On the analysis of double wishbone suspension regarding steering input and anti-dive/lift effect

Engin TANIK* and Volkan PARLAKTAŞ*
Department of Mechanical Engineering, Hacettepe University, Beytepe, Çankaya, 06800 Ankara, Turkey
E-mail: volkan@hacettepe.edu.tr

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Abstract
In this study, a novel kinematic analysis procedure of the double wishbone suspension mechanism regarding steering input is proposed. In the previous study, published by the authors, the double wishbone mechanism was investigated disregarding steering input. To the best of our knowledge, there is no analytical method available in the literature for the analysis of double wishbone suspension mechanism regarding steering input. Initially, analysis of the mechanism for variable steering input, while keeping wheel travel fixed is considered. Then, kinematic analysis is performed analytically for the ultimate case of two inputs. Analysis simultaneously for both variable steering and variable wheel travel is presented. The essential parameters; camber, caster, kingpin angles are defined according to the kinematic model. For verification, a synthesized mechanism is established in Catia software and same results with the analytical model are obtained. Another presented novel analytical approach is anti-dive/lift analysis of the double wishbone mechanism.

Key words: Suspension systems, Double wishbone suspension, RSSR-SSP mechanism, Suspension kinematics, Anti-dive, Anti-lift

1. Introduction

The purpose of a vehicle suspension is to maximize contact between tires and road surface for good road holding, to provide steering stability for good handling and to ensure comfort of passengers for good ride. Road holding is ability of a vehicle to grip the road (lateral acceleration capacity). Handling is ease and success in controlling a vehicle under various road conditions (curves, straight line, bumps). Ride is the ability of a vehicle to smooth out not perfectly flat roads. The most important kinematic parameters of suspension that affect handling, road holding and ride characteristics of a vehicle are: camber angle; the inclination of the wheel axis towards the road surface in the vertical plane, caster angle; the angle between the steering axis and the vertical plane as viewed from the side, kingpin angle (steering axis inclination); the angle between the steering axis and the vehicle's longitudinal plane, toe angle; the symmetric angle that each wheel makes with the longitudinal axis of the vehicle. A higher performance in the design is achieved as these kinematic parameters are improved (Reimpell and Stoll, 1998).

In most foundations suspension systems are analyzed via high cost software that are based on numerical methods. In the literature, there is no analysis study available for the double wishbone mechanism “regarding steering input” that is performed “analytically”. Recently, an analysis approach for the double wishbone mechanism disregarding steering input is introduced (Tanık and Parlaktaş, 2015a).

Kinematic design of double-wishbone suspension system using a multi objective dimensional synthesis technique is focused (Sancibrian, et al., 2010). A new design optimization framework for suspension systems considering the kinematic characteristics, such as the camber angle, caster angle, kingpin inclination angle, and toe angle in the presence of uncertainties is proposed (Wu, et al., 2014). A general method of the kinematic synthesis of suspension mechanisms is presented (Suh, 1989). Design and implementation of a double wishbone front suspension for a vineyard–orchard tractor is dealted (Uberti, et al., 2015). Two position synthesis method is applied to obtain desired
camber variation of an approximated double-wishbone mechanism (Tanık and Parlaktaş, 2015b). Formulation of a comprehensive kineto-dynamic quarter-car model to study the kinematic and dynamic properties of a linkage suspension, and influences of linkage geometry on selected performance measures is presented (Balike, et al., 2011).

From a kinematics point of view, a suspension system can be defined as a combination of links and joints (Russell, et al., 2009), (Tanık and Parlaktaş, 2011), (Tanık and Parlaktaş, 2015c) A double wishbone mechanism is essentially a RSSR–SSP (R:revolute, S:spherical, P:prismatic) linkage (Shen, et al., 2014) when steering function is required as shown in Fig. 1.

![Fig. 1. Double wishbone suspension mechanism](image)

In the previous study (Tanık and Parlaktaş, 2015a), an analysis procedure for RSSR-SSP linkage is presented when steering function is not required. Hence, position of the P joint was fixed (now it is variable) to a specific position. However, for a complete kinematic analysis, variable steering input and anti-dive/lift effect should be taken into account. In this study, RSSR-SSP linkage is analyzed initially for variable steering input, while keeping suspension travel fixed and an example is displayed. Then, kinematic analysis is performed for the ultimate case; both variable steering and variable wheel travel is investigated simultaneously. The essential parameters; camber, caster, and kingpin angles are defined according to the kinematic model. The same mechanism in the previous study is reconsidered; variations of these essential parameters with respect to steering angle and wheel travel are displayed. For verification, the same double wishbone mechanism is modeled in Catia. These essential angles are measured for specific values of wheel travel and steering angle. The results are compared in a tabular form.

It is well known that angled upper wishbone in transverse direction of a vehicle yields anti-dive or anti-lift characteristics. To the best of our knowledge, the amount of anti-dive rate is not calculated analytically in the literature. In this paper, we also propose an analytical method to calculate this effect. From the given example, it is observed that the amount of deflection of the suspension can be reduced remarkably by assigning a convenient anti-dive angle in our
analytical approach.

Furthermore, kinematic analyses of front and rear double wishbone suspension mechanisms are performed by using the methodology given in this study. After the mechanical design and manufacturing stages, these suspension systems are implemented to an electric vehicle which is also designed by the authors.

2. Analysis of the double wishbone mechanism for variable steering input

In this section, an analysis method is taken into consideration for variable steering input, while keeping suspension travel fixed in a specific position. Here, zero wheel travel (static) position is assigned for the fixed position. Analysis of the suspension system starts from determining fundamental parameters by using Eqns. (4)-(6) of the previous study (Tanık and Parlaktaş, 2015a). For a specified input angle ($\theta$), positions of upper and lower wishbones and the knuckle can be determined in closed-form equation. In this study, steering input or in another words X-coordinate of the spherical joint at point $F$ is variable. Since steering boxes are located in transverse direction of vehicles, the steering input only manipulates X-coordinate of point $F$ due to the translational freedom of the P joint (Fig. 1). Therefore, Eq. (7) in the previous study (Tanık and Parlaktaş, 2015a) is modified as: $F_{x0} = F_x$, note that $F_x$ is input variable now. The rest of the equations considering the loop $OADEFO$ remain the same, so Eqns. (8)-(10) in the previous study (Tanık and Parlaktaş, 2015a) are still valid. Steering angle of the suspension system can be determined from the projection of the hub direction on XY plane ($e_{hubp}$) and Y-axis. The angle between $e_{hubp}$ and Y-axis can be calculated from Eqns. (11), (13), (14), (15), and (22) in the previous study (Tanık and Parlaktaş, 2015a).

3. Example 1

Analyze steering variation of a front right double wishbone suspension mechanism for the following parameters: $l_1 = 330$mm, $l_2 = 260$mm, $l_3 = 240$mm, $p = 210$mm, $g = 32$mm, $f = -12$mm, $h_6 = 120$mm, $c_o = 0$, $h_{hub} = 60$mm, $E_0 = [260 -120 173]^T$, $F_0 = [16 -59 150]^T$, $\phi = 9^\circ$, $\xi = 0$, and $\xi_0 = -0.25^\circ$. Total stroke of the steering box is assigned as 110mm. The fixed position is the zero wheel travel position, where the lower wishbone angle $\theta$ is equal to zero. For variable steering input, X-coordinate of point $F$ is defined as: $F_{x0} - 55$mm $\leq F_x \leq F_{x0} + 55$mm. By using the approach summarized in Section-2, steering angle variation is determined and displayed in Fig. 2.

As seen in Fig. 2(a), for the maximum negative stroke of the steering box, the steering angle is equal to 29.7° and for the maximum positive stroke it is equal to 26.5°. In Fig. 2(b) difference of inside and outside wheel steering angle with respect to outside wheel angle is presented. Note that, blue line in this figure presents Ackermann geometry, where wheelbase and track width are taken as 2400mm and 1500mm respectively.

For verification purpose, the same double wishbone suspension mechanism is modeled in Catia software. As expected, the same results are obtained. In Table 1, a comparison chart is displayed for six different position of the steering box stroke. Here, $A$ represents the analytical model, $C$ represents Catia.
Table 1. Comparison chart for variable steering input, and fixed suspension travel

<table>
<thead>
<tr>
<th>Steering box input (mm)</th>
<th>A</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
<td>26.5</td>
<td>26.5</td>
</tr>
<tr>
<td>44</td>
<td>21.02</td>
<td>21.02</td>
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<tr>
<td>22</td>
<td>10.44</td>
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<tr>
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<td>-22.76</td>
</tr>
<tr>
<td>-55</td>
<td>-29.72</td>
<td>-29.72</td>
</tr>
</tbody>
</table>

4. Analysis of the double wishbone mechanism regarding both suspension travel and steering input

In this section, an analysis method is considered for both variable steering and variable suspension travel. Note that, there are two input variables in this approach. Therefore, in the following example, the suspension parameters are displayed with respect to variable steering and for specific wheel travel values with the following procedure:

Initially, for a specified suspension geometry, kinematic analysis is performed by using Eqns. (4)-(18) in the previous study (Tanık and Parlaktuş, 2015a). Then, considering \( \vec{e}_{\text{camber}} \) and \( \vec{e}_{\text{caster}} \) vectors which are perpendicular to each other on XY plane (Fig. 1), camber, caster, and kingpin angles are determined. It should be noted that, as the steering angle \( \beta \) alters, \( \vec{e}_{\text{camber}} \) and \( \vec{e}_{\text{caster}} \) vectors are rotated at the amount of steering angle \( \beta \) as follows (Fig. 1).

\[
\vec{e}_{\text{camber}} = \begin{bmatrix} \cos \beta & \sin \beta & 0 \end{bmatrix}^T
\]  \hspace{1cm} (1)

\[
\vec{e}_{\text{caster}} = \begin{bmatrix} -\sin \beta & \cos \beta & 0 \end{bmatrix}^T
\]  \hspace{1cm} (2)

In order to obtain caster angle, the angle between \( \vec{e}_{\text{caster}} \) and \( \vec{e}_k \) is determined as in Fig. 3:

\[
\tau = \cos^{-1}(\vec{e}_k \cdot \vec{e}_{\text{caster}}) - \pi/2
\]  \hspace{1cm} (3)
Kingpin angle is determined from the angle between $\vec{e_{\text{camber}}}$ and $\vec{e_k}$ as in Fig. 3:

$$
\sigma = \cos^{-1}(\vec{e_k} \cdot \vec{e_{\text{camber}}}) - \pi/2.
$$

(4)

In order to obtain the camber angle, variation of the kingpin angle can be used, since their variations are exactly the same. Thus, camber angle can be determined as:

$$
\varphi = \sigma_0 - \sigma + \epsilon_0
$$

(5)

where $\epsilon_0$ is the static camber at zero steering angle and $\sigma_0$ is the kingpin at static position corresponding to steering angle.

5. Example 2

Analyze the front right double wishbone suspension mechanism for the following parameters: $l_l = 330\text{mm}$, $l_u = 260\text{mm}$, $l_k = 240\text{mm}$, $p = 210\text{mm}$, $g = 32\text{mm}$, $f = -12\text{mm}$, $\epsilon = 0$, $l_{\text{hub}} = 60\text{mm}$, $E_0 = [260 \ -120 \ 173]^T$, $F_0 = [16 \ -59 \ 150]^T$, $\phi = 9^\circ$, $\xi = 0$, and $\epsilon_0 = -0.25^\circ$. Suspension travel is required to be ±58mm and steering angle is required to be ±30°. Let the lower wishbone working range be: -10° ≤ $\theta$ ≤ 10°. Initially, Eqns. (4)-(18) in the previous study (Tanık and Parlaktaş, 2015a) are solved accordingly and the wheel travel is determined as $-57.8\text{mm} \leq \Delta OJ_z \leq 58.1\text{mm}$. Then, by solving Eqns. (1)-(5) for -30° ≤ $\beta$ ≤ 30°, camber, caster, and kingpin angles are calculated and presented in Figs. 4, 5, and 6 respectively.

Fig. 4. Camber variation w.r.t steering angle and wheel travel

Fig. 5. Caster variation w.r.t steering angle and wheel travel
From Fig. 4, it can be observed that for $30^\circ$ toe-out angle the camber value is slightly affected from the wheel travel. However, for extreme toe-in angles camber angle values are radically different throughout the wheel travel.

From Fig. 5, it can be observed that the caster angle has nearly linear relationship with steering angle especially for negative valued wheel travels.

The same double wishbone suspension mechanism is modeled in Catia, then camber, caster, and kingpin angles are measured for specific values of wheel travel and steering angle. In Table 2, some of the results obtained from the analytical model (A) and Catia (C) are presented. As it should be, all of the results are the same. It can be concluded that, by the proposed design methodology, commonly desired suspension characteristics can determined easily and precisely.

<table>
<thead>
<tr>
<th>Steering angle (°)</th>
<th>-24</th>
<th>-12</th>
<th>-6</th>
<th>6</th>
<th>12</th>
<th>24</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel Travel (mm)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>29.1</td>
<td>A</td>
<td>C</td>
<td>A</td>
<td>C</td>
<td>A</td>
<td>C</td>
</tr>
<tr>
<td>-29</td>
<td>1.88</td>
<td>1.88</td>
<td>0.23</td>
<td>0.23</td>
<td>-0.45</td>
<td>-0.45</td>
</tr>
<tr>
<td>Caster</td>
<td>2.29</td>
<td>2.29</td>
<td>1.16</td>
<td>1.16</td>
<td>0.72</td>
<td>0.72</td>
</tr>
<tr>
<td>29.1</td>
<td>A</td>
<td>C</td>
<td>A</td>
<td>C</td>
<td>A</td>
<td>C</td>
</tr>
<tr>
<td>-29</td>
<td>6.85</td>
<td>6.85</td>
<td>6.93</td>
<td>6.93</td>
<td>3.96</td>
<td>3.96</td>
</tr>
<tr>
<td>Caster</td>
<td>6.13</td>
<td>6.13</td>
<td>4.56</td>
<td>4.56</td>
<td>3.7</td>
<td>3.7</td>
</tr>
<tr>
<td>29.1</td>
<td>A</td>
<td>C</td>
<td>A</td>
<td>C</td>
<td>A</td>
<td>C</td>
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<td>8.96</td>
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<tr>
<td>Caster</td>
<td>6.89</td>
<td>6.89</td>
<td>8.03</td>
<td>8.03</td>
<td>8.46</td>
<td>8.46</td>
</tr>
</tbody>
</table>

To explain how we performed the verification, Fig. 7 is displayed which is a snapshot from analysis with Catia. Here the wheel travel is 29.1mm and the steering angle is $6^\circ$. In Fig. 7, the angle between the $6^\circ$ rotated line (corresponding to the unit vector $\vec{e}_{\text{caster}}$ on XY given in Fig. 3) and the knuckle is measured as $93.823^\circ$. This angle corresponds to summation of caster angle and $90^\circ$ (refer to Fig.3), thus the caster angle is equal to $3.823^\circ$, which is given in Table 2 as $3.82^\circ$ (the values are rounded in the table).
6. Anti-dive/lift analysis

The anti-dive/lift angle reduces amount by which front end of a vehicle dips or rear end rises when the brakes are applied. It is well known that non-parallel upper and lower wishbone of a double wishbone suspension system yields anti-dive or anti-lift characteristics. However, to the best of our knowledge, the amount of anti-dive rate hasn’t been calculated analytically in the literature. In this section, we propose a method to calculate this effect analytically.

In Fig. 8(a), a front right double wishbone suspension system is presented with external forces $F_b$ and $F_{ad}$. $F_b$ is the braking force occurred during deceleration in $-Y$-direction and $F_{ad}$ is the anti-dive/lift force occurred in $Z$-direction. Rest of the external forces are not taken into consideration, since only the effect of braking force on anti-dive/lift characteristics is investigated. It should be noted that, force analysis can be performed with the principle of superposition, thus other forces can be added afterwards if necessary.

Fig. 8. Anti-dive/lift free body diagram
The analysis will be carried out for static equilibrium since masses of suspension links are assumed to be very low when compared to the loads that they are subjected to. Here, the method of virtual work will be employed since reaction forces of the joints aren’t required. Also, an equivalent simpler system can be determined if the forces are carried on the knuckle, exactly at point $H$. For our case, brake disc/drum is assumed to be on the knuckle. When the forces are carried to point $H$, the equivalent system will be as presented in Fig. 8(b), where the moment vectors are indicated with red colors. For the case where brake disc/drum is located on chassis of a vehicle, the moment vector $r_cF_b$ wouldn’t be on knuckle, which is the major loading that generates anti-dive/lift effect. Considering the loading given in Fig. 8(b), the virtual work is:

$$
\delta W = [F_{ad}\delta \vec{k} - F_{bd}]\delta \vec{OH} + [-r_cF_b\delta \vec{l} - c_0F_{ad}\delta \vec{j} - l_{hub}F_{ad}\delta \vec{j} - l_{hub}F_b\delta \vec{k}]\delta \vec{\lambda}
$$

(6)

Here, $\delta \vec{\lambda}$ is virtual rotation of the knuckle. According to the principle of virtual work $\delta W=0$ for equilibrium, thus we obtain the anti-dive/lift to braking force ratio, $\psi$, as:

$$
\psi = F_{ad}/F_b = \left[\frac{\delta \vec{\lambda}}{\delta \theta} + \frac{\delta \vec{OH}}{\delta \theta} + \frac{l_{hub}}{\delta \theta}\delta \vec{l} \frac{\delta \vec{\lambda}}{\delta \theta} - l_{hub}F_{ad}\delta \vec{j} - l_{hub}F_b\delta \vec{k}\right]
$$

(7)

The steering input is kept fixed; therefore the degree of freedom of this mechanism is equal to one. For this single degree of freedom system let the generalized coordinate be $\theta$. Thus, denominator and numerator of right hand side of Eq. (7) were multiplied by $1/\delta \theta$ in order to obtain derivatives of the virtual displacements with respect to $\theta$.

$\vec{OH} = l_i\vec{e}_i + l_k\vec{e}_k$, where $\vec{e}_k$ and $\vec{e}_i$ are previously derived (Tanık and Parlaktaş, 2015a). Thus, derivative of the virtual displacement of point $H$ is:

$$
\frac{\delta \vec{OH}}{\delta \theta} = l_i\frac{\delta \vec{e}_i}{\delta \theta} + l_k\frac{\delta \vec{e}_k}{\delta \theta}
$$

(8)

Taking derivatives of Eqns. (3), (5) and (6) in the previous study (Tanık and Parlaktaş, 2015a), we obtain:

$$
\frac{\delta \vec{OH}}{\delta \theta} = \frac{1}{l_k}\begin{bmatrix}
l_{hub}\sin \theta - l_kl_1\sin \theta - \Omega l_kl_k\cos \xi \sin \chi + \Omega l_kl_k\cos \chi \sin \xi \sin \phi
-l_kl_1\cos \theta + l_{hub}\cos \theta + \Omega l_kl_k\cos \chi \cos \phi
l_kl_1\cos \theta - l_{hub}\cos \theta + \Omega l_kl_k\cos \chi \cos \phi
\end{bmatrix}
$$

(9)

where, $\Omega = \delta \chi/\delta \theta$ ,

$$
\frac{\delta \chi}{\delta \theta} = -k_3\sin \theta + k_4\cos \theta - k_2\cos \chi \sin \theta + k_2\sin \chi \cos \theta - k_4\sin \chi \sin \theta
$$

$$
\frac{\delta \chi}{\delta \theta} = k_2\sin \chi \cos \theta - k_2\sin \chi \cos \theta - k_4\cos \chi - k_4\cos \chi \cos \theta - k_4\sin \chi \sin \theta
$$

(10)

In order to obtain $\delta \vec{\lambda}/\delta \theta$, $\delta \vec{\lambda}$ must be determined. The virtual angular displacement of the knuckle between consecutive positions “$\delta \vec{\lambda}$” can be determined from the cross product of the unit vector on the knuckle $\vec{e}_k$ and the differential of this vector $\delta \vec{e}_k$ (Fig. 9).

$$
\vec{e}_k \times \delta \vec{e}_k = [\vec{e}_k][\delta \vec{e}_k] \sin(\pi/2 + \delta \lambda)\vec{e}_n
$$

(11)

Fig. 9. Virtual angular displacement of the knuckle
Since, $\delta \lambda$ is the virtual angular displacement, its magnitude is very small. Thus, $\sin(\pi/2 + \delta \lambda) \approx 1$. The other vectors’ magnitudes are $|\vec{e}_k^o| = 1$ and $|\delta \vec{e}_k^o| = \delta \lambda$. Remarking that, $\vec{e}_n^o$ is the unit vector perpendicular to plane containing $\vec{e}_k$ and $\delta \vec{e}_k$, Eq. (11) yields:

$$\vec{e}_k \times \delta \vec{e}_k = \delta \lambda$$ \hspace{1cm} (12)

Differentiating this vector w.r.t. $\theta$, the derivative of virtual angular displacement of the knuckle between consecutive positions can be obtained as:

$$\frac{\delta \lambda}{\delta \theta} = \frac{\vec{e}_k \times \delta \vec{e}_k}{\delta \theta}$$ \hspace{1cm} (13)

Where,

$$\frac{\delta \vec{e}_k}{\delta \theta} = \frac{1}{l_k} \begin{bmatrix} l_1 \sin \theta - \Omega l_u \sin \chi \cos \xi + \Omega l_u \cos \chi \sin \phi \sin \xi \\ -\Omega l_u \sin \chi \sin \xi - \Omega l_u \cos \chi \cos \xi \sin \phi \\ -l_1 \cos \theta + \Omega l_u \cos \chi \cos \phi \end{bmatrix}$$

7. Example 3

For the suspension given in Example 2, calculate anti-dive to braking force ratio ($\mu$) for a wheel travel of 0 to +55mm. The car given in Fig. 10 is braking with maximum deceleration of $1g$, where the coefficient of friction between tires and road is $\mu = 1$. The dimensions are $m = 0.96m$, $n = 1.44m$, $h_{cg} = 0.42m$ and mass of the car is $M_{car} = 950$kg. If sum of two front suspensions has equivalent spring stiffness of $k_{eqf} = 32kN/m$, calculate the maximum amount of dive during braking both with and without anti-dive effect.

Initially Eqns. (4)-(6) in the previous study (Tanık and Parlaktaş, 2015a) are solved. Then, by solving Eqns. (9), (10), (13), and (7), percentage of anti-dive to braking force ratio w.r.t wheel travel is determined and displayed in Fig. 11.

Fig. 10. Free-body diagram of a vehicle during braking

Fig. 11. Percentage of anti-dive to braking force ratio
It can be observed from Fig. 11 that the percentage of anti-dive to braking force ratio doesn’t vary too much throughout the wheel travel; $11.95 \leq \psi \leq 12.1$. Thus, by using these lower and upper limits let $\psi_{\text{avg}} \approx 12$. According to the free body diagram of the car given in Fig. 10, the force acting on the front wheels is:

$$N_F = \left( \frac{m + h_{cg}}{m + n} \right) M_{\text{car}} g$$  \hspace{1cm} (14)

For the static case the front end loading, that corresponds to the force for zero wheel travel is:

$$N_{F_{\text{static}}} = \frac{M_{\text{car}} mg}{m + n}$$

In order to determine dive of the suspension without an anti-dive effect, the deflection is calculated as:

$$Def_{w/o\_\text{antidive}} = \frac{(N_F - N_{F_{\text{static}}})}{k_{eqf}}$$  \hspace{1cm} (15)

After the given variables are substituted, its numerical value is determined as: $Def_{w/o\_\text{antidive}} = 0.051m$.

If the anti-dive effect is included to the formulation, the deflection is calculated as:

$$Def_{w\_\text{antidive}} = \frac{(N_F - N_{F_{\text{static}}} - 0.01\psi_{\text{avg}}F_{bb})}{k_{eqf}}$$  \hspace{1cm} (16)

Note that, for 1g deceleration $F_{bb} = \mu N_F$

After the given variables are substituted, its numerical value is: $Def_{w\_\text{antidive}} = 0.031m$. When the anti-dive effect is included it can be observed that the amount of deflection is reduced by 20mm or 40%.

In this example anti-dive angle $\phi$ is equal to 9°. However, by keeping same suspension geometry and setting $\phi = 0$, average anti-dive ratio is calculated as $\psi_{\text{avg}} \approx 0$, so the deflection cannot be reduced. For negative values of $\phi$, $\psi_{\text{avg}}$ also takes negative values, so deflection increases and that corresponds to anti-lift characteristics. Thus, for rear suspension negative values of $\phi$ should be selected for better anti lift characteristics.

It should be also noted that when the term “$r_wF_b$” is set to zero in Eq. (6) which indicates a vehicle system carrying its brake disc/drum on its chassis rather than knuckle, $\psi_{\text{avg}} \approx 0$ for a wide variety of the suspension geometry we tried. This indicates that the anti-dive effect is gained both with the braking moment created on the knuckle of a suspension and with positive values of $\phi$.

8. Conclusions

In this study, a novel “complete” analysis of the double wishbone suspension mechanism is presented. Initially, an analysis method is taken into consideration for variable steering input while keeping suspension travel fixed in a specific position. Then, kinematic analysis is performed for the ultimate case; both variable steering and variable wheel travel is investigated. With this analytical approach, camber, caster, and kingpin angles of a synthesized mechanism are displayed with respect to variable steering and wheel travel values. This mechanism is established in CATIA software and same results with the analytical model are obtained for every steering angle and wheel travel values. Thus, it is verified that the proposed analytical model can be used precisely for the kinematic analysis and design of both front and rear double wishbone mechanism instead of commercial software.

To the best of our knowledge, the amount of anti-dive rate hasn’t been calculated analytically in the literature. In this paper, a novel analytical approach is also proposed to calculate this effect. It is observed that amount of deflection can be reduced remarkably by assigning a convenient anti-dive angle.

During analysis with expensive CAD programmers, a suspension mechanism must be re-drawn each time when
dimensions, orientations and hard points are changed. However, with this analytical approach by simply changing the input variables of the code, results can be observed instantly. Mechanisms of different dimensions can be analyzed and parameters can be optimized swiftly. Furthermore, an optimization routine algorithm cannot be used with a CAD programmer. Thus, time and money consumed during design stage decreases significantly by the aid of this analytical approach.

References