Fuzzy Based Stability Enhancement System for a Four-Motor-Wheel Electric Vehicle

Farzad Tahami
Jovain Electrical Machines Co.

Reza Kazemi
IKCO Automobiles

Shahrokh Farhanghi
University of Tehran

Behzad Samadi
IKCO Automobiles

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ABSTRACT

The stability of a four motor-wheel drive electric vehicle is improved by independent control of wheel torques. An innovative Fuzzy Direct Yaw Control method together with a novel wheel slip controller is used to enhance the vehicle stability and safety. Also a new speed estimator is presented in this paper, which is used for slip estimation. The intrinsic robustness of fuzzy controllers allows the system to operate in different road conditions successfully. Moreover, the ease to implement fuzzy controllers gives a practical solution for vehicle stability enhancement.

INTRODUCTION

Motor-wheels are the revolutionary new electric drive systems that can be housed in vehicle wheel assemblies. All-Wheel-Drive systems have been recognized as a break-through concept that will have a major impact on future electric and hybrid vehicle design. The drive system is versatile and can be configured for all types of electric vehicles including battery only, plug-in hybrid, autonomous hybrid and fuel cells. Motor-wheels permit packaging flexibility by eliminating the central drive motor and the associated transmission and driveline components, including the transmission, the differential, the universal joints and the drive shaft. In-wheel motors provide many advantages in manufacturing electric vehicles such as:

- Built-in 4WD capability.
- More tractive force and braking regenerative power due to all-wheel-drive capabilities.
- More space for the installation of batteries, or auxiliary power unit for fuel cells.
- Better weight distribution.
- Low floor possibility for urban buses due to elimination of the axle and differential.
- In-wheel motors also provide another important aspect of such a vehicle. That is the ability of vehicle dynamic control to assist the driver with path correction, thus enhancing cornering and straight-line stability and providing enhanced safety. In fact, an electric vehicle with independent driven wheels provides another steering control input, i.e. the torque steering. Controlling of the yaw rate of a vehicle by utilizing this steering method is usually addressed as Direct Yaw-moment Control (DYC). It has been proved that DYC is more effective in enhancing vehicle stability than four wheel steering [1]. Actually, the yaw moment resulting from differential longitudinal tire force in the left and the right wheels is insignificantly influenced by lateral acceleration. On the contrary, the yaw moment generated by four-wheel steering decreases as the lateral acceleration increases [2].

In this paper, a control strategy for a driver assist stability system for a four in-wheel drive electric vehicle with a conventional front wheel steering is presented. The system comprises of a fuzzy logic controller in order to control the yaw rate. The disturbances in yaw moment are counter-balanced by a differential torque applied to the driven wheels of both sides. Another fuzzy controller is employed in order to prevent the wheels to enter saturation region due to the additional torque applied by the yaw controller.

Fuzzy controllers have been already proved to have good performance in two-motor-wheel drive vehicles [3, 4].
VEHICLE MODELING

VEHICLE MODEL - A fourteen-degree of freedom model is used for the simulation purposes. Where, 6 degrees are devoted to the chassis motion and eight degrees are assigned to the wheels rotational and vertical movement. Figure 1 shows different coordinates that are used for the modeling purpose.

Figure 1. Coordinate systems for vehicle modeling

Where $u$, $v$ and $w$ are associated with longitudinal, lateral and vertical velocities and $\psi$, $\phi$ and $\theta$ denote the yaw, roll and pitch angles.

Suspension Model – A model of the suspension system is shown in Figure 2. It is assumed that the suspension is independent for each wheel. The suspension model consists of a spring and damper. Parameters $K_s$ and $C_s$ correspond to the spring stiffness and the damping factor. A virtual spring and damper assembly (corresponding to $K_{ui}$ and $C_{ui}$) is considered for the tire elasticity. $Z_s$ is the height of the center of gravity of the sprung mass. $Z_{ui}$ is assigned for the height of unsprung mass and $Z_{ri}$ is due to road roughness.

Figure 2. The model of suspension system

Tire Model – The accuracy of the simulation results is predominantly determined by the accuracy of the tire model used. The desired simulation for the application herein concerned can be carried out by a steady state tire characteristics. In this paper the simulation is performed using the well known magic formula [5]. The longitudinal force is carried out, in this formulation, from the following formula:

$$ F_x = D \sin(C \arctan(B\Phi)) + S_v \quad (1) $$

Where,

$$ \Phi = (1 - E)(\lambda + S_n) + \left(\frac{E}{B}\arctan(B(\lambda + S_n)) \right) $$

In which $\lambda$ is the wheel slip and other parameters depend on the normal load and factors, which are determined from experience.

There are similar formulas for the lateral force and the self-aligning torque. Figure 3 shows typical longitudinal and lateral forces obtained from the magic formula.

Figure 3. Typical tire characteristics at different slip angles

DYNAMIC EQUATIONS – Considering the forces acting on the vehicle, as depicted in Figure 4, one can write the vehicle motion equations in horizontal plane as below:

$$ M_x (\ddot{u} + qw - rv) = M_x \dot{h}(\dot{q} + pr) + F_{x1} \cos(\delta_{f1}) $$

$$ - F_{y1} \sin(\delta_{f1}) + F_{x2} \cos(\delta_{f2}) $$

$$ - F_{y2} \sin(\delta_{f2}) + F_{x3} + F_{x4} - F_{ax} $$

(2)
\( M_r (\dot{\gamma} + ru - pw) = -M_r h (p + qr) + F_{x1} \sin(\delta_{f1}) \\
+ F_{y1} \cos(\delta_{f1}) + F_{x2} \sin(\delta_{f2}) \\
+ F_{y2} \cos(\delta_{f2}) + F_{y3} + F_{y4} - F_{ay} \)

\( I_z \dot{\gamma} = (I_x - I_y) pq + L_f [F_{x1} \sin(\delta_{f1}) \\
+ F_{y1} \cos(\delta_{f1}) + F_{x2} \sin(\delta_{f2}) \\
+ F_{y2} \cos(\delta_{f2})] - L_r (F_{y3} + F_{y4}) \\
+ \frac{T_f}{2} [F_{x1} \cos(\delta_{f1}) - F_{y1} \sin(\delta_{f1}) \\
- F_{x2} \cos(\delta_{f2}) + F_{y2} \sin(\delta_{f2})] \\
+ \frac{T_f}{2} (F_{x3} - F_{x4}) + \sum_{i=1}^{4} M_{zi} \)

Where \( r, p \) and \( q \) denote the angular velocities corresponding to yaw, roll and pitch angles, \( M_r \) is vehicle total mass, \( M_s \) is the sprung mass and \( I_x, I_y \), and \( I_z \) are the vehicle moment of inertia about the \( x, y, \) and \( z \) axes. The parameter \( h \) is the distance from roll axis to sprung mass centre of gravity and \( F_{ax} \) and \( F_{ay} \) are the aerodynamic drag coefficients.

Now, considering the acting torques on the vehicle body, the sprung mass motion can be carried out as follows.

The vehicle roll motion satisfies the following equations:

\( I_{x\gamma} \dot{\gamma} + (I_{z\gamma} - I_{x\gamma}) qr = \sum_{i=1}^{4} \bar{R}_{ri} F_{si} + M_s g h \varphi \\
- M_s h (\dot{\gamma} + ru - pw) \)

Where \( l_{x\gamma}, l_{y\gamma}, \) and \( l_{z\gamma} \) are the moment of inertia of the vehicle sprung mass about different axes and \( \bar{R}_{ri} \) is a coefficient corresponding to suspension geometry of the \( i_{th} \) wheel as follows:

\( \bar{R}_{x1} = -d_1 + (d_1 - b_1) \frac{a_1}{a_1 + b_1} \)
\( \bar{R}_{x2} = d_2 - (d_2 - b_2) \frac{a_2}{a_2 + b_2} \)
\( \bar{R}_{x3} = -d_3 + (d_3 - b_3) \frac{a_3}{a_3 + b_3} \)
\( \bar{R}_{x4} = d_4 - (d_4 - b_4) \frac{a_4}{a_4 + b_4} \)

Dimensional parameters \( a_i, b_i \), and \( d_i \) have been depicted in Figure 2. \( F_{si} \) is the force acting on suspension system of the wheel number \( i \):

\( F_{s1} = K_{s1} (Z_{u1} - Z_s + L_1 \sin(\varphi) + d_1 \sin(\varphi)) \\
+ C_{s1} (w_{u1} - w + L_1 \cos(\varphi) + d_1 \cos(\varphi)) \)
\( F_{s2} = K_{s2} (Z_{u2} - Z_s + L_1 \sin(\varphi) - d_2 \sin(\varphi)) \\
+ C_{s2} (w_{u2} - w + L_2 \cos(\varphi) - d_2 \cos(\varphi)) \)
\( F_{s3} = K_{s3} (Z_{u3} - Z_s - L_1 \sin(\varphi) + d_3 \sin(\varphi)) \\
+ C_{s3} (w_{u3} - w - L_3 \cos(\varphi) + d_3 \cos(\varphi)) \)
\( F_{s4} = K_{s4} (Z_{u4} - Z_s - L_1 \sin(\varphi) - d_4 \sin(\varphi)) \\
+ C_{s4} (w_{u4} - w - L_4 \cos(\varphi) - d_4 \cos(\varphi)) \)

Where \( w_{ui} \) is the vertical speed of the \( i_{th} \) wheel.

Pitch rate \( (q) \) is carried out as in below:

\( I_{y\gamma} \dot{\gamma} + (I_{z\gamma} - I_{y\gamma}) pr = -L_f (\frac{F_{s1}}{R_{r1}} + \frac{F_{s2}}{R_{r2}}) \\
+ L_r (\frac{F_{s3}}{R_{r3}} + \frac{F_{s4}}{R_{r4}}) + h_{cg} \sum_{i=1}^{4} X_i \)

Where \( h_{cg} \) is the height of the center of gravity and the parameter \( R_{ri} \) is defined with respect to the suspension geometry of the \( i_{th} \) wheel:

\( R_{ri} = \frac{a_i + h_{ri}}{b_i} \)

The motion equation for the sprung mass in the vertical direction can be written as follows:

\( M_s (\dot{\gamma} + pv - qu) = \sum_{i=1}^{4} \frac{F_{si}}{R_{ri}} \)

The vertical movement of each unsprung mass is expressed by the following equation:
ELECTRIC MOTORS AND DRIVES MODELING

Permanent Magnet Synchronous (PMS) motors are the most popular motors for in-wheel applications. The developed torque of a salient pole PMS motor in d-q coordinates is:

\[ T = P \phi i_q + P(L_d - L_q) i_d i_q \]  \hspace{1cm} (9)

Where \( P \) is the number of poles and \( \phi \) denotes the magnetic flux linkage. \( L_d, L_q \) and \( i_d, i_q \) are the inductances and currents in \( d \) and \( q \) directions.

Figure 5 shows the block diagram of a typical electric drive system for PMS motors. Note that the flux control loop (\( i_{sd} \) control loop in figure 5) is slower than the torque control loop and the flux is a slow varying variable. Furthermore, since the dynamic response of modern motor drives are much faster than wheel dynamics, and considering the dominant poles of the closed loop system, an electric motor and its drive can be simply modeled as:

\[ G(s) = \frac{T}{T_s} = \frac{1}{(1 + 2\xi s + 2\xi^2 s^2)} \]  \hspace{1cm} (10)

CONTROLLER DESIGN

YAW RATE CONTROLLER - An uneven longitudinal tire force may have a significant undesired effect on yaw motion. In this paper a differential torque is applied to the driven wheels of the right and left sides in order to compensate the disturbance yaw moment.

Rearranging equation 4 and assuming small and equal steering angles for the front wheels and no steering for the rear wheels one can write:

\[ I_z = (I_x - I_y) p q + L_x F_{sy} - L_y F_{yr} + \frac{T_f}{2} \Delta F_{sf} + \frac{T_r}{2} \Delta F_{sr} + \sum_{i=1}^{4} M_{zi} \]  \hspace{1cm} (11)

Where \( \Delta F_{sf} \) and \( \Delta F_{sr} \) are the differences in the longitudinal force of left and right wheels in the front and the rear:

\[ \Delta F_{sf} = (F_{s1} - F_{s2}) \]
\[ \Delta F_{sr} = (F_{s3} - F_{s4}) \]

And,

\[ F_{sf} = F_{y1} + F_{y2} \]
\[ F_{sr} = F_{y3} + F_{y4} \]

Hence, the yaw rate can be directly controlled by applying a differential input torque to the right and left wheels (in tire stable region).

From the steady state cornering theory of a bicycle model, it is known that the desired yaw velocity of a vehicle satisfies the following equation [6]:

\[ r_d = \frac{V - L K V^2 \delta}{L} \]  \hspace{1cm} (12)

Where \( K \) is the under-steer gradient and \( L \) is the vehicle wheelbase:

\[ L = L_f + L_r \]

A fuzzy logic controller is used to keep the yaw rate in its desired value, \( r_d \).

The error \( e = r - r_d \) and the change in error are applied to a fuzzy controller. The output of the controller is the deviation in the applied torque to the motors. Figure 6 shows the block diagram of the vehicle model and the yaw rate controller.

The normalized membership functions for fuzzification of the controller inputs and defuzzification of the controller output are depicted in Figure 7.

Table 1 shows the rule base of the Fuzzy controller.
SLIP CONTROLLER - The additional torque applied by the yaw controller may saturate the tire force, hence a slip ratio controller is required in the yaw control loop.

A Fuzzy controller for each wheel is used to keep the slip ratio within its stable region. The inputs to the controller are the wheel slip and the wheel angular acceleration. The Fuzzy rules regard the latter input as a virtual criterion for the direction of slip variations. The output of the controller is the amount of torque weakening that should be devoted to the torque command of the motors, in order to prevent the tire to enter into its saturation region. A block diagram of the overall control system is depicted in Figure 8.
SPEED ESTIMATOR

All the required control signals, but the vehicle speed, can be obtained from sensors, at a reasonable price. These sensors measure steering wheel angle, wheel speeds and yaw rate. Vehicle speed however, should be estimated anyhow. In this paper a data fusion method is used for this purpose, the wheel linear speeds are used as pseudo-sources of vehicle speed. An additional accelerometer is embedded in the vehicle in order to measure the vehicle longitudinal acceleration. The integral of the acceleration is regarded as another input. The inputs are all fed into an estimator, where a fuzzy logic determines which input is more reliable. Inputs are weighted and averaged to give the estimated speed. Figure 10 shows the block diagram of the estimator. The estimator consists of two stages. In the first stage, which is addressed as preprocessing, the wheel slips are calculated using the previous estimated vehicle speed and the measured wheel speeds. Meanwhile, the offset of the accelerometer is rejected by subtracting the derivation of the estimated speed from the measured acceleration and then low-pass filtering.

In the next stage, the wheel linear speed and the vehicle speed, which has been carried out by integration, are weighted using fuzzy rules. The weighted values are averaged to give the vehicle speed.

Figure 11 depicts the membership functions used in fuzzy weight generators. In Table 3 the rule sets, which are used for the weight generators, are tabulated. These
rules have been established on basis of linguistic terms such as:

- In cruise driving, the integral of the vehicle acceleration is not reliable, since the accelerometer signal is comparable to offset and noise of the measuring circuit.
- In braking situation all wheel speeds are weighted low, because of large wheel slips.

<table>
<thead>
<tr>
<th>Vehicle acceleration</th>
<th>NB</th>
<th>N</th>
<th>Z</th>
<th>P</th>
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<tbody>
<tr>
<td>N</td>
<td>S</td>
<td>H</td>
<td>M</td>
<td>S</td>
</tr>
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<td>S</td>
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<td>M</td>
</tr>
<tr>
<td>P</td>
<td>S</td>
<td>M</td>
<td>H</td>
<td></td>
</tr>
</tbody>
</table>

Table 3. Weighting rule sets for: a) wheels speed, b) Integration

**SIMULATION RESULTS**

A series of computer simulations was carried out to evaluate the performance of the proposed control system. The parameters of the vehicle model are tabulated in Table 4.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle total mass</td>
<td>$M_v$</td>
<td>Kg</td>
<td>1482.7</td>
</tr>
<tr>
<td>Front Unsprung mass</td>
<td>$M_d$</td>
<td>Kg</td>
<td>95.5</td>
</tr>
<tr>
<td>Rear Unsprung mass</td>
<td>$M_g$</td>
<td>Kg</td>
<td>168.5</td>
</tr>
<tr>
<td>Moments of inertia of total car about $X$ axis</td>
<td>$I_x$</td>
<td>$Kg m^2$</td>
<td>346.73</td>
</tr>
<tr>
<td>Moments of inertia of total car about $Y$ axis</td>
<td>$I_y$</td>
<td>$Kg m^2$</td>
<td>1675.8</td>
</tr>
<tr>
<td>Moments of inertia of total car about $Z$ axis</td>
<td>$I_z$</td>
<td>$Kg m^2$</td>
<td>1808.8</td>
</tr>
<tr>
<td>Moment of inertia of wheels</td>
<td>$I_w$</td>
<td>$m^2$</td>
<td>2.11</td>
</tr>
<tr>
<td>Track width</td>
<td>$L_n$</td>
<td>m</td>
<td>1.4375</td>
</tr>
<tr>
<td>Distance from front axle to CG</td>
<td>$L_f$</td>
<td>m</td>
<td>1.2247</td>
</tr>
<tr>
<td>Distance from rear axle to CG</td>
<td>$L_r$</td>
<td>m</td>
<td>1.4373</td>
</tr>
<tr>
<td>Height of CG</td>
<td>$h_{CG}$</td>
<td>m</td>
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<tr>
<td>Effective radius of wheels</td>
<td>$R_e$</td>
<td>m</td>
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</tr>
<tr>
<td>Spring constant of front springs</td>
<td>$K_s$</td>
<td>N/m</td>
<td>15400</td>
</tr>
<tr>
<td>Spring constant of rear springs</td>
<td>$K_r$</td>
<td>N/m</td>
<td>19000</td>
</tr>
<tr>
<td>Spring constant of tires</td>
<td>$K_t$</td>
<td>N/m</td>
<td>175000</td>
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<tr>
<td>Damping coefficient of front dampers</td>
<td>$C_s$</td>
<td>N/m</td>
<td>1150</td>
</tr>
<tr>
<td>Damping coefficient of rear dampers</td>
<td>$C_r$</td>
<td>N/m</td>
<td>6000</td>
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<tr>
<td>Damping coefficient of tires</td>
<td>$C_t$</td>
<td>N/m</td>
<td>50</td>
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<td>Aerodynamic drag coefficient</td>
<td>$C_d$</td>
<td>$N/(m/s)^2$</td>
<td>0.32</td>
</tr>
</tbody>
</table>

Table 4. The specifications of the vehicle used in simulation

LANE CHANGE ON SLIPPERY ROAD - The results of a lane change maneuver for a 3-meter lateral displacement at 60 km/h on unpacked snow are illustrated in Figure 12. It is assumed that the driver applies the same steering effort as on an adhesive road. This figure compares lane change with and without controllers.

As depicted in Figure 12a, with the controller on board, the desired lateral displacement achieved after 60 meters of longitudinal distance. Without the control system, the vehicle cannot achieve the target lateral displacement and additional efforts are required to return the car into the path. Obviously the controller successfully helps the driver to handle the lane change.

Figure 12b shows how the controller keeps the vehicle’s yaw rate thoroughly near the yaw rate reference. The control system applies additional torques to wheels as depicted in Figures 12d to 12g. The torques in opposite sides are rather differential.

BRAKING ON $\mu$-SPLIT ROAD - Next simulation performs braking at 60 km/h on a $\mu$-split road (dry pavement on the right side and unpacked snow on the left). A regenerative braking is assumed; hence a constant torque is applied to the driven wheels by the electric motors. For better illustration of results, no driver model is considered and the steering wheel is assumed to be fixed.

Simulation results are depicted in Figure 13. One can see that the control system successfully prevents the car to deviate from its path. The lateral displacement and lateral acceleration is negligible when the car brakes on the split road. Even though, in the absence of slip controller the deviation is acceptable, but as shown in Figures 13d and 13e the left side tires are saturated and blocked even earlier than in case of no controller at all, which results in lacking of lateral stability.

Also, while the rate of speed reduction with no controller is slightly better than the controlled vehicle, but the car starts turning around after braking, as can be seen in Figure 13a.
Figure 12. Lane change on a slippery road: a) Vehicle trajectory, b) Yaw rate and yaw reference, c) Vehicle side slip angle, d) Applied Torque to the Rear-Right wheel, e) Applied Torque to the Rear-Left wheel, f) Applied Torque to the Front-Right wheel, g) Applied Torque to the Front-Left wheel.
CONCLUSION

A novel driver-assist stability system for a four in-wheel drive electric vehicle was introduced. The system is based on Fuzzy logic Direct Yaw-moment Controller and individual slip controllers for each wheel using fuzzy logic. A multi-sensor data fusion method was introduced in order to estimate the vehicle real speed. The effectiveness of the proposed controller was evaluated by simulation. A 14-degree of freedom vehicle model and the well-known Pacejka tire formula was employed in the simulation program. Simulation results show excellent performance of the proposed control system on slippery roads.

REFERENCES