STEERABILITY UNDER STRAIGHT BRAKING WITH TRUCKS AND VANS: A SIMULATION APPROACH

AUTHORS:

Jordi Arbiol, Alexandre Català

IDIADA Automotive Technology, SA

Vehicle Dynamics

Phone : + 34 977 166021

Fax : + 34 977 166036

E-mail : vehicle_dynamics@idiada.es

ABSTRACT

This paper presents a simulation approach to the control of the steerability of trucks and vans with leaf spring front suspensions.

The development of such suspensions has always needed a process for reducing the steer angles of front wheels under different driving conditions. This process has been basically based on the road and bench test results with prototypes, because of the difficulty of predicting the kinematic and compliance of the leaf spring with other methods.

The powerful simulation tools currently available for dynamics studies permits the handling of leaf spring models, with the corresponding number of degrees-of-freedom and the non-linearities of some elements.

The creation of the model, its validation with bench test data available and the final optimisation loop launched for the adjustment of the vehicle response will be presented. Therefore the two main effects involved on the steerability of the vehicle, kinematics and compliance, are widely explained and evaluated. The ultimate goal is to resolve the best co-ordinates for all the steering and knuckle joint points altogether with the optimal leaf spring settings for an optimal vehicle behaviour.

Hence, with the application of the simulation model presented, the straightability at high speed and stability under braking can be controlled and improved, and the overall development time and cost of the chassis reduced.

1. INTRODUCTION

The design and optimisation of suspensions has always been a challenging process for both test and design engineers. The big amount of parameters implied in the geometry and kinematics, the materials with its damping and stiffness, and the effect of all the components response with the total vehicle dynamic behaviour is a complex and delicate issue.

Certain suspension lay-outs present very well known drawbacks that the car manufacturer has to face for the right implementation. The optimisation loop launched once the suspension is defined can be an important time consuming task due to the construction of the prototypes and the consequent test work to be carried out.

The use of the simulation in the previous steps of the design and definition of the first geometry, is a great tool for shortening the entire development time. However, the application of such a method is not yet established for all the aspects, as the subjective assessment of the car behaviour is still a very crucial step for the definition and tuning of a new system.

One clear example of application arises with the implementation of leaf spring front suspensions, mainly used in light trucks and vans. The steerability of the vehicles with this type of suspensions is subjected to the vertical and torsional stiffness of the leaf spring, the allocation points of the suspension, the axle bending and the geometry and stiffness of the steering system.

The rather unequal interaction of the suspension and steering system under different driving conditions makes it necessary to execute a research process for the optimisation of the steerability performance.

Differences between the steering link and suspension trajectories will lead into a wheel turn versus its kingpin axis, so a toe angle variation. When this phenomena occurs in a pure vertical movement of the wheels the variable is called *bump steer*, and it receives the name *brake steer* when a brake effort is applied.

More specifically, the divergences between the *bump steer* and the *brake steer* magnitudes is a difficult concern to deal with because of the torsional movement of the leafs (*wind-up*). The use of either very effective simulation tools or complex bench and road test procedures turns out to be a necessity within the development process.

This paper is devoted to the application of the available dynamic simulation tools for the optimisation of the suspension and steering system interaction, with the aim of achieving the best possible lay-out and definition of both systems. The tool chosen for the study is the well-known multipurpose package for dynamic simulation ADAMS.

The two main effects involved on the steerability of trucks, kinematics and compliance, will be obviously considered. In this study the first prototype has been measured in a kinematics and compliance suspension rig, therefore the first steps of the project will consists in the correlation of the suspension model with the bench tests. Once the suspension is adjusted to the test values, the steering system is implemented in order to quantify the *bump steer* of the model.

The more complex '*wind-up*' phenomenon is then introduced through a brake torque input, altogether with the corresponding longitudinal load shift. All the trajectory differences of the suspension points are compared with the kinematics curves of the simple vertical movement. The optimisation loop of the suspension-steering interaction is meant to resolve the best co-ordinates for all the steering ball joint points altogether with the optimal leaf spring settings resulting from optimal *bump steer* and *brake steer* values.

The straightability at high speed and stability under braking can be controlled and minimised using the simulation model. With this study the test work and the total development time will be considerably reduced, and different possibilities and variations of the lay-outs can be easily evaluated without the need of testing new prototypes.

2. THE OBJECT OF THE STUDY AND ITS MODEL

The object of this study is a prototype of a light truck with front and rear leaf springs and rigid suspension axles. The modelled front suspension contains only one leaf in the spring. The steering system consists of a steering box with the proper connecting link to the spindle steering arm.

This apparently simple lay-out owns several particular points that have to be modelled making use of non-linear elements.

This altogether with the important deformations of the suspension axle makes the simulation modelling a quite delicate process.

The following points present the detailed description of the suspension and steering system and the way it has been simulated with ADAMS.

2.1. LEAF SPRING

Apart from being the main elastic component of the suspension, this element takes the role of the longitudinal and lateral guidance of the axle. This solution simplifies the number of components of the suspension system, however needs to be carefully designed for a proper behaviour.

The leaf spring of the suspension was modelled with ADAMS by means of the so-called beam elements. These elements create linear translational and rotational forces between two locations that define the end points of the beam. The spring subjected to the analysis of our project is a parabolic leaf spring type, so the profile of the leaf is basically a parabolic curve.

The main question that arises is the discretization required for a realistic model of the spring, while at the same time trying to minimise the number of degrees-of-freedom of the model. Figure 1 displays the different models considered.

So the first step here is to make different attempts of the same leaf spring varying the number of the beams and comparing the results obtained in different tests.

Figure 2 shows the results achieved in one of the most representative tests considered for the comparison of the leaf spring models: longitudinal effort applied to the wheel ground contact point (the vertical test was neglected as it delivers similar outputs for the models considered). Notice in Figure 2 the decreasing output variances with the higher discretization of the leaf. The leaf spring models with 42 and 80 beams give almost the same evolution of the longitudinal displacement of the wheel centre.

From the results of this first analysis it was agreed that smaller beams were needed where the variation of the leaf spring profile is higher. So the final leaf spring model presents a non-constant discretization mesh.

An in-house ADAMS routine was built in order to make the discretization of the spring from its geometry characteristics an automatic procedure. The parameters introduced were the co-ordinates, the width and thickness of every beam element and the program creates a command file that ADAMS will use for generating all the elements and links of the leaf spring.

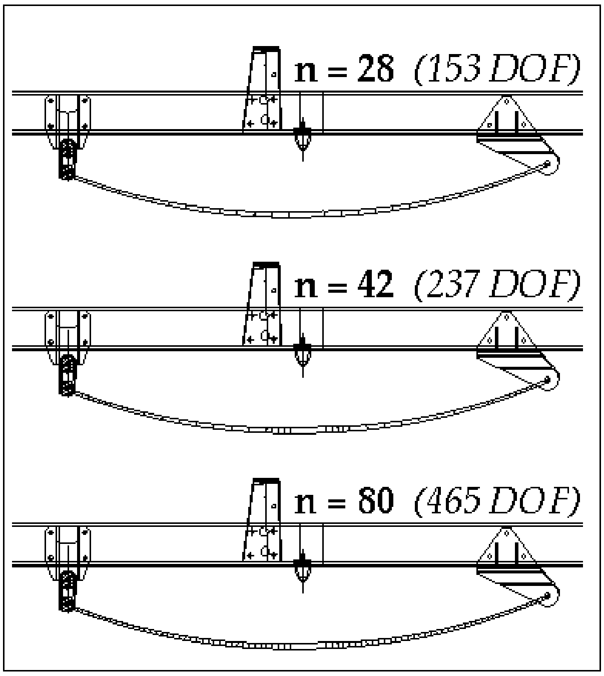

Figure 1. Leaf spring with different number of *beams*

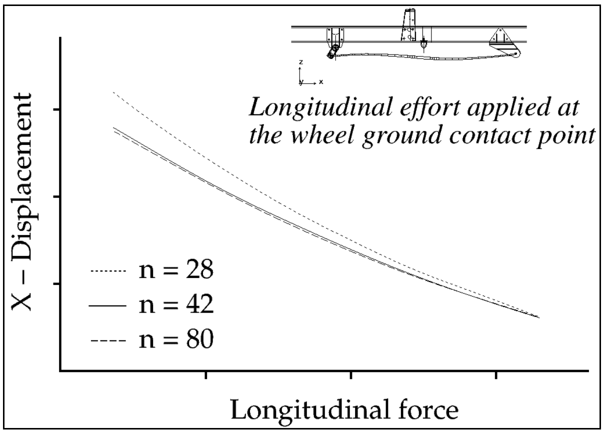

Figure 2. Test results with different discretization density of the leaf spring

2.2. SPRING BUSHES

The leaf spring forms eyes at its ends for the reception of the pivot parts. The rotational element of the front end consists of a metal bearing-bush, so it will be considered as a rigid component with the proper kinematics compliance.

The rear end point consists of two rubber bushes joined with a shackle. The two bushes of this end need to be characterized by the use of non-linear stiffness curves defined by splines or, in a more simplified model and previously checking the working zone of the bushes, assume a linear characteristic. Within this frame the main properties were the following:

- torsional rigidity (the axial torque of the bushes introduces a difference in the vertical rate of the suspension), and
- radial and axial rigidity.

2.3. SUSPENSION AXLE

The suspension axle was also modelled using beam elements. The important magnitudes of the axle deformations when braking forced the use of an accurate model with the proper stiffness.

The varying beam sections were defined by its proper inertia values. The information available from the bench test made it possible to run a very precise correlation task with this component.

The leaf spring is clamped between the seating pad and the axle by two U-bolts fixedly threaded. Due to the high efforts and torque generated at this point, it has to be carefully analysed, especially the rotational degree-of-freedom of the axle over the spring seating pad in the Z-axis of the vehicle.

The friction characteristics of the clamp joint responds to a non-linear behaviour, and it has to be accurately defined as it is one important contributing factor of the *brake steer* magnitude.

A full correlation loop will be executed as to precisely define the stiffness characteristics of this compliance.

2.5. STEERING SYSTEM AND TIRES

The steering system is another important system of the front axle that interacts with the suspension. The tests simulated were all made with the steering wheel fixed, so no movement of the steering box pit-man arm was introduced.

The efforts absorbed by the steering links are rather small so no flexible elements have to be considered.

No tire component was required as all the simulations executed in this study were only meant to analyse and evaluate the toe changes in the suspension. Hence, a simple body with linear longitudinal and vertical rigidity was used in the computations. The complete simulation model can be observed in Figure 3.

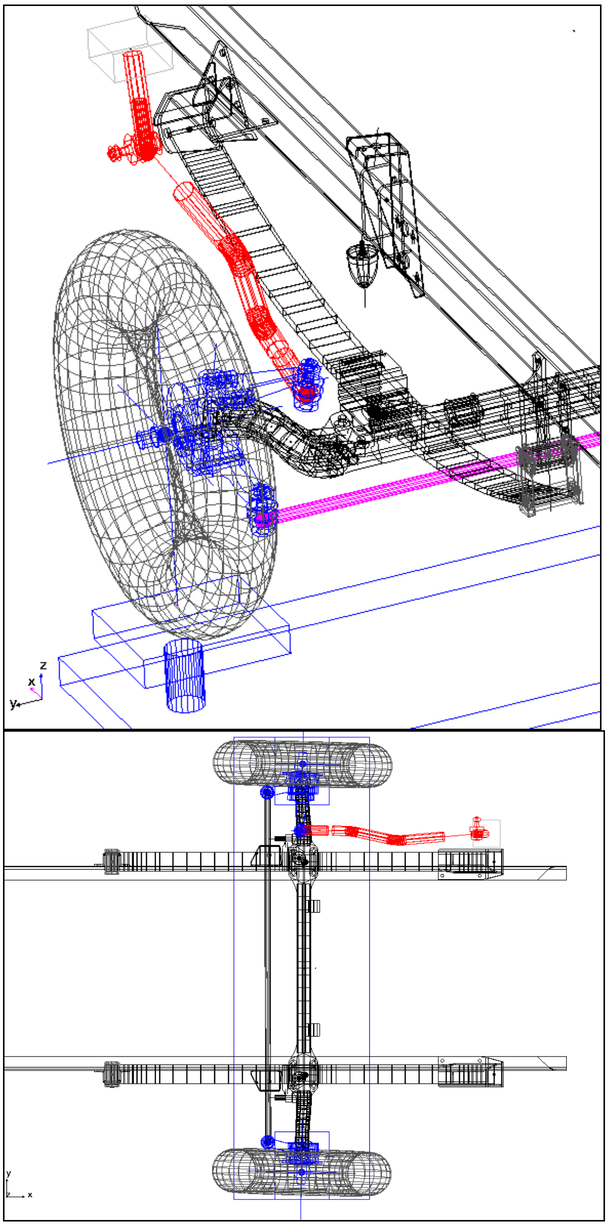

Figure 3. 3D and upper view of the ADAMS model

3. CORRELATION

Once the suspension model is completed it begins the correlation work using all the bench test results of the prototype measured. The data available was enough for assuring a rigorous validation of nearly all the parts involved in this study.

The main test data came from the kinematics and compliance test rig measures that the prototype was submitted to. Therefore the test conditions of the rig had to be reproduced in the model. Two tests were selected as the more representative and useful for the correlation:

- vertical displacement (pure bounce movement), and
- longitudinal compliance (brake effort without load shift).

Apart from these tests, the data of the longitudinal rigidity of the suspension axle were available, so a special simulation for this component was launched for its correlation.

3.1. VERTICAL DISPLACEMENT TEST

This test reproduces a pure bounce movement where no longitudinal effort is applied to the tires. In order to avoid steering wheel corrections of the driver when driving straight over bumpy roads, the toe variations in this test (*bump steer*) should be kept to minimum values.

The reason is that with this suspension the *bump steer* gives different signs at the left and right wheels, so the vehicle tends to pull whether the *bump steer* magnitude is high enough. This basically unstable situation forces the design engineer to find a solution when this parameter is set to zero.

Several graphs can be displayed, and they show the goodness of the model compared to the real prototype in this calculation.

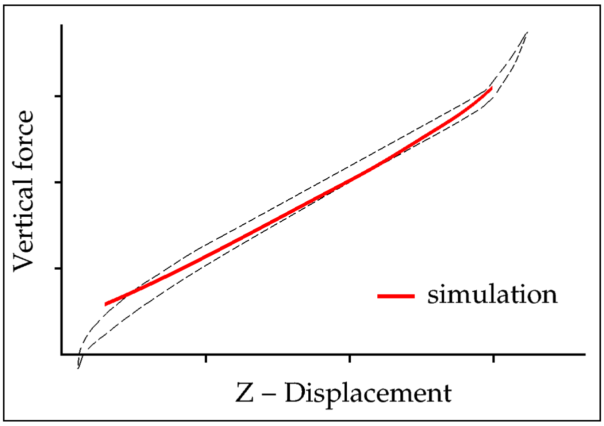

Figure 4. Suspension vertical rate

The suspension model was successfully correlated regarding the vertical stiffness (Figure 4) of the measured prototype, despite the fact that the construction tolerance of the suspension vertical rate is around 5%.

Figure 5 shows the *bump steer* evolution of one of the front wheels. The correlation of the curves highlights a good reliability of the measured prototype. Displaying the *bump steer* of the two front wheels, the differences between the left and right wheel can be seen. This can be explained by the deformation of the suspension axle, that origins a variation of the absolut distance of the side ends. As the connecting rod of the two wheels does not deflect with the axle, there is a secondary steering effect of the front-right wheel induced by the axle bending in the Z direction.

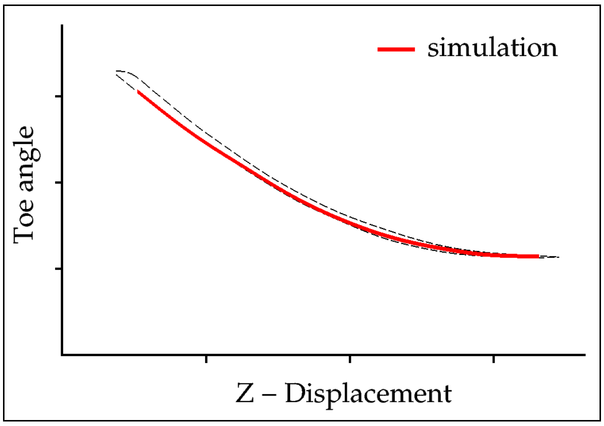

Figure 5. *Bump steer* graph

Figure 6 represents the wheel trajectory that is mainly influenced by the leaf spring deflection characteristics.

The overall correlation of the model shows a very accurate precision with the test results considering the construction tolerances. The main characteristics of the model implicated, the response of the axle and the leaf in the vertical axis will be considered in a realistic position from this point.

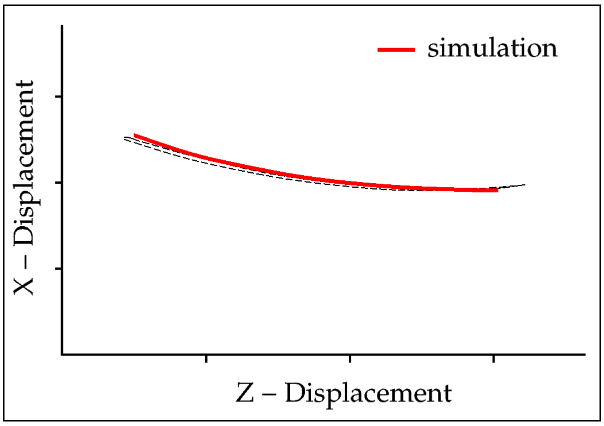

Figure 6. Wheel centre trajectory

3.2. SUSPENSION AXLE TEST

Because of the introduction of the brake longitudinal force, the suspension axle bends and twists proportionally to the axle stiffness.

Additionally the U-bolts clamped with the leaf springs tie up the elastic characteristic in the longitudinal direction and permit a degree of rotational movement. For the description of this joint two simulation restrictions were given:

- a kinematics restriction in the plane of the upper side of the axle,
- a reaction torque described by the function represented in Figure 7.

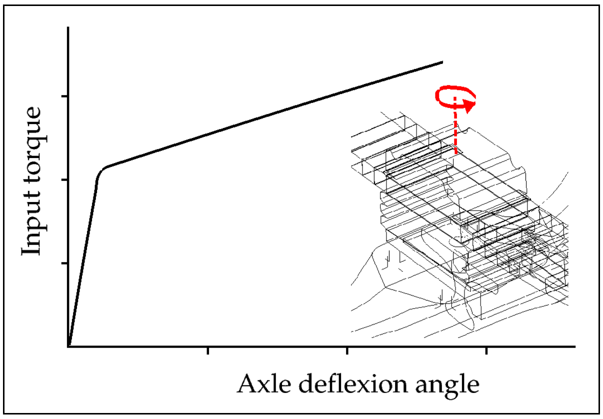

Figure 7. Spring-axle compliance characteristic

The values were computed from a specifically conceived bench test for the evaluation of this characteristic.

Note that when the input torque is under a certain magnitude, the reaction torque, which is the friction between the axle seating pad and the leaf spring, is very tight so the relative rotation between the axle and leaf springs becomes difficult.

Once the 'static friction' relieves because of the brake effort the resistance of the joint presents a softer rate. It should be noted that a high level of indetermination of the axle deflection depending on the boundaries and initial conditions of the test was found due to the big hysteresis.

3.3. LONGITUDINAL COMPLIANCE

The longitudinal compliance test reproduces the main effect when a brake torque is applied to the vehicle: the longitudinal effort. The leaf spring will act as the longitudinal guidance part for the axle and will absorb the braking torque introduced.

The torsional deflection of the leafs called *wind-up* movement, completely changes the kinematics properties of the suspension points, so the trajectory differences with and without the brake torque applied will be of great interest.

Besides, the longitudinal force will make the suspension axle bend and twist. The flexibility of the group axle-leaf joint takes here a crucial role for the total longitudinal displacement of the wheel centre point.

The first graph to evaluate is the castor rigidity, or the *wind-up* characteristic of the suspension system (Figure 8).

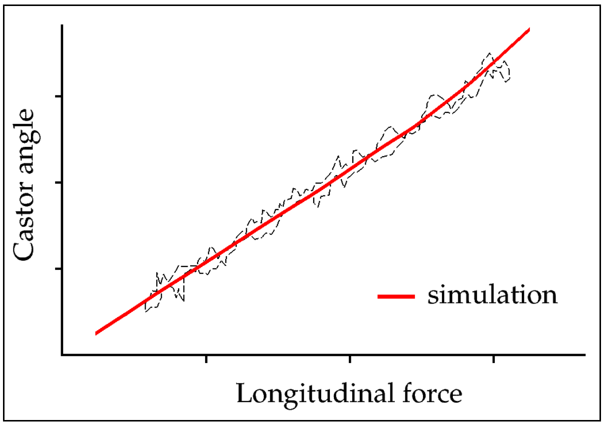

Figure 8. *Bump steer* graph

As it can be seen the adjustment of the model with the measurement evolution is rather precise. Therefore, it can be assumed that the leaf spring is correctly modelled for both vertical and torsional deflections.

Analysing the trajectory of the different points of the kingpin axis it can be seen how this axle turns around an effective centre point that shows a zero displacement. The height of this point will determine the steering arm ball joint position by means of the orthogonal projection from the kingpin axis. If the steering links fulfil this condition the kinematics characteristics of the steering-suspension will not be altered under braking conditions.

Another important point that has to be checked in this test is the different response of the system when varying the weight condition. The leaf spring deflects in the vertical direction and as a results exhibits different torsional stiffness. This deviation will lead to a compromise solution of the system, though the differences were not so important as eventually found out (see Figure 9).

It can be concluded that the overall longitudinal rigidity and the torsional deflection of the system affect the kinematics characteristic of the kingpin axis, hence it will also influence the steerability of the vehicle under braking conditions.

4. OPTIMISATION

All the analysis done till now with the correlation work will be used for the implementation of the optimisation routine.

The main objective of the optimisation is to control the *bump steer* and *brake steer* values, properly allocating the effective gyration centres of the suspension-steering system.

Additionally to the consideration of the *bump* and *brake steer* phenomenon, it has to be mentioned here that there are other lateral variables that also need a detailed study and control. Specially those related with the response of the system under lateral effort: *roll steer* and lateral compliance, so as to guarantee an optimum behaviour when turning.

As stated before the kinematics conditions have to be considered under both longitudinal and vertical wheel movements. Therefore, and with the calculations already done with the model the following steps will be followed:

1. Analyse and quantify the height of the effective rolling centre of the kingpin axis for a longitudinal effort applied at different weight conditions,
2. Re-locate the steering arm ball joint considering the information of the previous point,
3. Find the effective gyration centre point of the suspension end in the pure bounce movement,
4. Re-locate the pitman arm ball joint at the effective gyration centre point,
5. Check the evolution of the bump steer and brake steer magnitudes under real road conditions: bounce and brake with load shift,
6. Check the roll steer graph.

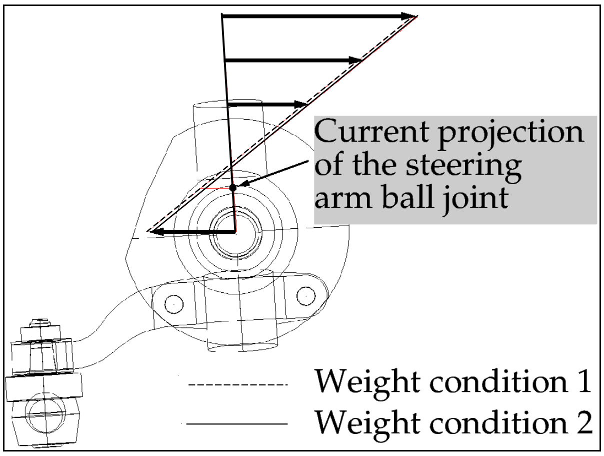

Figure 9. Kingpin axis movement under longitudinal effort and different weight conditions

The results of the first step can be observed in Figure 9. The velocity vectors of the kingpin axis points under longitudinal loads at the tire ground contact point will be plotted for two different weight conditions.

Figure 9 represents these vectors over the drawing of the spindle for a better comprehension. Notice how the neutral points for both weight conditions remain very close in the kingpin axis. This shows a similar torsional deflection of the leaf spring with different vertical deflections, and indicates that the system can be properly adjusted for a wide range of loading conditions.

The projection of the current steering arm ball joint only differs a small distance from the optimal solution. So, shifting the ball joint to the new position will be a simple task to execute in the real prototype, as well as in the simulation model.

Further in the action list, the effective gyration centre point described in step 3, as it is an instantaneous position, slightly varies along the vertical travel. However and averaging the most used vertical deflection range it is found that the point indicated in Figure 10 will be the one chosen for minimising the *bump steer*.

The big distance of this point in respect to the pit-man arm ball joint will force to implement a steering link with an intermediate ball joint located at the mentioned position.

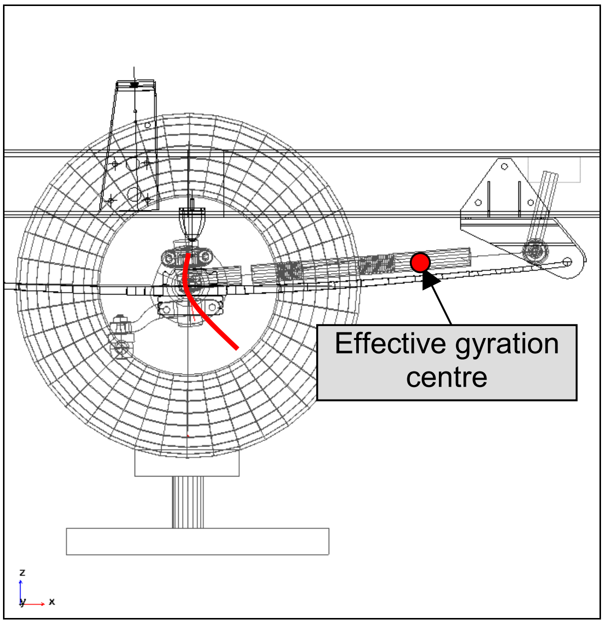

Figure 10. Trajectory and effective gyration radius of the suspension at pure bounce

With the new allocation of the steering ball joints, the front suspension can be considered as an optimised system for a proper vehicle response under wheel vertical movements and braking manoeuvres. In a further stage of the project a simplified model of the rear suspension was implemented altogether with tire components, and a rigid body element. The aim was to assess the lateral deviations of the vehicle under real driving conditions, and quantify the effect of the modifications executed during the study.

Figure 11 shows the pictures of the complete model in a calculation of a braking manoeuvre.

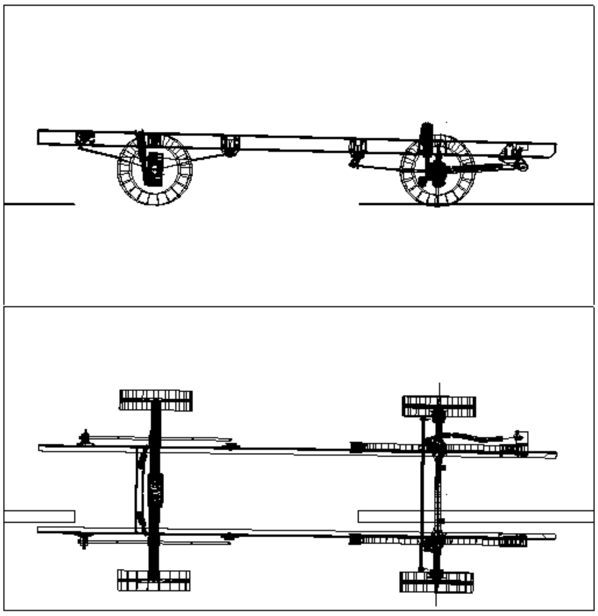

Figure 11.1. Steady state longitudinal velocity before a braking manoeuvre

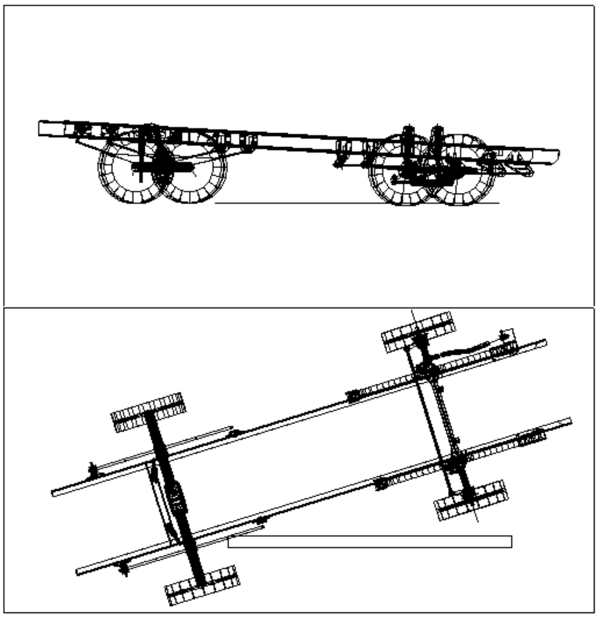

Figure 11.2. Final position of a braking manoeuvre

6. CONCLUSIONS

This paper summarises a practical and effective application of dynamic simulation in the chassis development process of a suspension.

The system object of the study owns, despite its simplicity, a delicate equilibrium. The steerability under different driving conditions has to be accurately tuned.

The modelization of leaf spring front suspension presents several non-linear elements, and results in a big number of degrees-of-freedom. Therefore, only a rigorous correlation work for validation offered the possibility to deal with the complex model within a wide margin of certainty.

The last step described in the paper demonstrates how, once the model is validated, the vehicle response can be improved by reducing the toe variations of the front wheels under different conditions. The proper adjustment loop required for the vehicle is executed with the simulation model.

Hence, it can be concluded that the simulation model, when it is appropriately validated, permits a reliable and fast improvement of the kinematics, better understanding of all the system response and effects, and the possibility to evaluate different geometries and solutions without the need of new prototypes construction.