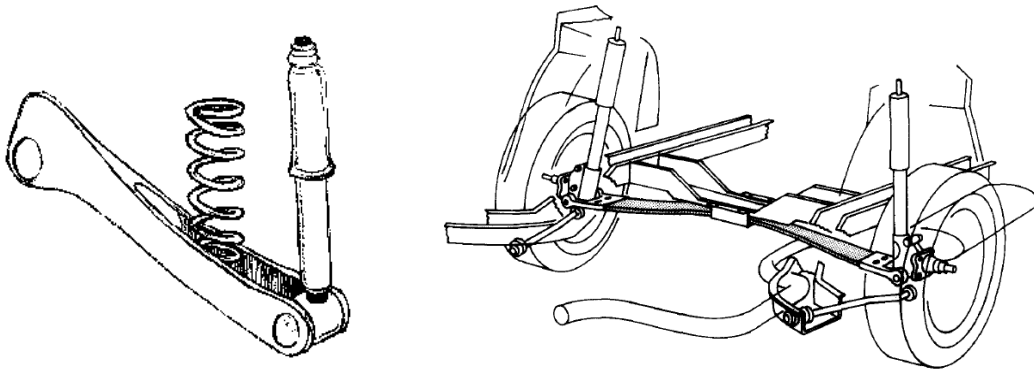


Figure 14 MacPherson strut and its compliant counterpart (2) (10)

2.3.4 Other Compliant Suspensions

U-link, TORCS and FLECS are some of the other compliant suspensions developed by Magneti Marelli. U-link is a modification to an existing suspension where the lower control arm was a H-shaped link. Figure 15 shows the original configuration and the U-link suspension concept developed. (12)

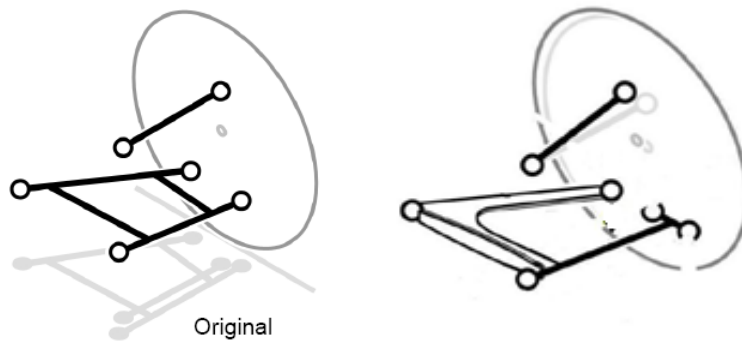


Figure 15 U-link original and modified configuration (12)

The original configuration of the suspension was transformed into a U shaped compliant link and a rigid link. The rigid link was determined to be the torsion reaction

link with the bushings at the wheel carrier separated into two, to allow for torque reactions around lateral axis to ensure non-linear behaviour of the bushings. A round cross-section for the rigid link is preferred for the rigid link to have high torsional stiffness. The U-shaped compliant member was designed to possess very low vertical stiffness at the wheel carrier attachment point and low bending stiffness throughout the flexible arm extending towards to chassis. Flat or open cross-sections for the compliant link can be used. The overall stiffness for the compliant control arm is the sum of compliances in vertical and lateral axis in series. Figure 16 shows the control of the compliant link around vertical and lateral directions. (12)

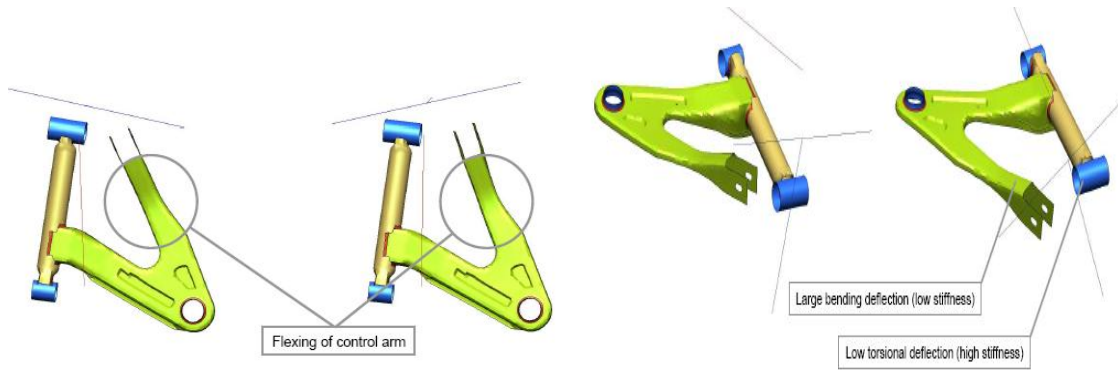


Figure 16 Prototype of compliant U-link showing large bending and low torsional deflection (12)

Figure 17 shows the lower link as a single stamping, where the compliant U-link exhibits large bending deflection and the rigid link represents low torsional deflection and also the prototype of the compliant U-link. Tests conducted on the compliant U-link showed similar performance compared to the original H-arm and the only limitation was to adhere to large anti-lift angles.

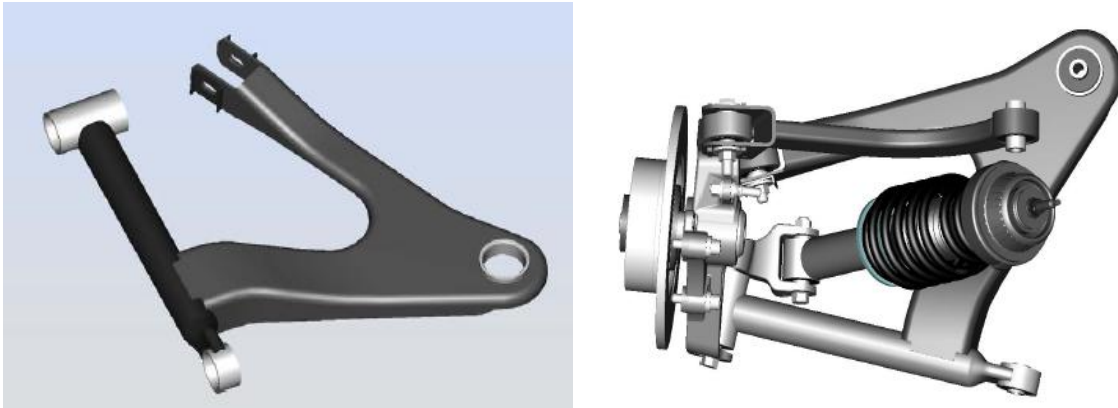


Figure 17 U-link compliant suspension prototype (12)

TORCS (TORsional Link Elevated Compliant Suspension) (13) is a compliant suspension built on the concept of Equivalent Elastic Mechanism (EEM) (14). TORCS was built entirely by imitating the EEM and suspension linkages were designed by subsequently removing each degree of freedom. The result was a compliant suspension system with the TORCS acting as the lower control arm of the suspension creating two torque reactions. TORCS does not rely on bushings for its performance and can even accommodate a spring on the torsion link. TORCS is ideal for small cars and FWD SUV's and not suited for powerful RWD vehicles as they create large traction forces. Figure 18 shows a schematic of the torsion link and the TORCS compliant suspension concept.

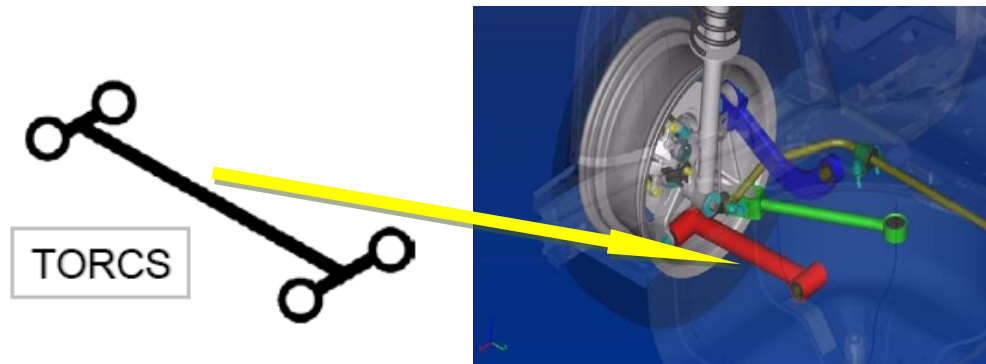


Figure 18 TORsional link Elevated Compliant Suspension (15)

FLECS (Flexible Links Elevated Compliant Suspension) (13), a concept of flexible control arms is similar to U-link suspension except that it has 2 separate compliant links to achieve the required elastokinematics. The U-link of the compliant suspension previously developed was slightly modified into 2 separate complaint links, connected to each other by a rigid link and the free end connected to the wheel carrier. The rigid link is connected to the chassis of the vehicle. FLECS acts as the lower control arm of the compliant suspension developed. Figure 19 shows a schematic of FLECS and the concept developed.

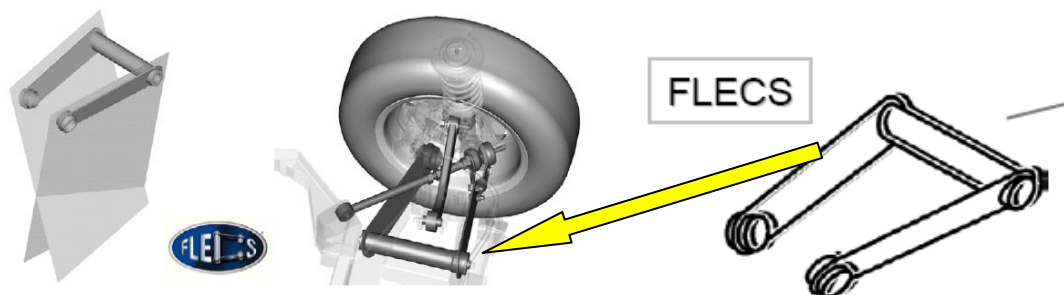


Figure 19 FLECS (Flexible Links Elevated Compliant Suspension)

Composite materials were also put to use in some of the light and low cost compliant suspension concepts for vehicle suspensions. Quality Industrial Product

proposed the use of composite compression C springs for automotive as well as industrial applications (16). Tests conducted on static measuring bench, fatigue and relaxation machines as well as road tests proved that composite C springs can successfully replace coil springs and trivial leaf springs. These C springs also showed cost and weight reduction with improvement in ride handling capability. Investigations on elliptic spring design using composite material also verified the use of composites in meeting the challenges of weight reduction and improved performance in the automotive industry (17). The use of composite materials were not only confined to replace rigid links, but were also extended to the use of axle beams which execute the function of axle suspension, wheel location and spring suspension at the same time (18). These composite axles can either extend to the full width of the vehicle or be restricted to one side in independent suspensions. Stress analysis on composite leaf springs having taper in thickness had constant stress with lower flexure stress, but higher nominal shear stress (19).

2.4 Suspension Characteristics

2.4.1 Camber Angle

Camber angle is defined as the angle between the wheel centre plane and the vertical to the plane of the road. Camber angle is usually measured as negative when the top of the wheels extend inwards from the road vertical plane and measured positive when the top of the wheels extends outwards from the road vertical plane. Normally

when a vehicle is cornering, the wheels on the outside of the turn go into positive camber relative to the ground reducing the lateral grip of the tire under load. To achieve this phenomenon, suspension designers tend to achieve positive camber in bump and negative camber in rebound motions, in their designs. Apart from this, a slight positive camber on a loaded vehicle would make the tires roll with maximum contact patch resulting in even wear and lower rolling resistance on a slightly curved road. The slip angle increases with increase in positive camber and the cornering force increases with increase in negative camber (20). Poor rough road tracking would occur for that suspension design which gives much camber change with wheel vertical displacement. Some of the advantages of reduced camber include less tire wear, smaller slip angles, better steering reaction at low lateral acceleration.

2.4.2 Toe Angle

Toe angle is defined as the angle between the vehicle centre plane in the longitudinal direction and the line intersecting the centre of plane of the wheel with the road plane. It is measured as positive when the front of the wheels tend towards the vehicle centre line and negative when the front of the wheels tends away from the vehicle centre line. Toe is also measured in distance instead of angles. It is measured as the difference between the front and rear edges of the right and left wheel rims at centre line level (21). In this case, toe is measured negative when distance between the front edges of the wheel rims is greater than that of the rear edges and is measured positive when the distance between the front edges of the wheel rims is lesser than that of rear edges. For

improved straight line stability of a vehicle, front wheel toe-in is usually preferred to counteract the compliances present in the suspension control arm bearings tending to push the wheel backwards. In front wheel drive vehicles, toe-out is used to balance the effect of traction forces and toe-in is preferred to balance the deterioration in the driving stability during coasting condition. Toe angles can be controlled with the help of toe control links which are often used with independent multi-link suspensions.

2.4.3 Track

Track is usually defined as the distance between the centers of right and left wheel. On twin tires the track is measured as the mean distance between them. Track usually should be as large as possible but should not exceed a threshold value relative to the vehicle width. Track on passenger cars is normally 1210 to 1602 mm (20). The width utilization ratio, which is the ratio of the track width to the vehicle width, should be large enough. Half track change is measured as the movement of one of the wheel in the lateral direction. Some of the effects of track width change in automobiles as given by Milliken and Milliken (22) are.

- Increasing the track width reduces the load transfer on turn entry. With the tire loads more evenly distributed the tires can produce more force.
- In steady state cornering the track width and the CG height determine the total lateral load transfer. Increasing the track reduces the load transfer improving lateral acceleration capability.

- Increasing the track will improve the braking in a turn performance by increasing the maximum lateral force available.
- Track width increase will help rough road cornering. Less lateral load transfer gives less body roll and this means there is more suspension travel available before hitting bump stops.

2.4.4 Wheelbase

Wheelbase, an important variable in ride and handling property, is measured from the centre of front axle to the centre of the rear axle. A relatively longer wheelbase compared to the overall length of the vehicle reduces the influence of load on axle. Ride comfort can be enhanced by reducing the short body overhangs to the front and rear, which also reduces the tendency to pitch oscillations. For the same steering input, smaller turning radius can be achieved with shorter wheelbase. Typically wheelbase lies in the range of 2160 mm to 3040 mm. The ratio of wheelbase to vehicle length, preferably large ranges from 0.57 to 0.67 on estate saloons, 0.56 to 0.61 on notchback saloons (22), less than 0.56 on coupes and around 0.72 for small cars (22).

CHAPTER 3

COMPLIANT MULTI-LINK SUSPENSION CONCEPTS

This chapter considers a generic independent type multi-link rear suspension model as a reference for the conceptual development of a compliant multi-link suspension. It discusses the type of compliant material used and the design constraints assumed for the compliant multi-link suspension. Finally, the results obtained from simulations carried out on the compliant suspension model and the reference suspension model are reported and analyzed.

3.1 Reference Multi-link Suspension Model

3.1.1 Description of the Model

One of the most commonly used types of independent rear suspension is the multi-link suspension. Compliance is incorporated in the suspension system either in the form of springs where they are used to provide compliance, but do not participate in the kinematics of the suspension, or in the form of leaf springs where they are used to provide compliance as well as partially or completely responsible for the kinematics of the suspension. One of them is the multi-link rear suspension which uses springs for its energy storage mechanism. Jozef et,al (23) discusses one such very commonly used multi-link suspension where the multibody system comprises of rigid links (wheel carrier, kinematic links) and compliant elements (springs), linked to each other by kinematic

joints. The architecture of the multi-link suspension is shown in Figure 20. This model has been a source for various researches on multi-link suspensions such as for developing analytical methods to determine the kinematics, optimizing the joint locations (24) and including compliance in the multi-link suspension through bushings.

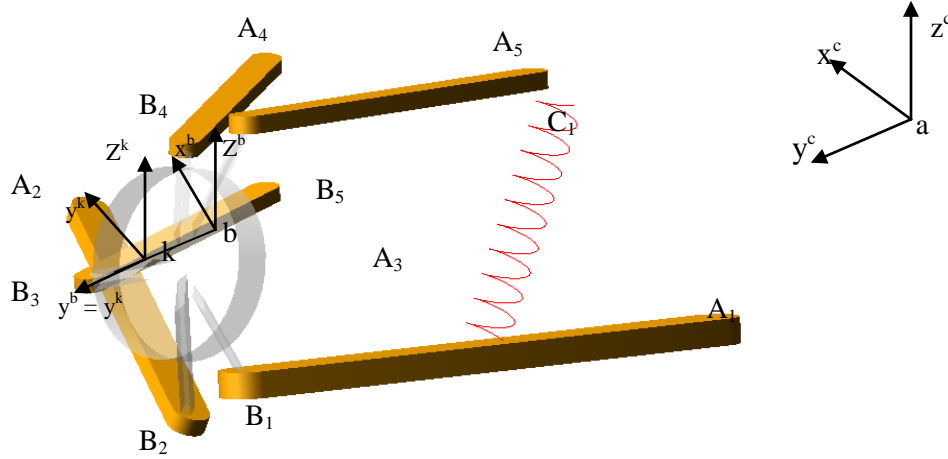


Figure 20 Multi-link suspension architecture

The coordinates of the reference multi-link suspension are given in Table 1. The chassis connection points for the links 1 to 5 and the spring ($A_1^a, \dots, A_5^a, C_1^a$) are described with respect to the 'a'-coordinate system arbitrarily fixed in space. The wheel carrier connection points for the links 1 to 5 (B_1^b, \dots, B_5^b) are described with respect to the 'b'-coordinate system, which is offset from the centre of the wheel carrier in the lateral direction. (23)

Table 1 Coordinates of multi-link suspension with respect to different coordinate systems

	B ₁ ^b	B ₂ ^b	B ₃ ^b	B ₄ ^b	B ₅ ^b	A ₁ ^a	A ₂ ^a	A ₃ ^a	A ₄ ^a	A ₅ ^a	C ₁ ^a
X	-43.8	40.9	140.7	72.2	0	-102.5	310	201.5	206.5	-1	-112.5
Y	-1.5	-30	1	-8.5	0	220	506	427	443	363	438
Z	-89.3	-130.4	-34.2	90.8	121	-64.5	-48.5	-9.5	96.5	119.5	115.5

Since it is a useful practice to express any subsystem of an automobile with respect to a single coordinate system, a fixed coordinate system attached at the centre of the wheel carrier was chosen to describe the complete model of the suspension. Transformation matrices were used to transform the coordinates from space fixed ‘b/a’-coordinate system to ‘k’-coordinate system.

The data used for transforming the coordinates from space fixed b and a-coordinate system to the fixed ‘k’-coordinate systems are (23)

$$o^{k,a} = [-1.9 \quad 711.7 \quad 20]^T \quad (1)$$

$$o^{b,k} = [0 \quad -50 \quad 0]^T \quad (2)$$

$$R^{k,a} = \begin{bmatrix} .9980 & -.0027 & .0637 \\ .0037 & .9999 & -.0165 \\ -.0636 & .0167 & .9978 \end{bmatrix} \quad (3)$$

$$R^{b,k} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad (4)$$

where $o^{k,a}$ is the distance between the origins of the k and a-coordinate system, $o^{b,k}$ is the distance between the origins of the b and k-coordinate system, $R^{k,a}$ is the rotation matrix between the k/a-coordinate system, $R^{b,k}$ is the rotation matrix between the b/k-coordinate system.

Following are the transformation matrices used to convert the coordinates of the links to 'k'-coordinate system (25).

$$\begin{bmatrix} x_k \\ y_k \\ z_k \\ 1 \end{bmatrix} = \begin{bmatrix} & & & x_{k,b} \\ & R^{b,k} & & y_{k,b} \\ & & & z_{k,b} \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_b \\ y_b \\ z_b \\ 1 \end{bmatrix} \quad (5)$$

$$\begin{bmatrix} x_k \\ y_k \\ z_k \\ 1 \end{bmatrix} = \begin{bmatrix} & & & x_{ka} \\ & R^{a,k} & & y_{ka} \\ & & & z_{ka} \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_a \\ y_a \\ z_a \\ 1 \end{bmatrix} \quad (6)$$

where $(x_k \ y_k \ z_k)$ represents the coordinates in 'k'-coordinate system and the right hand side of the equation represents the product of the transformation matrix and the coordinates in expressed in b and a – coordinate systems respectively.

Table 2 indicates the coordinates of all the locations of the links and the spring after transforming them to the 'k'-coordinate system.

Table 2 Coordinates of multi-link suspension with respect to single coordinate system

	B_1^k	B_2^k	B_3^k	B_4^k	B_5^k	A_1^k	A_2^k	A_3^k	A_4^k	A_5^k	C_1^k
X	-43.8	40.9	140.7	72.2	0	-96.8	314.8	203.8	202.1	-6.7	-117.4
Y	-51.5	-80	-49	-58.5	-50	-492.7	-207.6	-285.7	-267.9	-347	-271.7
Z	-89.3	-130.4	-34.2	90.8	121	-82.6	-45	-11.78	94	105	92.72

3.1.2 Kinematic Model of the Suspension

Once all the coordinates are transformed into a single coordinate system, the suspension is modeled in multi-body dynamics simulation software (MSC ADAMS). The kinematic model consists of 6 rigid elements (5 independent links and the wheel carrier). A schematic representation of the kinematic model is given in Figure 21. The multi-link suspension is constrained by kinematic joints. Universal joints are used to constrain the motion between the links and the chassis while spherical joints are used to constrain the motion between the links and the wheel carrier. Spherical joints can also be used instead of universal joints to constrain the motion between the links and the chassis, in which case the degree of freedom would increase to 6, instead of 1 degree of freedom obtained with universal joints.

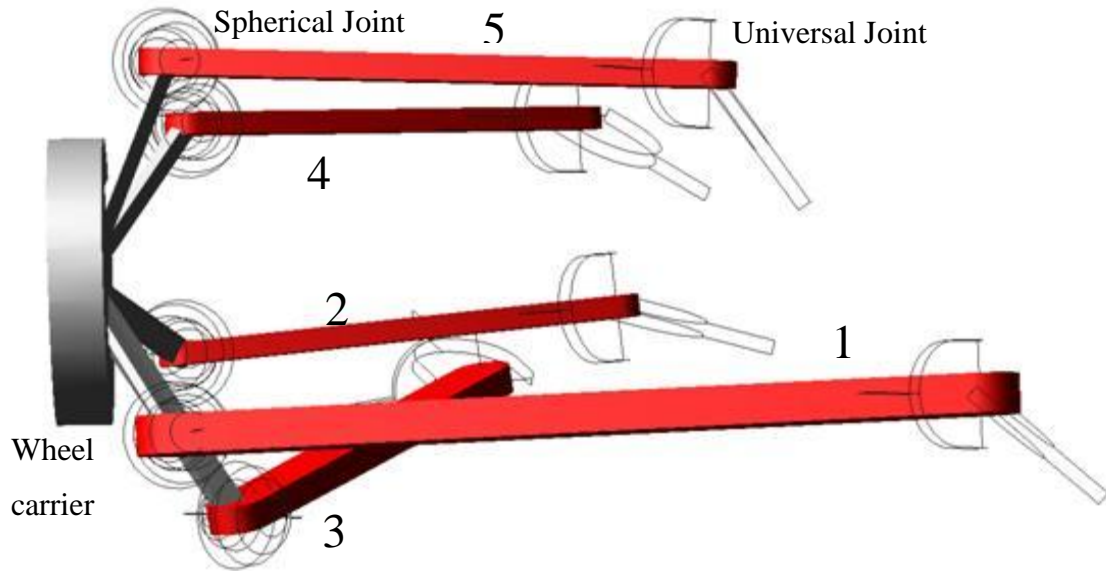


Figure 21 Kinematic model of the rigid multi-link suspension

3.1.3 Modeling of Non-Linear Spring

The kinematic model described above could only be used to determine the kinematic performance of the suspension. Apart from kinematics arising from orientation and location of the links, the other important characteristic of a suspension is its vertical compliance, which contributes to ride quality and comfort. Since the model described so far does not have any force elements to provide the right vertical displacement for a given vertical load, a non-linear spring was developed for the multi-link suspension model.

The concept behind modeling a non-linear spring in ADAMS for the kinematic model is that the spring is always in compression from its free length, giving a negative compressive force. A non-linear force deflection curve was used to simulate the non-linear behavior of the stiffness of the spring. The stiffness of the spring is set to be very

large when the wheel carrier reaches its full bounce position. The load acting on the spring at its reference position (when the vehicle is standing stationary), is considered as the load acting on the suspension of a quarter car model of typical automobile which uses a multi-link rear suspension. The force-deflection curve for the designed spring is given below in Figure 22.

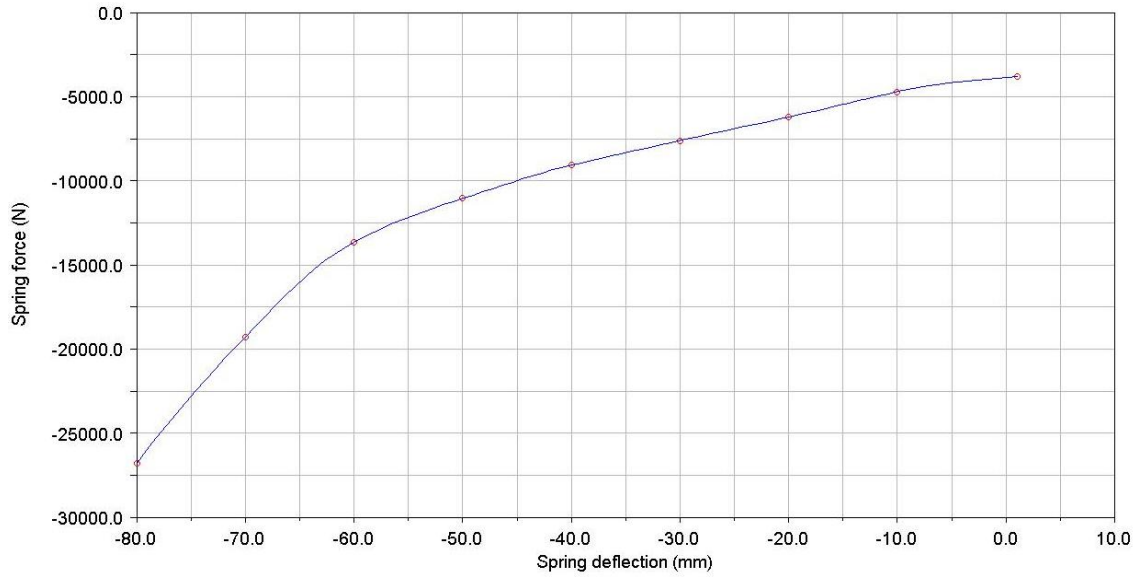


Figure 22 Non linear force-deflection curve for the spring used in the reference multi-link suspension model

3.1.4 Reference suspension model results

The kinematic suspension model along with the nonlinear spring is used as a benchmark or reference model for our research on compliant multi-link suspension. The performance characteristics of the suspension used for comparison are camber angle, toe angle half track change, half-wheelbase change and the vertical compliance of the

suspension. The kinematics of the suspension is unaffected by the stiffness of the spring and is measured by applying a motion, such that the system attains 80mm in bounce and rebound motion. The results of ADAMS simulation of the reference suspension model are given in Figure 23.

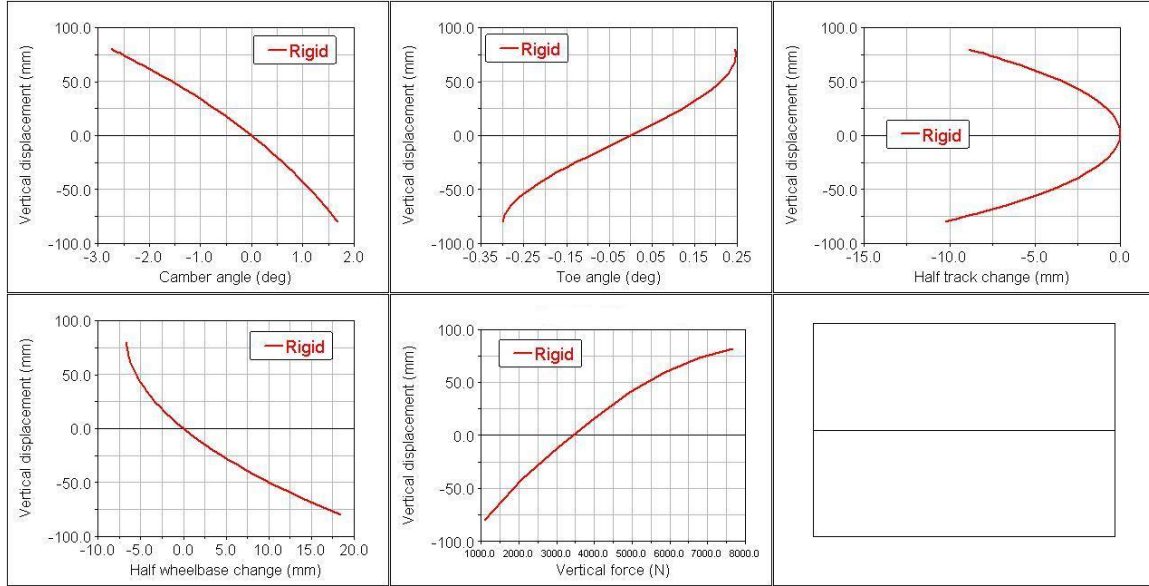


Figure 23 Results of reference rigid multi-link suspension

3.2 Compliant suspension model

After the results of the reference suspension are established, the development of the compliant multi-link suspension model is initiated. This section discusses in detail the type of material used, the design constraints and design of experiments considered in designing the compliant multi-link suspension model.

3.2.1 Type of Material

Apart from springs and bushings, the most common type of compliant member used in the suspension is the leaf spring. These leaf springs are basically made of steel and are either used to provide substantial springing by storing all the energy, or used to replace pivots, ball joints, bushings for kinematic guidance storing little or no energy. Intensive research conducted by Kirkham et al (9) shows the advantages of using a composite leaf spring instead of a steel leaf spring. Results from his research indicate that a composite leaf spring weighing about 3.6 kg was used to replace 10 steel leaf springs weighing about 18.6 kg. Apart from weight reduction, the composite leaf spring also offers improved ride, durability and packaging. Improved noise isolation can also be achieved by replacing coil springs with composite leaf springs. Figure 24 shows the use of a composite transverse leaf spring and a composite leaf spring replacing a coil spring (9).

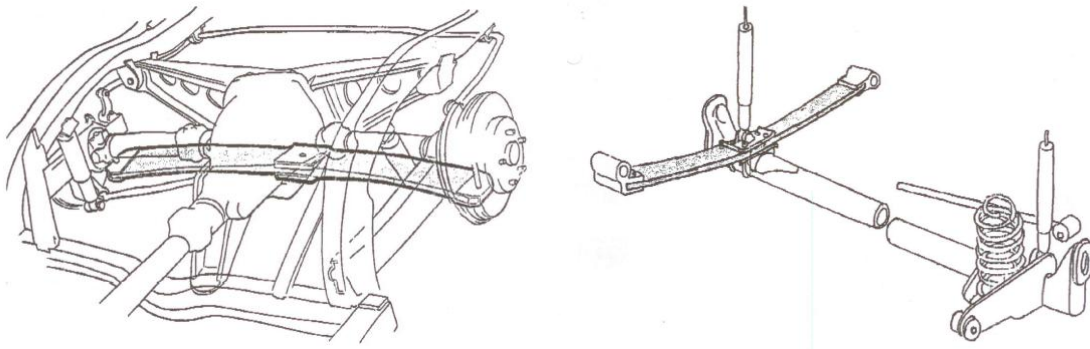


Figure 24 Composite transverse and longitudinal leaf springs, replacing a coil spring

Composite materials were chosen instead of the regular steel leaf springs to store energy as well as contribute to the kinematics of the suspension due to the advantages

mentioned above. Composites are a blend of materials which includes fiberglass, graphite, Kevlar, boron as reinforcement members and epoxy, vinyl ester, polyester, phenolics and thermoplastics (9). E-glass/Epoxy composites are used in our research on compliant multi-link rear suspension. The orthotropic properties of this composite material are given in Table 3.

Table 3 Orthotropic properties of the composite compliant member

E_1 GPa	E_2 GPa	E_3 GPa	ν_{12}	ν_{13}	ν_{23}	G_{12} GPa	G_{23} GPa	G_{13} GPa
41	28	13	.13	.29	.30	4.5289	4.1130	4.1212

where E_1 , E_2 , E_3 are the young's modulus, ν_1 , ν_2 , ν_3 are the Poisson's ratio and G_1 , G_2 , G_3 are the shear modulus of the composite material used.

Control over the stiffness property of the link is a function of the manufacturability of the composites and the type of epoxy used. The material property chosen are in accordance with tests conducted on a similar E-glass/Epoxy composite (26). A code is written in ANSYS for creating the flexible links with the above mentioned properties. The flexible links created in ANSYS are then imported into ADAMS. Creating beam elements for the flexible links in ANSYS is important at those locations where forces act on the flexible link when they are imported into ADAMS. Beam elements are also used to create joints on the flexible link in ADAMS. Once the material and its properties are fixed, the flexible link's geometric and kinematic constraints are determined considering other constraints.

3.2.2: Design Constraints

Given the design/packaging space, the links of the suspension are considered to be objects within that design envelope. Based on the packaging space available, a number of design variables exist for any suspension. Some of the design variables which could be considered for the compliant multi-link suspension are the length of the links, geometric dimensions of the links, shape of the links and the type of joint used to constrain the compliant members. The underlying assumptions for the design and implementation of the compliant members for the compliant multi-link suspension concepts are given below.

- Length of the compliant link is same as the rigid link being replaced.
- Orientation and location of the compliant link is same as the rigid link being replaced.
- A fixed joint is used instead of a universal joint which constrains the motion between compliant link and the chassis.
- A revolute joint is used instead of a spherical joint which constrains the motion between compliant link and the wheel carrier.
- With length and material property fixed, changes in the geometric dimensions (width and thickness) are done for those links, whose vertical compliance is not in accordance with the reference multi-link suspension model.

Assumption made on the type of joint used with the compliant link is derived from the compliant parallel 4-bar mechanisms previously used in automobiles (2) where different combinations of the lower and upper arms being compliant were discussed.

Fixed joints at the wheel carrier and pin joints at the chassis resulted in increased width and reduced thickness of the compliant member (2).

3.2.3: Design of Experiments:

This section discusses in detail the design of experiments carried out on compliant multi-link suspension models.

The first set of experiments carried out on the compliant multi-link suspension is done by replacing kinematic rigid links individually by their respective compliant members. Apart from suspension kinematics, vertical compliance is checked for the compliant multi-link suspension by applying a force at the centre of the wheel carrier, at its full droop position. Depending on the vertical compliance obtained from the compliant link suspension, the geometry of the corresponding link is modified so that the obtained vertical compliance is similar or close to the required vertical compliance of the reference suspension. Since the length of the compliant link is fixed to be the same as the corresponding rigid link, the width and thickness of the compliant member being the design variables, are modified depending on the stiffness of the compliant suspension. Once the required vertical compliance is achieved either from original dimensions or modified dimensions of the compliant link, the suspension is then checked for its kinematic characteristics. The compliant suspension is simulated to its ride height from its full droop position and is then checked for kinematics such as camber angle, toe angle, half track change and half wheelbase change. The kinematics of the compliant suspension is almost unaltered with change in geometry of the link to accommodate for vertical

compliance. Thus the compliant suspension is compared for its elastokinematics and vertical compliance against the reference multi-link suspension with individual rigid link replaced with the corresponding compliant link.

The second set and the third sets of experiments are carried out to test for similar elastokinematics by replacing two or three rigid links at a time. Since there are 5 links in the reference suspension model, different models consisting of combinations of 2 or 3 links replaced, are simulated and checked for its suspension characteristics. It is obvious that the compliant multi-link suspension obtained by replacing 2/3 links at a time would result in higher stiffness with its initial dimensions as compared to compliant multi-link suspension with just one compliant link. The geometric dimensions of the links are changed proportionately to deliver the required vertical compliance. Once the required vertical compliance is achieved by replacing 2 or 3 links, similar kinematic simulations are done as before.

Based on the assumed constraints on the compliant suspension model, the above explained design of experiments is conducted and the obtained results are compared with the reference rigid multi-link suspension. The results obtained and the procedure to converge on the best compliant suspension among the three sets of experiments conducted is explained in detail in the next section.

3.3 Compliant Suspension Results

This section discusses in detail the results obtained from the compliant multi-link suspension concept developed. It also discusses the method used to compare different

suspension characteristics between the reference rigid multi-link suspension and the compliant multi-link suspension. The results from the best compliant suspension configurations are presented in this section. Results from all other models simulated would be presented in later sections.

3.3.1 Cost Evaluation

The overall performance (cost) of the compliant multi-link suspensions developed is evaluated to determine the best suspension configuration. This is done by computing the root mean square deviation (RMSD), also known as root mean square error (RMSE) in performance characteristics at the successive positions in bounce and rebound of the wheel carrier between the reference rigid multi-link suspension and the compliant multi-link suspension. The equation used to calculate RMSD is given below (8).

$$RMSD = \sqrt{\frac{\sum_{i=1}^n (ref_i - comp_i)^2}{n}} \quad (7)$$

where n is the number of data points on respective performance characteristic for the reference and compliant multi-link suspensions respectively, the notations *ref* and *comp* are the individual performance characteristic for the reference and compliant multi-link suspensions respectively.

Since the root mean square deviation calculated for each individual performance characteristics have different units associated with them, non-dimensional forms of root mean square deviation (NRMSD) was used to calculate the overall performance of each

simulated compliant suspension model. NRMSD usually normalizes the RMSD to the range of the observed data or normalizes to the mean of the observed data (8). The mean of the observed data is used in our case to calculate the overall performance of each compliant multi-link suspension model. The formula for calculation NRMSD is given below.

$$NRMSD = \frac{RMSD}{comp_{mean}} \quad (8)$$

Before summing up the individual error deviations of each performance characteristic to get the overall performance cost, weights are added to each calculated RMSD. Vertical compliance is assigned a weight of 40, camber and toe is assigned a weight of 20 each and, track and wheelbase were assigned a weight of 10 each. Weights assigned to individual performance characteristic were based on their significance in determining the performance of the compliant suspension. In the reference rigid suspension model, a spring was used to provide vertical compliance for a given vertical load and since the compliant link was used to replace the spring and provide the right vertical compliance for the same force, which also contributes to ride quality and comfort, highest weight was assigned to vertical compliance characteristic. Kinematics of the suspension such as camber and toe angles do not change significantly with change in vertical stiffness of the flexible link, but contributes to the ride and stability of the vehicle and also independent of the type of vehicle, the second highest weights were assigned to them. Track and wheelbase also contributes to the ride and handling of the vehicle but are

dependent on the type of vehicle. In the present case a generic model is considered and the lowest weights were assigned to them.

Table 4 shows the deviation in performance characteristics for each compliant suspension model and their overall performance evaluation. The best suspension configurations (lowest cost) among the three sets of experiments conducted are highlighted in the table. In the notations used in the table, the number preceded by Comp represents the rigid links in the reference suspension, which are replaced by compliant links. WC, WT, WTR, WW, WVF represents weights for camber, toe, track, wheelbase and vertical force deviations respectively and cost represents the total performance cost for each compliant multi-link suspension model.

Table 4 Calculation of overall performance for each compliant suspension concept

	Camber Deviation	WC	Toe deviation	WT	Track deviation	WTR	Wheelbase deviation	WW	Vertical force deviation	WVF	Cost
Comp - 1	0.08843	20	0.38541	20	0.05197	5	0.01904	5	0.22481	40	18.8240677
Comp - 2	0.01298	20	0.50168	20	0.00775	5	0.12678	5	0.15325	40	17.0958594
Comp - 3	0.02338	20	1.35637	20	0.01457	5	0.01677	5	0.12121	40	32.6002471
Comp - 4	0.04272	20	0.63559	20	0.01015	5	0.31160	5	0.25026	40	25.1851255
Comp - 5	0.17531	20	0.34734	20	0.08133	5	0.08218	5	0.13066	40	16.4969
Comp - 54	0.18249	20	0.49716	20	0.08516	5	0.11030	5	0.23726	40	24.0607261
Comp - 53	0.16833	20	0.42733	20	0.08883	5	0.09203	5	0.13938	40	18.3926977
Comp - 52	0.17237	20	0.60718	20	0.09054	5	0.04417	5	0.14355	40	22.0067063
Comp - 51	0.05871	20	0.61141	20	0.18204	5	0.11735	5	0.10929	40	19.2710114
Comp - 43	0.03916	20	1.00195	20	0.02569	5	2.81990	5	0.14596	40	40.888333
Comp - 42	0.02630	20	0.62794	20	0.01762	5	2.79311	5	0.23119	40	36.3860283
Comp - 41	0.12991	20	0.71937	20	0.09915	5	0.16501	5	0.18265	40	25.6124145
Comp - 32	0.03530	20	0.22591	20	0.01977	5	0.12833	5	0.11485	40	10.5587669
Comp - 31	0.17422	20	0.48731	20	0.10132	5	0.04396	5	0.14209	40	19.6404623
Comp - 21	0.17361	20	0.68166	20	0.09969	5	0.09175	5	0.11331	40	22.5948501
Comp - 543	0.17618	20	0.57720	20	0.09684	5	0.11634	5	0.21060	40	24.5577638
Comp - 542	0.18427	20	0.65868	20	0.09772	5	0.25520	5	0.18550	40	26.0438173
Comp - 541	0.07409	20	0.71054	20	0.17998	5	0.06393	5	0.17320	40	23.8401471
Comp - 532	0.16014	20	0.33590	20	0.09909	5	0.06392	5	0.14731	40	16.6284003
Comp - 531	0.04071	20	0.45062	20	0.18461	5	0.12058	5	0.19858	40	19.295747
Comp - 521	0.06149	20	0.72478	20	0.17463	5	0.03455	5	0.11633	40	21.4245313
Comp - 432	0.04242	20	0.64059	20	0.03241	5	0.38831	5	0.19655	40	23.6258308
Comp - 431	0.14907	20	0.80161	20	0.11130	5	0.15952	5	0.14345	40	26.1056757
Comp - 421	0.13334	20	0.76414	20	0.10094	5	0.33515	5	0.15802	40	26.4511715
Comp - 321	0.17822	20	0.58862	20	0.10288	5	0.09241	5	0.14756	40	22.2159412

3.3.2 Compliant Suspension Concept Results

The final performance cost for all the compliant suspension model is obtained by the sum of the individual NRMSD calculated for each characteristic, along with their respective weights assigned to them. This performance cost is used to evaluate the best suspension configurations among all the suspension models developed and simulated. The three best compliant suspension configurations obtained, are from each set of design of experiments explained in the previous section. Figures 25, 26, 27 shown below represents the schematic views and results obtained for the best compliant multi-link suspension models. Comp – 5, Comp – 32 and Comp – 532 represents those compliant suspension concepts, in which the 5th, 3rd – 2nd, 5th - 3rd - 2nd rigid links were replaced with compliant links. The results for all other compliant suspension models simulated are given in the appendix.

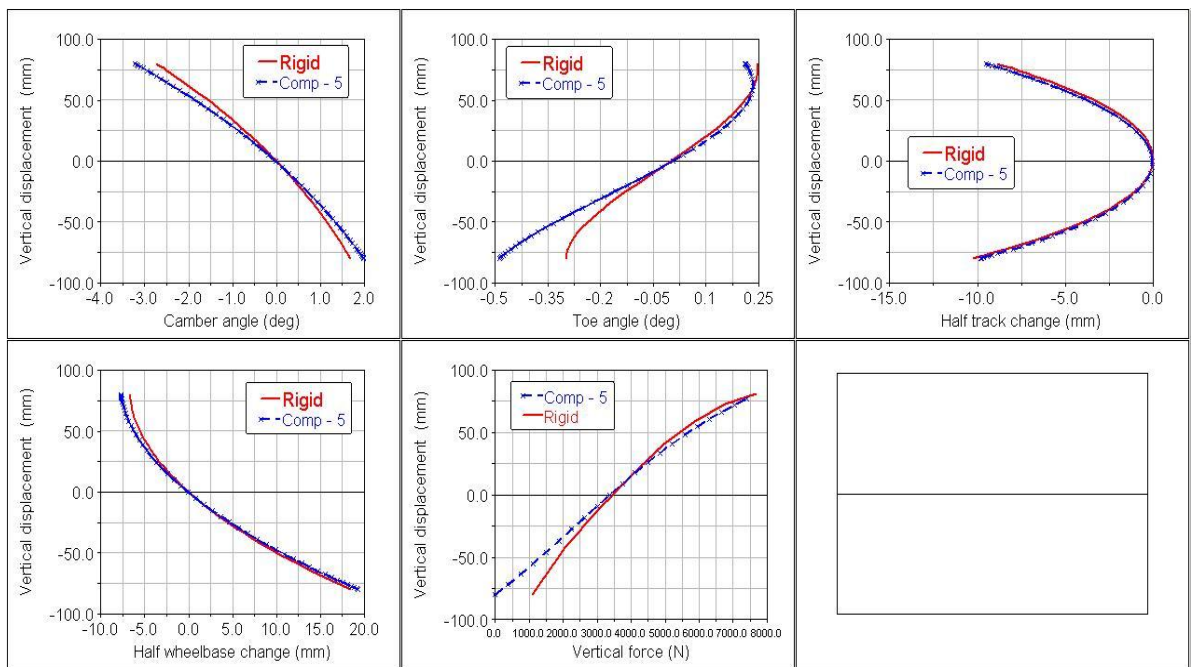
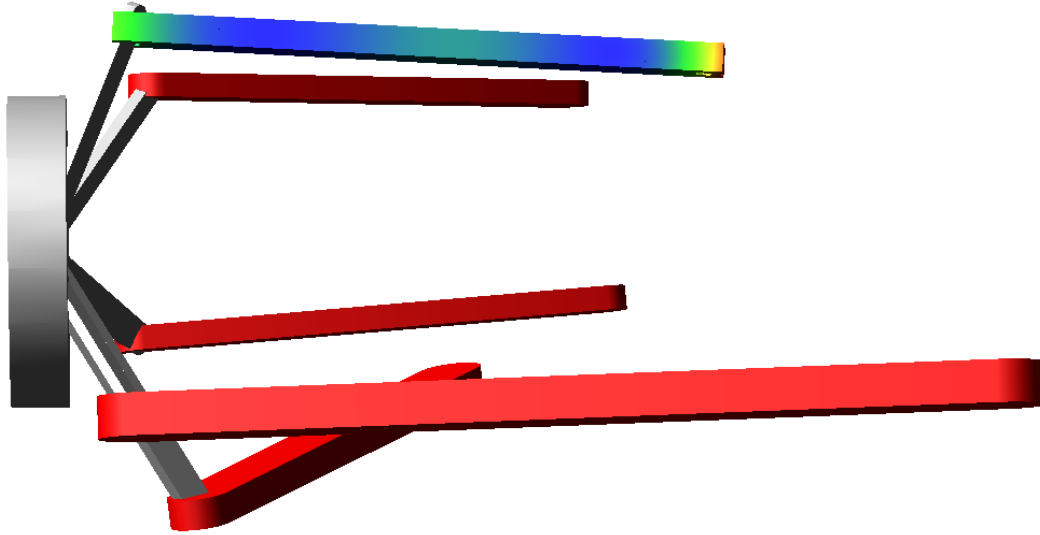


Figure 25 Compliant multi-link suspension and its performance characteristics with compliant 5th link

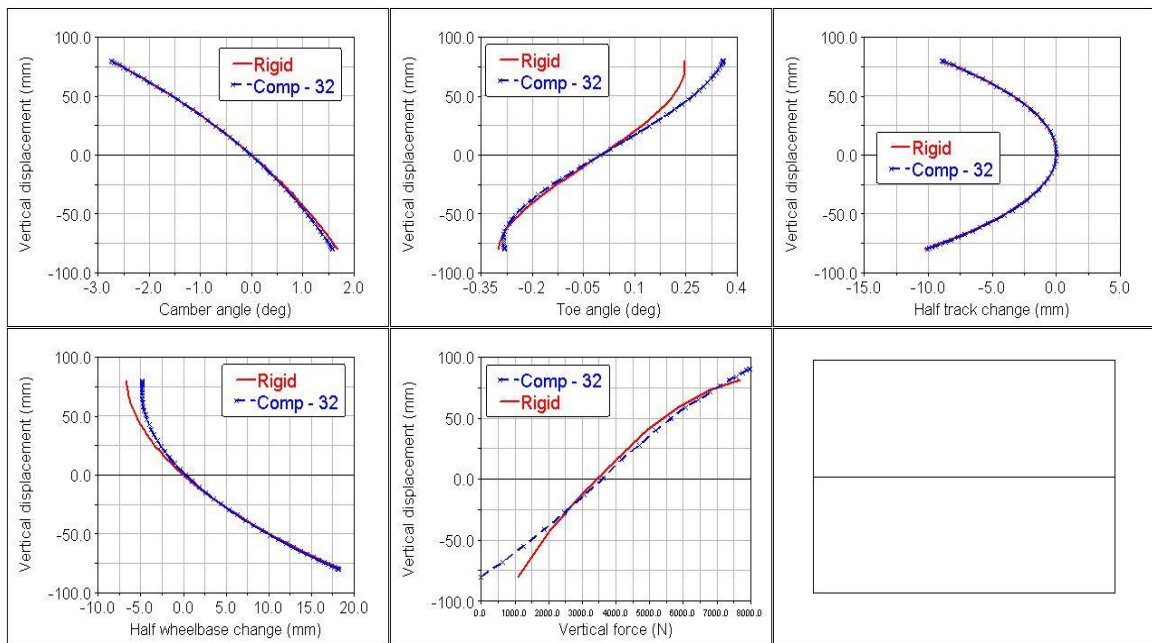
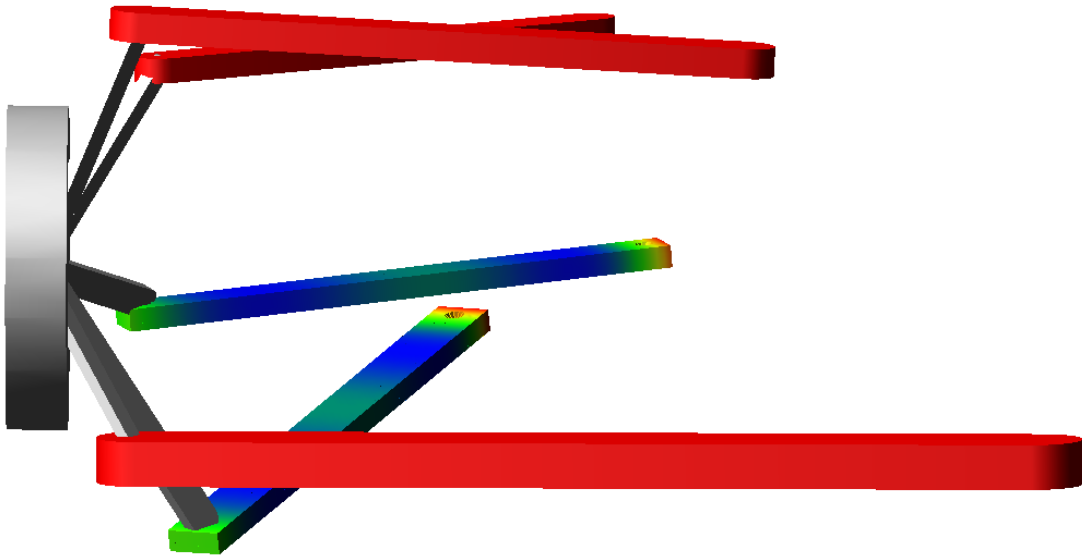


Figure 26 Compliant multi-link suspension and its performance characteristic for compliant 3rd and 2nd link

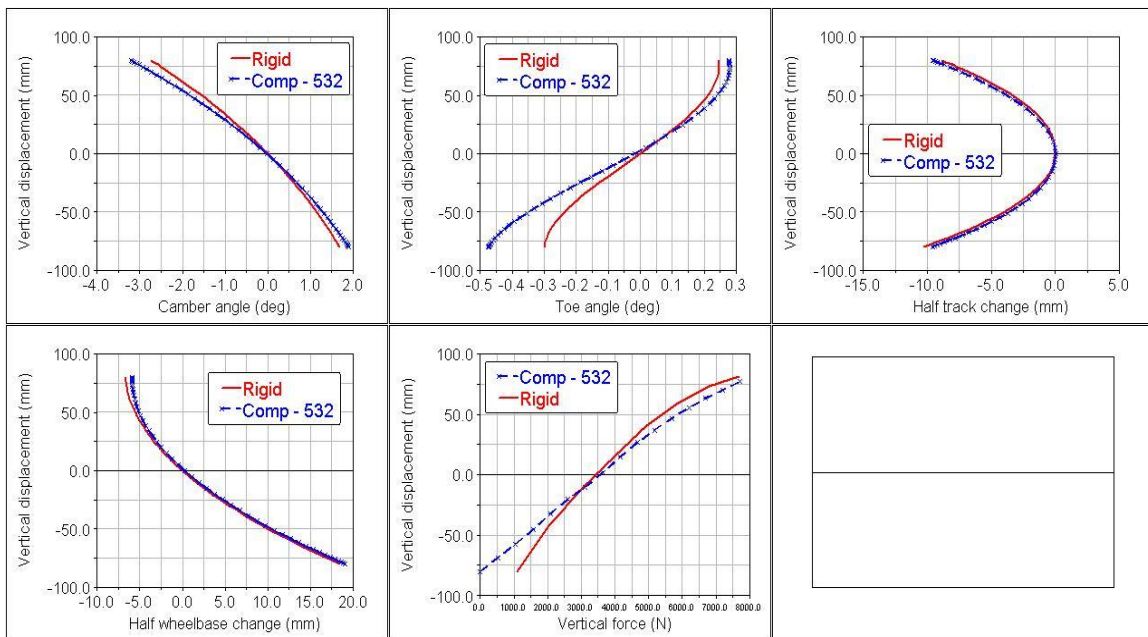
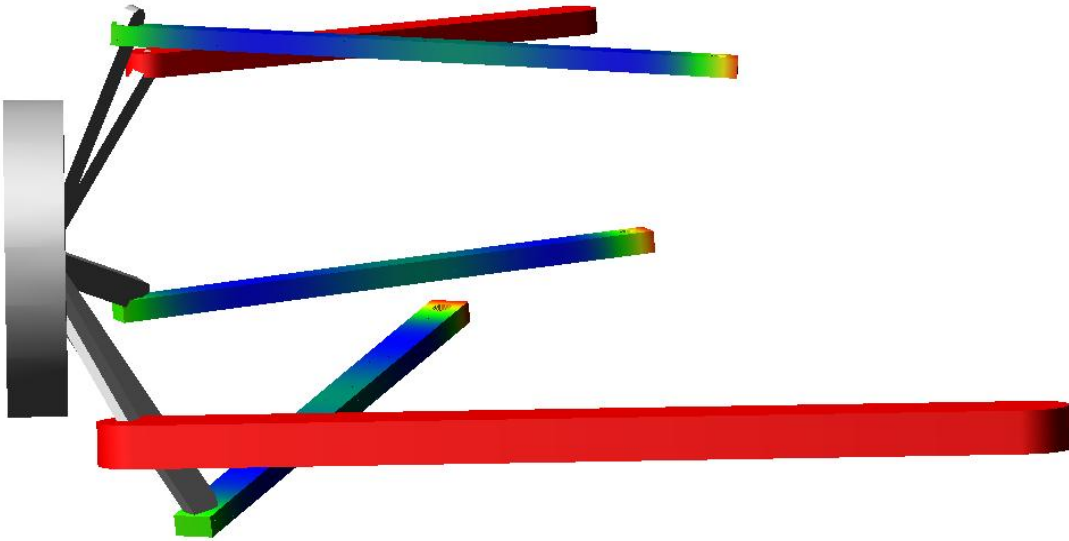


Figure 27 Compliant multi-link suspension and its performance characteristic for compliant 5th, 3rd and 2nd

3.4 Chapter Summary

This chapter discussed the compliant suspension concept designed for a generic multi-link suspension. The compliant suspension concept was aimed at investigating the potential use of compliant elements within the suspension which integrates the functions of energy storage mechanism and kinematic guidance for wheels. The reference suspension model was built in ADAMS, which included design of a non-linear spring with reference to an existing vehicle which uses multi-link rear suspension. Simulations were performed on the reference multi-link suspension to obtain a reference data for all the compliant suspension concepts designed later. Design of compliant multi-link suspensions were carried out by replacing individual and multiple kinematic rigid links along with the springs, by simple compliant links. Design of compliant members was focused on using composite materials instead of the regular steel leaf springs. These compliant suspension concepts with some assumptions made on the design, were simulated and the resulting performance characteristics were compared against the reference multi-link suspension results. Design of compliant multi-link suspension was explored by replacing multiple combinations of rigid links with compliant links to check for its kinematic performance against the reference suspension. Since the effect of kinematics of the suspension is nearly unaltered with the stiffness of the compliant link, modifications were done to the geometry of the compliant link to match the vertical compliance of the reference suspension. Deviation errors for each performance characteristic was non-dimensionalised to calculate the overall performance (cost) evaluation of each compliant multi-link suspension designed. Finally the best compliant

multi-link suspensions with individual and multiple links replaced was presented for further optimization of certain performance characteristic like the toe angle deviation following a vertical displacement of the wheel.

CHAPTER 4

COMPLIANT SUSPENSION CONCEPTS FOR A REFERENCE VEHICLE

4.1 Introduction

In this chapter we build on results obtained on a generic multi-link rear suspension in the previous chapter, by focusing on implementation of compliant members in the rear suspension of a reference high performance vehicle. The multi-link rear suspension of the reference vehicle was modeled in MSC ADAMS and the model is validated against results from K&C tests conducted on the reference vehicle. The suspension model was then used to iterate on compliant link suspension variants. The suspension analyzed the most includes a compliant ternary link upper arm replacing the rigid binary link upper arm in the reference suspension.

This chapter starts by describing the existing suspension and follow that up with a detailed simulation based analysis of the compliant suspension. We discuss concept evaluation using simulation and tests conducted on a mock-up of the proposed compliant link suspension.

4.2 Description of the reference suspension model

The multi-link rear suspension for the reference vehicle consists of a longitudinal arm with the wheel carrier as an integral part and 2 lateral arms extending from the wheel carrier along with a spring and damper system. Figure 28 shows a schematic of the reference suspension model built in ADAMS.

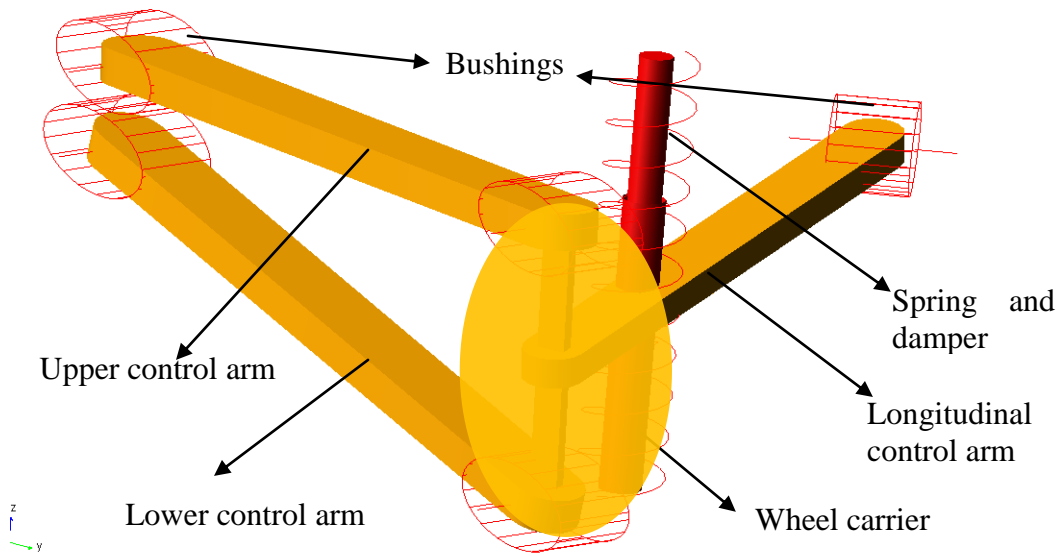


Figure 28 Schematic of the reference suspension

A virtual model of the reference rear suspension was built in multi-body dynamics simulation software (MSC ADAMS). The complete kinematic model built along with springs and bushings as shown in Figure 28 were simulated in ADAMS.

Once the results from the simulation and experiments conducted on a test rig were matched satisfactorily, new compliant suspension concepts were developed. The complete design process for the compliant link suspension concept and the final recommendations are explained in section 4.3.2.

4.3 Development of compliant suspension

The design process for the new compliant suspension concepts was initiated by having an insight of the functioning of basic compliant element mechanisms. Novel compliant elements such as using a hybrid universal joint or a ternary link for the replacement of rigid links were considered. The hybrid universal joint consists of two

rigid links with elastomer filled in between the two rigid links. Some of the advantages of using a hybrid universal joint are localized deformation and large range of motions. (27). Another compliant solution considered was to use a compliant ternary link instead of compliant binary link explained in the previous chapter, which replaces rigid links for kinematic guidance and energy storage.

4.3.1 Compliant suspension concepts considered

Compliant suspension concepts for the reference suspension were developed based on the design approaches mentioned in the previous section. A brief description of the design and their schematic are discussed below.

A range of compliant suspension concepts were initially considered to replace the rear suspension of the reference vehicle. A brief description of the design and their schematics of a few of them are included below.

I. Sliding clip concept

The sliding clip suspension concept was developed based on a simple design which works on a similar principle as an inverted nail clipper. Figure 29 shows the schematic of an inverted nail clipper and the suspension concept developed.

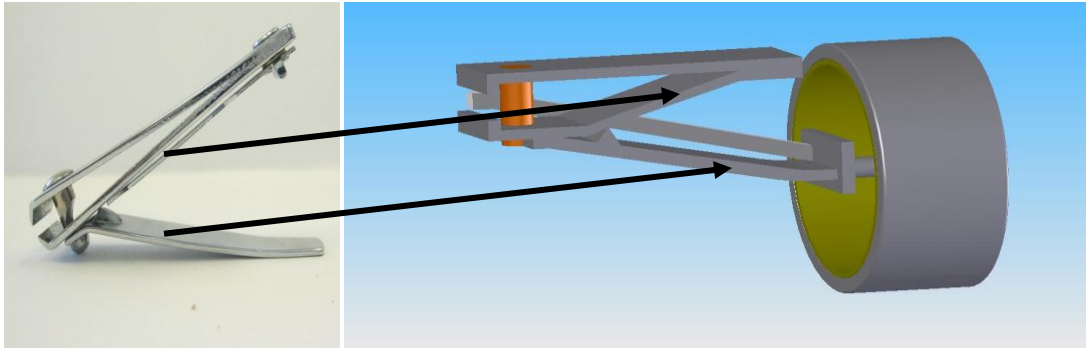


Figure 29 Sliding clip concept

This suspension concept consists of an upper arm, lower arm and a central arm connecting both the upper and the lower arm. The central arm, being the compliant member is fixed outboard to the upper arm and is pinned in-board. The lower control arm which is also a compliant member has a pivot point on the flexible central member and connects to the wheel carrier. During bump and rebound motions of the wheel carrier, the lower arm pivots about the compliant link, bending it and sliding along the pin attached to the upper arm. The distance between the upper arm and the lower can be used to control the vertical displacement of the wheel carrier, thereby making it suitable to different applications. During vertical displacement of the wheel carrier, energy is stored by the centre compliant member and the lower compliant member which acts as leaf springs in series connected to each other at an offset distance. A laterally and longitudinally inclined arm can be provided additional longitudinal and lateral support for this suspension. Some of the drawbacks of this suspension concept include the sliding friction associated with the pin, during wheel motions and the effect of longitudinal loads on the suspension. The longitudinal force experienced by the suspension is directly

transferred to the intersection of the central arm and the pin leading to wear issues at the intersection of the lower and central arm.

II. Binder clip concept

The binder clip suspension concept was developed based on a similar working principle of a paper binding clip. Figure 30 shows the paper binder clip and the compliant suspension concept design.

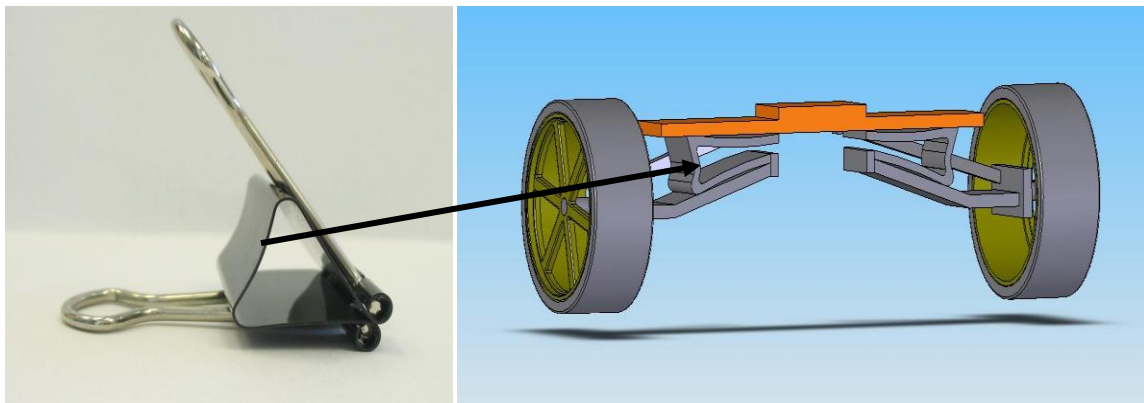


Figure 30 Binder clip concept

This suspension concept consists of an upper rigid member, which could act as the upper arm for either individual suspension or as an axle supporting both right and left wheel suspensions. The compliant member is in the form of a C-shaped member which is similar to the compliant member in the paper binding clip and can be stamped into a single piece. The compliant member is fixed to both the upper and the lower arm, where the lower arm connects to the wheel carrier. During the vertical displacement of the wheel carrier, the lower control arm flexes the C-shaped compliant member to expand, thereby acting as an energy storage member. Since the lower control arm can be assumed to act like a pivot at the chassis connection end of the suspension, higher camber angles

can be achieved during vertical displacement of the wheel carrier. In this case the C-shaped compliant member can also be used to limit the rebound motion of the suspension. One of the drawbacks of this suspension is the small range of motion obtained during bounce motions of the suspension with higher camber angles. Ride comfort and wear are also some of the issues associated with this suspension concept.

III. Cross-leaf spring concept

This suspension concept was developed by combining the traditional methods of springing using transverse and longitudinal leaf springs. Figure 31 shows the cross-leaf spring suspension concept developed.

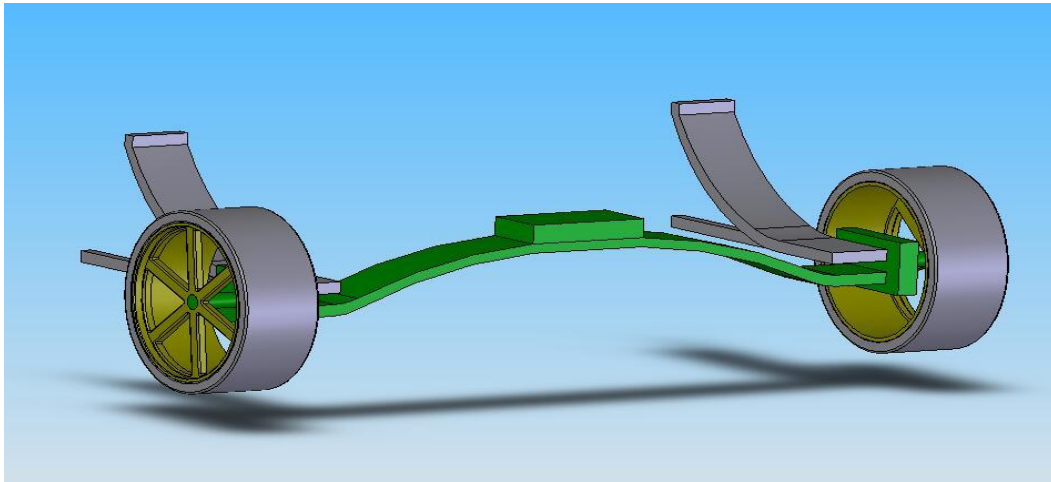


Figure 31 Cross leaf spring

This suspension concept consists of a semi-elliptic longitudinal leaf spring with one end connected to the chassis and the other end connected to the wheel carrier and the transverse leaf spring. The transverse leaf spring can either be a quarter-elliptic leaf spring or a semi-elliptic transverse leaf spring extending to the other end also. Compliant members with varied stiffness can also be used for suitable applications. The combined

effect of both the longitudinal and the transverse leaf spring contribute to the springing effect and the energy storage mechanism during vertical motion of the wheel carrier. This suspension concept can be referred to as a fully compliant suspension system. Shackles and toe control arms can be used to refine the response of the suspension. A longitudinal lower arm can be used as an additional member to support traction and braking forces. The drawback of this suspension is the semi-independent behavior of the suspension. If the transverse control arm is a quarter elliptic leaf spring, the lower control arm behaves like a simple cantilever beam accumulating large stresses in it. Since only leaf springs are used in this suspension concept, control of kinematics to the desired level may be an issue.

IV. Leaf springs and twist beam

This suspension concept is developed based on the twist beam suspension. Figure 32 shows the twist beam design with leaf springs.

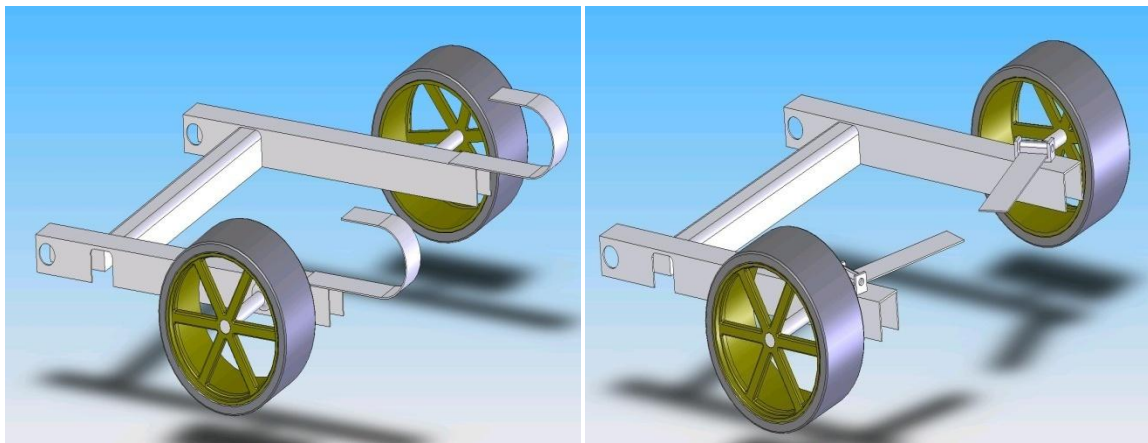


Figure 32 Leaf spring and twist beam

A typical twist beam suspension uses springs for its energy storage and twist beam which goes into torsion during vertical motion of the suspension. This suspension concept, apart from a typical leaf spring uses a C-shaped compliant member in the longitudinal direction or a ternary link in the lateral direction to store energy. With this design, packaging space is reduced and the use of a lateral compliant arm provides lateral support behind the contact patch and reduces lateral compliance oversteer. The main disadvantage of this suspension is the packaging issue with the compliant member taking up some part of the trunk space. Higher toe angles achieved would also contribute to the drawback of this suspension system.

V. Ternary link and trailing arm concept

This suspension concept was developed as a variant to the existing suspension of the reference vehicle. Figure 33 shows the leaf spring trailing arm suspension concept.

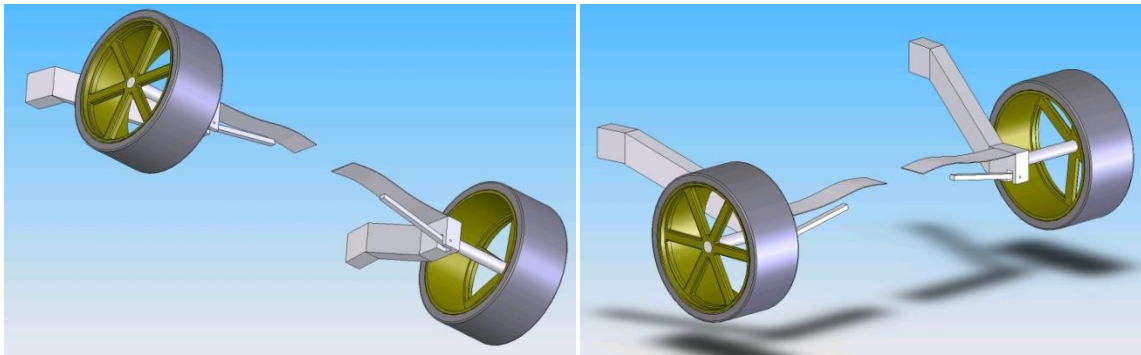


Figure 33 Ternary link and trailing arm

The leaf spring and trailing arm concept consists of an upper (or lower) transverse leaf spring or a compliant member instead of the transverse rigid trailing arm. The upper (or lower) compliant member provides energy storage and wheel guidance during vertical motion of the wheel carrier. Flexure members and leaf spring ternary links can also be

used instead of the standard leaf spring for the upper arm of the suspension. This suspension system would weigh less than the original reference suspension without any compromise in the performance of the suspension. Shackles can be used at the chassis connection point of the compliant member to allow for lateral elongation during bending of the compliant member. The major advantage of this suspension system is that similar performance characteristics can be achieved with the assigned package space used for the original suspension.

4.3.2 Recommended compliant suspension design

Figure 34 shows the schematic of the ternary compliant link used for the design of compliant suspension concept for the reference vehicle.

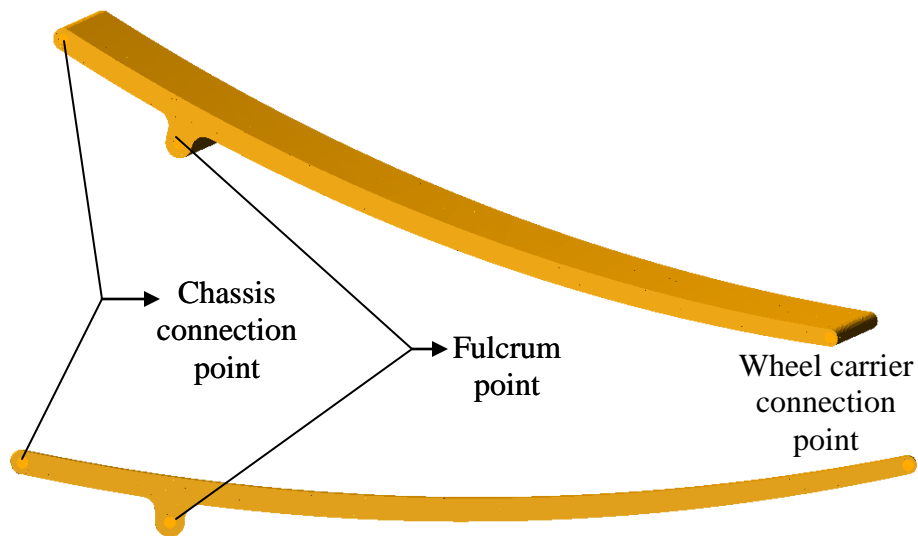


Figure 34 Schematic of ternary link

The term ternary link as one of the compliant solution considered is a simple link with three connections associated with it. One end of the ternary link is connected to the chassis through the shackle while the other end of the ternary link is connected to the wheel carrier and the intermediate fulcrum pivot point on the ternary link is also connected to the chassis. The intermediate fulcrum pivot point on the ternary link can be varied to obtain a shorter effective length of the compliant member from the fulcrum to the wheel carrier controlling the kinematics of the suspension. The distance behind the fulcrum allows the compliant member to have a longer effective length to accommodate for large range of motions. Bending stresses are distributed throughout the length of the link due to the effective longer length of the complaint link and also strain energy is stored throughout the length of the link. Control over the stiffness of the compliant link can be achieved for a given range of motion by moving the fulcrum point or tapering the thickness of the compliant link from the chassis connection point. Shackles are used to accommodate for change in effective length of the compliant ternary link during its motion.

Concept 4 which is the ternary link with trailing arm suspension was chosen among all the other compliant suspension designs for further development. The detailed explanation of the compliant suspension concept is described below.

Concept 4 chosen is a variant of the reference suspension. Figure 35 explains the modifications done to the reference suspension to arrive at its compliant counterpart.

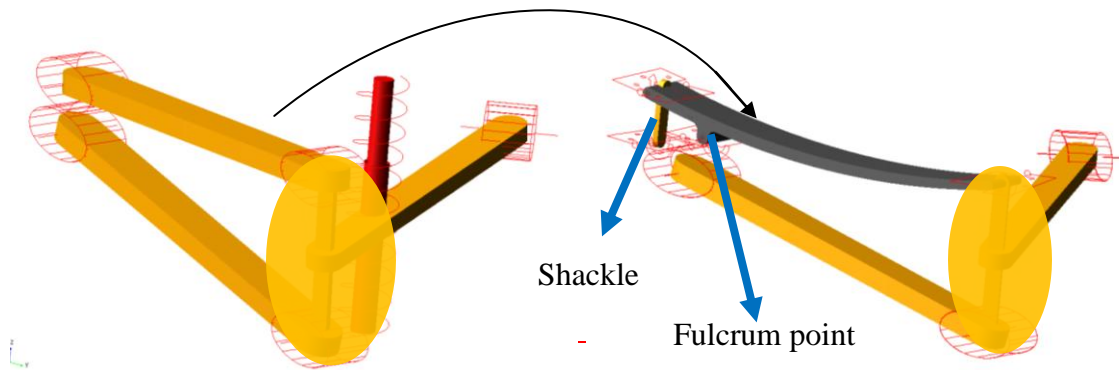


Figure 35 Reference suspension and its compliant suspension concept

The upper arm of the reference suspension was replaced by a compliant ternary link as shown in figure 35. This suspension concept has the dual functionality of both the energy storage and wheel guidance mechanism into a single compliant ternary link. The bushings at either ends of the upper arm in the reference suspension is replaced by pin joints on the compliant ternary link. The fulcrum point shown in the figure 35 is an intermediate joint location on the compliant ternary link which is connected to the chassis. Shackle is used to connect the compliant ternary link with the chassis. The length of the compliant link from the fulcrum point to the wheel carrier decides the amount of camber deviation during suspension vertical motion. The compliant member used for the ternary link was made up of composite material. The ADAMS model of the compliant suspension concept developed is as shown below with positions of the wheel carrier.

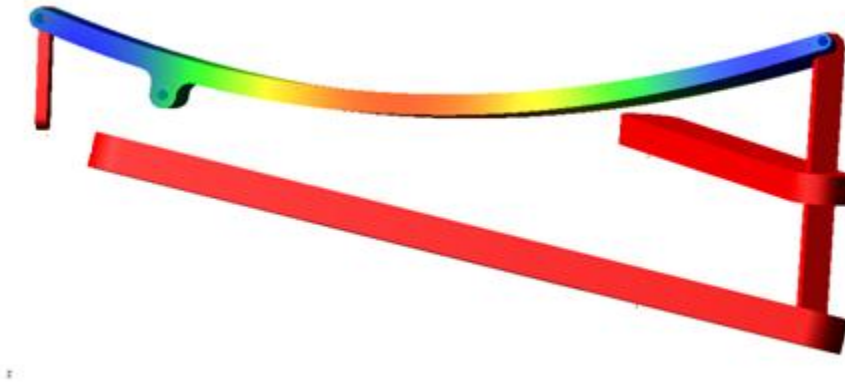


Figure 36 Compliant suspension concept, at its ride height

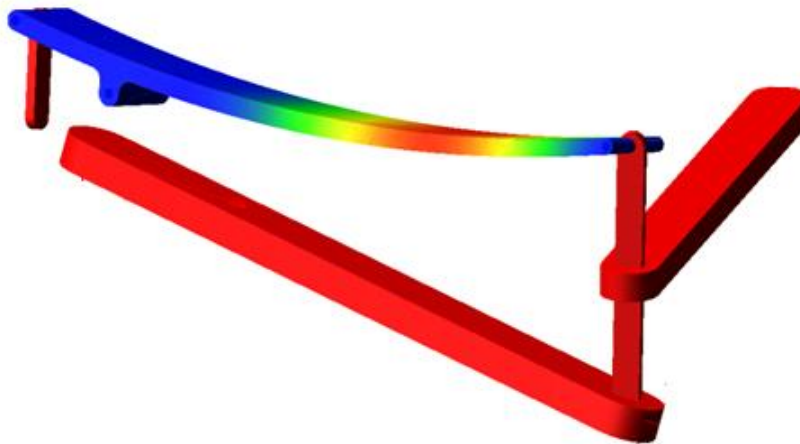


Figure 37 Compliant suspension concept, at its rebound position

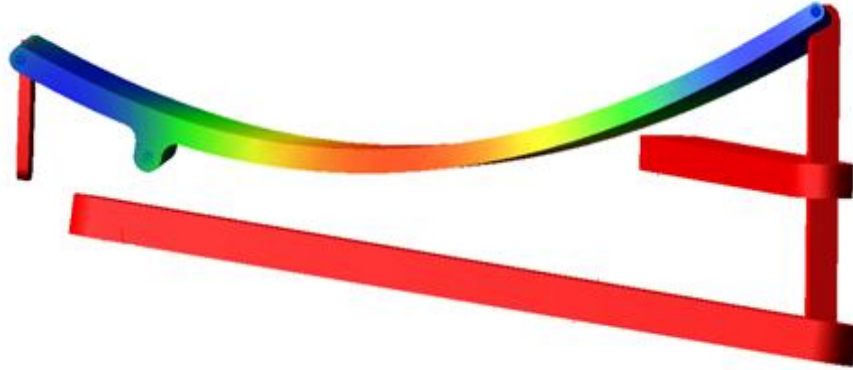


Figure 38 Compliant suspension concept at its full bounce position

Figure 36, 37, 38 shows the compliant suspension concept developed at ride height (reference height), complete rebound position and complete bounce position respectively.

4.3.3 Analysis of proposed suspension concept

This section discusses the results obtained from simulations performed on the compliant multi-link suspension concept developed for the reference suspension and mock-up of the suspension concept which uses an aftermarket fiberglass leaf spring, for the compliant member. The design of the compliant member was converged after numerous simulations performed in ADAMS and ANSYS. Apart from the shape of the compliant link, the design of experiments included variations in the length of the compliant member, in the locations of the effective pivot point of the ternary link and in the orientation of the complaint member. Once the final design was accepted, further

validation of the compliant suspension model developed by building a mock-up of the compliant suspension concept.

Other suspension performance characteristics are compared against the reference data. Even though the complete vertical displacement of the suspension is considered to be 80 mm in bounce and rebound motions, the compliant member performance was restricted to a minimal vertical displacement due to constraints in the test fixture. Individual performance characteristics for the compliant suspension are explained below which uses a curved ternary link with shackles at the chassis connection end.

1. Camber angle v/s vertical displacement

The figure 39 below shows the change in camber angle with respect to vertical displacement for the reference data, mock-up and simulations carried out in ADAMS. It can be seen that the camber angle deviation for the test conducted on the curved compliant member is in accordance with the mock-up as well as test data for the reference suspension.

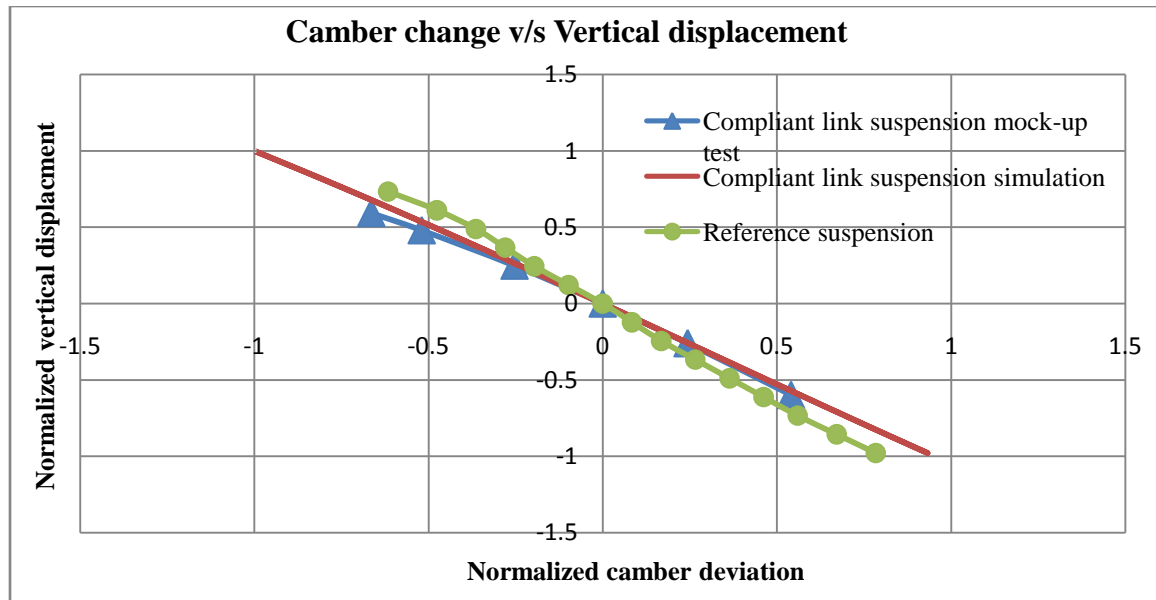


Figure 39 Camber change v/s vertical displacement

2. Vertical force v/s Vertical deflection

The figure shown below presents the results obtained for the vertical force v/s vertical deflection of the compliant link suspension concept. It can be inferred from figure 40 shown below that the aftermarket compliant link is close to the spring rate used currently on the high performance automobile. ADAMS simulation gives a satisfactory result in accordance with the reference data, but deviates from the results from the mock-up as shown in figure 40.

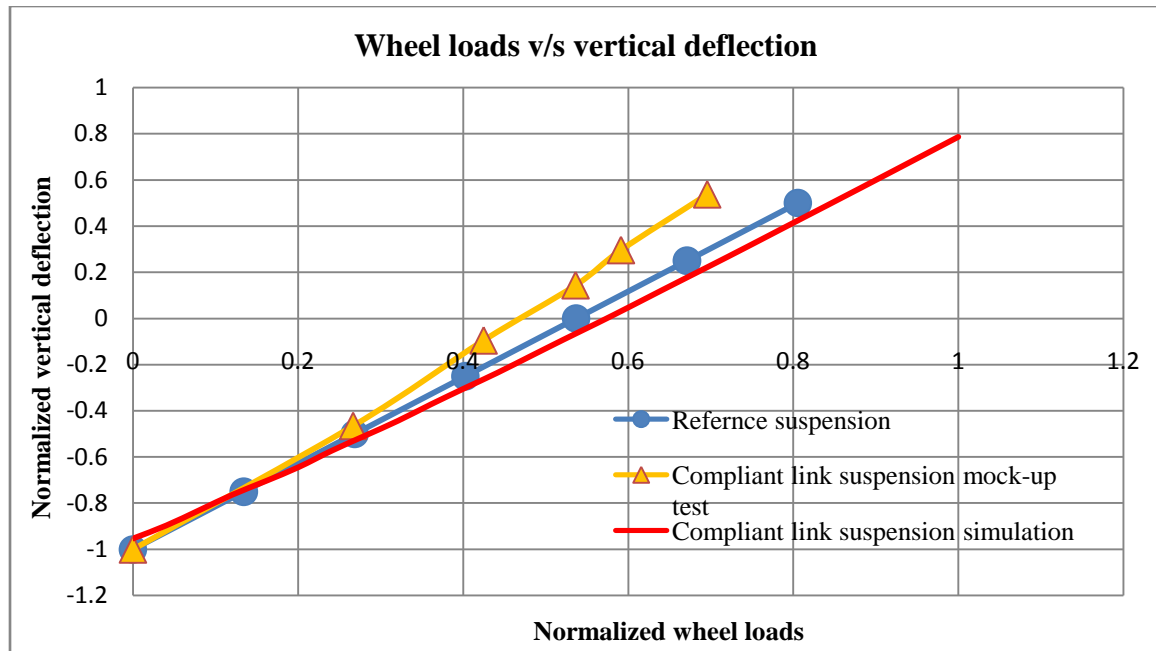


Figure 40 Wheel loads v/s vertical deflection

One of the possibilities for this deviation could be the assumed material property of the compliant member being used to simulate the suspension. Due to uncertainty in the material property of the compliant member used, further simulations conducted on the compliant link with its modified material property showed very close match with the mock-up, the result of which is shown in figure 41.

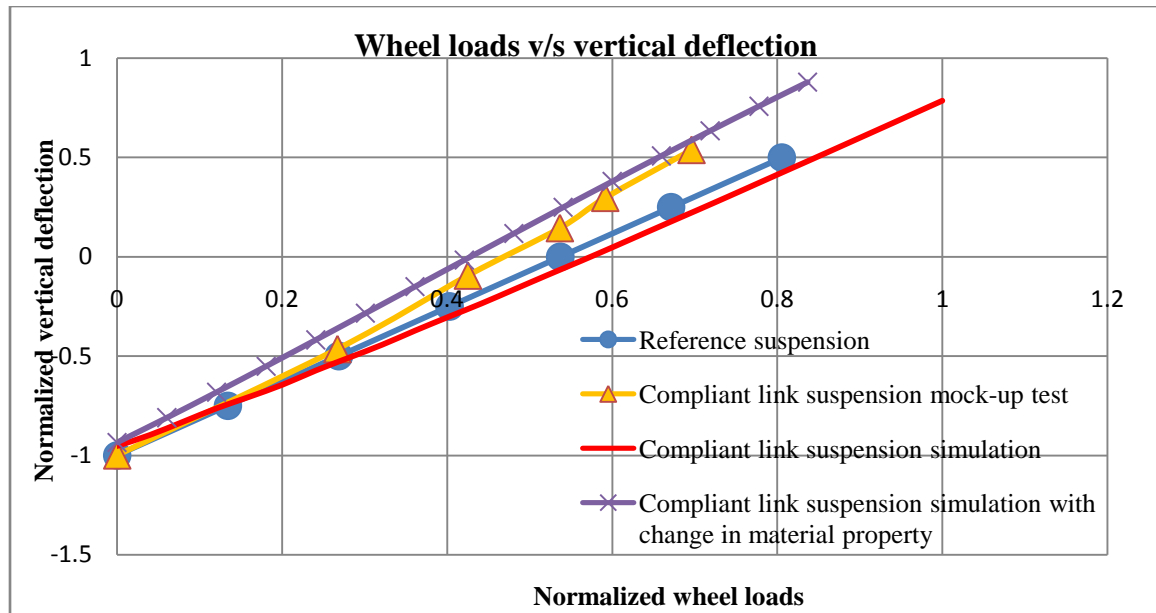


Figure 41 Wheel load v/s vertical deflection with change in material property

3. Lateral force response

Figure 42 below shows the lateral force response of the compliant multi-link suspension model.

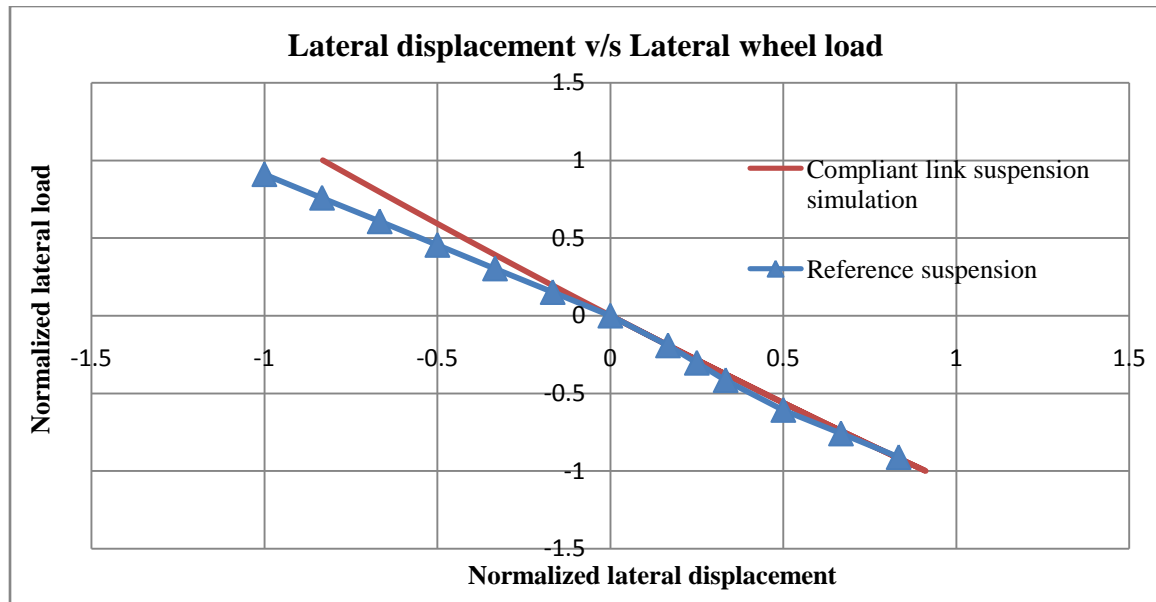


Figure 42 Lateral wheel load v/s lateral displacement

It is clear from figure 42 shown above that the lateral force response for the compliant link suspension developed for the reference OEM suspension is matching well with the data obtained for the reference suspension. The lateral force response is measured from the reference position of the suspension in ADAMS.

4. Longitudinal force response

Figure 43 below shows the longitudinal force response for the compliant link suspension.

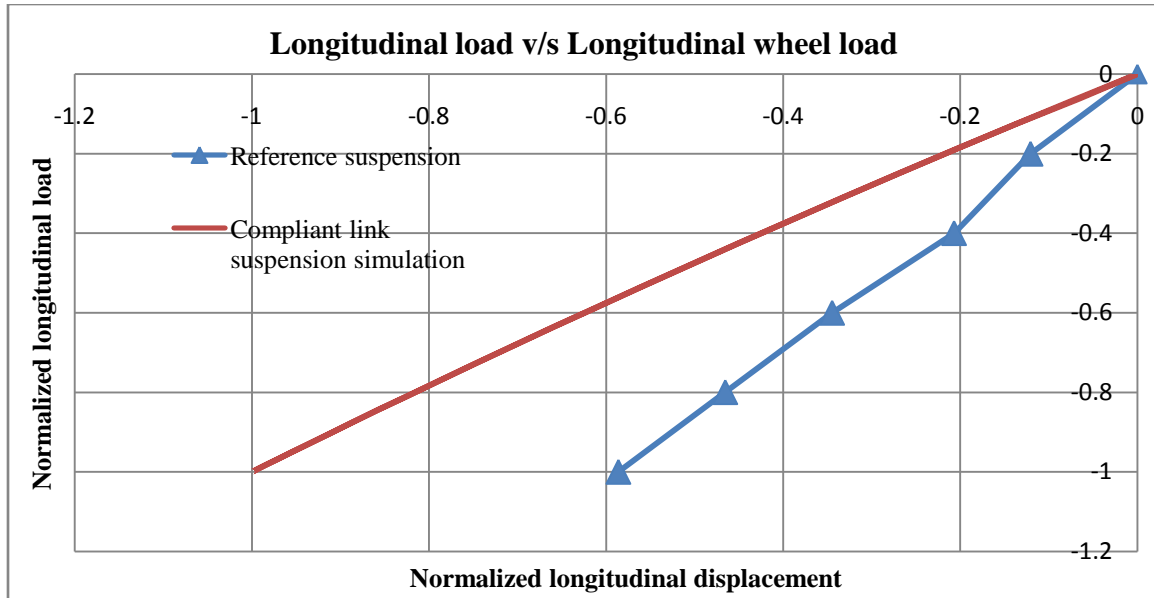


Figure 43 Longitudinal wheel loads v/s longitudinal displacement

The compliant link exhibits low stiffness in the longitudinal direction as compared to the longitudinal compliance of the reference suspension. The ternary compliant link chosen to build the mock-up of the suspension has not been optimized to perform better for longitudinal loads acting on the suspension. Further development of the ternary compliant link includes modifications to the thickness of the compliant link for a given stiffness of the material to attain the required longitudinal compliance.

5. Toe angle v/s vertical displacement

Figure 44 below shows the deviation in toe angle with respect to the vertical deflection of the wheel carrier.

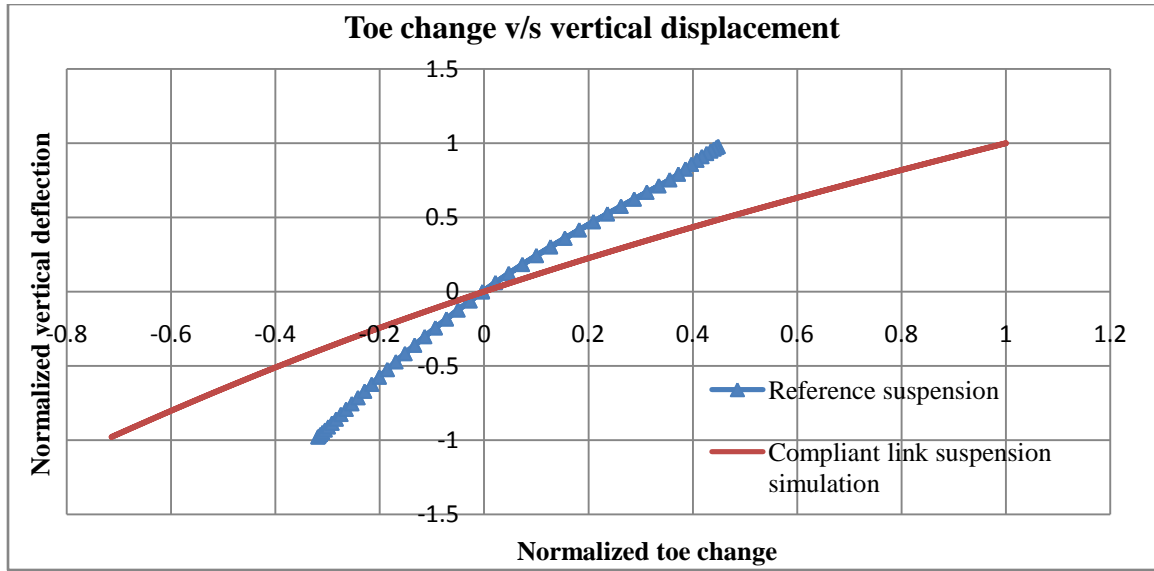


Figure 44 Toe change v/s vertical displacement

The toe angle deviation for the compliant link deviates from the reference suspension. It is also observed that the toe angle deviates from the reference data even for different sets of experiments including using a straight compliant member and also for optimizing the chassis connection point for the compliant link.

During the design phase of the ternary compliant link for the compliant suspension, simulations were also conducted on a straight ternary link to check for its kinematics. Results obtained from the straight ternary link are given below.

1. Camber angle v/s vertical displacement

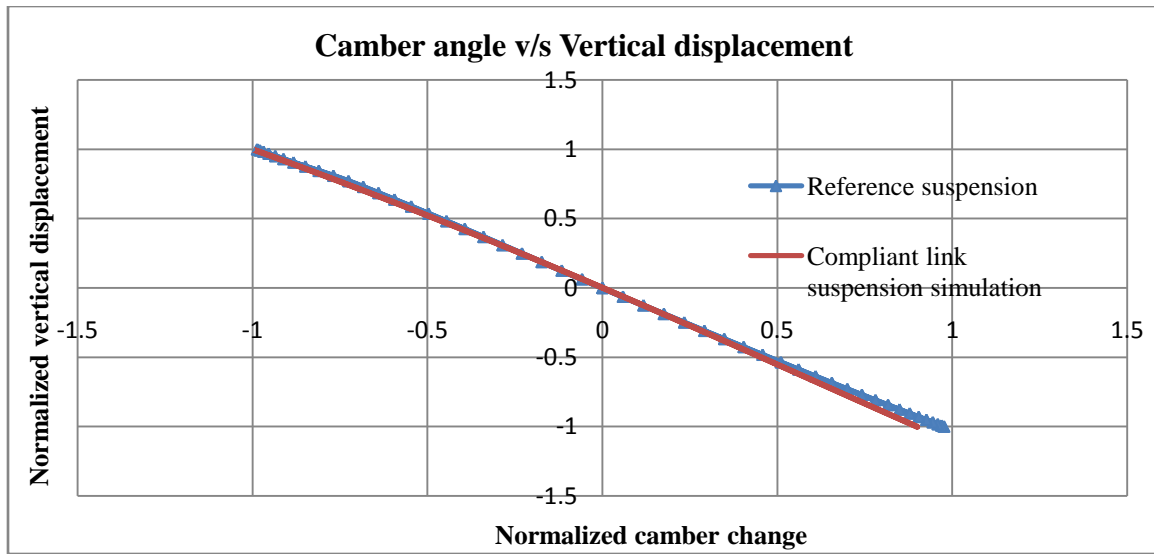


Figure 45 Camber angle v/s vertical displacement for straight ternary compliant link

2. Toe angle v/s vertical displacement

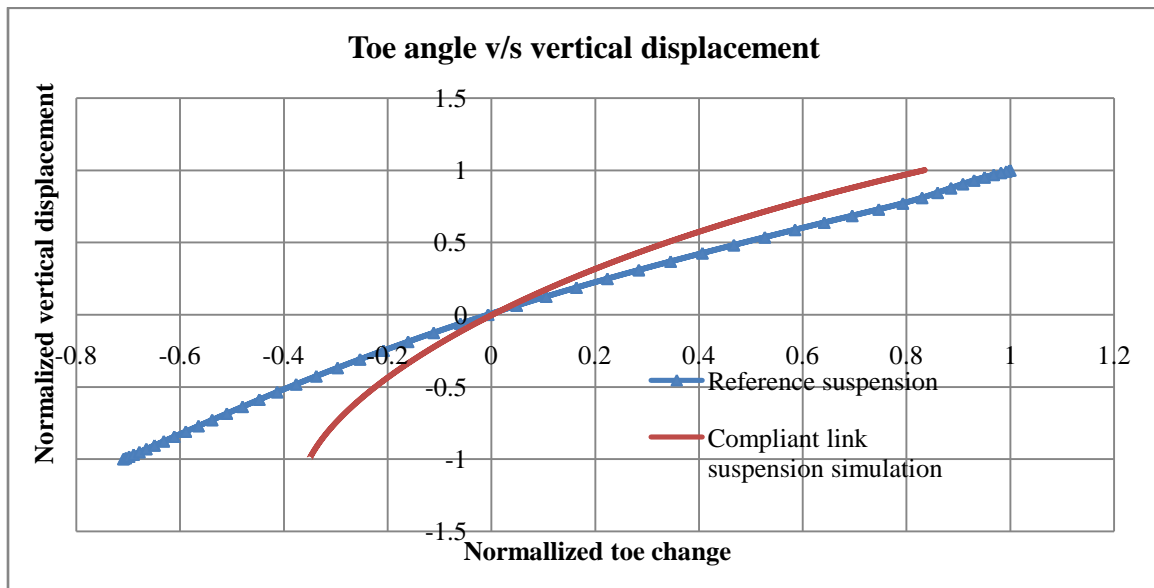


Figure 46 Toe angle v/s vertical displacement for straight ternary link

Figures 45 and 46 shows camber and toe angle change with respect to vertical deflection of the suspension. It was observed that camber angle showed good agreement

with the reference data, while toe angle change was found to be deviated to a smaller extent with very low toe change during rebound motion of the suspension. Since the aftermarket complaint link obtained from the local manufacturer had initial curvature to it, the design was concentrated on finding the optimized design for the curved compliant ternary link

Apart from simulations carried out on a straight ternary compliant link, several constraints on the compliant link suspension were modified to see the effects of those on the toe angle deviation. Figure 47 shows one such design of experiments conducted on the compliant suspension model. In this simulation, the length of the ternary compliant link was shortened compared to the length of the compliant link used in the recommended suspension and also the chassis connection point of the compliant member was shifted by 125, 155, 168 mm towards the rear. Results show that change in the location of the chassis connection point of the ternary compliant link towards the rear made the toe deviation better in rebound motion, whereas no change is observed in bounce motion.

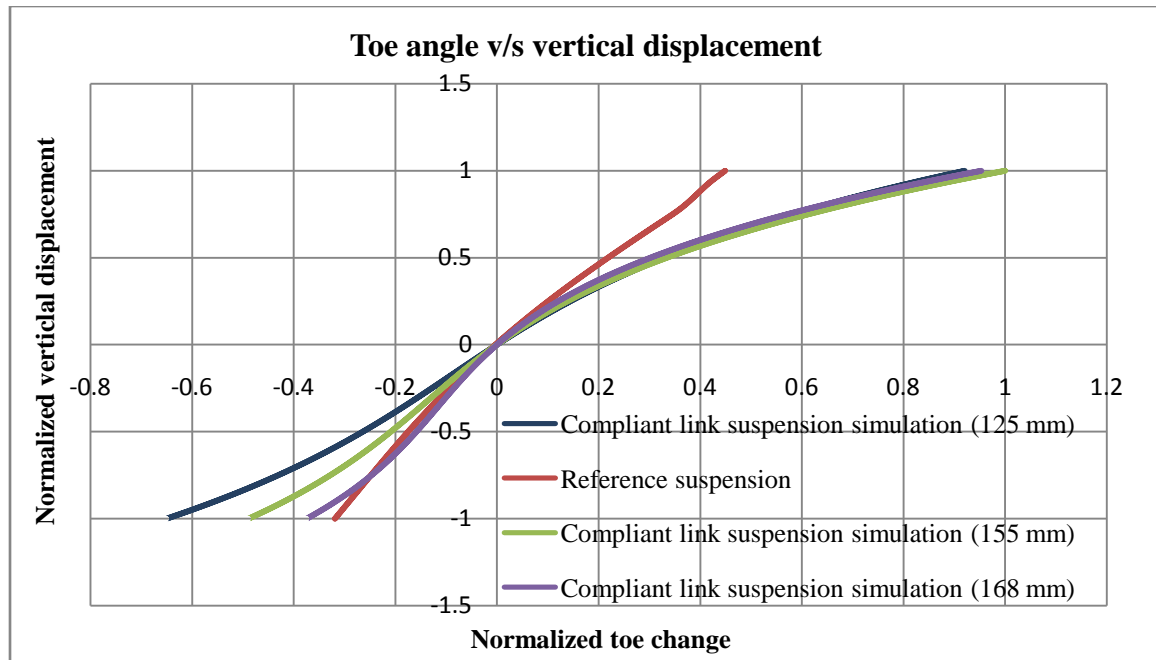


Figure 47 Toe angle v/s vertical displacement for modified chassis connection point

4.4 Chapter summary

This chapter discussed the compliant suspension concepts designed for a reference rear suspension of a high performance vehicle. Concepts designed were based on results obtained from simulations performed on a generic multi-link suspension model. Complaint suspension design was performed by looking into different compliant elements that could be possibly used to build a compliant suspension model. A variant of the reference suspension was chosen over a few compliant suspension concepts designed in which the upper arm of the reference suspension was replaced by a ternary complaint member thereby integrating the functions of energy storage and wheel guidance mechanism. Numerous simulations were carried out in ADAMS, to check for its performance characteristics against the reference suspension data. A mock-up of the

compliant suspension concept was built using an aftermarket compliant member to validate the simulation results. Simulation results for the ternary compliant member show good agreement with camber deviation and lateral force response. The vertical force response for the compliant link was also matched satisfactorily with the data from the mock-up after iterating on the material property of the compliant link to account for uncertainty the material property for the aftermarket product used for creating the mock-up. The longitudinal force response showed higher deviations from the reference suspension. Toe angle deviation was also significant. It is expected that the toe response and longitudinal compliance could be improved with further optimizations of the compliant link suspension design, such as with the use of straight compliant links of appropriate dimensions.

CHAPTER – 5

CONCLUSIONS AND FUTURE WORK

This chapter provides a summary of the conclusions drawn from the research conducted on compliant suspension concepts. It also outlines some remaining areas that need to be addressed to enhance the performance of the compliant suspension concepts.

5.1 Conclusions

The first objective of this thesis was to address the problem of integrating the functions of energy storage and wheel guidance mechanism into a single compliant member. One of the possible ways of achieving this was to replace both the springs which provide energy storage but not wheel guidance, and kinematic rigid links which provides wheel guidance but not energy storage, by a single compliant member within the suspension. Earlier forms of energy storage members were the leaf springs, where they contributed partially to the kinematics of the suspension. Leaf springs were stacked up to achieve the required vertical compliance, thereby increasing the weight of the suspension system. Compliance was achieved by making the leaf spring progressively wide and the load carrying capacity was accomplished with thicker cross sections, which again contributed to increased space and weight. This thesis has satisfactorily addressed the objective, by replacing kinematic rigid links and the spring by a binary and ternary compliant link. The binary compliant link used on the generic multi-link suspension was constrained to act as a simple cantilever beam, which successfully reproduced the

kinematics and stiffness illustrated by the reference suspension model. The binary link showed greater stress accumulation in the beam as compared to the ternary link. The ternary link allows for design adjustable ranges of motions and stiffness by modifying the fulcrum point on the link. Effective storage of strain energy and distribution of bending stresses are the main advantages of the ternary compliant link due to effective longer length of the link which extends from the fulcrum point to the chassis. ADAMS simulation results on compliant link suspensions analyzed shows good match with results from reference suspensions and also a mock-up of the compliant suspension. This proved the potential use of compliant links in the suspensions considered.

The second objective of this thesis was to develop compliant suspension concepts achieving similar or better performance characteristic compared to the reference suspension configuration chosen. Simulation results from chapter 3 shows promising results with the use of composite compliant members, replacing springs and kinematic rigid links on a generic multi-link suspension model. The best possible solutions to the compliant multi-link suspension with the assumed design constraints was obtained by varying the stiffness of the compliant link being replaced with their respective rigid links. The results obtained also indicate that the highest deviation in the results from the reference suspension was in the toe angle characteristic. The trend of the curve for toe characteristic is a function of the dimensions and orientation of the compliant links. The total vertical stiffness of the compliant multi-link suspension can be summarized as the sum of the stiffness of each kinematic rigid links replaced, which is a function of its geometric dimensions with fixed material properties.

Furthermore, compliant suspension concepts were developed for a reference existing suspension. Ternary compliant link with trailing arm suspension concept was chosen for further validation among compliant suspension concepts designed. Simulation results from ADAMS on both the geometric variations of the ternary compliant link and mock-up presented in chapter 4, signifies that the developed compliant suspension concept clearly achieves similar characteristics compared to the existing reference suspension with optimization of compliant links on characteristics like the toe angle.

5.2 Future work

The use of ternary links showed better results compared to binary links in carrying out the functionalities of energy storage and kinematic guidance for compliant suspensions. It was seen that both the binary and ternary compliant links showed great deviation in toe angle measurements. Performance of the compliant suspension model can be further enhanced with focus on the following issues.

Compliant suspension concepts designed for generic multi-link suspensions can be further investigated for optimal results by considering ternary links with different geometric profiles along with the use of shackles at the chassis connection end. Bushings can be used on the suspension models to validate its performance under lateral and longitudinal loads on the suspension. Optimization of the compliant links and the chassis connection points can also be carried out to control the toe angle deviations.

Ternary links used on the compliant suspension model developed for an existing rear suspension also showed greater toe angle deviations. Toe control links can be

incorporated in the compliant suspension system, which can better control the large toe angles. Other performance characteristics can be checked with the use of a straight ternary link on the mock-up of the compliant suspension built, to compare it against the ADAMS simulation and the reference suspension. Apart from upper control arms, lower control arm of the suspension can also be replaced with ternary links and checked for its overall performance. The current compliant link used can further be optimized for the effect of longitudinal forces on the suspension either by modifying the material property of the material or its geometric profile.

Composite materials used for the production of compliant links can also be considered for further research. Individual orthotropic properties of the composite materials could be modified to optimize its effect on control forces acting on the suspension and also to tune the kinematics of the suspension.

APPENDIX

Simulation results for generic multi-link suspension models

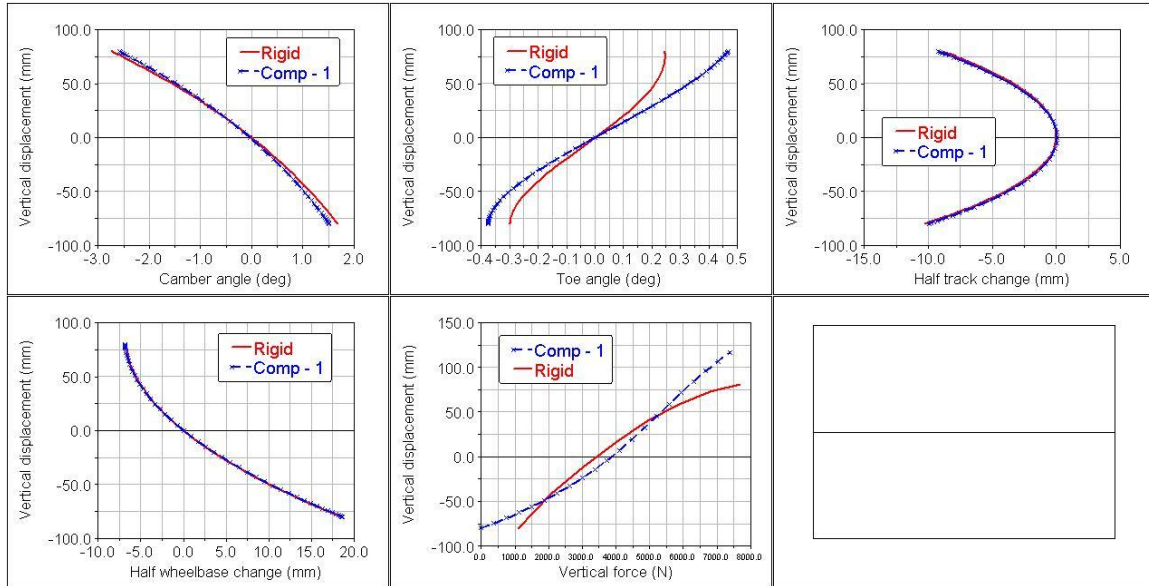


Figure 48 Compliant multi-link suspension performance characteristic with compliant 1st link

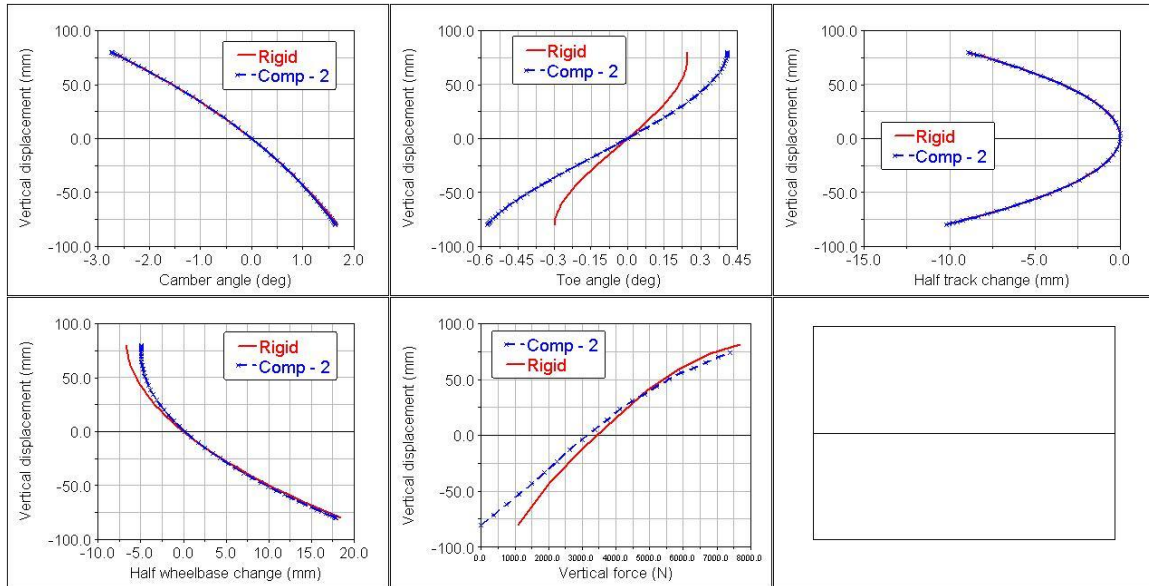


Figure 49 Compliant multi-link suspension performance characteristic with compliant 2nd link

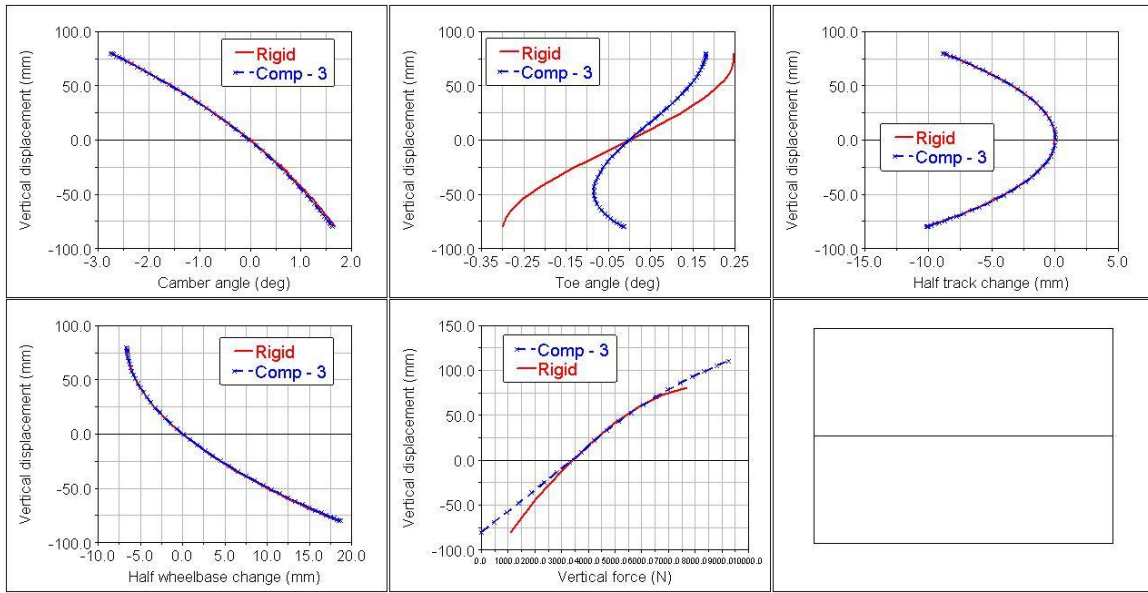


Figure 50 Compliant multi-link suspension performance characteristic with compliant 3rd link

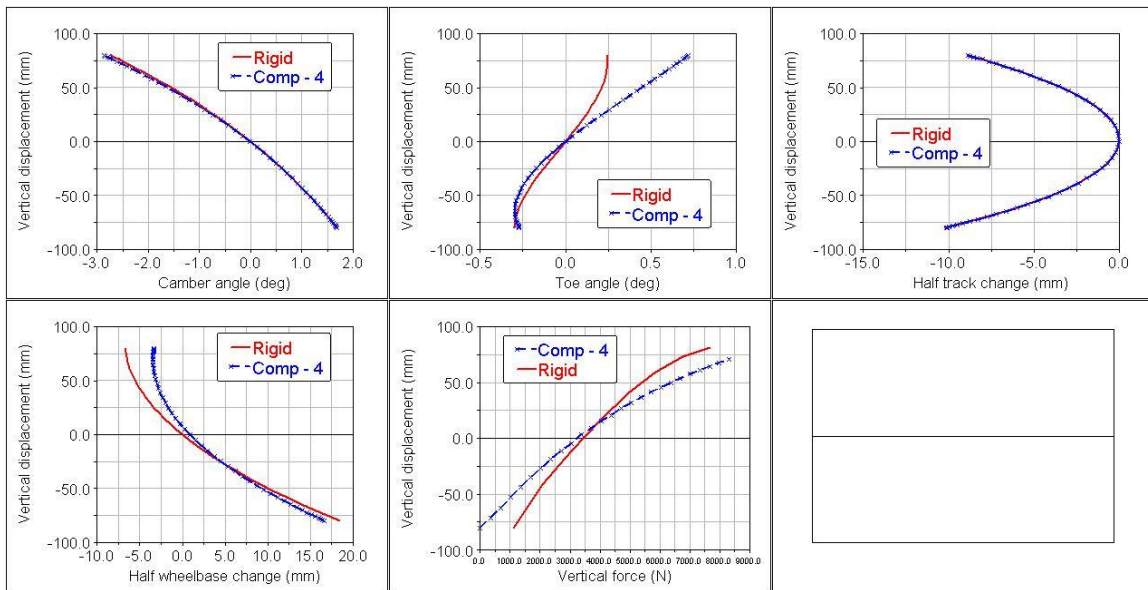


Figure 51 Compliant multi-link suspension performance characteristic with compliant 4th link

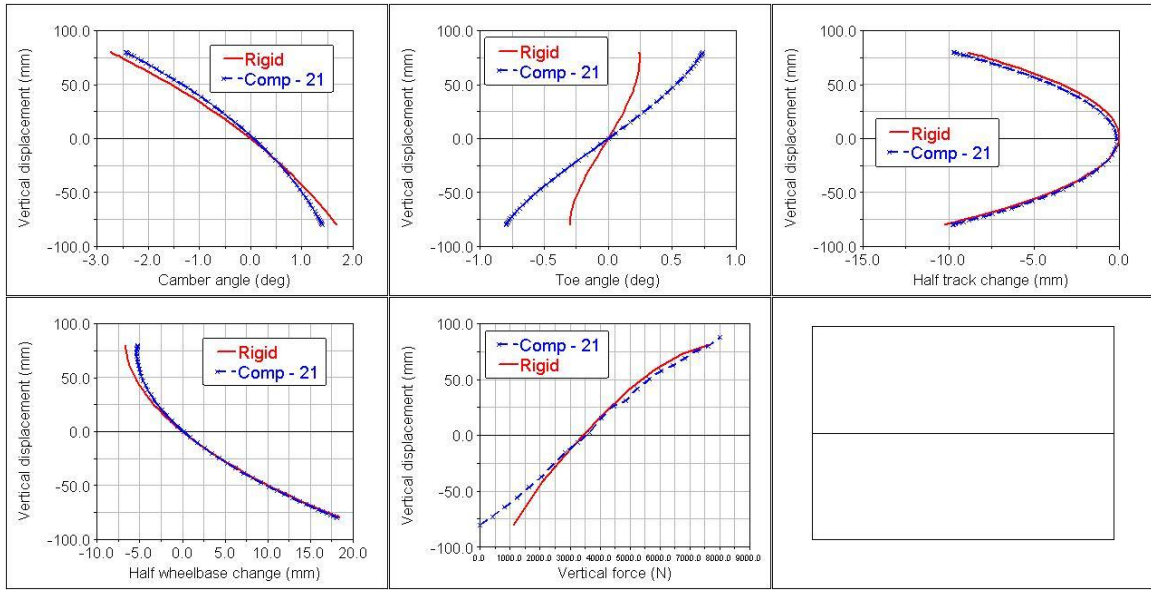


Figure 52 Compliant multi-link suspension performance characteristic with compliant 2nd and 1st link

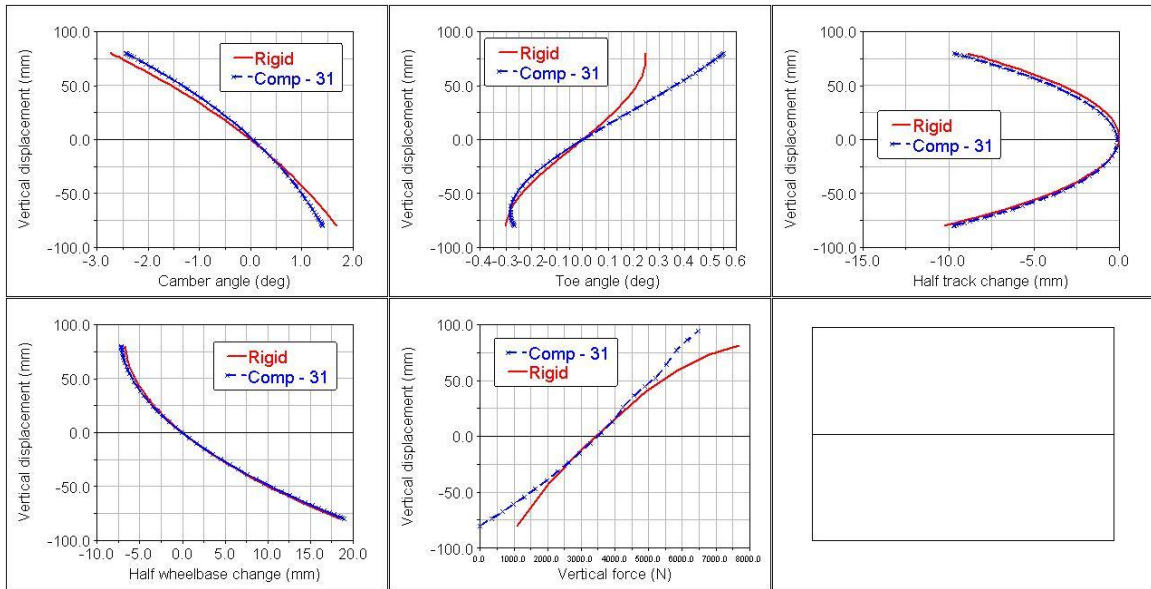


Figure 53 Compliant multi-link suspension performance characteristic with compliant 3rd and 1st link

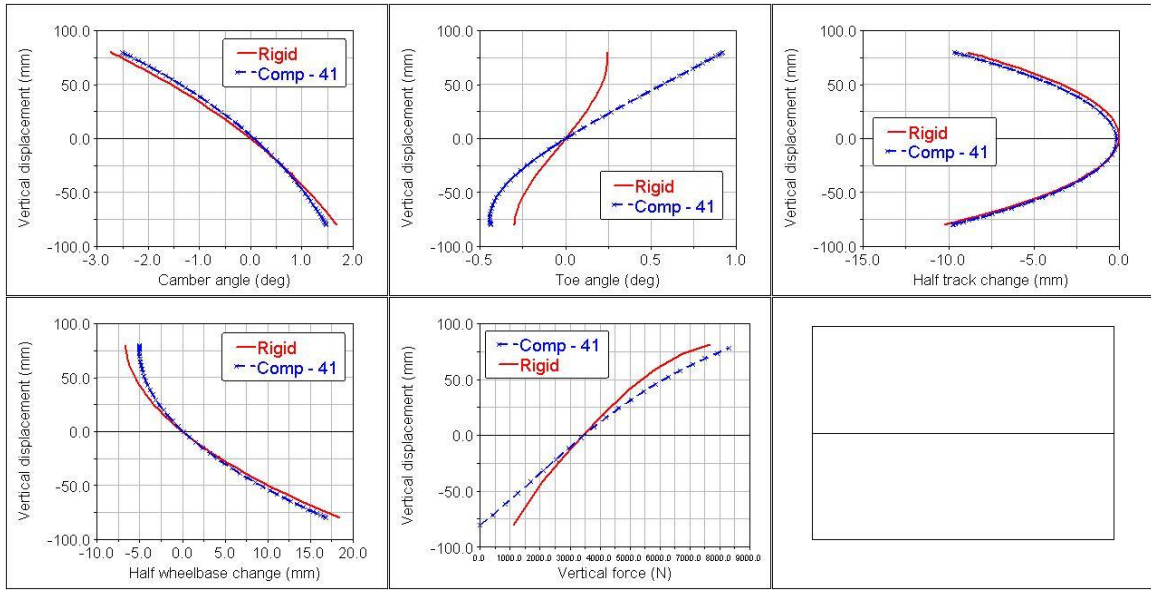


Figure 54 Compliant multi-link suspension performance characteristic with compliant 4th and 1st link

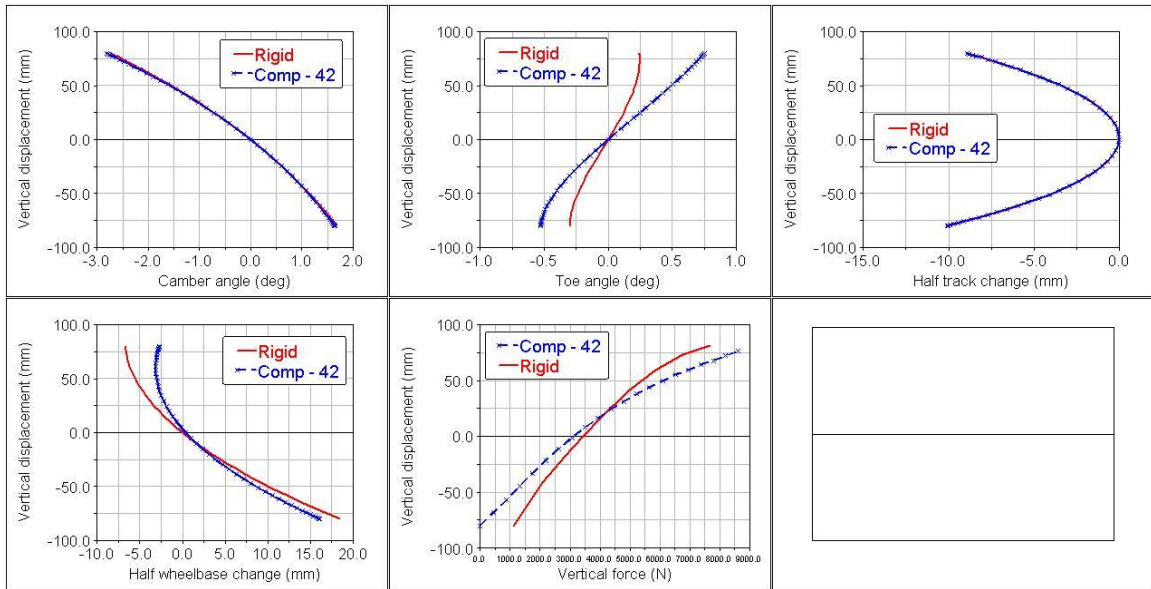


Figure 55 Compliant multi-link suspension performance characteristic with compliant 4th and 2nd link

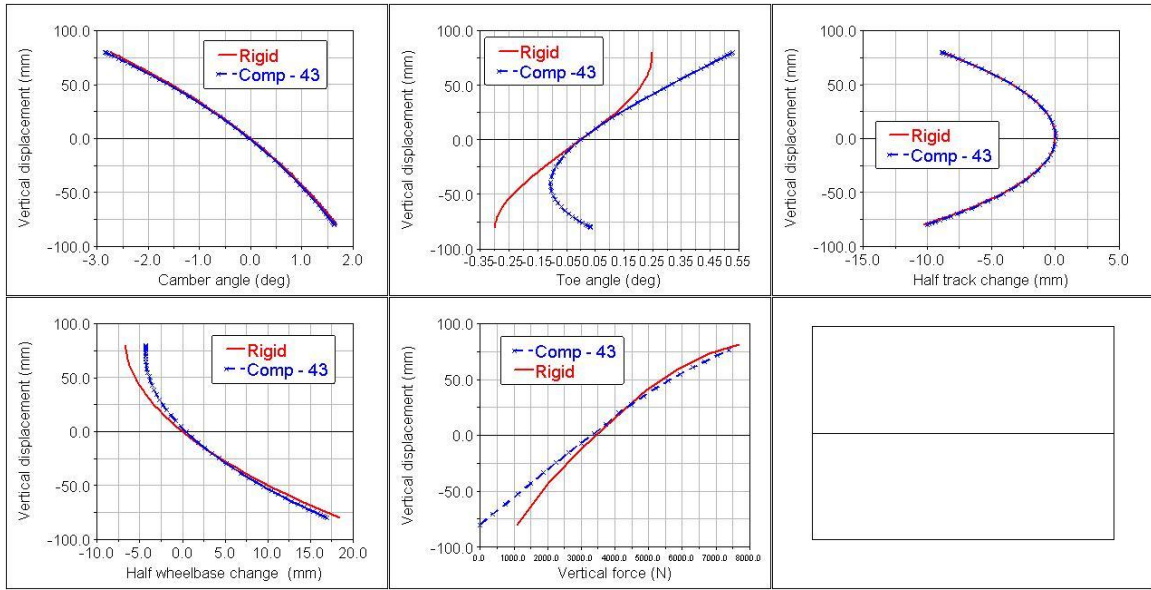


Figure 56 Compliant multi-link suspension performance characteristic with compliant 4th and 3rd link

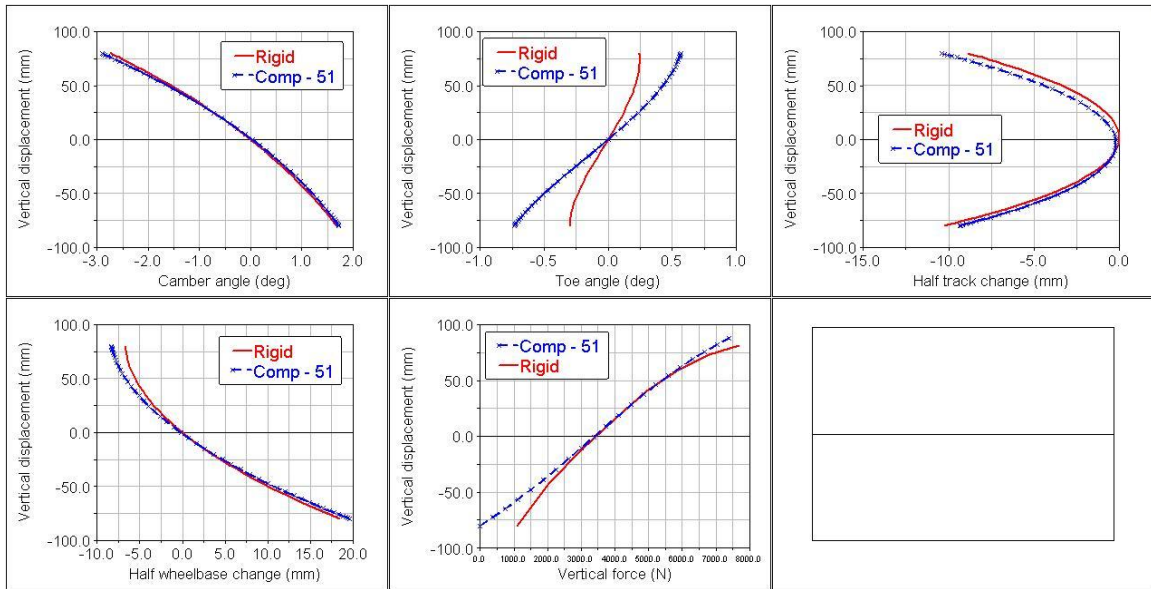


Figure 57 Compliant multi-link suspension performance characteristic with compliant 5th and 1st link

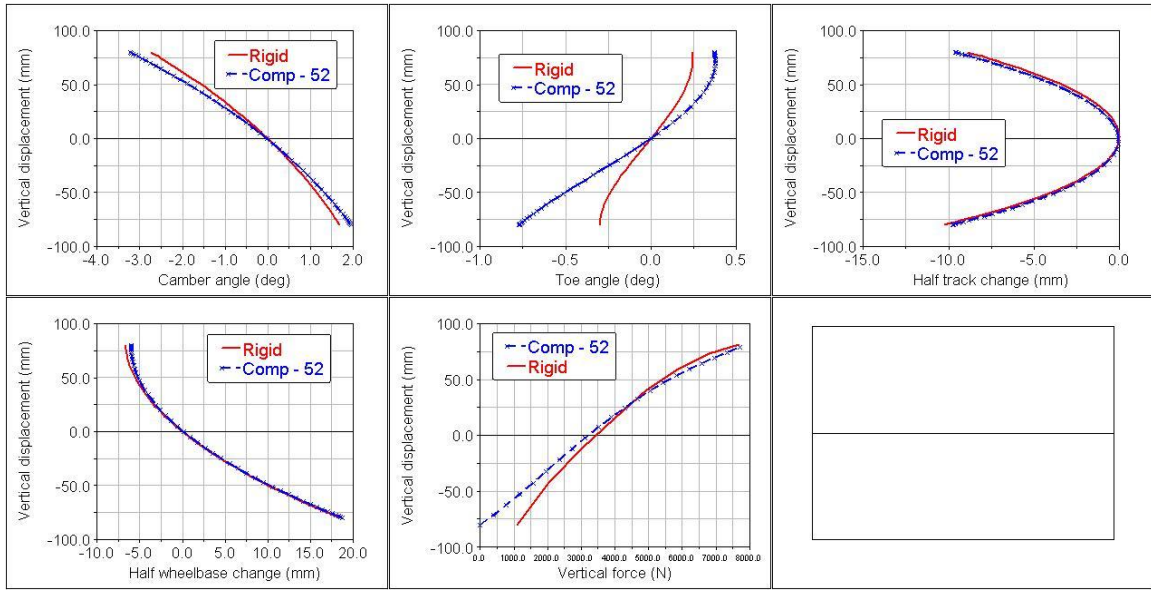


Figure 58 Compliant multi-link suspension performance characteristic with compliant 5th and 2nd link

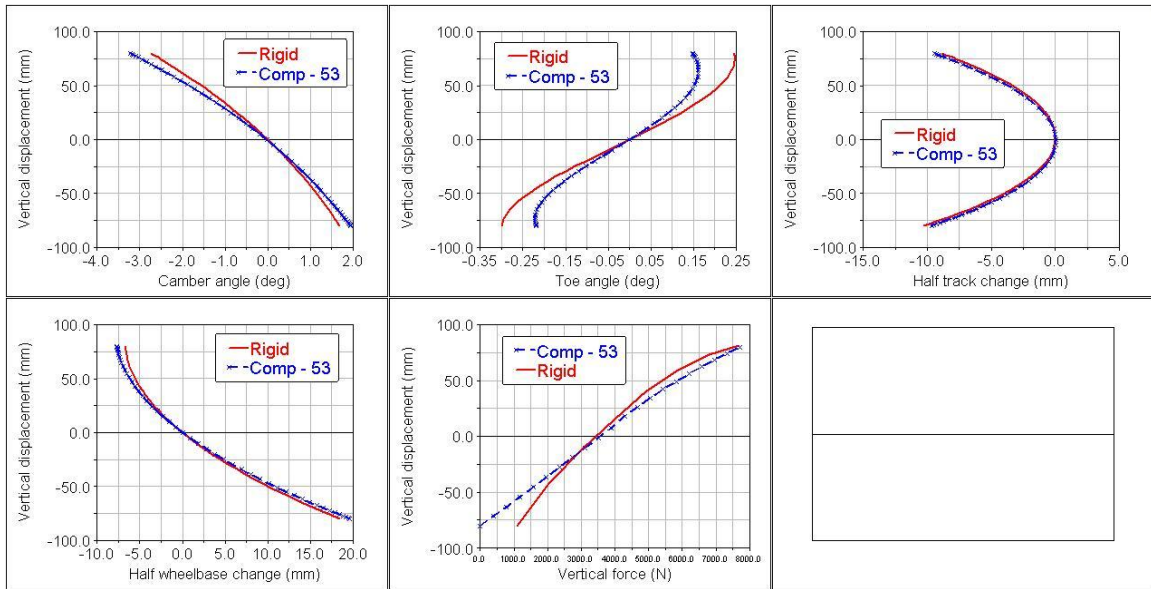


Figure 59 Compliant multi-link suspension performance characteristic with compliant 5th and 3rd link

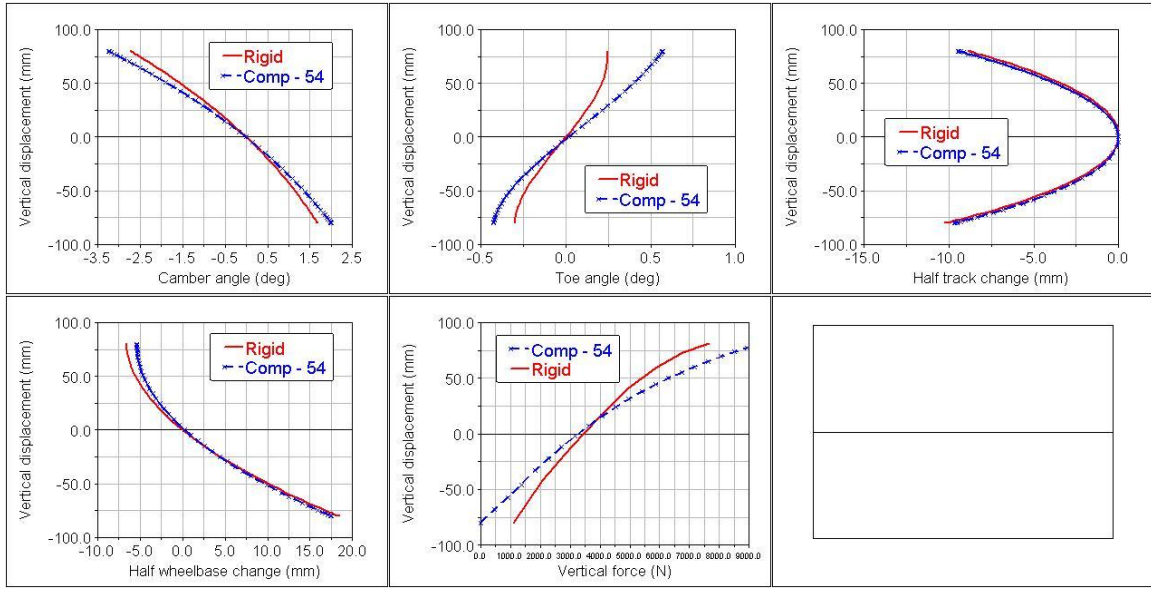


Figure 60 Compliant multi-link suspension performance characteristic with compliant 5th and 4th link

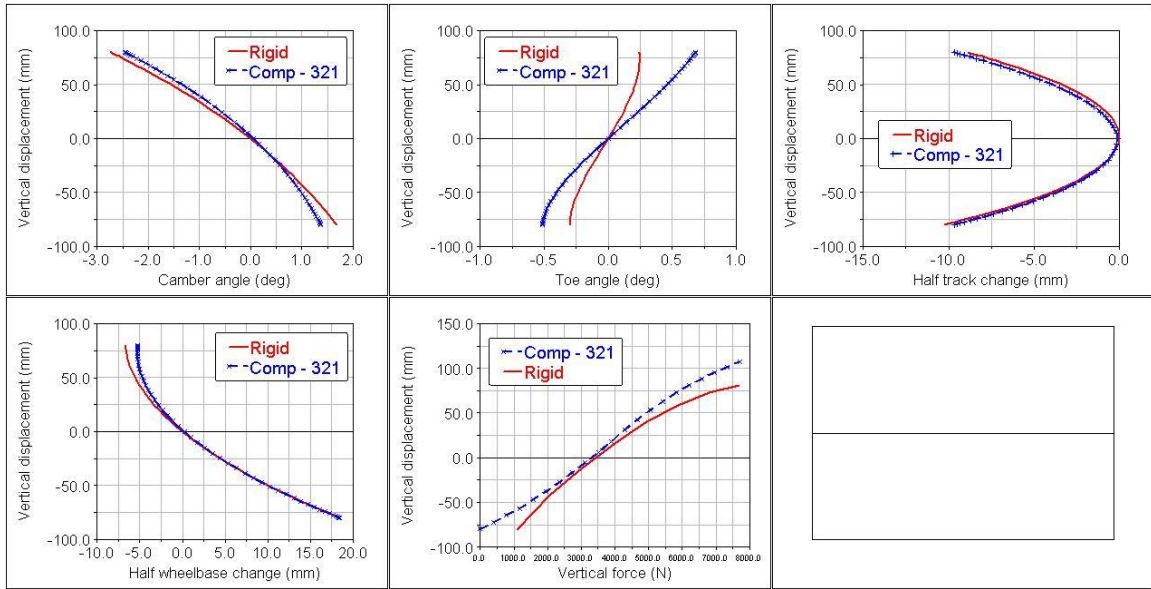


Figure 61 Compliant multi-link suspension performance characteristic with compliant 3rd, 2nd and 1st link

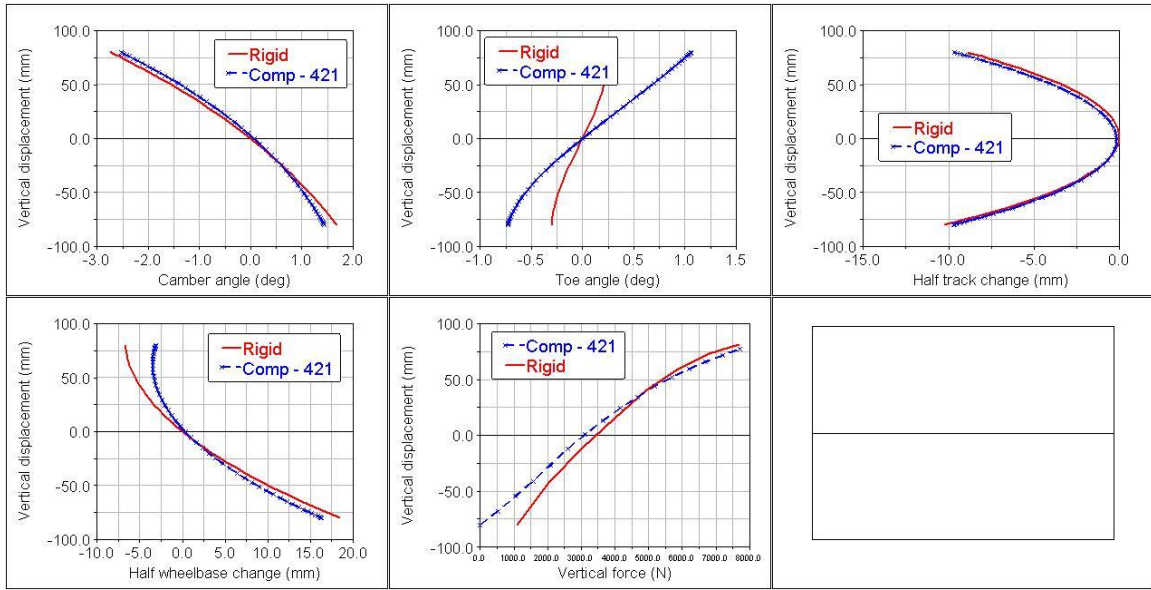


Figure 62 Compliant multi-link suspension performance characteristic with compliant 4th, 2nd and 1st link

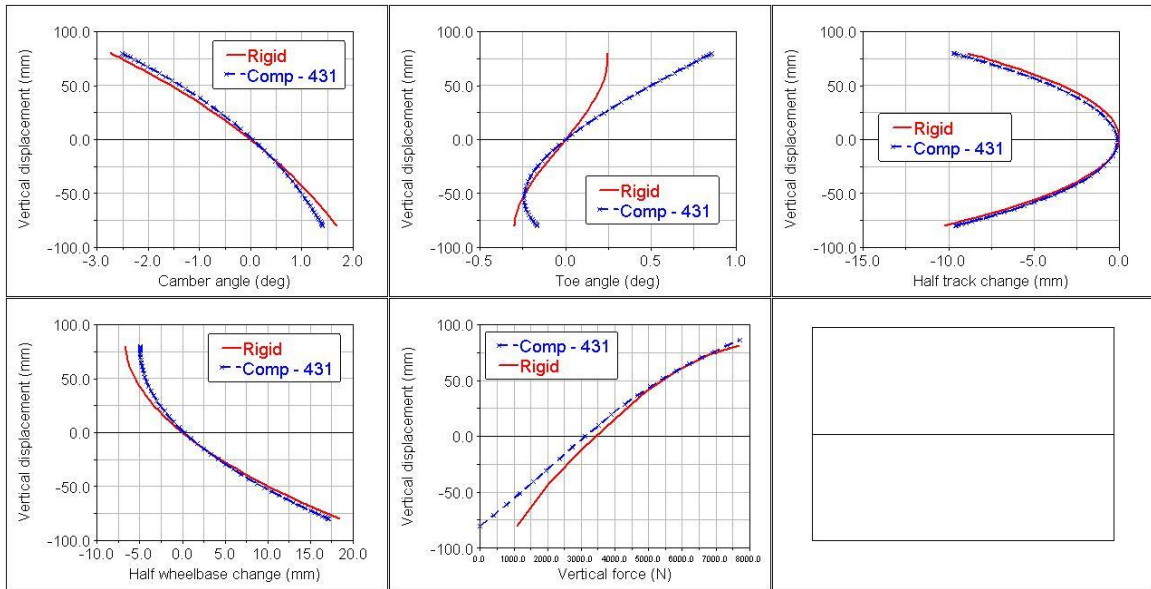


Figure 63 Compliant multi-link suspension performance characteristic with compliant 4th, 3rd and 1st link

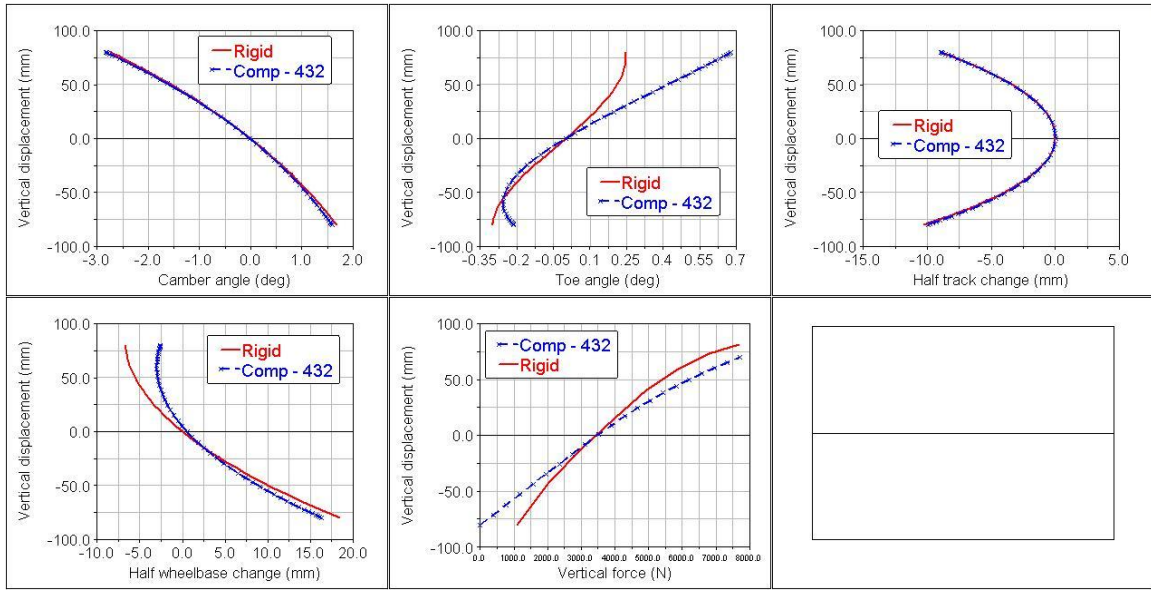


Figure 64 Compliant multi-link suspension performance characteristic with compliant 4th, 3rd and 2nd link

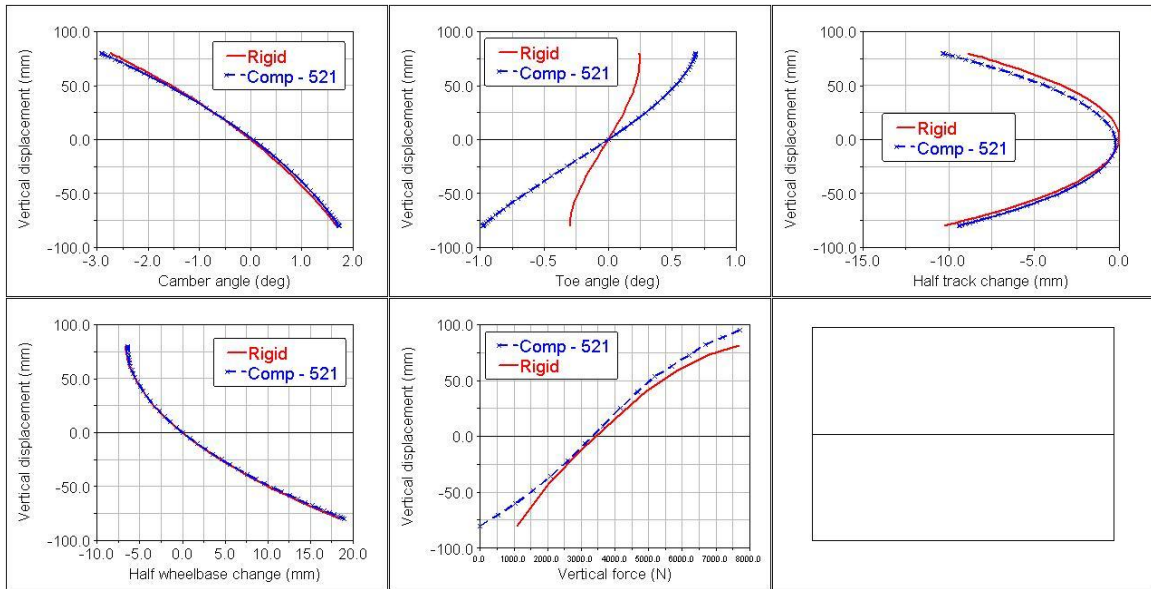


Figure 65 Compliant multi-link suspension performance characteristic with compliant 5th, 2nd and 1st link

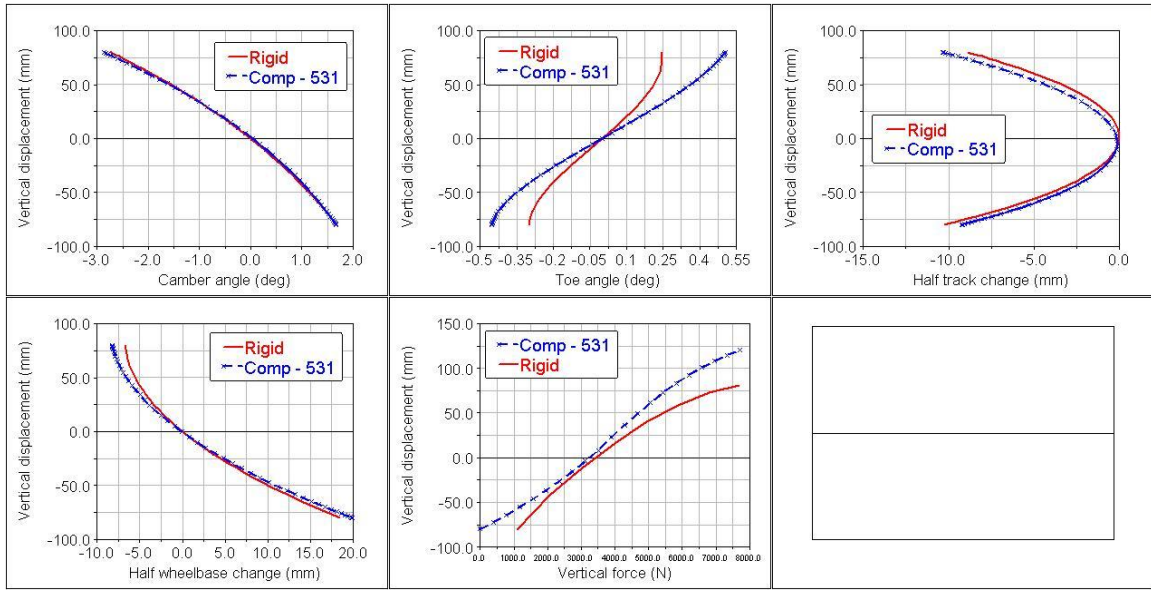


Figure 66 Compliant multi-link suspension performance characteristic with compliant 5th, 3rd and 1st link

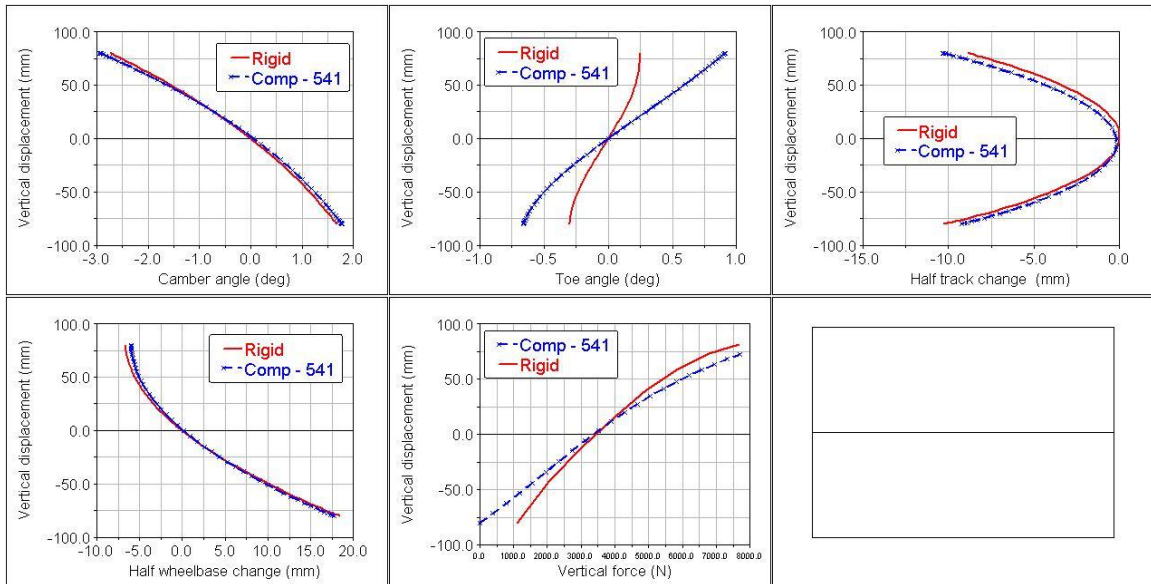


Figure 67 Compliant multi-link suspension performance characteristic with compliant 5th, 4th and 1st link

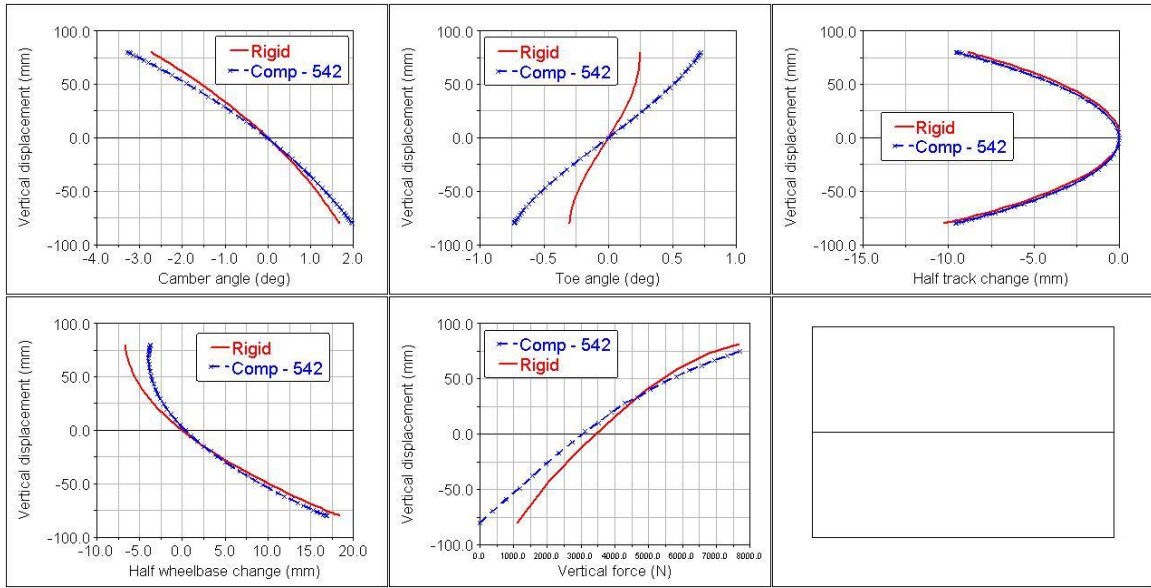


Figure 68 Compliant multi-link suspension performance characteristic with compliant 5th, 4th and 2st link

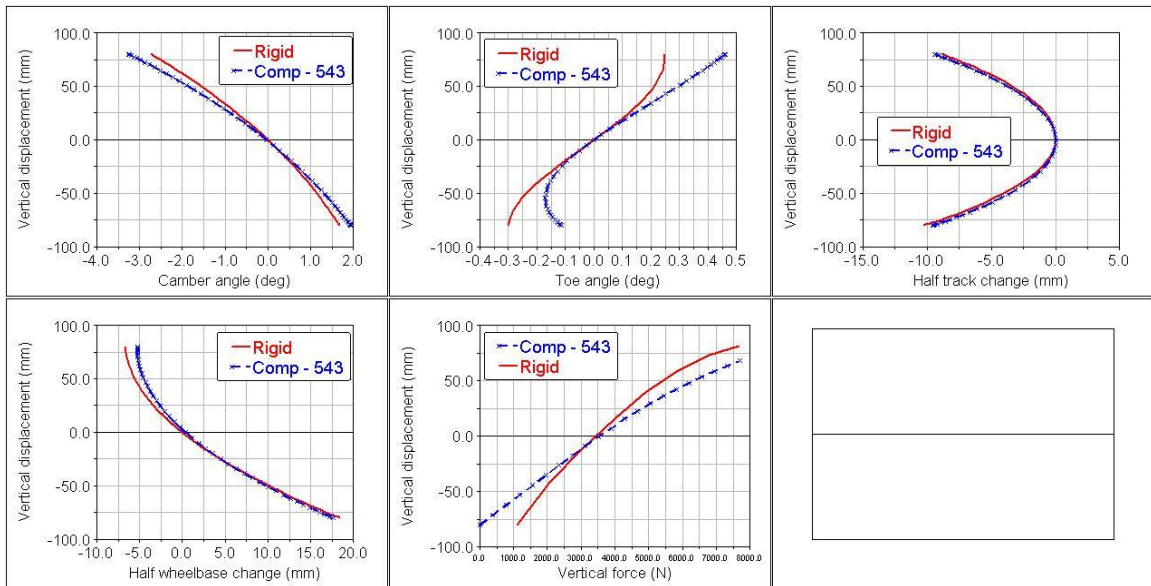


Figure 69 Compliant multi-link suspension performance characteristic with compliant 5th, 4th and 3rd link

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