Linear Programming Control for Hybrid 8-wheel-drive Vehicle

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ABSTRACT

For armored vehicle, the 0-32km/h and 0-60km/h acceleration time is a key tactical and technical driving performance index. In order to improve the driving performance of a hybrid 8-wheel-drive vehicle, the control method was proposed. The method uses linear programming to follow the demand driving torque so as to decrease the acceleration time and minimize the additional yaw moment so as to improve the handling performance. In order to verify the method, a simulation platform of 11 DOFs was built. The simulation results show that 0-32km/h and 60km/h acceleration time was decreased by 30% and 13% which is a significant improvement compared to the common level of main battle armored vehicle.

INTRODUCTION

In recent years, HEVs have made outstanding contributions to improve vehicle energy efficiencies and they are expected to save a lot of spending in the military field [1]. With the innovation of configuration, the margin of dynamic and control can be further extended [2]. Among different kinds of configurations, the driving performance of the HEVs equipped with independent motors can be further improved because of the flexible torque allocation strategy. Based on above advantages and strong demand of energy saving and driving performance especially on the off-road, the technology of hybrid electric distributed motor drive has been used in armored vehicle area [3].

Most researchers have concentrated on the power management strategy [4] and handling performance control [5], few researchers discussed the driving performance control based on the slip ratio. Generally, the previous researches focused on the conventional 4-wheel passenger cars. Hori et al. [6] proposed a technique of optimal slip ratio control method using the advantages of precise torque generation and the basic effectiveness is demonstrated by real experiment using the “UOT Electric March”. Ozkop et al. [7] proposed a fuzzy logic sliding mode controller on an electronic differential system and simulation results show that the controller has better efficiency and performance compared to those of PID and SMCs. Wu et al. [8] presented an acceleration slip regulation strategy to make the full use of road grip by controlling the slip ratio.
Due to the significant work of above authors, the Driving performance research of HEVs has been developed. The previous researches mainly focused on the conventional four-wheel passenger cars, few researches have concentrated on the armored hybrid 8WD vehicle with multi-axle.

**CONFIGURATION AND MODEL**

As the key member of military equipment, 8WD vehicles are mainly responsible for military investigation, personnel transportation, logistics support and other tactical functions. Besides, it plays an irreplaceable role in the complex and modern battlefield because of the great loading ability and off-road performance. Due to the increasing demand of the fuel efficiency and quiet working performance, the all E-drive vehicle technology has been widely used [9].

The conventional 8WD vehicles transfer the power form the engine to each wheel through mechanical device such as gear box and shaft which cases a typical problem of complex arrangement and long transmission train. Compared with the traditional configuration, the all E-drive vehicle transfer the power from the engine to each wheel through generator and motor, AC/DC converters and electric line, which contributes to a brief arrangement and short transmission train. Besides, the electric motor can provide responses and important Information such as torque and speed. So a variety of dynamics control functions can be easily implemented by controlling the drive or brake torque of the electric motor. However, the efficiency of motor in high rotation speed will be much lower.

Inspired by the comparison between traditional transmission and electric drive, the authors proposed a hybrid 8WD armored vehicle with a different powertrain configuration as following Figure 1 and many tests of component and vehicle are carried out on the prototype.

Vehicle power cabin is built including engine and AT, generator and gear box. The shaft connected to the Engine separates the power to double streams; one stream of power drives the AT and the other generator. The front two axles are driven by the power from AT. The power from generator drives the four independent motors or charges the battery through Motor/Generator controller. The M/G controller is an integrated inverter which controls the torque of four driving motors and output current of generator.

![Figure 1. Simulation Platform.](image-url)
For some previous researches, the control method was only analyzed by a vehicle body dynamic model with tire mechanical properties to investigate the driving behavior. However, for a hybrid vehicle, the performance of engine, generator, independent motors and battery have a great influence on the driving performance, which should also be focused on.

The external characteristics of the engine, generator and independent motors were obtained by the bench test. The parameters of the battery PNGV (the Partnership for a New Generation of Vehicle) model were obtained by the lithium batteries charge-discharge test and HPPC test. The double parameter gear shift logic of the AT was obtained by the bench test. Base on above tests, the transmission subsystem was built.

The power of the engine was separated by the simple gear1 and gear2, and input rotation speeds are equal. However, the input torques of the two gears is different, because the command of generator can increase and decrease the demand of torque.

For the initial SOC of the battery is 60%, the independent motors can drive the vehicle without the power of the generator.

Some main parameters of the vehicle are shown in Table I.

| Parameter                     | Value                  |
|-------------------------------|------------------------|
| Vehicle mass (m)              | 24,000 kg              |
| C.G height (H)                | 1.3 m                  |
| Track width (B)               | 1.7 m                  |
| Distance from 1st axle to C.G. (L₁) | 2.166 m               |
| Distance from 2nd axle to C.G. (L₂) | 0.866 m               |
| Distance from 3rd axle to C.G. (L₃) | 0.734 m               |
| Distance from 4th axle to C.G. (L₄) | 2.034 m               |
| Radius of wheel (r)           | 0.59 m                 |
| Engine max power              | 330 kW@2200RPM         |
| Electric motor max power      | 65 kW@2500RPM          |
| Generator max power           | 110 kW@4200RPM         |
| Battery rated voltage         | 550 V                  |
| Battery capacity              | 60 A.H                 |

Some main parameters of the vehicle are shown in Table I.
The vehicle dynamic model is the most important part in the simulation platform to investigate the vehicle driving behavior. The vehicle dynamic model has 11 DOFs, including translational motions (longitudinal and lateral), rotational motion (yaw) of the vehicle body, and rotational motions of eight wheels. During the modeling process of vehicle body, it is assumed that the vehicle is placed on a plane surface with a local coordinate frame denoted by \((X, Y, Z)\) assigned at its center of mass shown in Figure 3. The vehicle is assumed to be on a plane surface, regardless of the influence of the vertical displacement, pitch and roll angle of the body.

Define \(i\) the number of axle and \(j\) the mark of left or right. Define two following variables:

\[
X_{ij} = F_{xij} \cos \delta_{ij} - F_{yij} \sin \delta_{ij}
\]

\[\text{(1)}\]

\[
Y_{ij} = F_{xij} \sin \delta_{ij} + F_{yij} \cos \delta_{ij}
\]

\[\text{(2)}\]

Considering the force equilibrium along \(X\) and \(Y\) axis and the moment balance around \(Z\) axis, then the dynamic equations of the vehicle body expressed in \((X, Y, Z)\) coordinate frame can be written as:

\[
m(\dot{u}_x - u_x \omega) = \sum_{i=1}^{4} \sum_{j=1}^{2} X_{ij} - mg \sin \phi - \frac{1}{2} C_d \rho A (u_x - u_w)^2
\]

\[\text{(3)}\]

In equation (3) the \(C_d\) is aerodynamic drag coefficient, \(\rho\) is mass density of air, \(A\) is effective frontal cross-sectional area and \(u_w\) is headwind speed.

\[
m(\dot{u}_x + u_x \omega) = \sum_{i=1}^{4} \sum_{j=1}^{2} Y_{ij}
\]

\[\text{(4)}\]

\[
I_z \dot{\omega} = \sum_{i=1}^{4} (X_{i1} - X_{i2}) \cdot \frac{B}{2} + \sum_{j=1}^{2} (\sum_{i=1}^{2} Y_{ij} L_i + \sum_{i=3}^{4} Y_{ij} L_i)
\]

\[\text{(5)}\]

The dynamic equation of each wheel can be expressed as: (Figure 8)

\[
J \cdot \dot{w}_{ij} = T_f - F_{xij} \cdot r
\]

\[\text{(6)}\]

\[
T_f = N_{ij} \cdot f \cdot r
\]

\[\text{(7)}\]

The longitudinal slip ratio of driving wheel can be expressed as:

\[
\lambda_{ij} = \frac{W_{ij} r - V_{ij}}{W_{ij} r}
\]

\[\text{(8)}\]
The tire theory model of UA was established and the longitudinal and lateral force between tire and ground can be calculated by slip ratio, center speed, side slip angle and load of each wheel.

The influence of the load transfer is considered, and the steering angle of wheel is based on the test data. The center speed and side slip angle of each wheel can be calculated by the state variables of the vehicle body and steering angle of the wheel.

**CONTROL AND VERIFICATION**

Control strategy includes five modules, ‘Total Torque Demand Calculation’, ‘Identification of Optimal Slip Ratio’, ‘Set up Constraints’, ‘Set up Target’ and ‘Torque Allocation’.

1. In ‘Total Torque Demand Calculation’ module, total torque demand of the vehicle ($T_a$) will be calculated by the throttle ($\psi$) and vehicle speed ($u_x$).

2. In ‘Identification of Optimal slip ratio’, the adhesion coefficient of each wheel will be calculated:

$$\mu_{ij} = \frac{F_{xij}}{N_{ij}} \quad (9)$$

The adhesion characteristics between tire and road can be expressed as:

$$\mu = C_1 (1 - e^{-C_2 \lambda}) - C_3 \lambda \quad (10)$$

The fitting coefficient ($C_1 C_2 C_3$) was obtained by the experiments on different road and there is a maximum adhesion coefficient in a kind of road. The road can be identified by the fuzzy controller, which compares the $\mu_{ij}$-$\lambda_{ij}$ curve with the equation (10). Maximum driving torque of each wheel can be calculated based on the maximum adhesion coefficient ($\mu_{max}$) and load of wheel ($N_{ij}$).

3. In ‘Set up Constraints’ module, the constraints can be written as:

$$\begin{align*}
T_j &\leq \mu_{max} : N_{ij} \\
\frac{T_g w_e}{9550} - \sum_{i=3}^{4} \sum_{j=1}^{2} \frac{T_j w_{ij}}{9550} &\in (Pc(SOC), P_{disc}(SOC)) \quad (11)
\end{align*}$$

In equation (11), $T_e$ is torque of engine, $T_g$ is torque of generator, $w_e$ is rotary speed of engine, $P_c(SOC)$ is the charging power function of SOC, $P_{disc}(SOC)$ is the discharging power function of SOC. The range of values is shown in figure 4 between red and black line which is related with the capability and life of the battery.
The torque of front four wheels is always equal for the mechanical configuration which can be written as:

\[ T_{11} = T_{12} = T_{21} = T_{22} = \frac{T_e - T_g}{4} \cdot i_m \]  

(12)

In equation (12), \( i_m \) is mechanical transmission ratio. In order to simplify the constraints, we make following control.

\[ \begin{cases} T_{31} = T_{41} \\ T_{32} = T_{42} \end{cases} \]  

(13)

So the simplified constraints can be written as:

\[ \begin{cases} T_{31} \leq a \\ T_{32} \leq b \\ (T_e - T_g) \cdot i_m \leq 4c \\ \frac{T_g W_e}{9550} - \sum_{i=3}^{2} \sum_{j=1}^{2} \frac{T_{ij} W_{ij}}{9550} \in (d, e) \end{cases} \]  

(15)

In the equation of (15), \( a, b, c \) are value of torque, \( d, e \) are the value of power.

(4) In ‘Set up Target’ module, two functions are defined:

\[ \begin{cases} f_1 = \left| T_a - 4T_{11} - 2T_{31} - 2T_{32} \right| \\ f_2 = \left| \sum_{i=1}^{4} \left( F_{X\|i} - F_{X\|2} \right) \cdot \frac{B}{2} \right| \]  

(16)

If function 1 gets the minimum value, the total driving toque of the vehicle is the biggest. If the function 2 gets the minimum value, the influence of the additional yaw moment is the least.
In ‘Torque Allocation’ module, the linear programming method was carried out to get the torque of generator, torque of left independent motor and torque of right independent motor. The simulation verification was carried out by 0-32km/h and 0-60km/h acceleration test on well adhered pavement and the figure 5 is shown below. The blue line stands for our vehicle and the other is the common level.

![Figure 5. Result of acceleration test.](image)

**SUMMARY**

In this paper, a hybrid 8WD configuration and linear programming control was proposed to improve the driving performance of armored vehicle. The constraints and targets were set up and the control of motors and generator was solved through linear programming method. In the end, the simulation results showed that the acceleration of 0-32km/h was decreased by 30%, and the acceleration of 0-60km/h was decreased by 13% due to the configuration and torque allocation strategy.

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