A novel anti-lock braking strategy for high speed light weight dual-motor electric drive tracked vehicles

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Abstract. Anti-lock braking strategies are of great importance in improving manoeuvrability of vehicles. However, studies on that of tracked vehicles are rare compared with wheeled vehicles due to their complicated drivetrain and the sluggish of their braking system. Dual-motor electric drive tracked vehicles (DDTVs), whose drivetrain differs significantly from the traditional ones, have drawn a lot of attention in the trends of hybridization and electrification for tracked vehicles, and made it practical to apply advanced braking control strategies with the application of permanent magnet synchronous motors (PMSMs). This paper develops a novel anti-lock braking strategy with a sliding mode controller and a rule based braking torque allocating method to enhance the braking stability. Simulations are conducted under a typical low adhesion condition to compare the control performances of the proposed strategy with normal full braking strategy and traditional anti-lock braking strategy transplanted from wheeled vehicles. Simulation results show that the proposed control strategy performs better than the other two strategies in reducing stopping distances while still has the ability of regenerate braking energy.

1. Introduction

DDTVs have drawn lots of attentions in the trends of hybridization and electrification of tracked vehicles [1]. The tracks of a DDTV are driven separately by two PMSMs attached to the drive wheels through each semi axle with a mechanical brake installed on. The schematic diagram of it is shown in figure 1.

Figure 1. Schematic diagram of a DDTV
Combined braking system of tracked vehicles normally refers to hydraulic-mechanical braking systems with hydraulic retarders [2]. However, as for high speed light weight DDTVs, PMSMs can replace the role of them and enhance the dynamic response when braking. A strategy called full braking, which means the braking system provides as much braking torque as possible is widely used in tracked vehicles for emergency braking. Even in latest researches of DDTVs, the focus are still on how to coordinate the mechanical brakes and PMSMs produce enough braking torque [3, 4]. However, adopting this strategy will cause the locking of drive wheels and tracks, resulting in a poor braking performance.

To overcome similar drawbacks in the field of wheeled vehicles, traditional ABS system relies on high frequency switch of solenoid valves in braking circuits as well as advanced slip ratio control strategies are studied in previous researches [5-8]. Although research on traction control of tracked vehicles using sliding mode method has been conducted long before [9], to brake stably on a low adhesion surface is still difficult for a tradition tracked vehicle due to the sluggish response of a typical mechanical brake. However, with the participant of PMSMs with fast torque response ability, applying an advanced anti-lock braking control strategy is practical for DDTVs.

The organization of this paper is as follows. In section 2, models related to the braking process of DDTVs are introduced. In section 3, the design of proposed anti-lock braking strategy is introduced. Matlab/Simulink simulations of the proposed control strategy as well as full braking and traditional ABS strategy are shown and compared in section 4. Finally, conclusions are given in section 5.

2. Modelling

The walking structures of a tracked vehicle is significantly different from that of a wheeled vehicle, thus models related to the braking process of a tracked vehicle need to be built specially. Since the features we focused on of PMSMs and mechanical brakes in this case are their torque output capacity and response ability, simple models of first-order process based on experiments is suitable. As a result, what we need to focus on is the modelling of vehicle dynamics, track-ground conditions and driver operations.

2.1. Longitudinal dynamics of a DDTV

As shown in figure 2, the movement of a tracked vehicle can be divided into the rolling of drive wheels, the longitudinal motion of vehicle body, the rotational motion of the tracks, as well as the rolling of road wheels, idler and support rollers. Assuming that the track is uniform and soft, and there is no slip or slide between the track and any wheel, a semi-vehicle longitudinal dynamic model in braking situation can be established. Similarly with wheeled vehicle, the longitudinal motion of vehicle body and the rolling of drive wheel can be denoted as:

\[
\begin{align*}
F_x + F_{aero} + Mv_x &= 0 \\
j\dot{\omega} - F_tr + T_b &= 0
\end{align*}
\]  

(1)

where \(F_x\) is the track-ground resultant force of one track, \(F_{aero}\) is half of the air drag, \(M\) is half of the DDTV’s total mass, \(v_x\) is the longitudinal velocity of the vehicle, \(j\) is the rotational inertia (RI) of drive wheel and transmission system attached to it, \(\omega\) is the rotational speed of the drive wheel, \(F_t\) is the force applied on drive wheel by the track, \(r\) is the radius of the drive wheel, and \(T_b\) is the braking
torque applied on drive wheel by the transmission system.

In order to describe the motion of the track and other wheels attached to it, a method of calculating the equivalent rotational inertia (ERI) based on energy conservation theory is proved effective in [10, 11]. Taken the slip ratio defined in equation (2) into consideration, the ERI of track and wheels when braking can be expressed as equation (3).

\[
\begin{align*}
\lambda &= \frac{(v_x - \omega r)}{v_x} \\
J_{eri} &= n_{oi} J_{wi} r^2 / \omega_w \\
f_{eri} &= m_t f [1 + 1/(1 - \lambda)^2 - 2 \cos \phi_D / (1 - \lambda)] \\
f_{eri} &= m_t r [1 + 1/(1 - \lambda)^2 - 2 \cos \phi_D / (1 - \lambda)] \\
f_{eri} &= m_t u r^2 [2 - \lambda] / (1 - \lambda)^2 \]
\end{align*}
\]

where \( J_{oi} \) is the RI of the \( i \)th kind of wheel, \( n_{oi} \) is the quantity of this kind of wheel, and \( m_t f, m_t r, m_t u, m_t b \) are the masses of the front part, the rare part, the upper part as well as the bottom part of the track respectively. Thus, the longitudinal dynamics of the DDTV can be amended by replace \( J \) with \( J_{eri} \) which is the summation of equation (3) and \( J \) in equation (1).

2.2. Model of track-ground

Due to the differences between the contact form of track-ground and tire-road, there are no suitable analytical models of track-ground for tracked vehicle. Thus, the best way is establishing an experimental model. Some medium and low adhesion track-ground conditions are shown in figure 3.

According to figure 3, we can find that when the slip ratio nears 0.2, the adhesion coefficient achieves the largest value, thus the value 0.2 can be called the optimal slip ratio.

2.3. Model of driver operation

The travel of brake pedal can be interpreted as a demand deceleration and it is usually considered as meaningful after a small idle motion, as well as deemed to need a full braking when nearing the terminal. Thus the demanded deceleration can be defined proportional to the effective pedal travel.

3. Design of anti-lock braking strategy

According to the national military standard of China, the maximum deceleration must be higher than 5m/s\(^2\) and it is usually set as 5.5 m/s\(^2\) [4]. As shown in figure 3, it’s impossible to reach such deceleration when the vehicle runs on a low adhesion ground and will leads to the locking of tracks and wheels if braking torque remains unchanged. Thus, a sliding mode strategy which calculates demand braking torque in real time can overcome this problem.

Substituting \( \lambda \) and \( \dot{\omega} \) into the derivation of \( \lambda \) we can get that:

\[
\dot{\lambda} = [(1 - \lambda)^2 f_{equ} v_x + (1 - \lambda) F_{r x} r^2] / [\omega t J_{eri} + (1 - \lambda) t b \omega / (1 - \lambda)]
\]

A zero-order sliding surface can be chosen to track the demanded slip ratio and the sliding surface and its derivation with respect to time is:

\[
\begin{align*}
\dot{s} &= \lambda - \lambda_d \\
\dot{\lambda} &= \lambda - \dot{\lambda}_d
\end{align*}
\]

In order to converge quickly when \( s \) is large and reduce chattering when \( s \) is small, a sliding mode control method using exponential approaching law is suitable for this control system [12]. Considering that \( \lambda_d \) is the optimal slip ratio which is a constant, the derivation of sliding surface can be written as:

\[
\dot{s} = -\varepsilon \text{sgn}(s) - ks = \dot{\lambda}, \quad \varepsilon > 0, \quad k > 0
\]

where \( \text{sgn}(s) \) is symbolic function of \( s \), \( k \) should be designed relatively large and \( \varepsilon \) should be relatively small. Thus, the total demanded braking torque converted to drive wheel can be derived as:

\[
T_{bd} = r F_x + (1 - \lambda) J_{eri} v_x / \omega t + \omega t J_{eri} \text{sgn}(s) + ks / (1 - \lambda)
\]
However, the total demanded braking torque changes rapidly in the braking process, the performance will meet a significant degradation if there doesn’t have a proper allocating method. The braking system of a light weight DDTV consists of mechanical brakes and PMSMs and these actuators have their own advantages and drawbacks. The mechanical brake can offer a large braking torque, but its dynamic response is quite poor. As for PMSM, its torque response speed is much faster than mechanical brake, however, the maximum braking torque it can provide declines with the increasing of revolution speed when rotating faster than its rated speed, even if its rated torque is already much smaller than mechanical brakes. What’s more, the performances of the PMSM also get worse when the revolution speed is low.

Therefore, several fundamental allocating rules could be set as: 1. the total braking torque should be divided into a steady part and a dynamic part; 2. the steady part is provided by mechanical brake and the dynamic part is provided by the PMSM; 3. the PMSM exits braking process when the vehicle speed is slower than 5 km/h; 4. the PMSM does not output any positive torque when braking; 5. to supplement the shortage of the dynamic part of total demanded braking torque when \( s \) is large, a compensating coefficient is introduced to compensate the difference of them using mechanical brake.

Thus, substituting the definition of sliding surface into equation (7) can get that

\[
\begin{align*}
T_{bsd} & = rF_x + (1 - \lambda)\frac{\varphi_{err}\dot{v}}{r} + \chi(T_{bddd} - T_{mo}) \\
T_{bddd} & = \alpha\frac{\varphi_{err}(\epsilon + P_d)\text{sign}(s)/(1 - \lambda) + j_{err}(\varphi_{err}/r + \omega k/1 - \lambda)s}{(8)}
\end{align*}
\]

where \( T_{bsd} \) is the steady part of total demanded braking torque, while \( T_{bddd} \) is the dynamic part. \( T_{mo} \) is the real braking torque of PMSM converted to the drive wheel and \( \chi \) is the compensating coefficient. To reduce chattering, the symbolic function can be replaced by a saturation function \( sat(s/\Phi) \).

### 4. Simulations

Simulations are taken with the proposed control strategy as well as two other strategies to compare the braking performance and verify the effectiveness. All the three strategies are separately:

- **FB**: which is full braking strategy. Mechanical brake and PMSM are working simultaneously to provide enough braking torque to achieve demand deceleration despite the track-ground condition;
- **TABSC**: which transplant a traditional ABS with PWM driven solenoid valves added in wheel cylinders. When it is triggered, PMSM exits and all the braking torque is provided by mechanical brake.
- **SMSC**: which is the proposed control strategy that mechanical brake and PMSM work simultaneously.

#### Table 1. Main parameters of the DDTV for simulation

| Parameter                          | Value   | Parameter                          | Value   |
|-----------------------------------|---------|-----------------------------------|---------|
| Vehicle mass, \( m(t) \)          | 32      | Approach angle, \( \varphi_{A}(^\circ) \) | 27.3    |
| Drive wheel radius, \( r(m) \)    | 0.309   | Departure angle, \( \varphi_{D}(^\circ) \) | 35.6    |
| Frontal area of the vehicle, \( A(m^2) \) | 5.36 | Ratio of the side reducer, \( ir \) | 4.59    |
| Aerodynamic drag, \( C_D \)       | 1       | Ratio of the motor reducer, \( im \) | 2.2     |
| Air density, \( \rho(Nm^2/m^4) \) | 1.22 | Rated power of the PMSM, \( P_m(kW) \) | 375     |
| RI of drive wheel & powertrain,(kgm\(^2\)) | 158.8 | Rated speed of the PMSM, \( n_m(rpm) \) | 3000    |
| RI of idler,(kgm\(^2\))           | 31.2    | Max speed of the PMSM, \( n_{max}(rpm) \) | 9000    |
| RI of road wheel,(kgm\(^2\))      | 23.7    | Quantity of road wheels, one side | 6       |
| RI of support roller,(kgm\(^2\))  | 13      | Quantity of support rollers, one side | 2       |

Five indicators are chosen to evaluate the performances which are stopping time \( t \), stopping distance \( S \), mean value of slip ratio \( \bar{\lambda} \), standard deviation of slip ratio \( \sigma \) and regenerated energy \( E \).

The simulations are taken in the environment of Matlab/Simulink R2015b and the schematic diagram of the model is shown in figure 4. The DDTV is braked at a speed of 75km/h on snowfield.
and the brake pedal is set to increase from 0 to 100% in 10 ms half a second after the accelerator pedal decreased to 0. The start point of plotting and calculating the indicators is defined at when accelerator pedal started to decrease. The main parameters of the DDTV for the simulations are listed in table 1.

Simulation result with the proposed strategy is shown in figure 5, and the effect of these strategies can be reflected in table 2. In figure 5, VV, TV, MBT, MRT and TBT are the abbreviations of vehicle velocity, track velocity, mechanical brake torque, motor regenerative torque and total braking torque respectively.

Table 2. Value of the indicators

|       | t(s) | S(m) | $\lambda$ | $\sigma$ | $E$(kJ) |
|-------|------|------|-----------|----------|---------|
| FB    | 9.78 | 101.86 | 0.8413 | 0.3050 | 516.29 |
| TABSC | 9.01 | 97.24 | 0.1674 | 0.1087 | 0 |
| SMSC  | 8.65 | 92.19 | 0.1894 | 0.0396 | 316.7 |

It is obvious that whether transplanting a traditional ABS or applying the proposed control strategy can improve the braking performance, however, applying the proposed strategy end up with a much larger improvement. The reason can be found in figure 6 which shows the error of real slip ratio and the optimal one using TABSC and SMSC respectively. We can easily find the slip ratio of SMSC converges faster than that of TABSC without such oscillation. Hence, applying SMSC method can make the most use of the ground adhesion to decelerate the DDTV as the result shown in table 2.

5. Conclusion

The proposed anti-lock braking strategy utilizes the advantages of mechanical brake and PMSM, enables the braking system providing braking torque rapidly and accurately. When compared with FB, the stopping distance reduces by 9.49%. When compared with TABSC, the proposed one still has an advantage of reducing it by 5.19%. Although using the proposed strategy recovers less energy than using the FB strategy, it is negligible compared with the significant advances in other aspects. What’s more, applying the proposed strategy doesn’t have to change the original braking circuit of the DDTV, which can save the budget of adding solenoid valves of each wheel cylinder. In conclusion, applying the proposed anti-lock braking strategy can achieve best result with the minimal cost.

Figure 4. Schematic diagram of simulation model

Figure 5. Simulation results of emergency braking on snow

Figure 6. Error of slip ratio using TABSC and SMSC
References
[1] Yuan Z, Fengchun S, Chengning Z 2007 J. B. Inst. Techno. 27 29-33
[2] Guangjun Z, Jiangang L 2010 CMCE (Changchun) pp 515-518
[3] Hui S, Jiangtao G, et al 2017 Acta. Armamentarii. 38(5) 1027-1034
[4] Hui S, Jiangtao G, et al 2017 J. TYUT 48(1) 79-85
[5] Patra N, Datta K 2012 ICACCCT(Ramanathapuram) pp 385-391
[6] Hamzah N, Yahay M, Selamat H, et al 2012 CSPA(Melak) pp 138-143
[7] Jonathan M, David C 2010 Vehicle. Syst. Dyn. 48 373-392,
[8] Bera K, Bhattacharya K, Samantaray K 2011 P. I. Mech. Eng. J-J. Eng. 225 918-934
[9] Zhejun F, Yoram K, David W 1995 ACC(Seattle) pp 1176-1177
[10] Jiajun Y, Xiaojun Z 2013 J. Vib. Shock. 32 68-72
[11] Zhe W, Haoliang L, Xiaojun Z, Zhaomeng C, Yong Y 2018 Sensors. 18(7), 1993-.
[12] Jinkun L 2017 Sliding mode control using MATLAB. (Amsterdam: Elsevier)