Parameterisation analysis for a four in-wheel-motors drive and four wheels independent steering electric vehicle based on multi-body inverse kinematics

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Abstract: In this paper, a kinematics model with 14 degrees of freedom (DOF) is built for a four in-wheel-motors drive and four wheels independent steering electric vehicle based on the multi-body dynamics theory and modelling integration method. Considering the slippage control of wheels, the optimal speeds for the individual wheel and the optimal steering angles can be obtained for the desired trajectory through inverse kinematics analysis of the model. A typical case has been simulated and the useful driving parameterisation analysis for the vehicle can be obtained. The results of this paper can provide the basic and standard data for the coordination control among four driven in-wheel-motors and four all wheels independent steering motors according to the desired curve motion and will be validated by future test data results.

Keywords: 4WID-4WIS; kinematics model; inverse kinematics analysis; simulation; modelling integration method; parameterisation analysis; coordination control; in-wheel-motors.

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1 Introduction

In recent years many eco-vehicles have been researched, such as hybrid cars, fuel cell cars and electric cars. With the rapid development of many technologies such as electronic integration, advanced motor control, fuel cell and GPS navigation, the four independent in-wheel-motors drive and four wheels independent steering electric vehicle (4WID-4WIS) will become the trend of vehicle research focuses in the future.

In the process of 4WID-4WIS EV development, many similar and initial type vehicles have been studied and developed, for example front or rear wheel drive vehicle with two in-wheel-motors vehicle (2WD) and four in-wheels-motors drive vehicle (4WD). Kim and Park (2010) propose a control algorithm for an independent motor-drive vehicle with traditional steer system. The performance of the control algorithm is evaluated with a test car results, the vehicle with traditional steer system. The performance of the control algorithm is evaluated with a test car results, the control algorithm built improves vehicle performance.


By describing the configuration and studying the topological structure of 4WID-4WIS in Section 2 of the paper, the forward-inverse kinematics of the vehicle has been analysed and kinematics model has been developed by using multi-body dynamics theory and modelling integration method. In Section 3, vehicle movement of 4WID-4WIS for a typical condition has been simulated. Through the simulation results analysis, driving parameterisation analysis for 4WID-4WIS vehicle can be acquired. These useful results can provide the basic and standard data to the precise coordination control among driven motor speeds, steering motor angles and vehicle body for the desired path tracking.

2 4WID-4WIS kinematics modelling

In multi-body kinematics theory, the kinematics analysis that starts from driven motors motion to vehicle body motion is defined as the forward kinematics analysis. The inverse kinematics analysis is the process from vehicle body motion to motors motion. Multi-body kinematics modelling for 4WID-4WIS vehicle can provide united kinematics analysis, and simulation results are necessary for wheels coordination control, vehicle trajectory tracking and set-point control, etc.

2.1 4WID-4WIS multi-body kinematics analysis

Figure 1 shows the 4WID-4WIS vehicle manufactured by our research group. $OXYZ$ and $O_0X_0Y_0Z_0$ are the global coordination and body coordination located at the mass centre of the vehicle, $O_iX_iY_iZ_i$ ($i = 1, 2, \ldots, 22$) are local coordination located at the mass centre of each body. In kinematics analysis, the various joint constraints combined together in the model can determinate the motion characteristics of the system. There are 21 bodies defined in kinematics model of the vehicle shown in Table 1.
The developed 4WID-4WIS vehicle and coordination definition (see online version for colours)

Table 1  Bodies defined in 4WID-4WIS vehicle

<table>
<thead>
<tr>
<th>B</th>
<th>Ground</th>
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<tbody>
<tr>
<td>B0</td>
<td>Body</td>
</tr>
<tr>
<td>B1</td>
<td>Front left equivalent upper arm link of FL suspension</td>
</tr>
<tr>
<td>B11</td>
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<td>B10</td>
<td>Front left road wheel</td>
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<tr>
<td>B20</td>
<td>Rear left road wheel</td>
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</table>

The developed 4WID-4WIS vehicle and coordination definition (see online version for colours)

The topological structure of 4WID-4WIS vehicle is presented in Figure 2. There are three kinds of constraints, revolute, spherical and general. The degrees of freedom (DOF) of 4WID-4WIS vehicle is

\[ \delta = 6 \times n - 5 \times n_j1 - 3 \times n_j2 - 0 \times n_j3 = 14 \]

n is the total number of bodies, \( n_j1 \) is the total number of revolute joint, \( n_j2 \) is the total number of spherical joint, \( n_j3 \) is the total number of general joint.

2.2 Quarter vehicle modelling

The 4WID-4WIS vehicle manufactured is symmetrical in structure chassis between left and right, has same suspension structure at front and rear vehicle. So, left front quarter of vehicle (LF) is modelled firstly.

In multi-body theory (Kuan and Jianjun, 2012; Jiazhen, 2009), a revolute joint \( H_j \) between the body \( B_\alpha \) and \( B_\beta \) can be depicted as

\[
\Phi_j = \begin{bmatrix} \Phi_j(\theta^\alpha, \theta^\beta, \psi^\alpha, \psi^\beta) \end{bmatrix} = 0
\]

Matrix \( \Phi_j(\theta^\alpha, \theta^\beta, \psi^\alpha, \psi^\beta) = 0 \) has three analytical equations. Matrix \( \Phi_j(d_{\alpha}, d_{\beta}) \) is revolute constraint between \( B_\alpha \) and \( B_\beta \) in fixed rotate direction,

\[
\Phi_j(d_{\alpha}, d_{\beta}) = \begin{bmatrix} d_{\alpha}^T \theta_{\alpha} A_{\alpha} d_{\alpha}' \end{bmatrix} = 0
\]

\( d_{\alpha}' \) is local matrix of unit vector of \( B_\alpha \) in fixed rotate direction, \( d_{\beta}' \) are local matrix of unit vector of \( B_\beta \) in fixed rotate direction. \( d_{\alpha}', d_{\beta}' \) are vertical each other. Constraint variables of quarter model of vehicle are listed in Table 2.
Parameterisation analysis for a four in-wheel-motors drive and four wheels independent steering electric vehicle

Table 2 Variables values of quarter kinematics modelling

<table>
<thead>
<tr>
<th>$H_i$</th>
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<th>$\beta$</th>
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</tbody>
</table>

Variables matrix of each body $B_i$ is

$$q_i = [x_i, y_i, z_i, \varphi_i, \theta_i, \psi_i]^T$$

Variables matrix of quarter kinematics model of vehicle is

$$q_{1/4} = \begin{bmatrix} q_0^T & q_{LF}^T \end{bmatrix}^T$$

$$= \begin{bmatrix} q_0^T & q_1^T & q_2^T & q_3^T & q_4^T & q_5^T \end{bmatrix}^T$$

Constraints equations are

$$\Phi_{LF} = \begin{bmatrix} \Phi_{H1} \\ \Phi_{H2} \\ \Phi_{H3} \\ \Phi_{H4} \\ \Phi_{H5} \\ \Phi_{H6} \end{bmatrix} = 0$$

(4)

There are 28 dependent analytical equations and 36 variables. Constraints equations can simply be rewrote as $\Phi_{LF} = C_{LF} q_{1/4} + D_{LF}$, $C_{LF}$ and $D_{LF}$ are Jacobian matrix and constant matrix of quarter kinematics model of vehicle.

2.3 Vehicle kinematics modelling integration

By using integration modelling method, variable matrix $q$ of whole vehicle is $q = [q_0^T, q_{LF}^T, q_{RF}^T, q_{LR}^T, q_{RR}^T]^T$, it includes 126 variables.

Main constraints equations of vehicle is

$$\Phi^K = \begin{bmatrix} \Phi_{LF} \\ \Phi_{RF} \\ \Phi_{LR} \\ \Phi_{RR} \end{bmatrix} = 0$$

(5)

There are 112 dependent analytical equations and 126 variables. Main constraints equations can simply be rewrote as $\Phi^D = C q + D$, $C$ and $D$ are Jacobian matrix and constant matrix of kinematics model of the vehicle.

2.4 Vehicle kinematics model solution

Main constraints equations of vehicle $\Phi^K = 0$ include 112 dependent analytical equations and 126 variables. So there is no definite solution.
In inverse kinematics analysis, some hypotheses can be given. Defining slip ratio $\lambda$ of each driven wheel varies in certain limits, driven wheel variables $\theta_i (i = 5, 10, 15, 20)$ (reflecting in-wheel-motors rotations) the matrix becomes

$$\Phi_\theta = \begin{bmatrix}
(1-\lambda_1) \dot{h}_1(t) - \dot{x}_1(t) \\
(1-\lambda_2) \dot{h}_2(t) - \dot{x}_2(t) \\
(1-\lambda_{12}) \dot{h}_{12}(t) - \dot{x}_{12}(t) \\
(1-\lambda_{16}) \dot{h}_{16}(t) - \dot{x}_{16}(t)
\end{bmatrix} = 0$$  \hspace{1cm} (6)

Each steering carrier variable is $\psi_i (i = 5, 11, 17, 23)$ (reflecting steering motors rotations). They should have no interference with body in geometry, constraint matrix is

$$\Phi_\psi = \begin{bmatrix}
\dot{x}_4 - \dot{x}_6(t) \\
\psi_4 - \psi_6(t) \\
\dot{x}_{12} - \dot{x}_{16}(t) \\
\psi_{12} - \psi_{16}(t)
\end{bmatrix} = 0$$  \hspace{1cm} (7)

Supposing variables matrix of body is known as certain functions

$$\Phi_{\theta_0} = \begin{bmatrix}
\dot{x}_0 - \dot{x}_4(t) \\
\psi_0 - \psi_4(t) \\
\dot{x}_6 - \dot{x}_{12}(t) \\
\psi_6 - \psi_{12}(t)
\end{bmatrix} = 0$$  \hspace{1cm} (8)

From (6) to (8) can get driven constraint equations of vehicle. There are 14 independent equations.

$$\Phi^D = \begin{bmatrix}
\Phi_\theta \\
\Phi_\psi \\
\Phi_{\theta_0}
\end{bmatrix} = 0$$  \hspace{1cm} (9)

(5) and (9) can be integrated into constraints equations of whole vehicle

$$\Phi = \begin{bmatrix}
\Phi^K \\
\Phi^D
\end{bmatrix} = 0$$  \hspace{1cm} (10)

It includes 126 variables and 126 independent equations. It can be differential, then velocity constraints matrix is

$$\Phi_\theta \ddot{q} = -\gamma$$  \hspace{1cm} (11)

Expression (11) can be derivative, acceleration constraints matrix is

$$\Phi_\theta \dddot{q} = -\gamma$$  \hspace{1cm} (12)

(11) and (12) is linear algebra equations, $\Phi_\theta$ is Jacobian matrix of constraints equations of vehicle

$$\Phi_\theta = \begin{bmatrix}
\Phi^K \\
\Phi^D
\end{bmatrix},$$

$$\Phi = \begin{bmatrix}
\Phi^K \\
\Phi^D
\end{bmatrix}$$  \hspace{1cm} (13)

Numerical iterative process of inverse kinematics of 4WID-4WIS EV as follows:

Step 1 $\ddot{q}_{0\theta}$ is known at start time $t = t_0$, by using Newton-Raphson iterative method, numerical values of $\ddot{q}_{1\theta}$ can be got at $t_1 = t_0 + \Delta t$.

Step 2 By calculating (13), numerical values of $\gamma$ and $\ddot{q}_{1\theta}$ can be got, then by calculating (11), numerical values of $\ddot{q}_{1\theta}$ can be got.

Step 3 By using numerical values of $\ddot{q}_{1\theta}$ and $\dddot{q}_{1\theta}$ in (13), numerical values $\gamma$ can be got, through calculating (12) numerical values of $\dddot{q}_{1\theta}$ can be got.

Step 4 By choosing iterative time $T$ and iterative step $\Delta t$, $t_1 = t_0 + \Delta t$, $t_2 = t_1 + \Delta t$, $\cdots$, $T$, and after numerical iterative process circulation, vehicle kinematics model numerical simulation can be solved.

The flow chart of numerical iterative process is in Figure 3.

4 4WID–4WIS inverse kinematics analysis

In Section 2, vehicle kinematics model has been developed. By using software Adams/solver, 4WID–4WIS vehicle movement for the expected trajectory can be simulated and useful results can be acquired for many subsystems control of 4WID–4WIS dynamics in future research. Curving and straight line driving are two typical cases for the vehicle. Ideal driving condition needs no slip for every road wheels, and requires eight motors motion and vehicle body motion to coordinate each other precisely. Because of wheels difference on path radius and running velocities, the motion coordination among wheels must be analysed to meet the need for the control algorithms of many subsystems of the vehicle.
### 4.1 Typical condition simulation analysis

We choose the 5 m radius curve path, suppose that the 4WID-4WIS runs at constant velocity 6 km/h, motion trajectory can be obtained in Figure 4. In the end process of travelling curve, the mass centre of front left driven wheel deviates from its ideal path maximally, the deviation distance is 180.42 mm, and the deviation distance of the mass centre of front left driven wheel is 139.58 mm. At the beginning process of travelling curve, the mass centre of rear left wheel deviates from its ideal path maximally, the deviation distance is 1460.56 mm, and the deviation distance of the mass centre of rear right driven wheel is 113.26 mm. In the range of 15.96°~63.88° of travelling curve, or in the distance 1,413.84 mm~4,717.92 mm of mass centre of 4WID-4WIS in direction X, the trajectory of same lateral wheels is almost same. The state variables of the vehicle such as velocities change gradually in Figure 5 and mean no or little slippage of four driven wheel, the vehicle is in stable condition.

Figure 4 shows the velocity variations of mass centre of the vehicle and four driven wheels in curve travelling. At the beginning of curving, the velocities of mass centre of four wheels change violently comparing with that of the vehicle, and may give raise to wheels slip at this moment in extreme conditions. In order to finish 5 m radius curving, four driven wheels are required different velocities, average velocity of rear right wheel need be changed maximally from 1,666.67 mm/s to 1,831.96 mm/s, rear left wheel need be changed minimally from 1,666.67 mm/s to 1,588.69 mm/s. Figure 6 show the relative velocity for mass centre of four driven wheels to the mass centre of 4WID-4WIS.

By given the slip ratios of four wheels and supposing same rolling radius for four driven wheels, the model built can also calculate the rolling angular velocity of four driven wheels and angle of four steering wheels in the path.
tracking condition shown in Figure 7. Average angular velocity of rear right wheel need be changed maximally from 4.76 rad/s to 5.23 rad/s, rear left wheel need be changed minimally from 4.76 mm/s to 4.53 rad/s.

Figure 6  Relative velocity of four driven wheels to mass centre of 4WID-4WIS (see online version for colours)

Figure 7  The angular velocity and angle of four driven wheels (see online version for colours)

These inverse kinematics simulation results in Figures 5 and 7 can become the control velocity aims of driven wheel motors in order to finish 5 m radius curving.

The developed model can also calculate the steering angle of four wheels in the 5 m radius curve tracking in Figure 8. At the begin of curving $t = 0$ s, the angles of four steering motors change violently and get −0.4798 rad or 27.49° for rear left motor, −0.3388 rad or 19.41° for rear right motor, 0.4507 rad or 25.82° for front left motor and 0.3174 rad or 18.19° for front right motor. In the curving process, average steering angles of four wheels relative to body of 4WID-4WIS are −0.3166 rad, −0.2443 rad, 0.2951 rad and 0.2333 rad. Angle change of rear left wheel and front left wheel are greater than other two wheels in the whole process of driving radius curve. At the time $t = 5.32$ s the angles of four steering motors become zero, and finish curve travel.

4.2 Parameterisation analysis for the 4WID-4WIS

By using of 5 m radius curve path and constant velocity 6 km/h, Figures 5 and 6 show the simulation results of velocity or relative velocity of mass centre of the vehicle and four driven wheels. By parameterised running velocity, or choosing different constant velocities, parameterisation analysis for the steering angle and relative steering angles for four steering motors also can be shown in Figure 9. Parameterisation analysis for relative velocity of four driven wheels to mass centre of 4WID-4WIS can be obtained in Figure 10. We can get following results

- No matter how we vary driving speed, in order to finish 5 m radius path, the steering angles and relative steering angles for four steering motors of four driven wheels keep their constant values (shown in Figure 9).
- With the increasing running velocities equally, or choosing 6 km/h, 10 km/h, 14 km/h, 18 km/h, 22 km/h and 26 km/h, relative velocities of four driven wheels themselves to mass centre of 4WID-4WIS also increase equally, such as incremental quantity is 197.1 mm/s for mass centre of rear right driven wheel, and −89.2 mm/s, −162.8 mm/s and −105.8 mm/s for other three driven wheels themselves. Rear right driven wheel’s increment is maximally. So, rear right driven wheel of 4WID-4WIS may occur to slip most possibly in some extreme conditions (shown in Figure 10).
- With the increasing running velocities equally, the times of finishing curve path are 4.46 s, 2.66 s, 1.91 s, 1.48 s, 1.21 s, 1.03 s, by calculating deceleration trend of time values, we can acquire that the convergence time is 0.85 s, this time means maxim running velocity is 47.3 Km/h or so. If exceeding that speed, the vehicle cannot finish 5 m radius curving. This phenomenon means that kinetic energy of 4WID-4WIS is redundant. In order to finish curve path, brake operation must be applied, so, redundant energy can be used by regenerative device sets (shown in Figure 10).
Figure 9  Parameterisation analyses for the steering angle and relative steering angle for four steering motors (see online version for colours)

Figure 10  Parameterisation analysis for relative velocity of four driven wheels to mass centre of 4WID-4WIS (see online version for colours)
When 4WID-4WIS vehicle has been manufactured, its mass centre location is also fixed in unload condition. Because of different load, such as passengers or goods, mass centre location of 4WID-4WIS may be changed. By parameterised mass centre location of 4WID-4WIS, relative velocities of four driven wheels and the relative steering angles of four steering motors to mass centre of 4WID-4WIS can have been simulated in Figures 11 and 12.

- With deviation value of mass centre location varying equally, or choosing +200 mm, +100 mm, 0 mm, –100 mm and –200 mm, velocity differences of four driven wheels themselves to mass centre of 4WID-4WIS in Figure 11 show that when deviation value increase, or mass centre move forward, the velocity of rear right driven wheel increase. So, mass centre of 4WID-4WIS moves back in direction X can reduce slippage trend of rear right driven wheel which most possibly happens in high speeds (shown in Figure 11).

- When mass centre of 4WID-4WIS moves forward, relative steering angle of rear left steering motor increase slightly more than that of rear right steering motor. and that of front left steering motor decrease slightly more than that of front right steering motor and vice versa (shown in Figure 12).

5 Conclusions

In this paper kinematics model of 4WID-4WIS vehicle has been developed by using multi-body dynamics theory and integration modelling method. Many conclusions have been acquired.

By given slip ratio of driven wheels, many typical conditions such as cornering motion, line driving of 4WID-4WIS vehicle can be simulated for the expected trajectory in this model built.

By using parameterisation method in the model, driving parameterisation simulation of 4WID-4WIS can be obtained and has been analysed for one typical curving condition. The useful simulation results can provide the EV with guidance on precise coordination control among driven motor speeds, steering motor angles and vehicle motion for this typical curving, can predict maxim running velocity for stability and can decide regenerative control strategy.

By choosing different radius and different driving speeds, the simulation results have been collected and can be developed into driving database of 4WID-4WIS. Using fitting and looking up for method, driving database or driving MAPS can shows the movement characteristics of the 4WID-4WIS vehicle, and provide the EV with guidance on precise coordination control of vehicle motion.
The simulation results will be validated by future test of 4WID-4WIS vehicle manufactured by our research group. By changing the topologic form of 4WID-4WIS, the model built in the paper can also be reformed and simulate the movement characteristics of the 4WID-4WIS vehicle for travelling some extreme curve motions such as slalom motion, fault motor motion, etc. The research results will be released in the future.

The works of the paper is the preliminary research stage to 4WID-4WIS vehicle control for desired tracking path motion, set point motion, etc.

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