Approaches to diminish large unsprung mass negative effects of wheel side drive electric vehicles

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Abstract
Ride comfort and road holding ability of distributed drive electric vehicles are worsened due to large unsprung mass introduced by wheel side drive system. To diminish this large unsprung mass negative effects, three approaches (“Suspension and Reducer Integrated” Close-to-Wheel Drive System, “Dynamic Vibration Absorber” Close-to-Wheel Drive System and in-wheel drive system with “Active and Energy Regenerative Suspension”) are introduced. The corresponding dynamics are analyzed and the schemes are provided. Evaluations of suspension performance show the superiorities of these approaches in different magnitude compared with the conventional configuration which is equipped with hub motor and passive suspension. Overall comparisons between three approaches are carried out and the feasibilities are discussed. With proper design according to specific application, these approaches are promising to be applied on various wheel side drive systems.

Key words : Unsprung mass, Wheel side drive system, Close to wheel drive system, Suspension and reducer integration, Dynamic vibration absorber, Active and energy regenerative suspension

1. Introduction

Benefit from quick response of motor torque and torque vectoring control, distributed drive electric vehicles (EVs) have a good potential to realize ABS, ESP and Traction control, and thus to achieve better driving performance. The drivetrain layouts of distributed drive EVs can be mainly divided into on-board motor drive system and wheel side drive system. With elimination of classic differential and drive shaft, the transmission chain of the wheel side drive system is relatively short and it doesn’t occupy space of the car body, and thus it features high efficient and compact. Therefore, the wheel side drive system is a prospective application on distributed drive EVs. However, the wheel side drive systems are normally equipped with a low speed hub motor or a high speed motor with reducer. These would bring in a large unsprung mass. Research pointed out that moving the electric propulsion from the vehicle body to the wheels can add up to 50kg or more, per wheel, to the unsprung mass (Van Schalkwyk and Kamper, 2006). The driving characteristics of a vehicle can be greatly improved by reducing the unsprung mass (Heißing and Ersoy, 2010). With the increased unsprung mass, the ride comfort and especially the road holding ability (or in other word driving safety) of a vehicle are worsened (Han, 2002; Guo, 2012; Fang, 2012; Liu, et al., 2012; Rojas, et al., 2010; Ślaski et al., 2014). This is so called “large unsprung mass negative effects”.

Researchers and companies are devoting to diminish large unsprung mass negative effects. An axial flux disc motor is used to reduce unsprung mass of distributed drive EVs (Eastham et al., 1995; Hredzak, et al., 1998). As shown in Fig. 1(a), ZF released an early concept study of electric twist bean for Close-to-Wheel drive EVs at the IAA 2011(September) in Frankfurt. It features lightweight and compact (Yeh et al., 2013). Bridgestone developed dynamic-
damping in-wheel drive system to reduce the weight of in-wheel system and improve its vibration-damping performance (Nagaya et al., 2003). The system is shown in Fig. 1(b). Michelin released its active wheel technology on Paris Motor Show, 2008. Two motors are utilized in this in-wheel drive scheme as shown in Fig. 1(c). One is for traction and the other acts as an active suspension system to improve comfort, handling and stability (Evans, 2008).

![ZF electric twist beam](image1) ![Bridgestone developed dynamic-damping in-wheel drive system](image2) ![Michelin active wheel technology](image3)

Fig. 1 Approaches to diminish large unsprung mass negative effects.

Basing on the research (Ning, 2007), it can be concluded that four approaches are normally considered to diminish large unsprung mass negative effects. There are applications of newly developed light materials, structural integration design (Dai, et al., 2011; Chen, et al., 2014), “Dynamic Vibration Absorber” (“DVA”) design (Liang et al., 2008; Chen et al., 2014) and active/semi-active suspension design (Yin et al., 2015; Yin et al., 2015).

The rest of the paper is organized as follows. Section 2 introduces the idea of mass transfer together with integration design, and gives the corresponding dynamic model. Some schemes of the “Suspension and Reducer Integrated” (“SRI”) Close-to-Wheel Drive System (CWDS) are provided and a prototype is built based on the research. Section 3 introduces the idea of “DVA” design and gives the corresponding dynamic model. Some schemes of the “DVA” CWDS are provided to show the feasibility. In section 4, a dynamic model is given basing on the idea of active suspension design and energy harvest during vibration. And a scheme of electromagnetic actuated active suspension is proposed. Basing on given dynamic models, section 5 analyses the suspension performances of three approaches and compares with a benchmark car passive suspension both in frequency domain and in time domain. Then compares and discusses the complexity, cost and application of three proposed approaches. The last part comes to the conclusion.

2. “Suspension and Reducer Integrated” Close-to-Wheel Drive System

Given a specific mass on a sway bar, the equivalent inertia with respect to the joint would be smaller if the distance between the center of mass and the joint is getting shorter. This inspires the idea of mass transfer. In trailing arm suspension, moving the motor and gearbox from hub to somewhere near the joint, the equivalent unsprung mass may be reduced. So as other suspension structures. Focusing on research and development of distributed drive EVs for years,
Chen et al. proposed several schemes of “SRI” CWDSs according to different suspension configurations (such as single trailing arm suspension, single oblique arm suspension and single cross arm suspension) and theoretical analysis were provided. Corresponding patents (Chen et al., 2012; Chen et al., 2012; Chen et al., 2012) were applied in March, 2011 which were earlier than the releasing of ZF electric twist beam. The theoretical and schemes of “SRI” CWDSs are introduced and a prototype is designed basing on the schemes.

2.1 Theoretical analysis

Fig. 2 shows a quarter car model of “SRI” scheme suspension, where the sprung mass $M_b$, the unsprung mass $M_w$, the motor mass $M_m$, the moment of inertia with respect to the center of mass of the trailing arm assembly $J_1$, the suspension stiffness $K_s$, the suspension damping $C_s$, the tyre stiffness $K_t$, the length of the trailing arm $L$, the distance between motor and joint $L_1$, the angle between the trailing arm and line (which is connected with motor and joint of the trailing arm) $\beta$.

![Quarter car model of “SRI” scheme suspension](image)

Fig. 2 Quarter car model of “SRI” scheme suspension

The displacement of the sprung mass, unsprung mass and road input are denoted as $x_2$, $x_1$ and $x_0$. The dynamic equations of the quarter car model can be derived as below:

The derivation is based on the Lagrange’s equation.

$$
\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial \dot{q}_i} + \frac{\partial D}{\partial \dot{q}_i} = Q_i
$$

(1)

where the kinetic energy is:

$$
T = \frac{1}{2} M_b \dot{x}_2^2 + \frac{1}{2} M_w \dot{x}_1^2 + \frac{1}{2} M_m \dot{v}_m^2 + \frac{1}{2} J_1 \dot{\alpha}^2
$$

(2)

The potential energy is:

$$
U = \frac{1}{2} K_s (x_2 - x_i)^2 + \frac{1}{2} K_t x_i^2
$$

(3)

The dissipated energy is:

$$
D = \frac{1}{2} C_s \left(\dot{x}_2 - \dot{x}_i\right)^2
$$

(4)

Since the motor would rotate along the joint of the trailing arm during wheel bump and rebound, the velocity of the motor $v_m$ can be decomposed into longitudinal speed (which is denoted by subscript $x$) and vertical speed (which is denoted by subscript $y$), and it is described as:
\[
\begin{align*}
\dot{v}_m &= \sqrt{v_{mx}^2 + v_{my}^2} \\
v_{mx} &= \ddot{a}L_2 \sin \beta \\
v_{my} &= \ddot{a}L_2 \cos \beta + \dot{x}_2
\end{align*}
\] (5)

\[x_1 \text{ can be described as:} \]
\[x_1 = L \sin \alpha + x_2\] (6)

Since \(\alpha\) is micro, thus \(\sin \alpha \approx \alpha\) and Eqs.(6) can be simplified as:
\[x_1 \approx L\alpha + x_2\] (7)

Assuming that \(x_2\) and \(\alpha\) are two generalized coordinates and combine Eqs.(1)~(7), the mass matrix \(M_0\) can be obtained as Eqs.(8).
\[
M_0 = \begin{bmatrix}
M_0 + M_w + M_m & M_m L + M_m L_2 \cos \beta \\
M_m L + M_m L_2 \cos \beta & M_m L^2 + M_m L_1^2 + \int_1
\end{bmatrix}
\] (8)

Given a coordinate transformation matrix as shown below:
\[
P = \begin{bmatrix}
1 & L \\
1 & 0
\end{bmatrix}
\] (9)

The coordinates can be transformed as:
\[
\begin{bmatrix}
x_1' \\
x_2'
\end{bmatrix} = P \begin{bmatrix}
x_2 \\
\alpha \\
x_2
\end{bmatrix} = P \begin{bmatrix}
x_2' \\
\alpha
\end{bmatrix}
\] (10)

Thus \(M = (P^{-1})^T M_0 P^{-1}\)

From Eqs.(1)~(5) and in \(x_1, x_2\) coordinates, the stiffness matrix and damping matrix can be described as:
\[
K = \begin{bmatrix}
K_x + K_x & -K_x \\
-K_x & K_x
\end{bmatrix}, C = \begin{bmatrix}
C_x & -C_x \\
-C_x & C_x
\end{bmatrix}
\] (11)

The road excitation is the generalized force of the system, it can be described as \(k_{sys} \cdot x_0\) where \(k_{sys} = \begin{bmatrix} K_i \\ 0 \end{bmatrix}\)

Thus, the dynamic equation of the system can be described as shown below:
\[
M \begin{bmatrix}
\ddot{x}_1' \\
\ddot{x}_2'
\end{bmatrix} + C \begin{bmatrix}
\dot{x}_1' \\
\dot{x}_2'
\end{bmatrix} + K \begin{bmatrix}
\dot{x}_1' \\
\dot{x}_2'
\end{bmatrix} = k_{sys} \cdot x_0
\] (12)

Simulations of “SRI” scheme suspension are basing on these dynamic equations and the results are shown in the later paragraph.

2.2 System schemes

Several schemes of “SRI” CWDS had been proposed basing on different suspension structures (Chen et al., 2012; Chen et al., 2012; Chen et al., 2012) and the top view of the schemes are shown as Fig. 3.

Take the application on the trailing arm suspension for example. As shown in Fig. 3(a), instead of being placed in the center of a wheel, the motor is positioned more close to the joint. The gear box of the “SRI” CWDS also constitutes the trailing arm of the suspension system. During the vibration of the sprung and unsprung mass, the “SRI” CWDS would rotate along with the joint. Thus the shorter distance between the motor and the joint indicates less equivalent unsprung mass. Besides, compared with conventional trailing arm suspension, this “SRI” CWDS features anti squat during acceleration and the analysis is omitted for brief. The principles of the other two schemes shown in Fig. 3(b) and Fig. 3(c) are similar.
(a) Trailing arm suspension: 1. Motor; 2. Joint; 3. Trailing arm; 4. Output shaft; 5. Rim

(b) Oblique arm suspension: 1. Motor; 2. Joint; 3. Oblique arm; 4. Output shaft; 5. Rim

(c) Transverse arm suspension: 1. Motor; 2. Joint; 3. Transverse arm; 4. Output shaft; 5. Rim

Fig. 3 “SRI” CWDS on different suspension structures

A CAD model of the “SRI” CWDS which is applied on the trailing arm suspension was built (as shown in Fig. 4(a)) and a prototype was manufactured based on it (as shown in Fig. 4(b)). This prototype of the “SRI” CWDS was assembled onto a test car (as shown in Fig. 5(a)) and road tests had been done to prove the ability of vertical vibration during driving. Equipped with two “SRI” CWDSs (as shown in Fig. 5(b) and circled by red), the test car had been tested on a chassis dyno in an anechoic room. The acceleration test, climbing test and cruising test showed that the longitudinal dynamic performance of the “SRI” CWDSs meet the design requirements.

Besides lightweight design of the package and less equivalent unsprung mass, being part of Trailing/Oblique/Transverse arm of the suspension, this “SRI” CWDS not only benefits lower unsprung mass but also saves cost.

Fig. 4 CAD model and prototype of “SRI” CWDS and real car assembly

Fig. 5 Application and bench test of “SRI” CWDS
3. “Dynamic Vibration Absorber” Close-to-Wheel Drive System

The idea of “DVA” has been used to diminish the vibration in many applications. Even high buildings would utilize it to diminish vibration induced by wind (Chang and Qu, 1998). In a conventional configuration, the mass of the motor and gearbox of CWDS contributes to the unsprung mass. If this mass is suspended by a spring and a damper, it can be utilized as the mass of “DVA”. The following theoretical analysis and system schemes are basing on this idea.

3.1 Theoretical analysis

Fig. 6 shows a quarter car model of “DVA” scheme suspension, where the sprung mass $M_b$, the unsprung mass $M_w$, the moment of inertia with respect to the center of mass of the motor and gearbox assembly $J_2$, the suspension stiffness $K_s$, the suspension damping $C_s$, the absorber stiffness of “DVA” $K_{ab}$, the absorber damping of “DVA” $C_{ab}$, the tyre stiffness $K_t$, the distance between the motor and the wheel center $L_2$ and the rotation angle $\alpha$.

The displacement of the sprung mass, unsprung mass and road input are denoted as $x_2$, $x_1$ and $x_0$. Similar with the derivation of “SRI” scheme shown above, the kinetic energy is:

$$T = \frac{1}{2} M_b \dot{x}_2^2 + \frac{1}{2} M_w \dot{x}_1^2 + \frac{1}{2} J_2 \dot{\alpha}^2$$

The potential energy is:

$$U = \frac{1}{2} K_s (x_2 - x_1)^2 + \frac{1}{2} K_{ab} (x_1 - x_0)^2 + \frac{1}{2} K_t x_0^2$$

The dissipated energy is:

$$D = \frac{1}{2} C_s (\dot{x}_2 - \dot{x}_1)^2 + \frac{1}{2} C_{ab} (\dot{x}_1 - \dot{x}_0)^2$$

Assuming a micro $\alpha$, thus $\cos \alpha \approx 1, \sin \alpha \approx \alpha$ and $x_1, \dot{x}_1$ can be described as:

$$\begin{align*}
x_1 & \approx x_i + L_2 \sin \alpha \approx x_i + L_2 \alpha \\
\dot{x}_1 & \approx \dot{x}_i + L_2 \dot{\alpha}
\end{align*}$$

Thus the kinetic energy, the potential energy and the dissipated energy can be described as:

$$\begin{align*}
T &= \frac{1}{2} M_b \dot{x}_2^2 + \frac{1}{2} M_w \dot{x}_1^2 + \frac{1}{2} M_w (\dot{x}_1 + L_2 \dot{\alpha})^2 + \frac{1}{2} J_2 \dot{\alpha}^2 \\
U &= \frac{1}{2} K_s (x_2 - x_1)^2 + \frac{1}{2} K_{ab} L_2^2 \alpha^2 + \frac{1}{2} K_t x_0^2 \\
D &= \frac{1}{2} C_s (\dot{x}_2 - \dot{x}_1)^2 + \frac{1}{2} C_{ab} L_2^2 \dot{\alpha}^2
\end{align*}$$
Combine Eqs. (1) and Eqs. (17), the mass matrix, the stiffness matrix and the damping matrix can be described as Eqs. (18)–Eqs. (20).

\[
M = \begin{bmatrix}
M_b + M_m & 0 & M_m L_2 \\
0 & M_w & 0 \\
M_w L_2 & 0 & M_w L_2^2 + J_3
\end{bmatrix}
\]

\[\text{(18)}\]

\[
K = \begin{bmatrix}
K_b + K_s & -K_s & 0 \\
-K_s & K_s & 0 \\
0 & 0 & K_{ab} L_2^2
\end{bmatrix}
\]

\[\text{(19)}\]

\[
C = \begin{bmatrix}
C_b & -C_b & 0 \\
-C_b & C_b & 0 \\
0 & 0 & C_{ab} L_2^2
\end{bmatrix}
\]

\[\text{(20)}\]

The road excitation is the same as described above and \( K_{syr} = \begin{bmatrix} K_s \\ 0 \\ 0 \end{bmatrix} \).

Thus, the dynamic equation can be described as shown below:

\[
M \begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2 \\
\alpha
\end{bmatrix} + C \begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2 \\
\alpha
\end{bmatrix} + K \begin{bmatrix}
x_1 \\
x_2 \\
\alpha
\end{bmatrix} = K_{syr} \cdot x_0
\]

\[\text{(21)}\]

Simulations of “DVA” scheme suspension are basing on these dynamic equations and the results are shown in the later paragraph.

3.2 System schemes

Two schemes of the “DVA” CWDS with translational pair had been proposed and they are shown in Fig. 7(a) and Fig. 7(b) respectively. The motor is suspended on the unsprung mass by a spring and a damper. There could be relative motion between the unsprung mass and the motor which is guided by the translational pair. The mass of the motor is utilized as the mass of “DVA”. The vibration excited by uneven road surface would be transferred to the unsprung mass first, then the vibration would be transferred to the sprung mass and the motor respectively. Since some of the vibration energy is absorbed by the extra damper which is used to suspend the motor, the suspension performance is expected to be better. However, the relative motion between the motor and the gearbox could be an engineering challenge which needs to be taken good care of.

(a) 1. Motor; 2. Damper of “DVA”; 3. Spring of “DVA”; 4. Translational pair; 5. Output shaft; 6. Rim

(b) 1. Motor; 2. Damper of “DVA”; 3. Spring of “DVA”; 4. Translational pair; 5. Output shaft; 6. Rim

Fig. 7 Schemes of “DVA” CWDS with translational pair
To avoid the engineering challenge of the “DVA” CWDS with translational pair discussed above, a scheme of the “DVA” CWDS with rotational pair had been proposed (Chen, et al., 2014). Its top view is shown in Fig. 8(a) and its side view is shown in Fig. 8(b). As shown in Fig. 8(a), the motor is fixed on the gearbox1 while the gearbox2 is fixed on the unsprung mass. The gearbox1 together with the motor could rotate along with the rotational pair. Fig. 8(b) shows that the gearbox1 together with the motor are suspended on the gearbox2 by two springs and two dampers. The mass of motor and the mass of gearbox1 constitute the mass of “DVA”.

According to the scheme shown in Fig. 8 and basing on McPherson suspension structure, a CAD model of the “DVA” CWDS with rotational pair was built and it is shown in Fig. 9.

Furthermore, basing on the idea of “DVA” design, a lot more structures could be derived and much better suspension performance is expected.
4. Active and Energy Regenerative Suspension (AERS) system

Utilizing hydraulic or hydro pneumatic actuator to supply active force, traditional active suspension can compromise the requirements of ride comfort and road holding during different driving conditions. Thus the large unsprung mass negative effects of in-wheel drive EVs can be diminished in this way. But the large energy consumption of traditional actuator is not acceptable especially for EVs. In consideration of energy efficiency and to harvest energy from vibration, the electromagnetic actuator is utilized. While the linear motor is not mature at this moment, the mechanism design issue lies in how to transfer the linear motion of suspension into the rotation motion of a rotary motor within the design boundary of the suspension. Since various control strategies had been researched and developed, the ideal performance can be expected.

4.1 Theoretical analysis

Theoretical analysis is aiming to demonstrate the superiority of the AERS system as well as figuring out the design requirements of the electromagnetic actuator.

Fig. 10 shows a quarter car model with the AERS, where the sprung mass $M_b$, the unsprung mass $M_u$, the suspension stiffness $K_s$, the tyre stiffness $K_t$ and the force of the actuator $U$. To simplify the analysis, optimal control strategy is utilized. The method and simulation in MATLAB Simulink are provided in the previous work done by Yin, et al., 2015. However, in this demonstration, the weighting factors of the cost function are different from previous work due to difference of quarter car parameters.

The displacement of the sprung mass, unsprung mass and road input are denoted as $x_2$, $x_1$ and $x_0$. The dynamic equations of the quarter car model can be expressed as:

$$M_b \ddot{x}_2 = K_s (x_1 - x_2) + U$$  (22)

$$M_u \ddot{x}_1 = K_s (x_2 - x_1) + K_t (x_0 - x_1) - U$$  (23)

The ride comfort, road holding ability and suspension working stroke are indicated by the root mean square of body acceleration (BA), dynamic tyre load (DTL) and suspension working stroke (SWS) respectively. Less body acceleration means better ride comfort, less dynamic tyre load means better road holding ability and less suspension working stroke means more capability within determined suspension stroke. Thus, the suspension performance is indexed by root mean square of BA, DTL and SWS. The cost function of the optimal controller is given as follow and the related weighting factors are shown in Table 1:

$$J = \lim_{T \rightarrow \infty} \int_0^T \{ h_1 [x_1(t) - x_0(t)]^2 + h_2 [x_2(t) - x_1(t)]^2 + h_3 x_2^2(t) \} dt$$  (24)

| $h_1$ | $h_2$ | $h_3$ |
|-------|-------|-------|
| 40000 | 1000  | 1     |
Assuming a normal cruising condition and applying optimal control strategy, the nominal design requirements of the electromagnetic actuator can be estimated with parameters of the benchmark car. However, the electromagnetic actuator should satisfy extreme conditions. In a simulation, an extreme condition was achieved that the travel of the suspension had already reached its limit and it just hit the stop bump. The peak power and maximum force of the actuator as well as the maximum speed between sprung/unsprung mass are evaluated in this extreme condition. An electromagnetic actuator with these parameters is able to supply sufficient power and force within the full compression/rebound of suspension travel. One example had been done by Yin, et al., 2015 and the results are shown in Table 2.

### Table 2 Performance boundary in wheel side

| Description                                      | Value |
|--------------------------------------------------|-------|
| Peak power (kW)                                 | 6.5   |
| Maximum force (N)                               | 3500  |
| Maximum speed between sprung/unsprung mass (m/s) | 2     |

4.2 System scheme

Varies schemes have been researched and developed to realize the AERS design with a rotary motor. Ball screw mechanism (Suda, et al., 2000; Zhang, et al., 2008) and rack and pinion mechanism (Beno, et al., 1995; Hayes, et al., 2005; Zuo and Zhang, 2013) have been utilized to develop the electromagnetic actuator.

In this paper, a Permanent Magnet Synchronous Motor (PMSM) is chosen as the electromagnetic actuator for its mature technology and high efficiency. To match the output torque of the PMSM and required force in wheel side, a dual stages gearbox is utilized for its good reliability and high efficiency. The gear ratio is determined according to the parameters of the PMSM, performance requirement in wheel side and kinematics analysis of the suspension structure. The linear motion of the suspension and the rotation motion of the PMSM are transferred to each other via bell crank.

Benchmarked with double wishbone suspension, a test bench of the AERS is proposed and it is shown in Fig. 11.

The active control force of PMSM can be transferred to wheel side via gearbox, bell crank and push bar and vice versa. Thus the energy excited by uneven road can be harvested by the PMSM.

The kinematics and dynamics of the test bench are simulated using ADAMS. The structural parameters are optimized to ensure the achievement of ideal control force in wheel side. The structure is designed in consideration of diminishing the nonlinear kinematics characteristics. Structural properties and nonlinear characteristics are analyzed and thus provide a foundation for control strategy design (Yin, et al., 2015). It should be noted that, the volume of PMSM utilized in this test bench seems too large and unfeasible to real car. Actually, the peak power of this PMSM is 16kW and it is much beyond the requirements. The reason is we have this PMSM at hand and we can just make use of it for the test bench due to the limited resource we have. And we believe we may benefit more freedom in research of control algorithms due to the extra capacity of the PMSM. The volume of the actuator can be designed much more compact in real application.

![CAD Model of test bench](image1)

(a) CAD Model of test bench

1. Frame; 2. Actuator; 3. Upper A-Arm; 4. Knuckle; 5. Wheel; 6. Push bar; 7. Tie rod; 8. Lower A-Arm; 9. Spring; 10. Bell crank

![Real test bench of the AERS](image2)

(b) Real test bench of the AERS

Fig. 11 Test bench of the AERS
5. Comparisons among three approaches

Performance evaluations and overall comparisons of three approaches are carried out.

5.1 Performance evaluation

Although sharing the same objective to diminish large unsprung mass negative effects, three approaches introduced above are intended to be applied on different car categories due to complexity and cost. However, in order to evaluate and compare the superiority among three approaches, a middle class car model with wheel side drive system is chosen as a result of compromise and it is benchmarked with. As a middle class electric car, the total weight is assumed as 1600kg according to the electric cars available on the market. Thus a quarter of it is 400kg. The unsprung mass of a quarter car without wheel side drive system is assumed as 30kg and the sprung mass is 370kg. The sprung mass and unsprung mass ratio is 12.3:1 which is reasonable for a middle class car. The mass of motor, gearbox and accessories of wheel side drive system is assumed as 40kg. Since the powertrain and drivetrain are moved to wheel side from the car body, the sprung mass is 330kg. In benchmarked passive suspension car model, the wheel side drive system is represented by a hub motor and thus the total unsprung mass which represented by the mass of wheel is 70kg. In the AERS approach, the quarter car model is similar with the passive one except that the damper of the passive one is replaced by an active actuator.

The parameters of the passive suspension are calculated using Eqs.(25) and Eqs.(26).

\[
K_s = (2 \cdot \pi \cdot n)^2 \cdot M_b
\]

(25)

\[
C_s = \psi \cdot (2 \cdot \sqrt{K_s \cdot M_b})
\]

(26)

Where \( n = 1.25 \), \( \psi = 0.27 \) are chosen as a balance of ride comfort and road holding of benchmark middle class car.

The parameters of benchmark car model and car models of three approaches are shown in Table 3.

| Parameters                          | Symbol | Passive | Integration | Dynamic absorber | AERS |
|-------------------------------------|--------|---------|-------------|------------------|------|
| Mass of body (kg)                   | \( M_b \) | 330     | 330         | 330              | 330  |
| Mass of wheel (kg)                  | \( M_w \) | 70      | 30          | 30               | 70   |
| Mass of motor (kg)                  | \( M_m \) | /       | 40          | 40               | /    |
| Moment of inertia of trailing arm assembly (kg\(\cdot m^2\)) | \( J_1 \) | /       | 2.8         | /                | /    |
| Moment of inertia of motor assembly (kg\(\cdot m^2\))     | \( J_2 \) | /       | /           | 0.253            | /    |
| Suspension stiffness (N/mm)         | \( K_s \) | 20.4    | 20.4        | 20.4             | 20.4 |
| Suspension damping (N/s/mm)         | \( C_s \) | 1.4     | 1.4         | 1.4              | /    |
| Tyre Stiffness (N/mm)               | \( K_f \) | 192     | 192         | 192              | 192  |
| Suspension working stroke (m)       | SWS    | ±0.08   | ±0.08       | ±0.08            | ±0.08|
| Absorber Stiffness (N/mm)           | \( K_{ab} \) | /       | /           | 25               | /    |
| Absorber damping (N/s/mm)           | \( C_{ab} \) | /       | /           | 1.1              | /    |
| Length of trailing arm (m)          | \( L \) | /       | 0.4         | /                | /    |
| Distance between motor and joint (m) | \( L_1 \) | /       | 0.2         | /                | /    |
| Distance between motor and wheel center (m) | \( L_2 \) | /       | /           | 0.14             | /    |
| Angle between motor-joint and wheel-joint (°) | \( \beta \) | /       | 30°         | /                | /    |

Dynamic equations of introduced three approaches are described above. Optimal control is utilized in the AERS approach. The dynamic equations of passive suspension are basing on traditional quarter car model with two degrees of freedom and they are omitted for brief. The frequency responses of BA, DTL and SWS of different approaches are
compared in Fig. 12, Fig. 13 and Fig. 14 respectively.

Fig. 12 Frequency responses of body acceleration of different approaches

Fig. 13 Frequency responses of dynamic tyre load of different approaches

Fig. 14 Frequency responses of suspension working stroke of different approaches
The transient responses of body acceleration, dynamic tyre load and suspension working stroke are done in time domain using Matlab Simulink. The road excitation in time domain is generated using filtered white noise method and it is detailed in previous work (Yin, et al., 2015). The simulations simulate the car models running on C-class road condition for 2000 seconds at the speed of 5m/s, 10m/s and 20m/s respectively. The reason for this road condition and speeds is that the road conditions of most national roads in china are between B-class and C-class and the speed limits of these roads are normally between 60km/h and 80km/h. The simulation condition covers most of the running conditions. The results of simulation for 2000 seconds are shown in Table 4.

To show the responses in time domain, the first 20 seconds of the simulation results at the speed of 10m/s are taken as an example. The other two simulations at the speed of 5m/s and 20m/s provide similar results. The 20 seconds of road excitation in time domain is shown in Fig. 15. It represents a C-class road profile with the distance of 200 meters according to the simulation condition. With the road excitation, the transient responses of body acceleration, dynamic tyre load and suspension working stroke of different approaches are shown in Fig. 16, Fig. 17 and Fig. 18 respectively.

Fig. 15 20 seconds of road excitation in time domain

Fig. 16 Transient responses of body acceleration of different approaches
Fig. 17 Transient responses of dynamic tyre load of different approaches

![Dynamic tyre load graph](image1)

Fig. 18 Transient responses suspension working stroke of different approaches

![Suspension stroke graph](image2)

Table 4 Evaluations of suspension performance

| Speed: 5m/s, Road condition: C-class | Passive “SRI” CWDS | “DVA” CWDS | AERS |
|-------------------------------------|---------------------|-------------|------|
| BA(m/s²)s                           | 0.622               | 0.589       | 0.565| 0.477|
| DTL(N)                              | 418.0               | 405.1       | 320.3| 449.5|
| SWS(mm)                             | 5.91                | 5.95        | 5.72 | 5.51 |

| Speed: 10m/s, Road condition: C-class | Passive “SRI” CWDS | “DVA” CWDS | AERS |
|--------------------------------------|---------------------|-------------|------|
| BA(m/s²)s                            | 0.925               | 0.859       | 0.820| 0.732|
| DTL(N)                               | 660.4               | 626.3       | 497.7| 719.3|
| SWS(mm)                              | 8.36                | 8.38        | 8.03 | 7.78 |

| Speed: 20m/s, Road condition: C-class | Passive “SRI” CWDS | “DVA” CWDS | AERS |
|--------------------------------------|---------------------|-------------|------|
| BA(m/s²)s                            | 1.333               | 1.231       | 1.174| 1.049|
| DTL(N)                               | 968.3               | 911.5       | 732.1| 1057|
| SWS(mm)                              | 11.96               | 11.95       | 11.47| 10.72|
The simulation results show that the ride comfort of “Integration design” approach is slightly better than the passive one. During all the running conditions, the AERS approach realizes the best ride comfort while the “Dynamic absorber” approach has improved the ride comfort compared with the passive one as well as the “Integration design” approach. The “Dynamic absorber” approach features least dynamic tyre load which means the best road holding ability is obtained. The “Integration design” approach also improves the dynamic tyre load compared with the passive one. The dynamic tyre load of AERS is worst in compromise to achieve the best ride comfort. However, in some extreme conditions, the control strategy can be tuned to improve the road holding ability in sacrifice of ride comfort to ensure optimal driving safety. Three approaches utilize less or equivalent suspension working stroke in different magnitude compared with the passive one and less bump stops are expected thus the ride comfort and road holding ability are improved.

5.2 Overall comparison

The “SRI” CWDS is constituted by a motor and a gearbox. Besides being powertrain and drivetrain system, the gearbox housing also plays a role as the trailing arm of a suspension. The total cost of the powertrain, drivetrain and suspension is promising to be reduced compared with the conventional hub motor and passive suspension. Benefits from integration design, this configuration is easy to be designed and manufactured, and it saves efforts during the assembly process. However, design a proper mechanism to realize the steering function is complicated on this configuration and thus the “SRI” CWDS is more feasible on rear wheels. This configuration is especially suitable for A-class or lightweight vehicle which concerns cost efficient more.

The “DVA” CWDS is constituted by a motor, a gearbox and an extra set of spring and damper. The motor or motor with gearbox should have one degree of translational or rotational freedom. The layout of extra set of spring and damper, bump stops and guide mechanisms need to be taken good care of. Thus the complexity of the structural design is increased. The guide mechanism which ensures one degree of translational or rotational freedom needs to be manufactured in high accuracy since the transmission function should be ensured during the vibration. The “DVA” CWDS improves the suspension performance in all the aspects compared with the conventional hub motor and passive suspension. However, due to extra wheel side space occupied by the “DVA”, this configuration is not easy to realize steering function and thus it’s more feasible on rear wheels. Since the increasing cost of this configuration is inevitable, this is especially suitable for vehicles which require a premium ride comfort and driving safety, such as middle class vehicles.

The AERS provides the best ride comfort or driving safety and benefits the best potential to balance the requirement between ride comfort and driving safety. Besides, energy regenerative function provides a possibility to extend the drive range of EVs. However, the excess high accuracy sensors, high speed control systems and an electromagnetic actuator have much increased the cost of this system. Various driving situations also require a great effort on research and development of control strategies. Since the AERS could be applied on in-wheel drive system, the steering function is easy to realize and thus the AERS is feasible on both front and rear wheels. The AERS is especially suitable for customers who are pursing excellent ride comfort and driving safety experience and not so sensitive to the price of the vehicle. Thus the luxury or high performance vehicle is the target vehicle of the AERS.

Overall comparisons between the “SRI” CWDS, the “DVA” CWDS and in-wheel drive system with the AERS are listed in Table 5.

|                   | “SRI” CWDS | “DVA” CWDS | AERS      |
|-------------------|------------|------------|-----------|
| Simplicity        | ★★★★★     | ★★★★★     | ★★★★★★   |
| Cost efficient    | ★★★★★     | ★★★★★     | ★★★★★★   |
| Suspension performance | ★★☆☆☆       | ★★★☆☆     | ★★★★★    |
| Front wheel or rear wheel | Rear         | Rear       | Front and rear |
| Application       | A-class or light weight vehicle | Middle class vehicle | Luxury or high performance vehicle |

* ☆ represents lower score, ★ represents higher score
6. Summary and Conclusion

To diminish large unsprung mass negative effects of wheel side drive EVs, this paper introduces the “SRI” CWDS, the “DVA” CWDS and in-wheel drive system with the AERS basing on the idea of mass transfer together with integration design, DVA and active suspension design respectively. Corresponding dynamic models are built to analyze the suspension performance and related schemes are introduced to illustrate the feasibilities. Evaluations of suspension performance prove and show the superiorities in different magnitude of these three approaches compared with conventional passive configuration. The complexity, cost, suspension performance of three approaches are compared and discussed. It is concluded that the “SRI” CWDS, the “DVA” CWDS and in-wheel drive system with the AERS are suitable for rear wheels of A-class or lightweight vehicles, rear wheels of Middle-class vehicles and front and rear wheels of luxury or high performance vehicles respectively.

Basing on the ideas introduced, more structures are expected to be proposed in the future. And prototypes would be built according to specific wheel side drive EVs to diminish large unsprung mass negative effects of conventional configuration.

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Abbreviations

| Abbreviation | Description                          |
|--------------|--------------------------------------|
| EV           | Electric Vehicle                     |
| SRI          | Suspension & Reducer Integrated      |
| DVA          | Dynamic Vibration Absorber           |
| CWDS         | Close-to-Wheel Drive System          |
| AERS         | Active and Energy Regenerative Suspension |

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