Influence Mechanism of Electromechanical Parameters on Transient Dynamics of FWD-EVs: Part I: Co-Modeling of IWM and Vehicle System

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Abstract. In-wheel motor (IWM), as an ideal power source of independent four-wheel drive electric vehicles, has been paid more and more attention due to its high-power density, low starting current, wide speed adjustment range, simple control system and robustness. However, the electromechanical issue is enlarged in both longitudinal and vertical because of in-wheel driven scheme. In this paper, the electromagnetic multi-field characteristic of IWM is investigated based on Fourier series method. The negative vibration coupling on vehicle dynamics is discussed by proposing a conjoint electromechanical FWD-EV model. Results shows that the motor incentive coupled with the vehicle system in multi-degree of freedom, caused the body and wheel resonance in the low speed, meanwhile deteriorated the anti-rollover capability of the IWM-EV in the high speed.

Keywords. Four-wheel drive, electromechanically motivated harassment, vehicle system dynamics.

1. Introduction
IWM-EVs integrate multi-function assembly, which makes it a novel power source arrangement in electric vehicle and could achieve fast and accurate control in vehicle dynamics. However, electromechanically motivated harassment issues hinder its further application. The vertical component of the electromagnetic excitation, which is arisen by unavoidable road surface roughness excitation, is acting upon the tire-suspension system and bringing negative dynamic issues. In the previous work, the control approach of driving torque [1,2], the algorithm of vehicle state estimation [3, 4], the electric differential controller based on drive-by-wire method [5], as well as energy management were approached [6, 7]. Vibration and noise problems of SR-IWM were studied [8]. The impact of this problem on vehicle system dynamics is investigated [9]. Some IWM schemes and controllers were proposed to eliminate the vibration problem [10]. However, the mechanism of SR-IWM noise and vibration, and its influence on vehicle dynamic system was not studied deeply. Given this above, a research on the impact of SR-IWM on vehicle dynamics performance was carried on to reveal its coupling effect on vehicle system.
2. The Modelling of In-Wheel Driven Motor and Vehicle System Dynamics

A typical configuration of outer-rotor driven motor is shown in figure 1. This paper adopted the Fourier series equation to reveal the multi-field characteristic of in-wheel driven electric vehicles.

The co-energy $W_m$ of each phase winding in figure 1 is related the dependent-phase flux $\psi(\theta, i)$ and the phase current $i$, as shown in the following equation:

$$W_m = \int_0^i \psi(\theta, i) \, di$$

The generalized electromagnetic force is the partial differential of the generalized displacement $x$ by the magnetic total energy $W_m$. The k-phase broadening force $F_k$ of the motor is generated by electromagnetic attraction as:

$$F_k = \left. \frac{\partial W_m}{\partial x} \right|_{i=\text{const}} = \int_0^i \left( \frac{\partial \psi_k(\theta, i)}{\partial x} \right) \, di$$

By substitute the generalized displacement $x$ for the relative angular displacement $\theta$, the stator core thickness $l_z$ as well as the air gap $l_g$. The corresponding motor generalized force in equation (2) is developed in the driving torque $T_e$, the electromagnetic force in axial direction $F_h$ as well as the electromagnetic force in radial direction $F_r$:

$$T_e = \left. \frac{\partial W_m}{\partial \theta} \right|_{i=\text{const}} = \int_0^i \left( \frac{\partial \psi(\theta, i)}{\partial \theta} \right) \, di$$

$$F_h = \left. \frac{\partial W_m}{\partial l_z} \right|_{i=\text{const}} = \int_0^i \left( \frac{\partial \psi(\theta, i)}{\partial l_z} \right) \, di$$

$$F_r = \left. \frac{\partial W_m}{\partial l_g} \right|_{i=\text{const}} = \int_0^i \left( \frac{\partial \psi(\theta, i)}{\partial l_g} \right) \, di$$
The dynamics of vehicle model is adopted as shown in figure 2, which consist of a tire model, a vibration and roll model in vehicle as well as a plane motion model. The vibration model is developed to express the mechanism of electromechanically coupling between vibration negative effects and vehicle dynamics in vertical direction. The accordingly dynamic equation is derived by using D’Alembert’s principle:

\[ I_x \ddot{\phi} = (F_{s1} - F_{s3})L_3 + (F_{s2} - F_{s4})L_4 + m_s a_y d_r + m_s g d_r \phi \]  

(4)

where \( m_s \) and \( \phi \) are the sprung mass and the roll angle of the vehicle respectively. The inertia of roll moment of sprung mass, the length of vehicle centroid to roll axis are expressed as \( L_x \) and \( d_r \). In figure 2, \( L_3 \) and \( L_4 \) are the distance from vehicle front and rear halftrack to the centroid, and the lateral direction of inertial caused by acceleration and deceleration focused on vehicle centroid is expressed by \( a_y \).

The suspension force \( F_{si} \) are

\[ F_{si} = k_{si}(z_{ui} - z_{si}) + c_{si}(z'_{ui} - z_{si}) \]  

(5)

During running process of vehicle. In equation (5), the movement in vertical direction of vehicle sprung and unsprung mass are expressed as \( z_{si} \) and \( z_{ui} \); The stiffness damping characters of vehicle suspension are represented by \( k_{si} \) and \( c_{si} \).

Their geometrical relationship is obtained:

\[ z_{si} = \begin{cases} z - L_1 \theta + L_3 \phi \\ z + L_2 \theta + L_4 \phi \\ z - L_1 \phi - L_3 \phi \\ z + L_2 \phi - L_4 \phi \end{cases} \]  

(6)

In which, \( \theta \) is the vehicle pitch angle in figure 2; \( z \) is the displacement in vertical direction of vehicle sprung mass centroid during driving; \( L_1 \) and \( L_2 \) are the distance from vehicle front and rear axle to the centroid respectively.

Furthermore, the pitching equation of vehicle dynamics is derived:

\[ I_y \ddot{\theta} = (F_{s2} + F_{s4})L_2 - (F_{s1} + F_{s3})L_1 + m_s a_x d_p + m_s g d_p \theta \]  

(7)

where the inertia of pitch moment of sprung mass, the length of vehicle centroid to pitch axis are expressed as \( I_y \) and \( d_p \); the longitudinal direction of inertial caused by acceleration and deceleration focused on vehicle centroid is expressed by \( a_x \).

The vibration equation for unsprung and sprung mass are described:

\[ m_{ui} \ddot{z}_{ui} = F^d_{ti} - F_{si} \]  

(8)

\[ m_s \ddot{z} = \sum F_{si} \]  

(9)

Among them, the tire vertical force \( F^d_{ti} \) can be written:

\[ F^d_{ti} = c_t z (z_g - z_{ui}) + k_t (z_g - z_{ui}) + F_{r,zi} \]  

(10)

Which the road roughness excitation acted on each tire is described as \( z_g \); \( F_{r,zi} \) is the corresponding electromechanically motivated harassment introduced by the special layout of IWM-EV; \( k_t \) and \( c_t \) are tire stiffness and damping respectively.

According to equation (1-3) the electromagnetic excitation in the k-phase can be substituted as follows:

\[ F_{rk} = F_{rm} - F_{rn} = \frac{1}{2} l^2 \frac{k_k(\theta, i_k)}{I_g(m)} - \frac{1}{2} l^2 \frac{k_k(\theta + \pi, i_k)}{I_g(m)} \]  

(11)

For an IWM configuration, as shown in figure 1, the vertical component of electromagnetic excitation acts on vehicle dynamics is yielded:
\[ F_{rk,Z} = F_{rk} \cos \theta \]  
(12)

Furthermore, the unbalanced force in radial direction of all phases is derived:

\[ F_{r,Z} = \sum F_{rkZ} \]  
(13)

The tire load includes the static load and dynamic load from suspension system, in which the static load is expressed:

\[ F_{tx1}^s = F_{tx3}^s = \frac{L_2 m_2 g}{2(L_1 + L_2)} \]  
(14)

\[ F_{tx2}^s = F_{tx4}^s = \frac{L_1 m_2 g}{2(L_1 + L_2)} \]  
(15)

The total tire load can be expressed:

\[ F_{tx} = F_{tx}^d + F_{tx}^s \]  
(16)

3. Joint Modeling and Preliminary Analysis

The vertical excitation is obtained and applied to the vehicle dynamics module. The controller layout and logic flow chart of joint model are depicted in figure 3 and figure 4. The dynamic reactions of the vehicle are obtained by input acceleration/deceleration/steering/road excitation into the vehicle stability-related model, which contains the correlation and constraints between the vehicle degrees of freedom.

**Figure 3. Joint dynamics model.**

**Figure 4. Full vehicle stability analysis data flow logic.**
3.1. Modal Analyses

The equation (4-16) is written as the system of undamped vibration-free to carry out the research on the nature frequency of this system:

\[ M \ddot{Z} + KZ = 0 \]  

(17)

where

\[ M = \begin{bmatrix} M_s & 0_{3\times 4} \\ 0_{4\times 3} & M_u \end{bmatrix}, \quad K = \begin{bmatrix} H^T K_s H & -H^T K_s \\ -K_s H & (K_s + K_t) \end{bmatrix}, \quad Z = \begin{bmatrix} z \\ [z_u] \end{bmatrix} \]

\[ H = \begin{bmatrix} 1 - \frac{L_1}{2} & L_3 \\ L_2 & 2 & L_4 \\ 1 - \frac{L_1}{2} & -L_3 \\ \frac{L_2}{2} & -L_4 \end{bmatrix} \]

\[ K_s = \text{diag}(k_{s1}, k_{s2}, k_{s3}, k_{s4}), \quad K_t = \text{diag}(k_{t1}, k_{t2}, k_{t3}, k_{t4}), \quad M_u = \text{diag}(m_{u1}, m_{u2}, m_{u3}, m_{u4}) \]

\[ M_u = \text{diag}(m_s, I_y, I_x), \quad [z] = [z_{s1}, z_{s2}, z_{s3}, z_{s4}]^T, \quad [z_u] = [z_{u1}, z_{u2}, z_{u3}, z_{u4}]^T, \quad [z] = [z, \theta, \phi] \]

According to equation 17, the modal coupling results are listed in table 1. It is found out that the vertical force incentive causes vehicle body resonance at the start condition, body roll motion composes 78.52% of the vibration energy in the second modal, is decoupled with other degree of freedom. Moreover, when the vertical force frequency reaches 10Hz to tires resonant frequency, will seriously damage the grounding of the tires and make the vehicle easily lose stability.

| Modal order | Energy distribution (%) | Natural frequency (Hz) |
|-------------|------------------------|------------------------|
|             | (1)        | (2)        | (3)        | (4)        | (5)        | (6)        | (7)        |
| z           | 43.18      | 0.00       | 43.02      | 24.06      | 15.16      | 24.06      | 15.16      |
| \phi        | 0.00       | 78.52      | 0.00       | 11.45      | 17.41      | 11.45      | 17.41      |
| D \theta    | 38.57      | 0.00       | 38.74      | 22.51      | 12.09      | 22.51      | 12.09      |
| O z_{u1}    | 6.49       | 5.42       | 6.77       | 17.34      | 5.66       | 17.34      | 5.66       |
| F z_{u2}    | 2.64       | 5.31       | 2.34       | 3.65       | 22.02      | 3.65       | 22.02      |
| z_{u3}      | 6.49       | 5.42       | 6.77       | 17.34      | 5.66       | 17.34      | 5.66       |
| z_{u4}      | 2.64       | 5.31       | 2.34       | 3.65       | 22.02      | 3.65       | 22.02      |

The transfer characteristics of vehicle dynamics are depicted in figure 5, which reflects the influence of electromagnetic force excitation on the vehicle system. From the figure 5, the electromechanically motivated harassment is in 2nd mode of the main vehicle body. The impact will cause vehicle resonate at two degrees of freedom, and is both act on sprung and unsprung mass, which is comparable with the road roughness excitation. Furthermore, similar to the road surface excitation, the electromagnetic excitation is correlation with vehicle speed, mainly concentrated in the second-order 10Hz frequency resonance, dose more significant impact on non-sprung mass of the vehicle body.
6

(a) Vertical displacement of sprung mass  
(b) Roll angle  
(c) Pitch angle  
(d) Wheel bounce  

Figure 5. Vibration transfer characteristics of vehicle to the unbalanced radial force.

3.2. Coupling Effects between Road Surface Excitation and Unbalanced Radial Force

The simulation results at 15 m/s on B-class road roughness excitation are depicted in figure 6. It can be seen that because of the constant torque requirement of the vehicle, the relative eccentricity, which is stimulated by road surface excitation, is the only determining factor for the electromechanically motivated harassment impact on vehicle system. The degree of influence is positive correlation with the relative eccentricity and the road surface roughness. Furthermore, after the transient electromagnetic excitation is applied to vehicle system, a vicious circle was observed, which means the electromechanical coupling process worsen the non-uniform air-gap situation and further in turn intensifies the magnitude of the unbalanced electromagnetic excitation and motor vibration in vehicle interior dynamic system. The simulation results about other three wheels shown in figure 7, and the same conclusion can be drawn.

Figure 6. Coupling effects between road surface excitation and unbalanced radial force.
4. Conclusions

The special scheme and structural layout of IWM-EV leads the electromechanically motivated harassment directly acts on vehicle system, causes deterioration on vehicle ride comfort and handling stability. This issue has been studied and discussed by means of developing an electromechanical-dynamic coupling vehicle joint model. Some conclusions are drawn as follows:

(1) This vertical force frequency covers resonant frequency both sprung mass and un-sprung mass, easy to cause the resonance of body and wheel, especially in starting and low-speed running conditions.

(2) The transient electromagnetic excitation inevitable causes vicious circle between non-uniform air-gap situation and vehicle interior dynamic system, it is an important task for IWM-EV to investigate and overcome the negative effects caused by its own defects.

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