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A modular approach to the kinematics of the vehicle axle suspension linkages

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Abstract. The work deals with a modular approach on the kinematics of the suspension linkages used for the rear axle of motor vehicles. The multi-body systems theory/method was used to formulate the motion equations for the kinematic analysis, by individually considering the basic binary connections (which are defined by considering the types geometric constraints between the guiding arms/links and the adjacent bodies - car body and axle, respectively) that constitutes the suspension mechanism. Subsequently, by combining these binary connections in various combinations, the kinematic analysis of any type of axle suspension linkage (at least those commonly used) can be easily performed.

1. Introduction

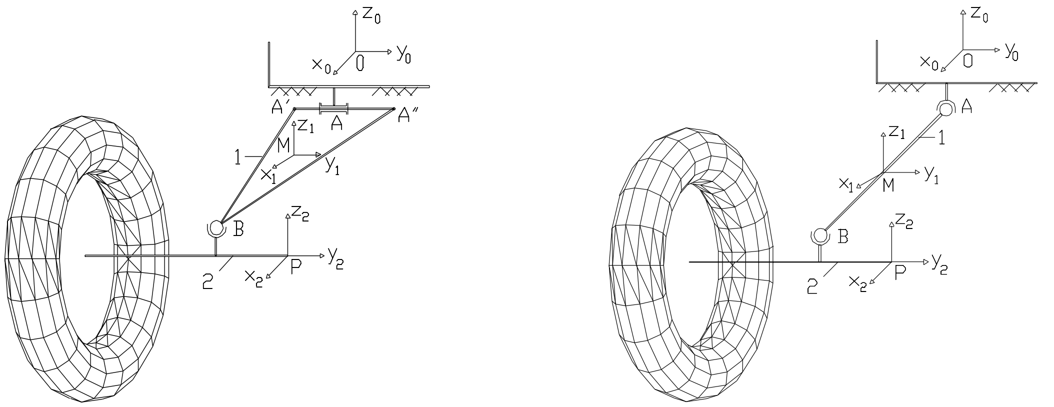
Relative to car body (chassis), the vehicle wheels can be guided independently - by means of a guiding mechanism for each wheel (independent suspension), or dependent - by a guiding mechanism of the rigid axle (dependent suspension). The first solution is frequently used for passenger cars (for both front and rear wheel suspensions), while the second one (which is addressed in this paper) is mainly used for the rear axles of larger gauge cars (e.g. commercial or off-road vehicles).

This paper deals with a study on the kinematics of the multi-link axle suspension linkages, which is a continuous research concern and challenge, the literature revealing various methods, more or less complex, and with a wider or narrower applicability [1-5]. A modular approach based on the multi-body systems (MBS) theory is proposed by decomposing the guiding mechanisms in the basic binary links by which is axle is guided in the relative movement to the vehicle chassis. Subsequently, by combining the binary links, the kinematics can be carried out for most types of axle guiding linkages.

2. The kinematics of the axle suspension linkages

The guiding links from the axle suspension linkages are hinged to axle and chassis by bushings (flexiblocks), which are compliant joints of rubber with 6 elastic restricted degrees of mobility. For the kinematic study, the bushings are frequently modeled by spherical joints, thus ignoring the linear deformations, which are generally insignificant [1, 3, 5-8]. In the case of the triangular guiding arms, which are double hinged to the vehicle chassis, the two corresponding spherical joints determine a rotational (revolute) joint, whose axis is defined by the centers of the spherical joints. A comprehensive systematization of the vehicle axle suspension linkages was carried out in [1].

From multiple possible variants of beam axle guiding, the guidance on circle (spherical - revolute binary connection between axle and chassis - Figure 1,a) and respectively the guidance on sphere (spherical - spherical binary connection between axle and chassis - Figure 1,b) are frequently used/implemented.



a. b.

Figure 1. The basic binary links/connections for guiding the vehicle rear axle.

One of the most used and performing methods for the kinematic, static and dynamic analysis of the mechanical systems is the one called in short MBS (the acronym stands from Multi-Body Systems), which treats the mechanical system as a set of bodies, connected by geometrical constraints (joints), elastic and damping elements, on which a varied system of external and reaction forces/torques can act. The MBS method underlies several powerful commercial software solutions, such as ADAMS or DYMES, which are used in a large variety of applications [9-14]. In the MBS concept, the spatial movement of a part is defined by 6 generalized coordinates: the coordinates of the origin of the local coordinate system attached to the part, and the orientation of the axes of this frame relative to the ones of the global (inertial) coordinate system.

The kinematic model of the axle suspension linkages contains several moving bodies (the axle and the guiding links/arms). For each moving body, there is defined a local coordinate system, namely $MX_1Y_1Z_1$ for the guiding arm "1" and "$PX_2Y_2Z_2$" for the rear axle "2", in the case of the basic binary connections shown in Figure 1. In the kinematic study, the car body (chassis) is considered to be rigidly connected to ground, so it is the reference part to which the global coordinate system $OX_0Y_0Z_0$ is attached, where X_0, Y_0 and Z_0 are the longitudinal, transversal and vertical axes of the vehicle.

Each geometric restriction (joint) in the axle suspension linkage replaces a number of degrees of freedom, by adding algebraic constraint equations of the following form (the first three equations constrain translational movements, while the last three restrict rotational movements):

- $X_i – X_j = 0$ - the global coordinate X (which is the longitudinal axis in Figure 1) of the "i"-body reference frame equals the corresponding coordinate of the "j"-body reference frame,
- $Y_i – Y_j = 0$ - the global coordinate Y (the transversal axis in Figure 1) of the "i"-body reference frame equals the corresponding coordinate of the "j"-body reference frame,
- $Z_i – Z_j = 0$ - the global coordinate Z (the vertical axis in Figure 1) of the "i"-body reference frame equals the corresponding coordinate of the "j"-body reference frame,
- $Z_i \cdot X_j = 0$ - the global coordinate Z of the "i"-body reference frame remains normal (perpendicular) on the X axis of the "j"-body reference frame,
- $Z_i \cdot Y_j = 0$ - the global coordinate Z of the "i"-body reference frame remains normal (perpendicular) on the Y axis of the "j"-body reference frame,
- $X_i \cdot Y_j = 0$ - the global coordinate X of the "i"-body reference frame remains normal (perpendicular) on the Y axis of the "j"-body reference frame.

In the following, for the main types of joints (connections) from the axle suspension linkages, the geometric constraint equations will be defined. For the spherical - revolute binary connection (see Figure 1.a) the corresponding equations are the ones shown in Table 1, while Table 2 corresponds to the spherical - spherical binary connection (see Figure 1.b). In these tables, in the constraint equations area, T stands from translational and R from rotational (the movements canceled by the binary links/connections).

Table 1. The constraint equations for the spherical - revolute binary connection.

Element / part			Car body (0)	Guiding bar (1)		Axle (2)
Joint			A - A'	A - A"	B	B
Coordinates	Local	X	X_A, $X_{A'}$	$X_{A(M)}$, $X_{A''(M)}$	$X_{B(M)}$	$X_{B(P)}$
		Y	Y_A, $Y_{A'}$	$Y_{A(M)}$, $Y_{A''(M)}$	$Y_{B(M)}$	$Y_{B(P)}$
		Z	Z_A, $Z_{A'}$	$Z_{A(M)}$, $Z_{A''(M)}$	$Z_{B(M)}$	$Z_{B(P)}$
	Global	X	X_A, $X_{A'}$	$X_M + m_{11} \cdot X_{A(M)} + m_{12} \cdot Y_{A(M)} + m_{13} \cdot Z_{A(M)}$ $X_M + m_{11} \cdot X_{A''(M)} + m_{12} \cdot Y_{A''(M)} + m_{13} \cdot Z_{A''(M)}$	$X_M + m_{11} \cdot X_{B(M)} + m_{12} \cdot Y_{B(M)} + m_{13} \cdot Z_{B(M)}$	$X_P + w_{11} \cdot X_{B(P)} + w_{12} \cdot Y_{B(P)} + w_{13} \cdot Z_{B(P)}$
		Y	Y_A, $Y_{A'}$	$Y_M + m_{21} \cdot X_{A(M)} + m_{22} \cdot Y_{A(M)} + m_{23} \cdot Z_{A(M)}$ $Y_M + m_{21} \cdot X_{A''(M)} + m_{22} \cdot Y_{A''(M)} + m_{23} \cdot Z_{A''(M)}$	$Y_M + m_{21} \cdot X_{B(M)} + m_{22} \cdot Y_{B(M)} + m_{23} \cdot Z_{B(M)}$	$Y_P + w_{21} \cdot X_{B(P)} + w_{22} \cdot Y_{B(P)} + w_{23} \cdot Z_{B(P)}$
		Z	Z_A, $Z_{A'}$	$Z_M + m_{31} \cdot X_{A(M)} + m_{32} \cdot Y_{A(M)} + m_{33} \cdot Z_{A(M)}$ $Z_M + m_{31} \cdot X_{A''(M)} + m_{32} \cdot Y_{A''(M)} + m_{33} \cdot Z_{A''(M)}$	$Z_M + m_{31} \cdot X_{B(M)} + m_{32} \cdot Y_{B(M)} + m_{33} \cdot Z_{B(M)}$	$Z_P + w_{31} \cdot X_{B(P)} + w_{32} \cdot Y_{B(P)} + w_{33} \cdot Z_{B(P)}$
Constraint equations	T		$X_A = X_M + m_{11} \cdot X_{A(M)} + m_{12} \cdot Y_{A(M)} + m_{13} \cdot Z_{A(M)}$		$X_M + m_{11} \cdot X_{B(M)} + m_{12} \cdot Y_{B(M)} + m_{13} \cdot Z_{B(M)} = X_P + w_{11} \cdot X_{B(P)} + w_{12} \cdot Y_{B(P)} + w_{13} \cdot Z_{B(P)}$	
			$Y_A = Y_M + m_{21} \cdot X_{A(M)} + m_{22} \cdot Y_{A(M)} + m_{23} \cdot Z_{A(M)}$		$Y_M + m_{21} \cdot X_{B(M)} + m_{22} \cdot Y_{B(M)} + m_{23} \cdot Z_{B(M)} = Y_P + w_{21} \cdot X_{B(P)} + w_{22} \cdot Y_{B(P)} + w_{23} \cdot Z_{B(P)}$	
			$Z_A = Z_M + m_{31} \cdot X_{A(M)} + m_{32} \cdot Y_{A(M)} + m_{33} \cdot Z_{A(M)}$		$Z_M + m_{31} \cdot X_{B(M)} + m_{32} \cdot Y_{B(M)} + m_{33} \cdot Z_{B(M)} = Z_P + w_{31} \cdot X_{B(P)} + w_{32} \cdot Y_{B(P)} + w_{33} \cdot Z_{B(P)}$	
	R		$\dfrac{\left(Y_{A'}-Y_A\right)\cdot \sum_{i=1,K=X}^{3,Z} m_{3i}\cdot\left(K_{A''(M)}-K_{A(M)}\right) = \left(Z_{A'}-Z_A\right)\cdot \sum_{i=1,K=X}^{3,Z} m_{2i}\cdot\left(K_{A''(M)}-K_{A(M)}\right)}{\left(Y_{A'}-Y_A\right)\cdot \sum_{i=1,K=X}^{3,Z} m_{1i}\cdot\left(K_{A''(M)}-K_{A(M)}\right) = \left(X_{A'}-X_A\right)\cdot \sum_{i=1,K=X}^{3,Z} m_{2i}\cdot\left(K_{A''(M)}-K_{A(M)}\right)}$		–	
Kinematic parameters			–	X_M, Y_M, Z_M φ_{1X}, φ_{1Y}, φ_{1Z}		X_P, Y_P, Z_P φ_{2X}, φ_{2Y}, φ_{2Z}
Geometrical parameters (input data)			X_A, Y_A, Z_A $X_{A'}$, $Y_{A'}$, $Z_{A'}$ φ_{0X}, φ_{0Y}, φ_{0Z}	$X_{A(M)}$, $Y_{A(M)}$, $Z_{A(M)}$; $X_{A''(M)}$, $Y_{A''(M)}$, $Z_{A''(M)}$ $X_{B(M)}$, $Y_{B(M)}$, $Z_{B(M)}$		$X_{B(P)}$, $Y_{B(P)}$, $Z_{B(P)}$

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Table 2. The constraint equations for the spherical - spherical binary connection.

Element / part			Car body (0)	Guiding bar (1)		Axle (2)
Joint			A	A	B	B
Coordinates	Local	X	X_A	$X_{A(M)}$	$X_{B(M)}$	$X_{B(P)}$
		Y	Y_A	$Y_{A(M)}$	$Y_{B(M)}$	$Y_{B(P)}$
		Z	Z_A	$Z_{A(M)}$	$Z_{B(M)}$	$Z_{B(P)}$
	Global	X	X_A	$X_M + m_{11} \cdot X_{A(M)} + m_{12} \cdot Y_{A(M)} + m_{13} \cdot Z_{A(M)}$	$X_M + m_{11} \cdot X_{B(M)} + m_{12} \cdot Y_{B(M)} + m_{13} \cdot Z_{B(M)}$	$X_P + w_{11} \cdot X_{B(P)} + w_{12} \cdot Y_{B(P)} + w_{13} \cdot Z_{B(P)}$
		Y	Y_A	$Y_M + m_{21} \cdot X_{A(M)} + m_{22} \cdot Y_{A(M)} + m_{23} \cdot Z_{A(M)}$	$Y_M + m_{21} \cdot X_{B(M)} + m_{22} \cdot Y_{B(M)} + m_{23} \cdot Z_{B(M)}$	$Y_P + w_{21} \cdot X_{B(P)} + w_{22} \cdot Y_{B(P)} + w_{23} \cdot Z_{B(P)}$
		Z	Z_A	$Z_M + m_{31} \cdot X_{A(M)} + m_{32} \cdot Y_{A(M)} + m_{33} \cdot Z_{A(M)}$	$Z_M + m_{31} \cdot X_{B(M)} + m_{32} \cdot Y_{B(M)} + m_{33} \cdot Z_{B(M)}$	$Z_P + w_{31} \cdot X_{B(P)} + w_{32} \cdot Y_{B(P)} + w_{33} \cdot Z_{B(P)}$
Constraint equations		T	$X_A = X_M + m_{11} \cdot X_{A(M)} + m_{12} \cdot Y_{A(M)} + m_{13} \cdot Z_{A(M)}$		$X_M + m_{11} \cdot X_{B(M)} + m_{12} \cdot Y_{B(M)} + m_{13} \cdot Z_{B(M)} = X_P + w_{11} \cdot X_{B(P)} + w_{12} \cdot Y_{B(P)} + w_{13} \cdot Z_{B(P)}$	
			$Y_A = Y_M + m_{21} \cdot X_{A(M)} + m_{22} \cdot Y_{A(M)} + m_{23} \cdot Z_{A(M)}$		$Y_M + m_{21} \cdot X_{B(M)} + m_{22} \cdot Y_{B(M)} + m_{23} \cdot Z_{B(M)} = Y_P + w_{21} \cdot X_{B(P)} + w_{22} \cdot Y_{B(P)} + w_{23} \cdot Z_{B(P)}$	
			$Z_A = Z_M + m_{31} \cdot X_{A(M)} + m_{32} \cdot Y_{A(M)} + m_{33} \cdot Z_{A(M)}$		$Z_M + m_{31} \cdot X_{B(M)} + m_{32} \cdot Y_{B(M)} + m_{33} \cdot Z_{B(M)} = Z_P + w_{31} \cdot X_{B(P)} + w_{32} \cdot Y_{B(P)} + w_{33} \cdot Z_{B(P)}$	
		R	–		–	
Kinematic parameters			–	X_M, Y_M, Z_M $\varphi_{1X}, \varphi_{1Y}, \varphi_{1Z}$		X_P, Y_P, Z_P $\varphi_{2X}, \varphi_{2Y}, \varphi_{2Z}$
Geometrical parameters (input data)			X_A, Y_A, Z_A $\varphi_{0X}, \varphi_{0Y}, \varphi_{0Z}$	$X_{A(M)}, Y_{A(M)}, Z_{A(M)}$ $X_{B(M)}, Y_{B(M)}, Z_{B(M)}$		$X_{B(P)}, Y_{B(P)}, Z_{B(P)}$

The connection matrix M_{10} between the local ($X_1Y_1Z_1$) and global (XYZ) coordinate systems has the following form:

$$M_{10} = \begin{bmatrix} m_{11} & m_{12} & m_{13} \\ m_{21} & m_{22} & m_{23} \\ m_{31} & m_{32} & m_{33} \end{bmatrix} = \begin{bmatrix} \bar{i} \cdot \bar{i}_1 & \bar{i} \cdot \bar{j}_1 & \bar{i} \cdot \bar{k}_1 \\ \bar{j} \cdot \bar{i}_1 & \bar{j} \cdot \bar{j}_1 & \bar{j} \cdot \bar{k}_1 \\ \bar{k} \cdot \bar{i}_1 & \bar{k} \cdot \bar{j}_1 & \bar{k} \cdot \bar{k}_1 \end{bmatrix} = \begin{bmatrix} \cos(X, X_1) & \cos(X, Y_1) & \cos(X, Z_1) \\ \cos(Y, X_1) & \cos(Y, Y_1) & \cos(Y, Z_1) \\ \cos(Z, X_1) & \cos(Z, Y_1) & \cos(Z, Z_1) \end{bmatrix}. \quad (1)$$

3. Results and conclusions

By linking in parallel the above presented guidance cases, a large variety of axle suspension linkages can be obtained, as they are systematized in [1], in which the geometric constraints equations have the form of those shown in Tables 1 and 2. For example, by interposing between axle and chassis two spherical - spherical connections and one spherical - revolute connection, the so-called 2S1C (where S means guidance on sphere , while C - guidance on circle) suspension mechanism is obtained, which is shown in Figure 2 (as structural model and semi-constructive solution).

This mechanism has two mobilities, which correspond to the left and right wheels vertical movements, which can be transposed into the vertical displacement of the axle center (Z_P) and the axle roll rotation around the longitudinal axis (φ_{4X}).

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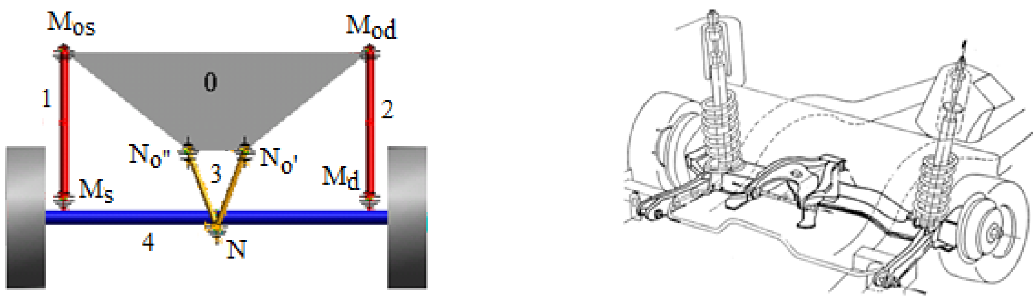

a. b.

Figure 2. Structural model (a) and semi-constructive solution (b) for the 2S1C axle guiding linkage.

By transposing the general forms of the constraint equations from Tables 1 and 2 for this particular type of axle suspension linkage, the following equations are obtained:

- for the joints of the guiding bars "1-3" to axle "4":

$$F_{14}=\begin{bmatrix}X_P\\Y_P\\Y_P\end{bmatrix}+M_{40}\cdot\begin{bmatrix}X_{Ms}\\Y_{Ms}\\Z_{Ms}\end{bmatrix}_4-\begin{bmatrix}X_{Mos}\\Y_{Mos}\\Z_{Mos}\end{bmatrix}-M_{10}\cdot\begin{bmatrix}X_{Ms}\\Y_{Ms}\\Z_{Ms}\end{bmatrix}_1=0, F_{24}=\begin{bmatrix}X_P\\Y_P\\Y_P\end{bmatrix}+M_{40}\cdot\begin{bmatrix}X_{Md}\\Y_{Md}\\Z_{Ms}\end{bmatrix}_4-\begin{bmatrix}X_{Mod}\\Y_{Mod}\\Z_{Mod}\end{bmatrix}-M_{20}\cdot\begin{bmatrix}X_{Md}\\Y_{Md}\\Z_{Md}\end{bmatrix}_2=0,$$
$$F_{34}=\begin{bmatrix}X_P\\Y_P\\Y_P\end{bmatrix}+M_{40}\cdot\begin{bmatrix}X_N\\Y_N\\Z_N\end{bmatrix}_4-\begin{bmatrix}X_{No}\\Y_{No}\\Z_{No}\end{bmatrix}-M_{30}\cdot\begin{bmatrix}X_N\\Y_N\\Z_N\end{bmatrix}_3=0; \quad (2)$$

- for the joints of the guiding bars "1-3" to car body "0":

$$F_{10}=\begin{bmatrix}X_{Mos}\\Y_{Mos}\\Z_{Mos}\end{bmatrix}-\begin{bmatrix}X_O\\Y_O\\Z_O\end{bmatrix}-\begin{bmatrix}a_1\\a_2\\a_3\end{bmatrix}=0, F_{20}=\begin{bmatrix}X_{Mod}\\Y_{Mod}\\Z_{Mod}\end{bmatrix}-\begin{bmatrix}X_O\\Y_O\\Z_O\end{bmatrix}-\begin{bmatrix}b_1\\b_2\\b_3\end{bmatrix}=0, F_{30}=\begin{bmatrix}X_{No}\\Y_{No}\\Z_{No}\\\delta_{3x}\\\delta_{3z}\end{bmatrix}-\begin{bmatrix}X_O\\Y_O\\Z_O\\\delta_{ox}\\\delta_{oz}\end{bmatrix}-\begin{bmatrix}c_1\\c_2\\c_3\\c_4\\c_5\end{bmatrix}=0. \quad (3)$$

The matrices M_{i0} (i=1…4) have similar forms with the one in equation (1). The geometric constants a_i, b_i, c_j (i=1…3, j=1…5) can be obtained from the initial configuration of the suspension system. The equations (2) and (3) define a system of 20 scalar equations, with 24 generalized coordinates (6 per moving body). As mentioned, the independent parameters are Z_P and φ_{4x}, while the rotations of the guiding bars "1" (φ_{1x}) and "2" (φ_{2x}) around their own axes are kinematically passive. Therefore, 20 unknown kinematic parameters will be established by processing the system of equations (2) and (3), thus determining the kinematic behavior of the axle suspension linkage, as it is defined in [1].

The global coordinates (X, Y, Z) of the joints in the initial position of the guiding mechanism/vehicle are input parameters for the kinematic study, as follows (in correlation with the notations in Figure 2): $M_{0s/d}$ (2014.5, ±536.0, 40.0), $M_{s/d}$ (2523.5, ±536.0, 40.0), $N_{0'/''}$ (2362.0, ±100.0, 168.0), N (2629.0, 0.0, 168.0). In the same position, the axle center has the global coordinates P (2596.0, 0, 111.0).

To exemplify, it was considered the case where the wheels move vertically from the initial position with the same rate, but in opposite directions (up/down to the altitude of ±80mm), and then in reverse, until the wheels perform a full up-down / down-up race, with a return to the starting (initial) position.

These vertical displacements of the wheels are equivalent with the following variations of the two independent kinematic parameters: Z_P = 111.0 mm (it remains constant during the analysis, so $\Delta Z_P = 0$), and $\varphi_{4X} \in [-6.676°, 6.676°]$. From the results of the analysis thus carried out, Figure 3 shows the variation diagrams for the displacements of the axle centre along the longitudinal (ΔX_P) and transversal (ΔY_P) axes, as well as the twisting angle of the axle (φ_{4Y}) and the pivoting angle around the vertical axis (φ_{4Z}), depending on the roll angle of the axle (φ_{4X}).

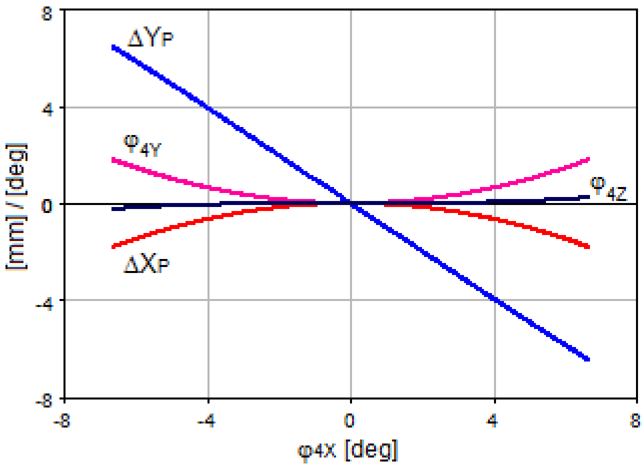

Figure 3. Kinematic analysis results.

The kinematic analysis method can be implemented for most types of axle guiding linkages, and it avoids the disadvantages of the numerical methods based on nonlinear equation systems. At the same time, the method is easy to implement for the independent wheel suspension mechanisms.

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