OPTIMAL KINEMATIC DESIGN OF A CAR AXLE GUIDING MECHANISM IN MBS SOFTWARE ENVIRONMENT

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Abstract: *This work deals with the optimal kinematic design of the guiding mechanism used for the rear axle of a motor vehicle. A guiding mechanism of type 2S1C (S - sphere, C - circle) is taken into consideration, the optimization study being conducted by using the multi-body systems (MBS) software environment ADAMS of MSC. The global cartesian coordinates of some design points (the locations of the joints between the guiding arms and car body) are used as independent design variables for the optimization study. The main design objectives refer to the longitudinal displacement of the axle and its proper rotation (around the transversal axis), the kinematic optimization goal being to minimize these variations.*

Keywords: axle guiding mechanism, kinematics, multi-body system, optimal design.

1. Introduction

For the guidance of the rear axle of the motor vehicles, two solutions are used: independent guidance of the wheels, case in which each wheel is guided by its own linkage, and dependent guidance of the wheels, case in which the rear axle is guided relative to car body [1]. The dependent guidance of the rear axle, which is frequently used for the off-road and commercial vehicles, is assured by spatial linkage mechanisms, on which between axle and car body a number of binary links or kinematic chains are interposed. The links connections to axle and car body are made through compliant joints (bushings) with 6 elastic restricted degrees of freedom. Usually, for the kinematics of the axle guiding linkages, the bushings are modeled as spherical joints, neglecting the linear deformations. At the same time, the car body is attached to ground. In this way, the axle guiding linkages have low degree of mobility, m=1 or m=2.

The guidance of the axle is made by driving a number of its points on suitably chosen surfaces and curves (fig. 1): sphere (s), circle (c), and coupler curve (cc). The guidance on sphere (fig. 1,a), respectively circle (fig. 1,b), is achieved by a binary link, interposed between axle and car body, with spherical joints in both ends, respectively a rotational joint to car body and a spherical joint to axle. The guidance on coupler curve (fig. 1,c) is performed by a spherical joint between axle and coupler; in this case, watt mechanism configuration is frequently used, but roberts, chebyshev or evans straight-line linkages can also be used.

Joining in parallel the basic chains, various axle guiding linkages with m=1 and m=2 can be obtained. The structural systematization of the guiding linkages was presented in [3], taking into account the type of joints (spherical and/or revolute), the type of kinematic chains, and the number of kinematic chains connected in parallel. The axle guiding linkage by three points (so called 2s1c) is obtained by joining three binary links of type 2[a] + 1[b], in compliance with figure 1. The mechanism contains two lower rods (1_s, 1_d) and one upper triangular arm (3), which are longitudinal disposed (fig. 2). The connections of the upper arm to car body can be disposed in front (fig. 2,a) or in rear (fig. 2,b) relative to axle. The two spherical joints between the upper arm and car body determine, in fact, a revolute joint.

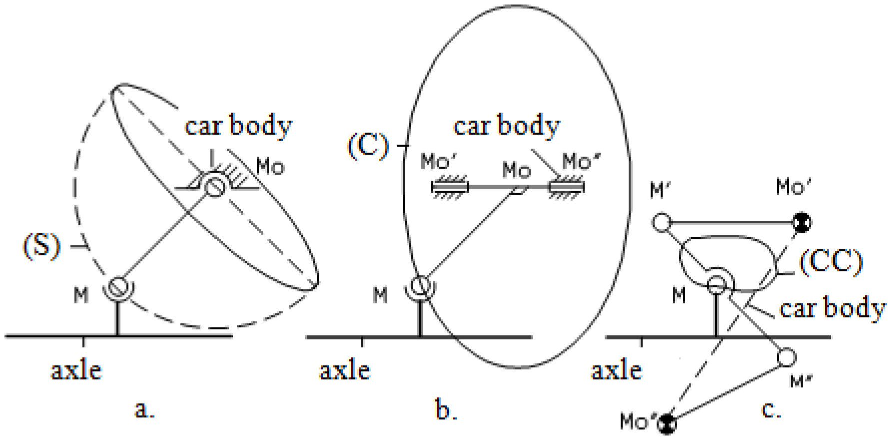


Fig. 1. The basic types of guidance of the axle points.

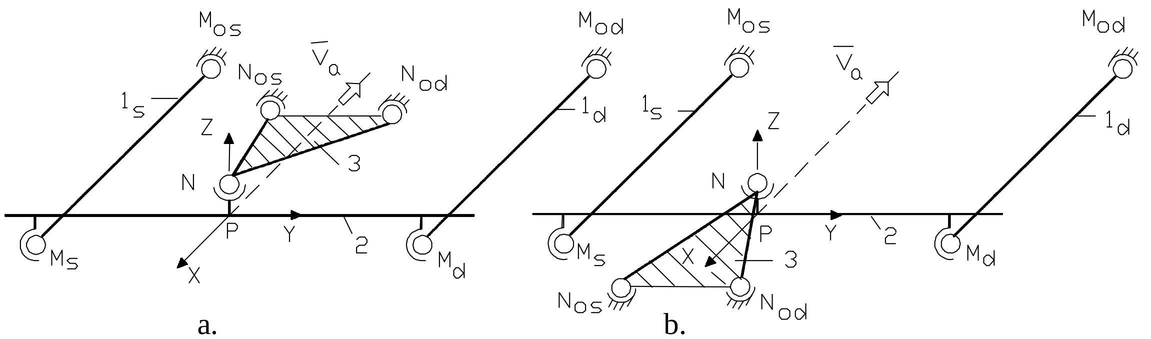


Fig. 2. The axle guiding mechanism by three points (2s1c).

Relative to car body, the rear axle must have the possibility of vertical motion and rotation around the longitudinal axis of the car. The modification of the vertical position of the wheels (when the vehicle passes over bumps) determines, besides the above-described necessary motions of the rear axle, secondary undesirable motions, as follows:

- displacements of the axle center P along the longitudinal (ΔX_P) and transversal (ΔY_P) directions;
- rotations of the axle around the vertical (η_Z) and transversal axes (proper rotation, η_Y).

The minimization of the undesirable motions can be transposed into kinematic optimization criteria, as follows: $\Delta X_P \to 0$, $\Delta Y_P \to 0$, $\eta_Z \to 0$, $\eta_Y \to 0$. These criteria cannot be equally satisfied, and for this reason a certain criterion has priority, or, usually, a compromise is accepted, such as: $\Delta X_P \in [\Delta X_{P\ min}, \Delta X_{P\ max}]$, $\Delta Y_P \in [\Delta Y_{P\ min}, \Delta Y_{P\ max}]$, $\eta_Z \in [\eta_{Z\ min}, \eta_{Z\ max}]$, $\eta_Y \in [\eta_{Y\ min}, \eta_{Y\ max}]$. The boundaries can be established depending on the top speed of the car, type of tires, type of car, and other criteria.

2. The kinematic model of the axle guiding linkage

ADAMS of MSC.Software is a powerful modeling and simulation environment that provide build, simulate and refine models of mechanical system. In ADAMS, the kinematic model of a mechanical system is characterized as a constrained, multi-body system, in which the parts are connected through geometric and kinematic constraints (joints and motion generators) [4, 5].

The mechanism is modeled and analyzed in a global coordinate system (GCS), which it is an inertial system. The ground body acts as the global coordinate system that defines the global origin and axes about which the model is created. For any mobile body (guiding arms, axle), a local coordinate system (LCS) is assigned, which moves with the body and its original position defaults to that of the global coordinate system. It must be mentioned that - for the kinematic model - the car body is attached to ground.

The 2S1C guiding mechanism contains 4 mobile parts (lower guiding arms - $1_{s/d}$, upper arm - 3, axle - 2, see the notations in figure 2) that are connected through 6 geometric constraints, and 2 kinematic constraints that control the vertical displacements of the wheels. For modeling the vertical motion of the left/right wheels, there are used motion generators, which dictate the part (wheel/axle) motion as a function of time. Having in view that the independent kinematic parameters are the vertical positions of the centers of the left/right wheels, the motion generators have been modeled as single point motions (fig. 3), which prescribe the motion of the mobile parts (left/right wheels) relative to the ground body along the vertical axis (Z). Each motion generator removes one degree of freedom (DOF).

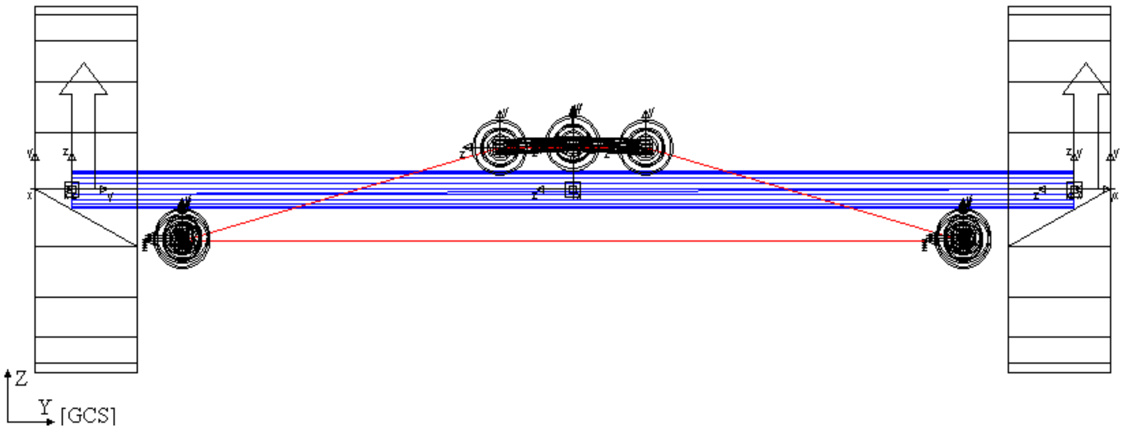


Fig. 3. The motion generators that drive the kinematic model.

The total number of degrees of freedom, which represents the number of undetermined motions (generalized coordinates), is equal to the difference between the number of allowed part motions and the number of constraints (Gruebler count), DOF = $6 \cdot n - \Sigma r$, as follows:

- generalized coordinates for 4 mobile bodies: $4 \times 6 = 24$;
- degrees of freedom restricted by the geometric constraints (joints):
  - spherical joints between axle and lower/upper arms (M_s, M_d, N): $3 \times (-3) = -9$,

- spherical joints between car body (ground) and lower arms (M_{0s}, M_{0d}): $2 \times (-3) = -6$,
- rotational joint between car body and upper control arm (N_0): $1 \times (-5) = -5$,

- degrees of freedom controlled by the kinematic constraints:
  - motion generators (point motions): $2 \times (-1) = -2$,

resulting DOF = 24 - 22 = 2, meaning the two passive proper rotations of the lower guiding arms (namely 1_s and 1_d - see fig. 2). These passive degrees of freedom can be eliminated by applying motion generators in one of the corresponding spherical joints that are used to connect the guiding arms (for example, in M_s and M_d - see fig. 2), obtaining in this way DOF = 0; therefore the model is kinematically determined.

Given the aim of the paper (the kinematic optimization of the axle guiding linkage), the mechanism has been modeled by using parameterization tools. Parameterizing the mechanism simplifies changes to model because it helps to automatically size, relocate and orient the design objects (e.g. bodies, joints). In this way, relationships into the model are created, so that when a modeling object is changed, ADAMS updates any other objects that depend on it.

For the guiding linkage in study, the design points are the easiest way to parameterize the model. Using design points, important locations can be specified in the model, and then other objects (e.g. joints) can be attached to these points. According to figure 4, in the 2S1C guiding mechanism there are the following design points: M_{0s}, M_{0d} - the locations of the joints between the lower guiding arms and car body; N_{0s}, N_{0d} - the points that define the axis of the revolute joint between the upper control arm and car body; M_s, M_d, N - the locations of the joints between the lower & upper guiding arms and axle; G_s, G_d - the centers of the wheels.

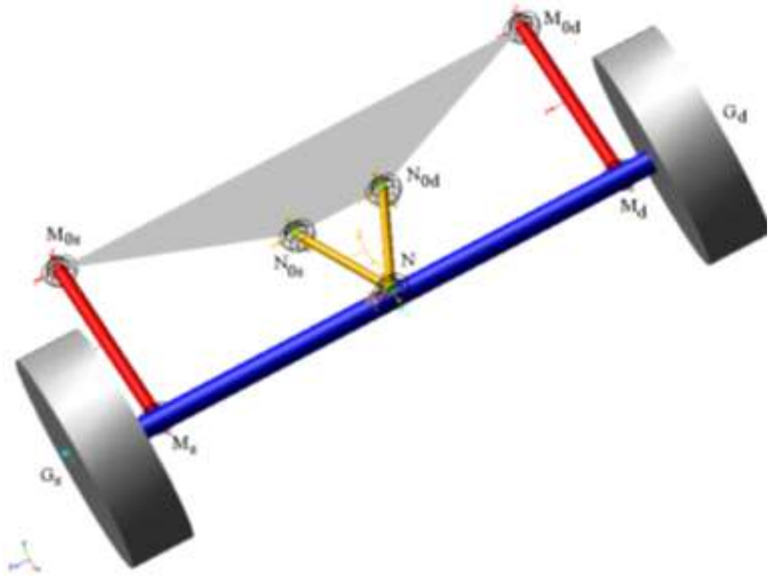


Fig. 4. The virtual model of the 2S1C guiding linkage.

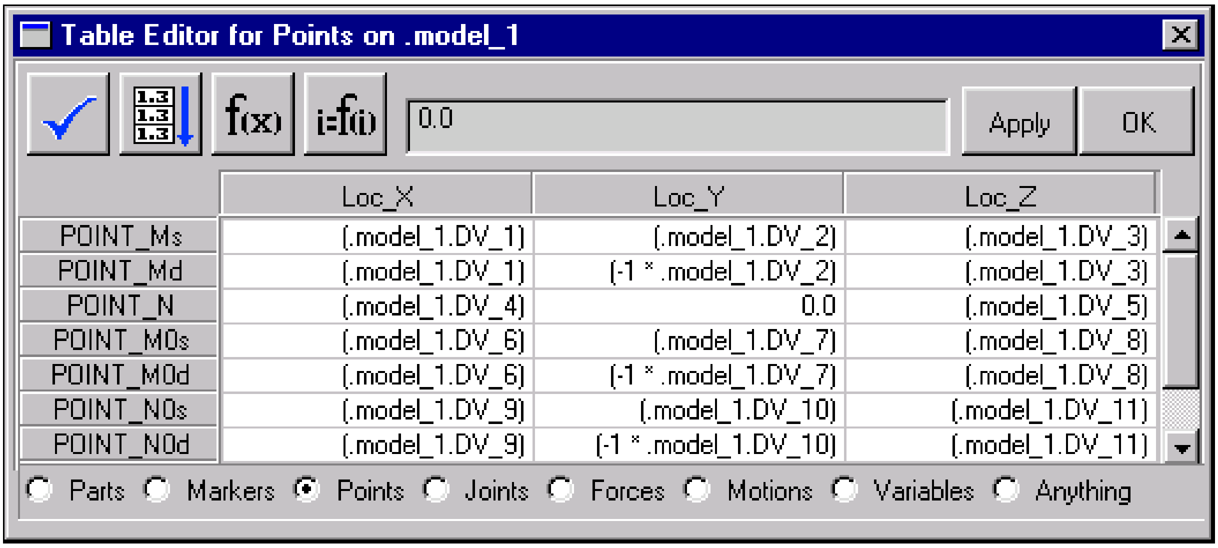
The geometries of the bodies have been attached to these points, as follows: the axle - between G_s and G_d; the left lower arm - M_{0s} and M_S; the right lower arm - M_{0d} and M_d; the upper control arm - N_{0s} and N, respectively N_{0d} and N. The left/right wheels are fixed connected / attached to axle.

3. Results and conclusions

To optimize a mechanical system, the following steps are necessary [2]: defining the design variables, defining the design objectives for optimization, performing design studies, optimizing the model on the basis of the main design variables. Design variables allow creating independent parameters and tie modeling objects to them. Design study describes the ability to select a design variable, sweep that variable through a range of values and then simulate the motion behavior of the various designs in order to understand the sensitivity of the overall system to these design variations. Before running the design study, the range (list) of values for each design variable must be specified. Design optimization represents the capability to define design objectives, constraints and variables, and then have the software iterate automatically to the optimally - performing configuration.

In the first step, the coordinates of the points that parameterize the model (shown in fig. 4) have been transformed in design variables (DV): $X_{Ms} \rightarrow$ DV_1, $Y_{Ms} \rightarrow$ DV_2, $Z_{Ms} \rightarrow$ DV_3 and so on. Having in view the symmetry of the mechanism relative to the longitudinal axis of the car, between the points that define the topological scheme of the mechanism the following expressions have been modeled: $X_{M0d} = X_{M0s}$, $Y_{M0d} = -Y_{M0s}$, $Z_{M0d} = Z_{M0s}$, $X_{N0d} = X_{N0s}$, $Y_{N0d} = -Y_{N0s}$, $Z_{N0d} = Z_{N0s}$, $X_{Md} = X_{Ms}$, $Y_{Md} = -Y_{Ms}$, $Z_{Md} = Z_{Ms}$, $Y_N = 0$. When an expression is created, ADAMS stores the expression and updates the value whenever a value in the expression changes. In these terms, the design variables that control the axle guiding linkage during the optimization are presented in figure 5.

It must be mentioned that for this paper only the coordinates of the joints on car body ($M_{0s/d}$, $N_{0s/d}$) have been chosen as design variables for the optimization (in other words, DV_6 $\rightarrow$ DV_11 are the selected design variables) the locations of the joints on axle ($M_{s/d}$, N) remaining established by constructive criteria.



	Loc_X	Loc_Y	Loc_Z
POINT_Ms	(.model_1.DV_1)	(.model_1.DV_2)	(.model_1.DV_3)
POINT_Md	(.model_1.DV_1)	(-1 * .model_1.DV_2)	(.model_1.DV_3)
POINT_N	(.model_1.DV_4)	0.0	(.model_1.DV_5)
POINT_M0s	(.model_1.DV_6)	(.model_1.DV_7)	(.model_1.DV_8)
POINT_M0d	(.model_1.DV_6)	(-1 * .model_1.DV_7)	(.model_1.DV_8)
POINT_N0s	(.model_1.DV_9)	(.model_1.DV_10)	(.model_1.DV_11)
POINT_N0d	(.model_1.DV_9)	(-1 * .model_1.DV_10)	(.model_1.DV_11)

Fig. 5. The design variables in the optimization process.

The design functions for optimization are represented by measures that define the kinematic behavior of the guiding mechanism (see section 1): ΔX_P, ΔY_P, η_Z and η_Y. The objective is to minimize these motion variations of the axle. With this end in view, for each design variable and objective function, design studies were performed to determine the design variables that have great influence on the kinematic optimization criteria. Thus, a list of values was specified for each design variable, the simulation of the mechanism being performed for all values of the design variable. Finally, the design study report offers information about the sensibility of the objective function on the variations of a certain design variable. The report allows selecting the main variables that will be used for optimization.

On the basis of the design studies done for the above-described guiding linkage, the kinematic optimization is performed taking into account the criteria $\Delta X_P \in [\Delta X_{P\,min}, \Delta X_{P\,max}]$, and $\eta_Y \in [\eta_{Y\,min}, \eta_{Y\,max}]$ (the other two variations, ΔY_P and η_Z, are minimal for the initial configuration of the mechanism), for $\Delta Z_{GS} = \Delta Z_{Gd}$ (the values of the generator motions simulate the vertical displacement of the axle, without roll motion). In the case of opposite displacements of the left and right wheels ($\Delta Z_{GS} = -\Delta Z_{Gd}$), the influence of the design variables on the objective functions are insignificant, considering rational-constructive modifications of the design variables.

The optimization study was conducted in ADAMS/View by using the GRG (Generalized Reduced Gradient) algorithm from the OPTDES code of Design Synthesis [6]. In these terms, the diagrams shown in figure 7 (in which the independent kinematic parameter is the vertical position of the wheels/axle) present the variation of the longitudinal displacement of the axle (ΔX_P) and its proper rotation around the transversal axis (η_Y), for the initial (before optimization) and optimal guiding mechanism. The corresponding initial and optimal values of the selected design variables are presented in table 1.

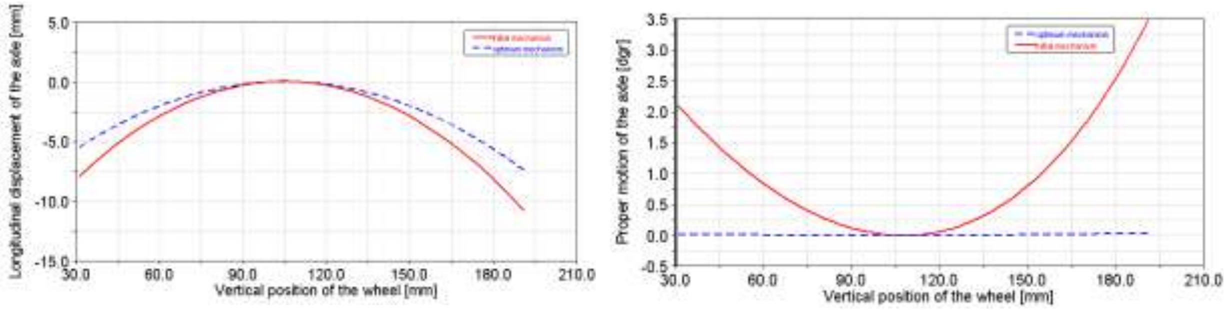


Fig. 7. Results of the optimization study.

Table 1. The initial and optimum values of the design variables (in mm).

DV	$X_{M0s(d)}$	$Y_{M0s(d)}$	$Z_{M0s(d)}$	$X_{N0s(d)}$	$Z_{N0s(d)}$
initial value	2014.5	± 536.0	40.0	2362.0	168.0
optimal value	2026.0	± 536.0	41.5	2130.0	167.5

As can see, the optimal design study leads to significant improvements in the kinematic behavior of the axle guiding linkage, with minimal changes in the geometric configuration of the mechanism, and this proves the viability (usefulness) of the adopted optimization strategy.

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