Improving traction and active safety of the wheeled vehicle by the distribution of the driving torque between its axles

O I Chudakov, V A Gorelov and E B Sarach

Bauman Moscow State Technical University (BMSTU), 105005, Moscow, Russian Federation, 2nd Baumanskaya St., 5

Abstract. One method to improve dynamic qualities, stability and cross-country ability of wheeled vehicles is to use Active Torque Transfer Systems (ATTS). An operating algorithm for the ATTS that ensures improved traction and dynamic properties, road-holding and trajectory stability of a two-axle wheeled vehicle has been suggested. Efficiency and feasibility of the suggested algorithm have been proved through simulation modeling. Lines of future research have been defined.

1. Introduction

There has been observed a trend in the modern automobile industry to improve power-to-weight ratio of wheeled vehicles (WV), which is followed by issues related to efficient implementation of increased tractive capacity. In bad weather conditions and, first of all, in vehicles with high-performance engines, significant wheel sliding can appear due to low friction coefficient combined with high tractive capacity. As a result, not only the created tractive capacity is decreased, but the response of vehicle as well. Solution of this problem requires new design concepts of the powertrain, which would allow the full use of the tractive potential and improvement of the powertrain characteristics by the optimization of its control laws and algorithms.

Currently, there are many known all-wheel drive systems providing redistribution of the driving torques between both the axles and the individual wheels of the vehicle.

One of the most popular options to implement four-wheel drive on two-axle passenger WV’s is connection of the second driving-axle (front or rear axle depending on the vehicle layout) using a multi-plate frictional clutch [1-5].

The following companies produce multi-plate clutches for WV: Borg Warner (used on Honda, Hyundai, Porsche), Magna Steyr (BMW, Mercedes-Benz), GKN Driveline (Nissan, Renault), Haldex Traction (Volkswagen, Volvo). Clutch plates pack compression may be performed using mechanical (BMW xDrive system), hydraulic (Haldex) and electromagnetic forces (Porsche PTM system).

The use of the Part-time 4WD systems have the following primary objectives:
1) increasing tractive capabilities of the vehicle;
2) improving stability of the vehicle driving in curves;

A possible criterion of the meeting of the first objective is the longitudinal acceleration value and the capability to drive over the low friction or non-homogenous surfaces, including the situation of driving up-slope. The second objective achievement can be assessed by the speed of the vehicle performing the test without loosing yaw and trajectory control. Efficiency of the Part-time 4WD systems mostly depends on their multi-disk frictional clutch control algorithms and can be assessed by means of the mathematical simulation [6-14]. Operating algorithm of Part-time 4WD systems made by foreign
manufactures is a commercial secret of particular manufacturing companies and it can not be found in public domain.

Operating algorithm of existing designs depends on many parameters, including temperature conditions of a clutch and its interaction with other electronic active safety systems (Anti-lock Braking System, Electronic Breakforce Distribution, Dynamic Stability Control, etc.). On the initial stage of development, we can evaluate the operating algorithm regarding its impact on tractive and dynamic properties, as well as WV stability [15-16].

The objective of this paper is to increase tractive and dynamic properties, as well as stability of two-axle WVs with a part-time front axle through torque biasing.

2. Theory

The authors developed an algorithm of connecting the first axle of the two-axle vehicle. As the initial data for the calculation of the torque that should be transmitted by the friction clutch, the algorithm uses the following sensors signals: crankshaft speed of the internal combustion engine (ICE), current gear in the gearbox; longitudinal and lateral accelerations of the vehicle center of mass; rotational rate of the vehicle; steer angle; rotational speeds of the wheels. The placement of the sensors is shown in figure 1.

![Figure 1](figure1.png)

In case of linear motion (it is generally thought that a linear motion is motion at steering-wheel angle of $\theta_{SW} \leq 15^\circ$), the system operates in a mode preventing rear axle skidding. We should calculate the relative difference between angular velocities of the front and the rear axles:

$$\Delta \omega_{rel} = 1 - \frac{\omega_{FL} + \omega_{FR}}{\omega_{RL} + \omega_{RR}},$$  \hspace{1cm} (1)

where $\omega_{FL}, \omega_{FR}$ – angular velocities of the front left and right wheels respectively, $s^{-1}$; $\omega_{RL}, \omega_{RR}$ – angular velocities of the rear left and right wheels respectively, $s^{-1}$.

Apart from the relative difference of angular velocities, torque ratio transmitted to the front axle ($h_M$) depends on the accelerator pedal position ($h_A$). Value $h_M$ is calculated according to the curve in figure 2.
Figure 2. Dependence $h_M(\Delta \omega_{rel})$: 1 – $h_A$ is less than 30%; 2 – $h_A$ is 60%; 3 – $h_A$ is more than 90%.

During nonlinear motion ($\theta_{SW} > 15^\circ$), all-wheel-drive system operates in Dynamic Stability Control mode. One of the main problems in developing Dynamic Stability Control (DSC) operation algorithm is to receive accurate information on quantitative values of WV movement parameters, which would allow to consider how such parameters go in lines with the ones chosen by a driver, predict when an emergency situation occurs and diagnose such situations (for instance, front or rear axle drift, a risk of roll-over, etc.).

In this paper we use the following definitions from Global Technical Regulation No. 8. Electronic Stability Control Systems:

**Ackerman Steer Angle** means the angle whose tangent is the wheelbase divided by the radius of the turn at a very low speed;

**Oversteer** means a condition in which the vehicle's yaw rate is greater than the yaw rate that would occur at the vehicle's speed as result of the Ackerman Steer Angle;

**Understeer** means a condition in which the vehicle's yaw rate is less than the yaw rate that would occur at the vehicle's speed as result of the Ackerman Steer Angle;

**Yaw rate** means the rate of change of the vehicle's heading angle measured in degrees/second of rotation about a vertical axis through the vehicle's centre of gravity.

Considering the definitions above, WV motion state is diagnosed, i.e. based on sensor values it should be determined to which of three possible states belongs the current data set:

– standard situation, no correction required
– understeer in progress, correction required
– oversteer in progress, correction required

Expressions to calculate theoretical values for WV wheels angular velocities for a WV steering at the Ackerman angle:

$$
\begin{align*}
R_i &= L / \tan(\theta_{SW}) ; \\
R_{RL} &= R_i - 0.5 \cdot B \cdot \text{sgn} \theta_{SW} ; \\
R_{FL} &= \sqrt{R_{RL}^2 + L^2} ; \\
R_{RR} &= R_i + 0.5 \cdot B \cdot \text{sgn} \theta_{SW} ; \\
R_{FR} &= \sqrt{R_{RR}^2 + L^2} ; \\
\omega_{FR} &= [\omega_{FR} + (\omega_{FL} / R_{FL} + \omega_{RR} / R_{RR} + \omega_{RL} / R_{RL}) \cdot R_{FR}] / 4 ; \\
\omega_{FL} &= (\omega_{FR} / R_{FR}) \cdot R_{FL} ; \\
\omega_{RR} &= (\omega_{FR} / R_{FR}) \cdot R_{RR} ; \\
\omega_{RL} &= (\omega_{FR} / R_{FR}) \cdot R_{RL} ;
\end{align*}
$$

(2)
where \( R_t \) – theoretical turning radius WV, m; \( L \) – wheelbase, m; \( B \) – gauge, m; \( R_{FL}, R_{FR}, R_{RL}, R_{RR} \) – radiuses of trajectories of wheels, m; \( \omega_{FL}, \omega_{FR}, \omega_{RL}, \omega_{RR} \) – actual values of angular velocities of the wheels, \( s^{-1} \); \( \omega_{FLt}, \omega_{FRt}, \omega_{RLt}, \omega_{RRt} \) – theoretical values of angular velocities of the wheels, \( s^{-1} \).

Based on the wheels angular velocities theoretical values, a WV linear velocity is calculated:

\[
V_{aw} = 0.25 \cdot \left( \omega_{FLt} + \omega_{FRt} + \omega_{RLt} + \omega_{RRt} \right) \cdot r_{st},
\]

where \( r_{st} \) – static wheel radius, m.

Then, estimated value of WV yawing velocity is calculated:

\[
\omega_{zt} = \frac{a_y}{V_{aw}},
\]

where \( a_y \) – lateral acceleration of the WV, \( m/s^2 \).

Relative difference between the actual and calculated WV yawing velocities is considered to be taken as an indicator to diagnose oversteer or understeer occurrence:

\[
\Delta \omega_z = \frac{|\omega_{zt} - \omega_{zt}|}{|\omega_{zt}|}.
\]

If \( \Delta \omega_z \) is less than 0, a understeer is diagnosed, if \( \Delta \omega_z \) is greater than 0, a WV oversteer is diagnosed.

Value \( h_M \) is calculated according to the curve in figure 3.

![Figure 3. Dependence \( h_M(\Delta \omega_z) \).](image)

Calculation of the torque value transmitted by a multi-plate clutch is shown in [9]. It may be interpreted to be used within a numeric model as follows:

1. Using readings from the ICE crankshaft rotation velocity sensor and the ration of depressing of accelerator, as well as partial characteristics of the engine (figure 4), the current engine torque is defined.
Figure 4. External and partial speed characteristics of the engine.

2. Input torque on the transfer case is calculated:

\[ M_{TC} = M_e \cdot i_{GB} \cdot \eta_{GB}, \]  

where \( i_{GB} \) – ratio of the current gear in the gearbox; \( \eta_{GB} \) – efficiency of the gear.

3. Using the relations above the front axle torque ratio \( (h_M) \) is calculated, and then the torque transmitted by the multi-plate clutch:

\[ M_{ndf} = M_{TC} \cdot h_M. \]  

The developed algorithm is integrated within a numeric model of non-linear motion of a WV, which is described in detail in papers [10]. Equations of transmission gear motion of a two-axle WV with a part-time front axle (figure 1):

\[
\begin{align*}
J_k \cdot \dot{\omega}_{FL} &= 0.5 \cdot i_{FD} \cdot M_{CF} - M_i; \\
J_k \cdot \dot{\omega}_{FR} &= 0.5 \cdot i_{FD} \cdot M_{CR} - M_i; \\
J_k \cdot \dot{\omega}_{RL} &= 0.5 \cdot i_{FD} \cdot M_{CR} - M_i; \\
J_{CF} \cdot \dot{\omega}_{CF} &= M_{ndf} - M_{CF}; \\
J_{CR} \cdot \dot{\omega}_{CR} &= M_e \cdot i_{GB} - M_{ndf} - M_{CR}; \\
\dot{\omega}_{CF} &= 0.5 \cdot i_{FD} \cdot (\dot{\omega}_{FL} + \dot{\omega}_{FR}); \\
\dot{\omega}_{CR} &= 0.5 \cdot i_{FD} \cdot (\dot{\omega}_{RL} + \dot{\omega}_{FR}),
\end{align*}
\]

where \( J_k \) – moment of inertia of the wheel, kg·m²; \( i_{FD} \) – final drive ratio; \( M_i \) – moment of resistance on the \( i \)-th wheel, N·m; \( \dot{\omega}_{CF} \) – angular acceleration of the front cardan shaft, s²; \( \dot{\omega}_{CR} \) – angular acceleration of the rear cardan shaft, s²; \( J_{CF}, J_{CR} \) – moments of inertia of the front and rear cardan shafts, respectively, kg·m²; \( M_{CF}, M_{CR} \) – torque on the front and rear cardan shafts, respectively, N·m.

3. Research results

The mathematical model of the two-axle rear-drive vehicle with connectible first axle and the algorithm of the part-time all-wheel drive are implemented in the Matlab Simulink.

The results of numeric modeling of a two-axle WV motion with the total weight of 2,100 kg with a part-time front axle are provided. To study how the algorithm influences tractive and dynamic properties of a WV, an acceleration on a bearing surface with non-homogenous traction properties has been
simulated (the interaction ratio of the mover with the bearing surface at full skidding under the rear axle was $\mu_{s100\%} = 0.3$, under the front axle $\mu_{s100\%} = 0.75$). The modeling results are provided in figure 5.

Figure 5. Change in linear velocity on a level track (a) and a 7.5° elevating track (b):
1 – a rear-wheel drive WV; 2 – a WV with a part-time front axle.

Applying algorithm of four-wheel drive connection while accelerating on a level track allowed a 3 seconds decrease in time of acceleration to reach maximum speed in 1st gear in the WV (figure 5, a). Maximum angle of the elevation being overcome for the two-wheel drive vehicle equals 7.5° (figure 5, b), and the acceleration time to the maximum speed in 1st gear reached 19 seconds (11 seconds greater than for the 4-wheel drive one). For the WV with a part-time front axle, the maximum angle of the elevation being overcome increased up to 13°.

The impact of the algorithm performance in a WV stability has been evaluated based on the modeling results of two maneuvers: going into corner and turning movement with a fixed position of the steering wheel; changing a traffic line (lane change). Movement along a surface “wet asphalt” was studied (interaction ratio of the mover with the surface at full skidding was $\mu_{s100\%} = 0.4$). The results of the turning maneuver $R = 35$ m are shown in figure 6 below.

Figure 6. The trajectory of the movement when performing turning maneuver $R = 35$ m:
a – a rear-wheel drive WV; b – a WV with a part-time front axle.
When the 4-wheel drive system is isolated, a WV loses its road-holding ability at the speed of 38 km/h (figure 6, a). With the 4-wheel drive system activated, a WV maneuvers with the speed of 40.7 km/h (figure 6, b), which meet the sweep critical speed at a given interaction ration. Further acceleration of a WV will lead to breaking the marked tunnel limits, however, no loss of road-holding ability is observed.

The results of the lane change maneuver $S = 20$ m are shown in figure 7. When the 4-wheel drive system is isolated, a WV loses its road-holding ability at the speed of 49 km/h (figure 7, a). With the 4-wheel drive system activated, a WV is able to maneuver with the speed of 54 km/h (figure 7, b). Further acceleration of a WV will lead to breaking the marked tunnel limits, however, no loss of road-holding ability is observed.

![Figure 7](image)

**Figure 7.** The trajectory of the movement when performing lane change maneuver $S = 20$ m:
a – a rear-wheel drive WV; b – a WV with a part-time front axle.

Simulation results showed the following: During acceleration on the road with non-uniform friction, the algorithm eliminates the wheel spin, decreases the time needed for the acceleration up to the given speed, and increases gradeability. When driving in curves, the developed algorithm considerably increases the speed of the vehicle during the tests in comparison to the single-axle drive variant. If the vehicle speed is close to the critical speed, the cornering ability of the vehicle changes from oversteer to understeer which means improvement in the yaw stability and results in better safety. Thus, the mathematical simulation proved efficiency of the developed algorithm of the part-time all-wheel drive.

4. Conclusions
1. We have suggested an algorithm of the Part-time 4WD to increase tractive and dynamic properties and stability of two-axle wheeled vehicles with a part-time front axle.
2. Efficiency and feasibility of the developed algorithm have been proved through simulation modeling of movement of such two-axle wheeled vehicles.
3. The next line of the study is the algorithm improvement oriented at limitations imposed by real-life constructions (temperatures, delay, etc.), as well as better interaction with other active safety systems.

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