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KINEMATIC OPTIMISATION OF AN ARTICULATED TRUCK INDEPENDENT FRONT SUSPENSION BY USING RESPONSE SURFACE METHODOLOGY

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Abstract
Conceptual kinematic design study of a double wishbone suspension that will be used as the front axle of a 40 metric tonnes capacity articulated heavy commercial vehicle is summarised. Design problem is basically established on two main goals namely, minimum deviations of the wheel track and the camber angle under all drive and load conditions. In the light of the design constraints such as the ground clearance of the tractor, chassis location, structural elements of the wheel-end package and the steel wheel, the possible design volume of the suspension was predicted. Subsequently a primary kinematic model of the suspension was built via Adams/Car™ multibody dynamics (MBD) software and the deviation characteristics of this model were computed in case of wheel jounce and rebound. Targeted optimal ranges of the kinematic parameters during the wheel travel were also obtained by using response surface methodology (RSM). For this purpose, a central composite design (CCD) based multiobjective optimisation process was performed to the primary model by using Adams/Insight™ tool of MSC.Adams® commercial software. 3D plots of the kinematic parameters including the effects of the wheel travel and wheel steering are presented for primary and optimised designs. Results showed that the optimal geometry of the double wishbone suspension obtained from the multiobjective optimisation study satisfies the track change limitation given in the literature.

Keywords: Independent front suspension (IFS), Design of experiments (DOE), Multibody dynamics (MBD), Multi-objective optimisation, Kinematic optimisation, Central composite design (CCD)

1. INTRODUCTION
Through their high loading capacity and ease of manufacturing, solid axles have a broad application area as heavy commercial vehicle suspension. Moreover, constant track and camber requirements have a much greater priority for commercial vehicles than for passenger cars due to the minimisation of tyre wear, rolling resistance and fuel consumption (Bramberger and Eichlseder, 1998 p.1). These targets can be highly attained via solid axle designs. An example of this system can be seen in Figure 1.a (Topaç et al., 2011 p.149). On the other hand, solid axles have higher unsprung mass in comparison with the independent suspensions. High unsprung mass affects the ride comfort, wheel-road contact and handling dynamics adversely (Gillespie, 1992 p.165), (Gysen et al., 2010 p.1159). In addition, this may cause structural noise problems. As a result of the comfort and control requirements, one of the main targets to be reached at the end of the design process of a vehicle suspension is to keep the unsprung mass as small as possible. In order to satisfy these requirements, independent suspensions are chosen as front suspensions of busses and trucks by the heavy commercial vehicle manufacturers increasingly (Timoney and Timoney, 2003 p.426). Applied sample of the truck IFS which was designed in the scope of this study is seen in Figure 1.b. These systems also have some advantages such as little space requirement, easier steerability, low weight and no mutual wheel influence which are important for good road-holding characteristics. Aforementioned advantages are most easily achieved by using double wishbone suspension (Reimpell et al., 2001 p.7). On the other hand, kinematic characteristics of the suspension should also be taken into account during the design process. For instance, variation range of the track change during the vertical wheel travel should not be higher than certain design limits. Deviation characteristic of the IFS kinematic parameters during the wheel travel is highly dependent on the hardpoint positions. Essentially, kinematic design of the independent suspension mechanism is determination of the hardpoint positions (Hwang et al., 2007 p.1).
In this work, optimal kinematic design study of a double wishbone suspension that will be used as 7100 kg capacity front axle of a 40 metric tonnes articulated truck was carried out by using multibody dynamics (MBD). In order to do that, hardpoint positions of the front axle which give optimal variation range of the kinematic parameters during the wheel travel were determined via multiobjective optimisation. Firstly, physical design limitations were determined by taking the structure of the vehicle into account. By this way, a primary kinematic model of the IFS was built. By using the initial hardpoint positions, a multibody model of the mechanism was performed by using MSC. Adams™ commercial software. Variation of the kinematic parameters such as camber (\(\sigma\)), track (\(s_{\text{RV}}\)) and kingpin inclination angle (\(\delta\)) were computed for parallel wheel travel and steering. After that, structural design constraints which constitute the possible design volume of the suspension mechanism were determined. The possible ranges of a hardpoint positions were determined by taking the design limitations such as the ground clearance, roll centre height, structural elements of the wheel-end package etc. into account. With the use of these limitations, a Design of Experiments-Response Surface Methodology (DOE-RSM) -based optimisation study was carried out via Adams/Insight™ multi-objective optimisation tool. By this way, optimal hardpoint locations which satisfy the design targets namely, minimum deviations of \(\sigma\) and \(s_{\text{RV}}\) during the wheel travel were obtained.

### 2. KINEMATIC DESCRIPTION OF THE IFS

General view and the kinematic parameters of the double wishbone suspension are seen in Figure 2.a. Here, \(O_y\) is the midpoint of the front track. In this model, double wishbone suspension is basically represented via 8 hardpoints, \(A_1\) to \(A_8\). The points \(A_1\) and \(A_2\), also give the axis of the suspension spring. Knuckle is in connection with the upper and lower wishbones via \(A_3\) and \(A_4\) spherical joints. \(A_3-A_4\) line represents the geometric steering axis. \(A_7\) is the intersection point of steering axis and wheel rotation axis that is described by the unit vector \(\{e_{\text{RV}}\}\). The unit vector \(\{e_1\}\) represents the steering axis. Wishbones are mounted to the vehicle body via bearings \(A_1\), \(A_2\), \(A_3\) and \(A_4\) which can be considered as revolute joints. Axes of the hard points \(A_1\), \(A_2\), \(A_3\) and \(A_4\) which are mounted on a sub frame are parallel to \(\xi\); the longitudinal axis of the vehicle. Figure 2.b also shows the kinematic model of the double wishbone suspension. Reference frame \(\xi-\eta-\epsilon\) (frame 1) is defined at the centre of the mass \(S_a\) of the vehicle body. In this model, U-V-W frame (frame 2) was also defined for each wishbone where subscripts O and U indicate upper and lower wishbones respectively. It can be assumed that \(A_3\) and \(A_4\) rotate around the points \(O_0\) and \(O_0\) which are at the origin of the U-V-W frames. Here, the unit vectors \(\{e_{\text{RV}}\}\) and \(\{e_{\text{VW}}\}\) also represent the rotation axes of the wishbones. Hardpoints \(A_9\) and \(A_{10}\) represent the tie rod. Tie rod and knuckle are connected via \(A_9\) ball joint. Steer angle of the wheel, \(\beta_1\) (Figure 3.a) is adjusted by the position of \(A_{10}\). Position vector of \(A_3\) can be written in terms of U-V-W frame as:

\[
[R_{A3}]_{2/1} = [O_0A_3]\{e_{\text{VW}}\}
\]

(1)

As a result of the vertical wheel displacement \(\epsilon_{\text{RV}}\), \(\{R_{A3}\}_{2/1}\) can also be expressed in terms of \(\xi-\eta-\epsilon\) frame as:

\[
[R_{A3}']_{1/1} = [O_{0U}']_{1/1} + [T_2']_{1/1} [R_{A3}]_{2/1}
\]

(2)

where, the sign '*' indicates the displaced position of the hardpoint.
Here, $[T_{12}]$, is also defined as the transformation matrix between U-V-W and $\xi$-$\eta$-$\varepsilon$ frames. Position of $A_4$ can be obtained from the solution of the following constraint equations (Kuralay, 1985 p.19), (Simionescu and Beale, 2002 p.818), (Hwang et al., 2007 p.2):

$$
(\xi - \xi_{A3})^2 + (\eta - \eta_{A3})^2 + (\varepsilon - \varepsilon_{A3})^2 = (\xi_{A4} - \xi_{A3})^2 + (\eta_{A4} - \eta_{A3})^2 + (\varepsilon_{A4} - \varepsilon_{A3})^2
$$

(3)

$$
(\xi - \xi_{OO})^2 + (\eta - \eta_{OO})^2 + (\varepsilon - \varepsilon_{OO})^2 = |O_O A_4||e_{vO}|
$$

(4)

By using the same notation, new positions of $A_7$, $A_8$, and $A_9$ can be computed. Camber angle and track variations caused by the wheel displacement $\varepsilon_{A8}$ are shown in Figure 3.b (Topaç, 2010 p.47), (Topaç et al., 2010 p.10).

Here, the displaced positions of the points $A_7$ and $A_8$ are shortly denoted as $A_7'$ and $A_8'$. In order to determine the change in i.e., camber angle, the new positions $A_7'$ and $A_8'$ can be compared with the initial positions $A_7$ and $A_8$. Hence the camber change can be written as the angle $\Delta \sigma$ between the projection of $A_7A_8$ and $A_7'A_8'$ onto the $\eta\varepsilon$ plane that is assumed as global frame as:

$$
\cos \Delta \sigma = \frac{[R_{A7A8}]_1 \cdot [R_{A7A8}']}{|R_{A7A8}||R_{A7A8}'|}
$$

(5)
Kinematic parameters $\delta$, $\varepsilon$, and $\beta$ can also be represented by using the same formulation (Blundell and Harty, 2006 p.43). Moreover, by using the $\xi$ and $\eta$ co-ordinates of the points F and S which are given in Figure 4, the scrub radius, $r_s$ and the caster arm, $n_c$ can be written respectively as (Kuralay, 1985 p.28):

$$ r_s = |\eta_F - \eta_S| $$

$$ n_c = |\xi_F - \xi_S| $$

3. MULTIBODY MODEL

Although a parallel arm mechanism satisfies the design targets such as minimum deviations of $\sigma$ and $s_{RV}$ in jounce and rebound modes of the wheel, this design configuration is not applicable in many cases because of the design constraints. Figure 4.a shows the produced final prototype of the IFS which was designed in the scope of this study. Figure 4.b also summarises some of the design limitations which were used for the kinematic design and optimisation phases. Initial positions of the hardpoints $A_3$ and $A_4$ were selected in the light of the design constraints such as the brake system, steel wheel, steering knuckle and connection element for the suspension spring. Moreover, the effective distance $c$ between $A_3$ and $A_4$ should be as large as possible to achieve small reaction forces in the link bearings and to limit the deformation of the rubber elements fitted (Reimpell et al., 2001 p.7). Primary positions of $A_3$ and $A_4$ were selected by considering the chassis position, design of the subframe, ground clearance of the vehicle and the roll centre height of the front axle. It is known from the literature that the roll centre of a double wishbone suspension should be as low as possible to obtain minimum deviations of $\sigma$ and $s_{RV}$ during the wheel travel (Reimpell, 1988 p.206), (Reimpell, 2001 p.156).

By using the selected initial hardpoint positions, a three dimensional multibody model of the draft suspension design was composed by using Adams/Car™ module of MSC. Adams™ as shown in Figure 5.a. In this model, the structural elements namely, upper and lower wishbones, knuckle arm and tie rod were assumed as rigid. For the pre-load of the suspension spring used in this model, total sprung load of the 7 metric tonnes loading capacity front axle was calculated in static loading condition by using the equation:

$$ P_s = m_g (m - m_u) g \quad \text{(N)} $$

Here, $m_s$ is sprung mass per front wheel. $m$ is total mass that is carried by a single wheel. Unsprung mass per wheel of the front axle which includes wheel-end, brake system and tyre is assumed as $m_u = 250$ kg. In this design, suspension spring is directly mounted to the kingpin via a revolute joint (Figure 4). By this was spring rate of the suspension can be assumed as $i_p=1$ (Matschinsky, 2007 p.96,97), (Blundell and Harty, 2006 p.228-231). Upper and lower wishbones of the truck suspension are also connected to the vehicle body via elastic bushings at $A_1$, $A_2$, $A_5$ and $A_6$ hardpoints. It is known from the literature that elastic bushings affect the variation range of the kinematic properties of the suspension and the dynamics of the vehicle.
Therefore, bushing elasticity was also taken into the account in this model. A schematic for wishbone bushings can also be seen in Figure 5.b.

3. MULTI OBJECTIVE OPTIMISATION

In this work, DOE-RSM was employed to obtain the proper positions of the IFS hardpoints for the given design targets. For the optimisation process, Adams/Insight™ multiobjective optimisation tool was utilised. Principal target of the response surface experiments is to obtain a proper model to estimate and analyse the relationship between design variables and system response. For a second order response surface, regression model can be expressed as (Han and Park, 2004 p.170):

$$\begin{align*}
\epsilon & = \beta_0 + \sum_{i=1}^{k} \beta_i x_i + \sum_{i=1}^{k} \beta_{ij} x_i x_j + \epsilon 
\end{align*}$$

This model can also be expressed in matrix form as:

$$\{y\} = [X][\beta] + \{\epsilon\}$$

Here, \(\{y\}\) is vector of observations, \([X]\) is the model matrix, \([\beta]\) is the vector which includes the interception parameter \(\beta_0\) and the partial regression coefficients and \(\{\epsilon\}\) is the vector of random errors (Myers et al., 2009 p.46). Estimated value of \([\beta]\) which minimises \(\epsilon\) can be expressed as:

$$\hat{\{\beta\}} = ([X]^T[X])^{-1}[X]^T\{y\}$$

ADAMS/Insight™ uses the method of least squares to estimate the \(\beta\) coefficients in the regression model (Aydin and Ünlüsoy, 2012 p.747), (ADAMS/Insight™, 2013 p.43). In this study, Central Composite Design (CCD) type which is offered in the design specification table of Adams/Insight™ was utilised for this purpose. In order to figure out the kinematic multi-objective optimisation problem, at the first stage, parallel wheel travel simulation was applied to the initial kinematic model for \(e_{a8} = \pm 100\text{mm}\) vertical wheel displacement in Adams/Car™. Analysis was completed in 100 steps. In the next step, \(\sigma_{SRV}\) and \(\delta\) were defined as design objectives. Absolute maximum values of the design objectives obtained from initial design were also defined in Adams/Car™. Assembly of the initial design was exported to Adams/Insight™. In the light of the design constraints, the variation range of the \(\eta\) components of the hardpoints \(A_1, A_2, A_5\) and \(A_6\) were assumed as \(\pm 30\text{ mm}\) relative to the initial value. Variation range was chosen as \(\pm 15\text{ mm}\) for the rest of the factors. Investigation strategy was chosen as DOE Response Surface. By this way, a total number of 88 design samples (trials) were generated for 8 factors. Successive analyses were performed and the results of these trials were reported in workspace. Flow diagram of the optimisation process is given in Figure 6 (Topaç et al., 2015, p.35).
4. RESULTS AND DISCUSSION

4.1. Optimisation of camber and track alterations

Histograms shown in Figure 7 were generated by using the results provided from the workspace. These diagrams present the dissipation of the design points for the various values of the kinematic parameters, namely, \( \sigma \), \( s_{RV} \) and \( \delta \). Diagrams give the number of design samples for a certain value of a parameter. Main effect plots for \( \sigma \), \( s_{RV} \) and \( \delta \) are also seen in Figure 8. In the light of the design limitations, suspension geometry which gives minimum variation of \( \sigma \) and \( s_{RV} \) during jounce and rebound was chosen by the software among the design samples.

Parallel wheel travel simulation for \( \varepsilon_{A8} = \pm 100 \text{mm} \) and steering simulation for \( \beta_{L} = \pm 50^\circ \) were applied to the optimised MBD model simultaneously. These simulations are also illustrated by using the solid model of the IFS in Figure 9. Results were generated via Adams/PostProcessor™. Camber deviation ranges for \( \varepsilon_{A8} = \pm 100 \text{mm} \) and \( \beta_{L} = 0^\circ \) were obtained for original and optimised designs as \((-2.6^\circ; +1.8^\circ)\) and \((-1.8^\circ; 1.3^\circ)\) respectively. It is known from the literature that total track variation \( \Delta s_{RV} \) of an IFS should not be higher than 25 mm for \( \varepsilon_{A8} = \pm 40 \text{mm} \) (Reimpell, 1976 p.149, 150). \( \Delta s_{RV} \) was calculated as 42 mm for the primary design and 25 mm for the optimised design as seen in Figure 10. In order to improve the ease of the steerability, variation of the kingpin inclination angle and the scrub radius are also reduced. 3D response plots obtained from MATLAB® are also given in Figure 11.a and 11.b. Here, the comparisons of the camber angle and kingpin angle variations for initial and optimised designs are presented as functions of \( \varepsilon_{A8} \) and \( \beta_{L} \).

The main kinematic parameter that affects the roll dynamics of the vehicle body is the roll centre height, \( h_{MV} \). Determination of \( h_{MV} \) is given in Figure 11.a. Roll centre of the optimised design was obtained as \( h_{MV} = 58 \text{mm} \) at \( \varepsilon_{A8}=0 \text{mm} \) as seen in Figure 11.b. Results of the multibody analyses indicated that reducing the \( h_{MV} \) decreases the variation of \( \sigma \) and \( s_{RV} \) which satisfies the design targets. However, lower \( h_{MV} \) also increases the roll moment and the roll angle \( \psi \) of the vehicle body during a cornering manoeuvre.
Figure 8. Main effect plots for $\sigma$, $s_{RV}$, and $\delta$

![Figure 8](image)

Figure 9.a. Wheel travel mode b. steering mode

![Figure 9](image)

Figure 10. a. Comparison of the initial and optimised models in wheel travel mode (exaggerated) b. Comparison of the maximum total track change range for initial and optimised designs ($\varepsilon_{AB} = \pm 40$ mm).
4.2. Minimisation of toe deviation

One of the main targets in the kinematic design of the independent front suspensions is to minimise the $\beta_v$ (Figure 3.a) variation during the service. By this way uncontrolled steering effects can be prevented. In order to do that, angular position of the tie rod ($A_{y}-A_{10}$) should be determined with sufficient accuracy. Figure 13.a summarises the kinematic requirement to adjust the appropriate tie rod position (Reimpell, 1974 p.150). In this study, the correct position of the hardpoint $A_3$ for the initial design is determined by using this principle. First, the instant centre $P_3$ was assigned by using the positions of $A_2$, $A_3$, $A_4$ and $A_6$. Extension of the path $A_2-A_{10}$ should also intersect $P_3$, $\alpha$ is the angle between $A_4$-$P_2$ and $A_3$-$P_3$ lines. The instant centre $P_3$ was also determined by using $A_2$, $A_3$, $A_4$ and $A_6$ as seen in Figure 13. Then the third instant centre $P_3$ was found by using the extension of the lines $A_3-A_{10}$ and $P_2-P_3$. Here, the angle between $P_2$-$P_3$ and $P_2$-$P_3$ lines should be equal to $\alpha$. Finally, the correct position of the hardpoint $A_3$ was found where the extensions of $A_{10}$-$P_3$ and the extension of $A_4$-$P_3$ intersect. Figure 13.b also shows the toe angle variation during jounce and rebound. For the final kinematic design, the calculated variation range of $\beta_v$ is $(-0.02^\circ; +0.02^\circ)$.

Figure 11. Response surfaces for a. camber angle b. kingpin angle

Figure 12.a. Determination of the roll centre b. Roll centre height deviations for initial and optimised kinematic models
Figure 13. a. Determination of the angular position of the tie rod b. $\beta_V$ deviation of the final IFS geometry during wheel travel

4. CONCLUSIONS

In this work, multi objective kinematic optimisation stage of an articulated truck IFS design study is presented. At the first stage, primary hardpoint positions of the IFS were determined by taking the physical design constraints of the truck front axle. By using this draft geometry, a primary kinematic model of the suspension was built via Adams/Car$^\text{™}$ multibody dynamics (MBD) software. The deviation characteristics of the kinematic properties of this model were computed for wheel jounce and rebound. With the use of the Design of Experiments- Response Surface Methodology (DOE-RSM), targeted optimal ranges of the kinematic parameters for the wheel travel were also obtained. For this purpose, a central composite design (CCD) - based multiobjective optimisation process was also performed to the primary model by using Adams/Insight$^\text{™}$ tool of MSC.Adams$^\text{®}$ commercial software. Results obtained from this study are summarised below:

- Results of the optimisation study showed that the camber variation range during the wheel travel was reduced about 30% in comparison with the initial model.
- The final geometry of the truck IFS prototype obtained from the multiobjective optimisation study satisfies the track change limitation given in the literature for the $\varepsilon_{A8} = \pm 40 \text{ mm}$ range.
- Toe angle variation range of the suspension system was obtained for the final kinematic model as $\beta_V = (-0.02^\circ; +0.02^\circ)$ in the range of $\varepsilon_{A8} = \pm 100 \text{ mm}$

Force analyses, mechanical design, mechanical optimisation and the manufacturing process of the final prototype of the truck IFS was carried out by using the results obtained from this study.

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