Analysis of Multilink Suspension Characteristics due to the Rods Length Variations

Ernesto ROCCA, Francesco TIMPONE

Abstract—**This paper reports a multilink suspension analysis aimed to improve the performances in terms of car handling or comfort, by acting on the components of the suspension itself. In particular the analysis focuses on the effects produced on the suspension characteristic curves by some modifications in rods length.**

The technique can be adopted in the design phase of a multilink suspension to obtain a desired characteristic, in terms of camber or steering, by optimizing the proper length of each single rod.

Index Terms — **multilink, car suspensions, kinematical analysis**

I. INTRODUCTION

THE road vehicles need to constantly maintain contact with track while ensuring a reasonable level of comfort for passengers. Moreover, since ancient times, it is well known the importance for any type of terrestrial vehicle, both in civil and military fields, to be equipped with a suitable mechanism which allows to follow correct trajectories, in presence of particularly uneven road surfaces [1].

The above requirements can be achieved through the use of a suitable suspension system.

The use of multi-link suspensions on cars is today widely diffused allowing good performances both in terms of handling and comfort. The regulation of this two characteristics can be properly done by setting the geometrical parameters and compliances of the suspension.

In the design phase of a vehicle it would be useful to dispose of information about the possible effects on the vehicle dynamic produced by the variation of some parameters such as the suspension rod length or the bushing compliances [2,3].

The multi-link suspensions are mechanisms with one d.o.f., constituted by five rods, each one eliminating one d.o.f., connecting the wheel carrier to the car body, through joints that, in the most general case, can be considered as spherical [2,5].

The joints' compliance contributes to the ride comfort thanks to the capability of absorption of the forces deriving

from the unevenness of the road. For a correct adjustment of the compliance of these components may be also possible to assign to the suspension, under dynamic conditions, some appropriate displacements or a particular steering effect to improve the stability of the vehicle [6, 7].

The study of the mechanism, firstly, can be conducted under the hypothesis that all the elements, i.e. rods, wheel carrier and joints, are rigid.

In the present paper, with reference to a rear car suspension, an iterative procedure is described, that enables to develop kinematical and static analyses under the hypothesis of rigid rods and joints.

The procedure allows to draw the whole characteristic curves of the suspension.

The aim of the paper is to analyze the kinematical behaviour of this type of suspension by varying the lengths of the rods, in particular by varying the joints coordinates on the chassis, and keeping constant the joints positions on the wheel carrier. In this way it is possible to analyze how the length of a single link can influence a particular characteristic of the suspension.

This technique could be particularly useful in an identification procedure, during the design stage of a new suspension, starting from a desired characteristic, as example in terms of camber or steering curves, to identify the correct length of each single rod of the suspension itself.

II. KINEMATIC ANALYSIS

A. The multilink suspension model

The analysis considers a multi-link suspension (fig. 1) with 5 rods and a spring interposed between the car body (A_6) and the lower link d_1 (C_1). The system presents then 6 spherical joints on the car body and 5 on the wheel carrier. This kind of mechanism refers in particular to a vehicle rear suspension.

Considering the hypothesis of rigid links and with no clearances in the joints, this mechanism has one d.o.f.

It is therefore possible to assume the spring extension s, as the independent co-ordinate, and a frame Oxyz fixed to the car body as the base co-ordinate system.

The kinematics of this mechanism can be resolved through a vectorial method [2, 3, 4], also used in the study of parallel mechanisms with one d.o.f., of the Stewart platform type. This analogy is legitimate since the wheel carrier displacements are comparable to the displacements of a platform with respect to a base that, in the specific case of the suspension, is represented by the vehicle body.

Details about the numerical solution method are reported in Appendix.

The kinematic analysis then makes it possible to obtain the trajectory of any point of the suspension, and in particular, of the wheel centre B_0, or of the wheel-road contact centre N, in the reference fixed to the vehicle body.
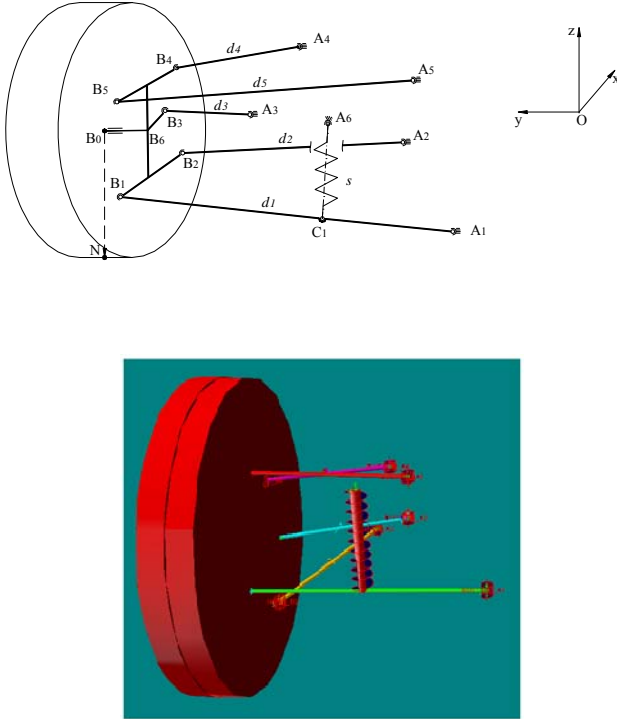

Fig.1 The multi-link suspension scheme

Moreover, the steering and the camber angles for a suspension rebound can be obtained. The numerical procedure evaluates the reactions in the links by applying in the point N a vector of external forces which resultant and moment are respectively **F** and **M**. The spring reacts so with a force: $R_{c1}=K_s \, s$.

B. The numerical analysis

The unknown value of s, that gives equilibrium at the system, can be calculated by a iterative procedure that converges through the resolution of the following vector equation system:
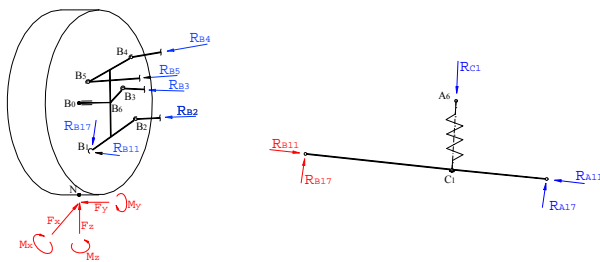

Fig.2 Forces on wheel and on link-1

$$\overline{F}_0 + R_{B11}\hat{d}_1 - R_{B17}\hat{d}_7 + R_{B2}\hat{d}_2 + R_{B3}\hat{d}_3 + R_{B4}\hat{d}_4 + R_{B5}\hat{d}_5 = 0$$
$$\overline{M}_0 + \hat{b}_{10} \wedge \overline{F}_0 + R_{B2}\hat{b}_{12} \wedge \hat{d}_2 + R_{B3}\hat{b}_{13} \wedge \hat{d}_3 + R_{B4}\hat{b}_{14} \wedge \hat{d}_4 + R_{B5}\hat{b}_{15} \wedge \hat{d}_5 = 0$$
$$(1)$$

where with R_{Bi} the reactions on the links are indicated (see fig. 2), while $\boldsymbol{F_0}, \boldsymbol{M_0}$ are the external forces in static position, and $\hat{d}_i$ e $\hat{b}_{ij}$ indicate the versors of vectors $\mathbf{d_i} = \overline{A_i B_i}$ e $\mathbf{b_{ij}} = \overline{B_i B_j}$.

The numerical code starts from the following input data:
• Co-ordinates of the A_i points, which are the joint centre situated on the chassis, in the base reference frame;
• Co-ordinates of the B_i points, which are the joint centre situated on the wheel carrier, in the reference frame fixed to the latter;
• The rods lengths d_i;
An example of the results is reported in fig. 3 where the trajectories described by the point N, i.e. the wheel-road
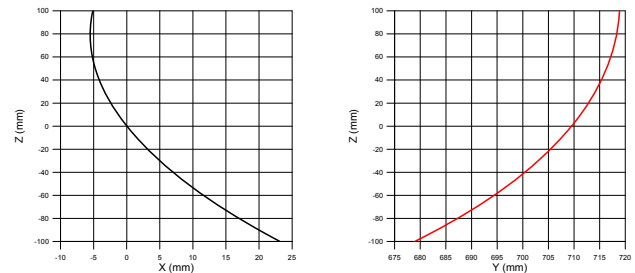

Fig. 3 Trajectories described by the N point in longitudinal Oxz (a) and transversal Oyz (b) plane

contact centre, both in the longitudinal Oxz and the transversal Oyz reference plane, for a vehicle rear suspension whose parameters are reported in the tables 1,2.

Table 1 – link coordinates in the Oxyz reference

Spring stiffness: K_s	60 (N/mm)
Spring free length: s_0	282 (mm)
Tired wheel radius: R	302 (mm)
wheel and wheel carrier total mass: m	30 (kg)

Table 2 – wheel carrier and suspension parameters

	x	y	x		x	y	z
A1	-102	205	231	**B1**	-43.	662.	208.
A2	321	496	251	**B2**	44.	632.	163.
A3	209	404	296	**B3**	141.	666.	269.
A4	209	437	402	**B4**	78.	662.	397.
A5	-5	356	416	**B5**	-5.	666.	427.
A6	-109	428	411	**B6**	-.4	605	300.5
	x	y	z				
C1	-72.5	433.5	219.5				
B0	0.	705.	302.				

The suspension model, in a previous analysis, has been numerically validated by using a multibody dynamics code (ADAMS) employed as a sort of "experimental" data source in order to test the accuracy of the theoretical code [3].

III. VARIATION OF THE ROD LENGTHS

In order to conduct an analysis about the performances of the system under investigation, when varying the lengths of the suspension rods, a numerical procedure has been implemented under the following hypotheses: no-compliances in the joints and rigid rods. In the multi-link suspension only link d1 has been kept unchanged in length and location, because it is connected to the suspension spring. The length of all the other links has been varied, in turns, to see what characteristic is more influenced, and so to see how vary the performances of the entire suspension system.

Starting from the assigned original geometry of this suspension, provided by an actual design project, the coordinates of the joints in correspondence of the chassis have been modified by increasing or decreasing the rod length of links d_2, d_3, d_4 and d_5, keeping constant their direction, in order to test the effects of such modifications on the characteristic curves. The first rod (link d_1) has been kept in the original position, how above said.

In this study the geometrical parameters related to the attachment points of the links on the chassis have been modified, without modifying the direction of the links themselves in the early configuration; so, the characteristic curves necessary to study the suspension behaviour, have been obtained. Each modified configuration of the suspension represents a mechanism cinematically and geometrically different from the previous one, but it is characterized by the same values of camber and steer angle in the nominal original configuration with respect to the early position.

In particular the numerical analysis has been conducted by varying in turn the lengths of each link, in lengthening and in shortening, of 4 cm by steps of 0.5cm.

IV. RESULTS OF THE ANALYSIS

The numerical investigation presented in the paper has been carried on by means of a code for kinematical analysis of multilink suspensions developed by the authors. In the following figures the various characteristics in terms of steer angle, camber angle, wheel base and wheel track have been reported as a function of the rod length, assuming as 0 the starting original length. The coordinates and the lengths of each single link are reported in tables (the coordinates and the relative length of the links in the early unmodified configuration are highlighted in green).

Many characteristics in terms of steer, camber, wheel base variation, semi-track, have been performed for each configuration of the modified system. In the following figure the characteristic curves of the multilink modified in the links d_2, located in the lower part of the suspension system, have been reported. Figure 4 and 5 report, as an example, the curves of steer and camber angle for the modified configurations of the rod d_2.

Tables 3 and 4 reassume the percent variations of the steer angle characteristic in rebound and in bumps showing a large sensitivity especially in the latter movement.
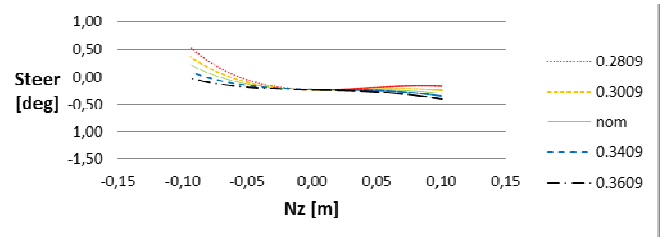

Fig.4 Steer angle values for different rod-d_2 lengths vs. N_z

Figure 6, 7 and 8 report the characteristic curves of steer angle variations for the modified configurations of the link d_3, d_4 and d_5, respectively.

lenght	δ	var. (%)
0,3609	-0,3994	34%
0,3509	-0,3755	26%
0,3409	-0,3519	18%
0,3309	-0,3254	9%
0,3209	-0,2989	0%
0,3109	-0,2707	-9%
0,3009	-0,2388	-20%
0,2909	-0,2065	-31%
0,2809	-0,1697	-43%

Table 3 – Parameters variation of the rod d_2 for rebound

lenght	δ	var. (%)
0,3609	-0,0228	-110%
0,3509	0,0326	-86%
0,3409	0,0943	-58%
0,3309	0,1578	-30%
0,3209	0,2265	0%
0,3109	0,2992	32%
0,3009	0,3748	65%
0,2909	0,4568	102%
0,2809	0,5405	139%

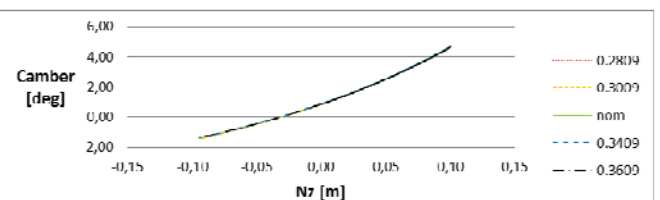Table 4 – Parameters variation of the rod d_2 for bumps

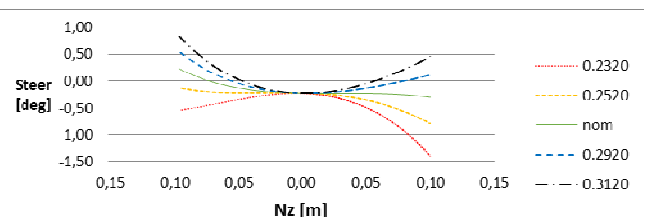Fig.5 Camber values for different rod- d_2 lengths vs. N_z

Fig.6 Steer angle values for different rod- d_3 lengths vs. N_z
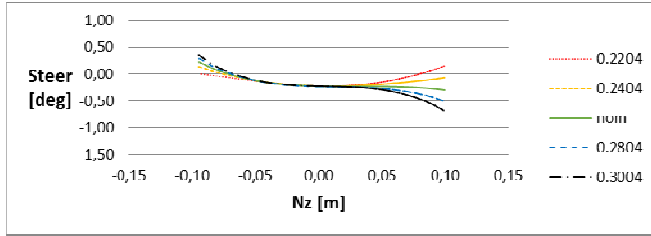
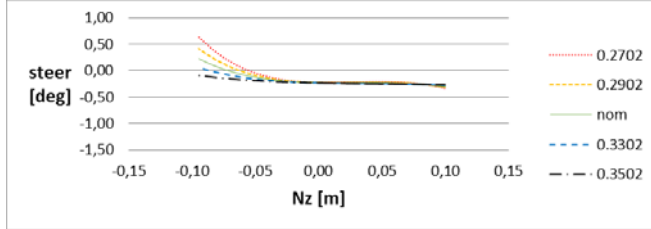Fig.7 Steer angle values for different rod- d_4 lengths vs. N_z

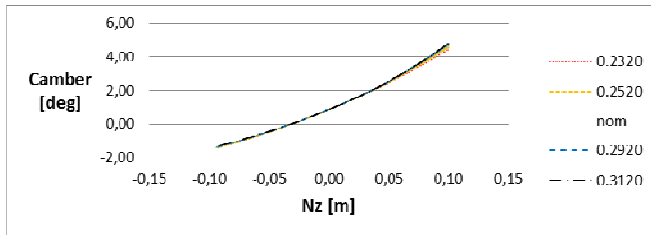Fig.8 Steer angle values for different rod- d_5 lengths vs. N_z

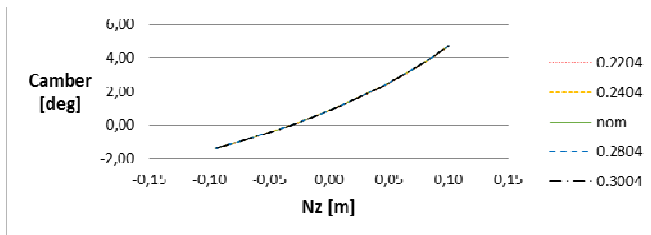Fig.9 Camber values for different rod- d_3 lengths vs. N_z

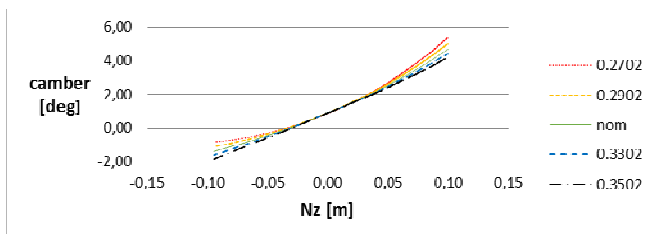Fig .10 Camber values for different rod- d_4 lengths vs. N_z

Fig.11 Camber values for different rod- d_5 lengths vs. N_z

Figure 9, 10 and 11 report, for the same rod modifications, the characteristic curves of camber angle, respectively.

How it can be noted by observing the characteristic curves for the link d_2, posed in the lower part of the suspension, the steer angle varies enough as the length of the rod increases or decreases, while the camber angle seems to be insensitive to this variations. This insensitivity is probably due to the position of the link itself which appears to be quite centered respect the wheel carrier and transversally oriented, so its position doesn't influence the camber characteristic.

As regard to the link d_3 a large variation in the steer angle is observed, both in rebound and in bumps, due to its spatial position, i.e. almost a "steer rod". The upper links, d_4 and d_5, show a trend similar of the link d_2 as regard the steering characteristic. As regard to the camber angle characteristic, it shows a sensitive variation only in the curves of the modified rod d_5. In this case the spatial position of link d_5 in

the suspension layout has great influence in the camber characteristic when the wheel carrier oscillates.

V. CONCLUSIONS

In the paper the behaviour of a multi-link suspension by varying the lengths of the rods has been analyzed, in particular by varying only the joints coordinates on the chassis, in order to understand the influence of the length of a single link on a specific characteristic of the suspension, such as the steer angle and camber.

From the kinematical analysis, conducted by using data coming from a vehicle rear multilink suspension, it has been highlighted the possibility of changing the various characteristic curves, and so the performances in terms of handling, by modifying the length of the various rods, eventually combining their effects.

Generally the position and the length of each link can influence significantly one of the particular characteristics of the suspension.

In particular, for the multilink system examined in the paper, the modifications in rods d_3 have shown a very sensitive variation in the steering characteristic. About the camber variation, the rod d_5 seems to be the more influent, due to its spatial position in the suspension layout.

The technique could be particularly useful in an identification procedure, during the design stage of a new suspension, when starting from a desired characteristic in terms of steer angle or camber, it permits to identify the correct length of each single rod of the suspension itself.

APPENDIX

As reference frame, Oxyz, it is assumed a frame fixed to car body, with axis x and y oriented respectively in the longitudinal and transversal directions (see fig.1). The Eq. (1) projected along the reference frame axes generates a system of 6 algebraic equations. In matrix notation is possible to highlight the unknown forces on the links (R_{Bi} vector).

The vector of the known terms is represented by the external forces, so:

$$\begin{bmatrix} d_{1x} & d_{2x} & d_{3x} & d_{4x} & d_{5x} & -d_{7x} \\ d_{1y} & d_{2y} & d_{3y} & d_{4y} & d_{5y} & -d_{7y} \\ d_{1z} & d_{2z} & d_{3z} & d_{4z} & d_{5z} & -d_{7z} \\ 0 & (\hat{b}_{12} \wedge \hat{d}_2)_x & (\hat{b}_{13} \wedge \hat{d}_3)_x & (\hat{b}_{14} \wedge \hat{d}_4)_x & (\hat{b}_{15} \wedge \hat{d}_5)_x & 0 \\ 0 & (\hat{b}_{12} \wedge \hat{d}_2)_y & (\hat{b}_{13} \wedge \hat{d}_3)_y & (\hat{b}_{14} \wedge \hat{d}_4)_y & (\hat{b}_{15} \wedge \hat{d}_5)_y & 0 \\ 0 & (\hat{b}_{12} \wedge \hat{d}_2)_z & (\hat{b}_{13} \wedge \hat{d}_3)_z & (\hat{b}_{14} \wedge \hat{d}_4)_z & (\hat{b}_{15} \wedge \hat{d}_5)_z & 0 \end{bmatrix} \begin{Bmatrix} R_{B11} \\ R_{B2} \\ R_{B3} \\ R_{B4} \\ R_{B5} \\ R_{B17} \end{Bmatrix} = \begin{Bmatrix} -F_{0x} \\ -F_{0y} \\ -F_{0z} \\ -(M_{0x}+b_{10}F_{0x}) \\ -(M_{0y}+b_{10}F_{0y}) \\ -(M_{0z}+b_{10}F_{0z}) \end{Bmatrix}$$

(2)

Now the equilibrium in link-1 (fig.2) is solved to obtain the reactions in the A_1 joint on the car body, and the R_{C1} spring reaction, starting from the previously calculated reactions R_{B11} and R_{B17}; indicating with the versor of $\mathbf{c_{61}} = \overline{C_1 A_6}$, and with c_{11} the modulus of $\mathbf{c_{11}} = \overline{C_1 A_1}$, it gives:

$$-R_{B11}\hat{d}_1 + R_{B17}\hat{d}_7 + R_{A11}\hat{d}_1 + R_{A17}\hat{d}_7 + R_{C1}\hat{c}_{61} = 0$$

$$-R_{B17}d_1\hat{d}_1 \wedge \hat{d}_7 + R_{C1}c_{11}\hat{d}_1 \wedge \hat{c}_{61} = 0$$

(3)

At this stage it is verified whether the spring reaction value R_m, for the assigned deformation s, matches the R_{C1} value obtained from the solution of Eq. (3).

The procedure is so iterated until the convergence is reached.

REFERENCES

[1] Cesare Rossi, Stefano Pagano - A Study on Possible Motors for Siege Towers - ASME, Journal of Mechanical Design, Vol. 133, 2011.

[2] Knapczyk, J., Dzierzek, S. - Displacement and force analysis of five-rod suspension with flexible joints – ASME, Journal of Mechanical Design, vol. 117, pp.532-538, 1995.

[3] E. Rocca ,R. Russo. A feasibility study on elastokinematic parameter identification for a multilink suspension. Journal of Automobile Engineering, 2002, Vol. 216, p. 153-160.

[4] C. Rossi, S. Savino – "Robot Trajectory Planning by Assigning Positions and Tangential Velocities", Robotics And Computer Integrated Manufacturing (2012) doi:10.1016/j.rcim.2012.04.003, vol.29, issue 1, February 2013, pp 139-156, ISSN 0736-5845.

[5] Knapczyk, J., Dzierzek, S. - Displacement analysis of five-rod wheel guiding mechanism by using Vector Method – Proc. Of 25th ISATA, Florence, v. Mechatronics, p. 920893, pp. 687-695.

[6] Lee D.M.A., Pascoe, D.M. ElMaraghy W.H. – An analysis of the multi-link independent suspension system – 1993, Int. J. of Vehicle design, vol. 14, n°1, pp.44-58.

[7] Bastow, D. – Car suspension and handling - 1987, Pentech Press Ltd.