Automatic Traction Control for Articulated Off-Road Vehicles

Johan Markdahl

ABSTRACT—Construction equipment is designed to maintain good traction while operating in difficult off-road conditions. To curb wheel slip, the vehicles are equipped with differential locks. The locks may be engaged/disengaged to switch between two distinct operating modes: the closed mode is characterized by greater off-road passability, while the open mode allows better maneuverability. This brief compares three on/off differential lock control algorithms. The algorithms are based on the same kinematic vehicle model, but each relies on the availability of output signals from different sensors. The validity of the kinematic model and the algorithms’ sensitivity to the values assumed by a couple of unobservable states, the wheel slip angles, is investigated by comparison to a realistic articulated hauler model in the multibody physics simulator MSC ADAMS.

Index Terms—Articulated hauler, articulated vehicles, construction equipment, differential locks, heavy equipment, off-road, on/off control, traction control, wheel loader.

I. INTRODUCTION

The design of articulated haulers is optimized for traversing difficult terrain [1], [2]. This allows the vehicles to take the shortest route on a load, haul, dump run, thereby minimizing fuel consumption and time expenditure. Part of this optimization is automatic traction control (ATC), which eases the decision-making burden of the driver, protects the tires from unnecessary wear, and reduces fuel consumption by up to 6% [3]. Note that the price tag for a single tire is US $5 000 and tires represent 20%–25% of haulers operating costs [4]. Manufacturers, such as Caterpillar, John Deree, Komatsu, and Volvo, use ATC [5], but the details of their algorithms are unknown to the public except for glimpses gained from ads [3] and patents [6], [7], [8], [9], [10], [11], [12]. This brief proposes three traction control algorithms based on different sensor outputs. The algorithms are validated against a model of a Volvo Construction Equipment (VCE) A40X articulated hauler in the multibody physics simulator MSC ADAMS [13].

A. Background

Articulated haulers are heavy equipment, used to transport large quantities of loose materials such as sand, gravel, and liquids in off-road environments. The articulated steering (see Fig. 4) ensures high maneuverability although at the cost of a lower maximum payload. Articulated haulers may, for example, be employed at mines to transport ore over grounds not traversable by ordinary vehicles or on construction sites to transport building material.

Traction is an adhesive friction force in the tire/road interface that serves to drive the vehicle forward. A tire may sometimes lose its grip and slip rather than roll over the road, e.g., if subject to full throttle, icy conditions, or a steep inclination. This is referred to as lost traction. Lost traction is undesirable since it reduces the vehicle’s traversability and increases tire wear. Means must therefore be taken to curb wheel slip and regain lost traction, preferably at the onset of wheel slip.

A differential is a driveline component that distributes power, i.e., torque and rotational speed, from an input shaft to two output shafts. The differentials on articulated haulers have two distinct operating modes: open and closed. An open differential distributes rotational speed freely and torque evenly. A locked differential forces the output shafts to assume the lowest of the two-wheel speeds, while torque is distributed freely. A succession of locks may be engaged to curb the slip of multiple wheels.

The differential locks are of the dog-clutch variety: a pair of face gears that are locked together pneumatically and pulled apart by a spring. Being locked together, the two shafts assume the same speed. The dog clutch has a range of angular velocity differences over which it is safe to engage; otherwise, engagement risks damaging the gear teeth. If the angular speed differences do not satisfy this constraint, then some articulated hauler models allow individual wheel brakes to reduce the rotational speed of selected shafts.

B. Problem Statement

The key question for ATC is when to lock the differentials. This comes down to comparing the revolutions per minute (RPM) of driveline shafts that are coupled through open differentials. A difference in RPM indicates slip, unless the vehicle is turning or braking. In this brief, we use the hauler geometry to develop a kinematic model that lets us discern between these cases. Moreover, we explore the advantages that may be gained from utilizing information obtained from a ground speed sensor (e.g., a ground speed radar or a GPS receiver) and individual wheel tachometers (angular speed sensors). These sensors are not standard, so this brief also addresses questions that are of interest to the manufacturers.
C. Literature Review and Contribution

The literature on traction control and various related topics such as antilock breaking systems, electronic stability control, and antislip regulation mainly concerns automobiles, see, e.g., the survey [14] and the survey [15] on traction control for electric vehicles. Traction control for articulated off-road vehicles is largely unexplored in the academic literature, although there are some exceptions [4]. Rather, the state of the art exists as in-house software solutions that are unavailable to the public. To gain an idea of what ATC algorithms are used by manufacturers, we may turn to commercials, product specifications, white papers, gray literature, and patents. An overview of the patents we review is shown in Table I.

Some algorithms presented in patents are based on braking the slipping wheel, including [6] and [8]. This actuation is not without limitations, including additional wear on the brakes. It requires sensors that can detect which a pair of wheels that is slipping, i.e., individual wheel tachometers. Moreover, the heat generated by braking can only be sustained by the vehicle for a limited time [12]. The majority of relevant patents use differential locks for actuation [7], [9], [12]. The algorithm [9] is unique in which it uses detection of oscillations in the driveline shafts to detect the presence or absence of wheel slip.

Most algorithms make use of a steering angle sensor [6], [8], [9], [10], [12] or a related output signal [7], but unlike this brief, they do not use the steering angle derivative. For motor graders, an additional angle sensor for the front wheel pair is needed [11], [12]. The algorithm [7] uses data from the pressure sensor in the steering hydraulics to decide when to lock or unlock the differentials. This algorithm also assumes that readings from a ground speed sensor are available. Some patents use wheel speed sensors [6], [8], and however, this does not necessarily mean that such algorithms are employed at present.

The contribution of this brief is to provide three novel traction control algorithms based on a kinematic model of an articulated hauler. The kinematic model and the algorithm’s effectiveness are proved by comparison to a realistic articulated hauler model in MSC ADAMS. The results also apply to wheel loaders, which are obtained as a special case with the articulated hauler model in MSC ADAMS. The results of this study are validated by simulations of an articulated hauler. The kinematic model and the algorithm’s effectiveness are proved by comparison to a realistic articulated hauler. The results also apply to wheel loaders, which are obtained as a special case with the articulated hauler model in MSC ADAMS. The results also apply to wheel loaders, which are obtained as a special case with the articulated hauler model in MSC ADAMS.
where $\omega$ is the wheel angular speed, $r$ is the outer tire radius, $\alpha$ is the tire slip angle, and $v$ is the ground speed (see [1], [20], [21]). This implies that $v_\alpha = v \cos \alpha$ is the ground speed along the longitudinal plane of the wheel. The two cases in (1) correspond to driving and braking. Together, they imply $\lambda_i \in [0, 1]$. We get back to wheel slip angles $\alpha$ in Section III-B, see also Figs. 3 and 4. For now, we just consider longitudinal slip. For an alternative, simpler longitudinal slip definition, consider the nonnormalized quantity

$$s_l \equiv \omega r - v \cos \alpha$$  \hspace{1cm} (2)$$

where $s_l \in \mathbb{R}$ can be interpreted as the distance a slipping wheel slips per second along its direction of orientation.

The normalized slip quantity $\lambda$ is of interest since it relates to the tire/road interface friction coefficient $\mu(\lambda)$, as detailed in a body of empirical studies, see, e.g., the survey [22]. The friction coefficient $\mu(\lambda)$ in turn enters vehicle dynamics, including simple models such as the one-wheel and bicycle models as well as more advanced models [20]. Since the algorithms developed in this brief do not use tire dynamics, there is little benefit in adopting $\lambda$ as a measure of slip.

The nonnormalized slip quantity (2) has the advantage of being more directly related to tire wear. In fact, some amount of wheel slip always exists in the tire/road interface, but small-to-moderate amounts of wheel slip are tolerable. To see how small and large amounts of slip affect our definitions, let $\alpha = 0$, $r = 1$ and consider two situations.

1) $\omega = \epsilon$ and $v = 0$.
2) $\omega = 2M$ and $v = M$.

Here, $\epsilon$ is a small angular velocity and $M$ is a large angular velocity. Note that (2) may result in significant tire wear, whereas small amounts of slip as in case 1) are tolerable. Definition (1) and $r = 1$, $\alpha = 0$ yields $s_l = 1$ in case 1) and $s_l = 1/2$ in case 2). Definition (2) yields $s_l = \epsilon$ in case 1) and $s_l = M$ in case 2). The definition (2) is preferable for our purposes since it clearly distinguishes between cases 1) and 2) in a way that captures the fact that 2) could potentially result in significant tire wear, whereas 1) could not.

### III. Main Results

#### A. Control Strategy

Certain sensor output signals are barely affected by wheel slip levels, while others may change rapidly. Ground speed calculated from a GPS receiver’s position readings is an example of an unaffected output. Tachometer measurements of the rotational speed of a drive shaft are an example of an affected output. This notion of either a static or a transient behavior of sensor output signals at the onset of wheel slip forms the basis of the traction control algorithms in this brief.

The rotational speed $\omega_{db}$ of a driveline shaft and the ground speed $v$ are proportional to each other under the assumption of zero slip and a rigid driveline, $\omega_{db} r / i = v$, where $i$ is a gear conversion ratio (possibly equal to 1) and $r$ is the outer tire radius. In the presence of wheel slip, this relation is replaced by our definition (2) of wheel slip, $s = \omega_{db} r / i - v \cos \alpha$.

We can generalize this idea to other kinematic equations that involve the vehicle velocity.

Let $y \in \mathbb{R}^k$ be the sensor output signals and $z \in \mathbb{R}^l$ denote relevant states that are not measured. Let $g(y, z) : \mathbb{R}^k \times \mathbb{R}^l \to \mathbb{R}$ be a function of the states and suppose that a kinematic equation

$$g(y, z)|_{z = 0} = 0$$  \hspace{1cm} (3)$$

holds in the absence of slip, i.e., when $s = 0$. The equality (3) is not guaranteed to hold, while $s \neq 0$. Any difference in the left- and right-hand side of (3) can therefore be used as an indicator of wheel slip. Our approach to traction control is hence: if the left- and right-hand sides of (3) differ beyond some preset tolerance, then engage a differential lock.

The basic idea of our control strategy is summarized in Fig. 2. The controller monitors the value of $g(y, z)$ to see whether slip beyond a preset threshold is detected. Before the locks are engaged, a check is performed to ensure that the difference in angular velocity over the dog clutch is not too large. Engaging the clutch while the difference is large risks damaging the teeth. Hence, apply the brakes prior to locking. After locking, it may not be possible to disengage the locks due to wind-up torques in the driveline. The dog clutch remains locked until the conditions are such that the locks can be disengaged.

Equation (3) is likely to be a kinematic relation. Most manufacturers own patented traction control algorithms based on articulated vehicle kinematics [6], [7], [8], [9], [10], [11]. Ideally, we would be able to measure the rotational and ground speed of each wheel, setting $y = [\omega_i, v_i]^T$ and $g_i(y) = \omega_i r_i - v_i$, where $i \in \{1, \ldots, 6\}$ denotes the $i$th wheel. However, neither individual wheel tachometers nor ground speed sensors form a part of the basic sensors’ configuration on VCE articulated haulers. As such, we need to consider alternatives.

As a more feasible example, consider

$$\omega_{db, in} - \omega_{db, out} = \begin{cases} 0, & \text{if } g(y, z) = 0, \forall i \in \{1, \ldots, 6\} \\ g(y, z), & \text{otherwise} \end{cases}$$

where $\omega_{db, in}$ and $\omega_{db, out}$ are the dropbox in/out shafts angular speeds (see Fig. 1), respectively, $\gamma$ is the steering angle between the tractor and trailer unit, and $g$ is some function. However, note that if $\gamma = 0$ and a tractor and trailer wheel should slip simultaneously, the above equation could still hold. This illustrates a limitation in our control design strategy: $s = 0 \Rightarrow g(y, z) = 0$, but $g(y, z) = 0 \Rightarrow s = 0$. 

![Fig. 2. Schematic control strategy. The focus of this brief is on the condition $l(y, z) \leq g(y, z) \leq u(y, z)|_{z = z}$ for locking, where $Z$ is some nominal value of the unobserved states $z$. The functions $l(y, Z)$ and $u(y, Z)$ bound the tolerance for error in the kinematic relation $g(y, Z) = 0$.](image)
Introduce the functions \( u(y) \) and \( l(y) \) to bound the error in the kinematic equation. Relax the no slip constraint (3) as
\[
geq g(y, Z)_{|s=0} \geq l(y) \\
\leq g(y, Z)_{|s=0} \leq u(y)
\] (4)
where \( Z \) is a constant nominal value of the unknown quantity \( z \). The functions \( l \) and \( u \) are included to account for the errors arising from setting \( z = Z \). Note that there is a tradeoff between functions \( l \) and \( u \) with small magnitudes, which make control action fast but increase the risk of unnecessary engagements and functions \( l \) and \( u \) with larger magnitudes, which reduce the risk of unnecessary engagements but also delay control action. Ideally, \( g(y, z) = g(y) \) so that \( u \) and \( l \) may be set to small values. If the chosen expression \( g \) depends on unobservable states \( z \), then setting \( z = Z \) may require the functions \( u, l \) to have large magnitudes. In practice, as we see in Section III-G, \( u \) and \( l \) can be defined as piecewise linear functions of \( y \) based on data from simulations.

Finally, we provide a formal statement that encompasses the family of traction control algorithms we consider.

Algorithm 1: Let \( g(y, z)_{|s=0} = 0 \) be a kinematic relation that holds in the absence of slip. Suppose that we have two bounds \( l(y, z) \) and \( u(y, z) \) such that
\[
l(y, z)_{|s=0} \leq g(y, z)_{|s=0} \leq u(y, z)_{|s=0}
\]
for tolerable amounts of slip \( s \approx 0 \). Replace \( z \) with a nominal value \( Z \) since the exact value of \( z \) is unknown to us. Lock the differentials if
\[
l(y, Z) \leq g(y, Z) \leq u(y, Z)
\]
do not hold. Unlock the differentials if a prespecified time \( \Delta t \) has passed since locking.

To limit the scope of this brief, we focus on the simultaneous engagement of all differential locks to curb wheel slip. The control of brakes and individual differential locks can be achieved using similar strategies to the ones we present here.

B. Kinematic Model of an Articulated Vehicle

The kinematics of load–haul–dump vehicles (a kind of low set articulated wheel loaders) is discussed in the literature on path tracking in underground environments, see, e.g., [23], [24]. The model [23] includes wheel slip angles. We cannot apply it directly to our setup since the vehicle geometry of wheel loaders and articulated haulers differ in key respects. A procedure for deriving kinematic models for the planar motion of articulated vehicles is presented in [24]. This brief generalizes the models of [23] and [24] by accounting for a broader range of ground vehicles, including but not limited to articulated haulers with wheel slip, using the technique from [24].

Consider the \( k \)th transverse axle of an articulated vehicle with \( n \) joints, as shown in Fig. 3 (see the figure for definitions of the notation used here). The velocity \( v_k = \begin{bmatrix} \cos \alpha_k \\ \sin \alpha_k \end{bmatrix}^{T} \) in the local coordinate origin \( O_k \) is related to the velocities \( \mathbf{v}_{Ak} \) and \( \mathbf{v}_{Bk} \) in the points \( A_k \) and \( B_k \) by the equations \( \mathbf{v}_{Ak} = \mathbf{v}_k + \Omega_k \times r_{O_k Ak} \) and \( \mathbf{v}_{Bk} = \mathbf{v}_k + \Omega_k \times r_{O_k Bk} \). The velocity \( \mathbf{v}_{Bk+1} \) is equal to \( \mathbf{v}_{Ak} \) rotated by \( \gamma_k \) degrees since \( \gamma_k \) is the difference between the coordinates in \( O_k-1 \) and \( O_k \) (see Fig. 3). Write these relations on matrix form. First, introduce a screw, \( \Psi_k \equiv [v_k \ \Omega_k]^{T} \in \mathbb{R}^{2} \), and then,
\[
\mathbf{v}_{Ak} = \mathbf{v}_k + \Omega_k \times r_{O_k Ak} \\
\begin{bmatrix} v_k \\ \Omega_k \end{bmatrix} = \begin{bmatrix} \cos \alpha_k \\ \sin \alpha_k \end{bmatrix}^{T} \]
resulting in the system of equations
\[
M_{Ak+1} \Psi_{k+1} = R_{Ak} M_{Ak} \Psi_k
\]
which may be solved for \( \Omega_k \), \( v_{k+1} \) and \( \Omega_{k+1} \) as functions of \( v_k \), \( \gamma_k \), and \( \gamma_k \) by using the relation
\[
\Omega_{k+1} = \Omega_k - \gamma_k
\]
and a few trigonometric identities.

Moreover, the velocity \( v_i \) of a single wheel with slip angle \( \alpha_i \), expressed in the \( k \)th axle local coordinates, may be calculated as
\[
v_i = v_k + \Omega_k \times r_{O_k i} \\
\begin{bmatrix} v_k \\ \Omega_k \end{bmatrix} = \begin{bmatrix} \cos \alpha_k \\ \sin \alpha_k \end{bmatrix}^{T} \]
where \( \alpha_i \) is the \( i \)th wheel slip angle. Note that the sign before \( c_k \) implies that the wheel \( i \) is on the right of the vehicle.
steering angle, and steady-state turning radius (i.e., $p$) can be solved to yield

$$
\begin{align}
\dot{\alpha}_{ij} &= \frac{(l_2 \cos(\gamma + \alpha_{12}) + l_1 \cos \alpha_{12}) v_{12}}{l_2 \cos \alpha_{34} + l_1 \cos(\gamma - \alpha_{34})} \\
& \quad \quad + \frac{l_2 \cos \alpha_{34} + l_1 \cos(\gamma - \alpha_{34})}{l_2 \cos \alpha_{34}} \sin(\gamma + \alpha_{12} - \alpha_{34}) v_{12} \\
\Omega_1 &= \frac{l_2 \cos \alpha_{34} + l_1 \cos(\gamma - \alpha_{34})}{l_2 \cos \alpha_{34}} \\
& \quad \quad + \frac{l_2 \cos \alpha_{34} + l_1 \cos(\gamma - \alpha_{34})}{l_2 \cos \alpha_{34}} \sin(\gamma + \alpha_{12} - \alpha_{34}) v_{12} \\
\Omega_2 &= \frac{l_2 \cos \alpha_{34} + l_1 \cos(\gamma - \alpha_{34})}{l_2 \cos \alpha_{34}} \\
& \quad \quad - \frac{l_2 \cos \alpha_{34} + l_1 \cos(\gamma - \alpha_{34})}{l_2 \cos \alpha_{34}} \left(\gamma \right)
\end{align}
$$

where the notation is that of Figs. 3 and 4.

Note that (7)–(9) encompass the models of a wheel loader with and without slip in [23] and [24], respectively, as special cases.

Introduce the functions

$$
\begin{align}
p(\alpha_{12}, \alpha_{34}, \gamma) &= \frac{\sin(\gamma + \alpha_{12} - \alpha_{34})}{l_1 \cos(\alpha_{34} - \gamma) + l_2 \cos \alpha_{34}} \\
q(\alpha_{12}, \alpha_{34}, \gamma) &= \frac{l_2 \cos(\alpha_{34} - \gamma) + l_1 \cos \alpha_{12}}{l_1 \cos(\alpha_{34} - \gamma) + l_2 \cos \alpha_{34}}
\end{align}
$$

where $p$ can be interpreted as the inverse of the tractor unit steady-state turning radius (i.e., $\gamma = 0$) and $q$ as the quotient of the trailer and tractor unit steady-state turning radii.

Equations (7)–(9) may be tidied up by writing

$$
\begin{align}
v_{34} &= q(\alpha_{12}, \alpha_{34}, \gamma) v_{12} + q(-\pi/2, \alpha_{34}, \gamma) \gamma l_1 \\
\Omega_1 &= p(\alpha_{12}, \alpha_{34}, \gamma) v_{12} + p(-\gamma + \pi/2, \alpha_{34}, \gamma) \gamma l_2 \\
\Omega_2 &= p(\alpha_{12}, \alpha_{34}, \gamma) v_{12} + p(-\pi/2, \alpha_{34}, \gamma) \gamma l_1
\end{align}
$$

with, e.g., $\gamma l_1$ in (10) being interpreted as a tractor unit velocity with a $-\pi/2$ side slip angle.

To validate the kinematic model, we compare it to an existing articulated hauler model in MSC ADAMS, an environment for simulation of multibody dynamics. The ADAMS model has previously been described in [13] and [25]. The articulated hauler in ADAMS is run repeatedly on a curved road, see Fig. 5, while varying the gear and load mass.

We use two error metrics

$$
\begin{align}
d_1(x, \hat{x}) &= \frac{1}{n} \sum_{k=1}^{n} |x_k - \hat{x}_k| \\
d_{\infty}(x, \hat{x}) &= \max_{k \in \{1, \ldots, n\}} |x_k - \hat{x}_k|
\end{align}
$$

with $\hat{x}$ being the estimate of $x$. The index $k$ is a discrete time instance and $n$ is the total number of time instances considered. Table II shows the accuracy of an estimate obtained from (10), i.e.,

$$
\hat{v}_{34} = q(\alpha_{12}, \alpha_{34}, \gamma) v_{12} + q(-\pi/2, \alpha_{34}, \gamma) \gamma l_1
$$

compared to the velocity $v_{34}$ obtained from a simulation in ADAMS. The columns illustrate the error caused by setting $\gamma$ and $\alpha_{ij}$ to zero. Observe that the error increases with gear choice (speed) and with load. Based on Table II, we conclude that only the models with nonzero steering angle velocity $\gamma$
are close to the ADAMS model. Slip angles are helpful, but not crucial. Slip angles are not measured, wherefore we look for relations (3) that do not depend on them.

D. Basic Sensor Network

The basic sensor network is based on [3] and consists of a steering angle sensor for the angle between the tractor and the trailer unit and four tachometers that measure rotational velocities at various points on the driveline (see Section II-A and Figs. 1 and 6). There are no tachometers on the wheels since they engage directly with the terrain, which would drastically affect their expected lifespan.

In practice, these five sensors provide enough information for a complete traction control algorithm [3]. However, we will also consider algorithms based on additional information. Of the basic sensors, the steering angle \( \gamma \) (and its derivative \( \dot{\gamma} \)) is the most interesting since its readings relate to our kinematic model of the hauler [see (7)]. Traction control for the case of \( \gamma = 0 \) is fairly straightforward from a theoretical point of view (see [25] for details), so we focus on the case of \( \gamma \neq 0 \).

Recall that our traction control approach described in Section III-A is based on comparing quantities that are affected and not affected by slip. The kinematic equation (10) can be used to obtain

\[
g(y, z)|_{z=0} = \frac{\omega_3 + \omega_4}{2} r - q(0, 0, \gamma) \frac{\omega_1 + \omega_2}{2} r - q(\pi/2, 0, \gamma) l_1 \dot{\gamma} = \omega_{bg, in}^r / i - q(0, 0, \gamma) \omega_{dbx, out}^r / i - q(\pi/2, 0, \gamma) l_1 \dot{\gamma} \tag{13}
\]

where \( i \) is the gear conversion ratio from the differentials over the hub reductions to the wheels (\( i = i_{\text{diff}} \cdot i_{\text{bg}} = 3.09 \cdot 6 = 18.54 \)), \( y = [\omega_{12} \omega_{34} \dot{\gamma} \dot{\gamma}]^T \) and \( z = [\alpha_{12} \alpha_{34}]^T \). Note that we used \( (\omega_1 + \omega_2) r / 2 = v_i \) as we have set \( z = 0 \).

The error in (13) should not be larger than that in Table II with the slip angles \( \alpha_{12} \) and \( \alpha_{34} \) set to zero. If they are given, we can conclude that it is due to wheel slip and engaging the differential locks. Note that the errors for setting \( \dot{\gamma} = 0 \) in Table II are rather large. As such, with only basic sensors, it is important to use an accurate kinematic model.

E. Ground Speed Sensor

GPS receivers and ground speed radars are examples of sensors that can be used to estimate or measure the ground speed. With a two-antenna GPS receiver, one antenna on the tractor unit, and one on the trailer, it is possible to estimate wheel slip angles [26]. Note that the 1-Hz sample rate of a GPS is rather slow, but the ground speed can be assumed piecewise constant. What is worse, a hauler may be put to work in GPS-denied environments, e.g., mines or tunnels. Still, a GPS is probably a more likely option than a ground speed radar due to the former’s multipurpose versatility. A ground speed radar also needs a clear line of sight to the ground. It is hence exposed to mud and can be rendered inoperable.

Output from a ground speed sensor can be compared to tachometer readings of the driveline shaft’s angular speed to calculate the slip according to the definition (2). An algorithm that knows the vehicle ground speed would hence be able to detect most occurrences of wheel slip.

Assume that the speed is measured somewhere on the tractor unit and that it is recalculated to correspond to the mean 1st and 2nd wheel speed (see Fig. 4). The expression

\[
\frac{\omega_1 + \omega_2}{2} r - v_{12} = \omega_{dbx, out}^r / i - v_{12} \tag{14}
\]

where \( i \) is the gear conversion ratio from the dropbox output to the wheel \( (i = i_{\text{diff}} \cdot i_{\text{hub}} = 3.09 \cdot 6 = 18.54) \), is an equation of the type (3), which can be rewritten as a system of inequalities of the type (4) to detect a slip of the front mean wheel. Note that we have set \( z = [\alpha_{12} \alpha_{34}]^T = 0 \). A slip of one of the front bogie wheels is detected as

\[
\frac{\omega_3 + \omega_4}{2} r - v_{34} = \omega_{bg, in}^r / i - q(0, 0, \gamma) v_{12} + q(-\pi/2, 0, \gamma) \dot{l}_1 \tag{15}
\]

where \( i \) is a gear conversion ratio and \( \omega_{bg, in}^r \) is the input tachometer on the bogie axle, see Fig. 1.

F. Individual Wheel Tachometers

The VCE ATC system tachometers (see Fig. 1) measure the angular speed of driveline shafts. These speeds are proportional to the speeds of the mean wheels under the assumption of a rigid driveline. Measurements of the wheel angular speeds facilitate slip detection but are difficult to carry out since tachometers placed at those locations have short expected lifetimes. To protect the sensors, they need to be encapsulated into the wheel hub. In this section, we assume that the output from tachometers in all wheel hubs is readily available.

Consider wheel \( i \) in Fig. 3. From \( v_i = \Omega_k \rho_i \), we get \( v_i \cos \alpha_i = \Omega_k \rho_i \cos \alpha_i \). Moreover, by definition (2),

\[
s_i = v_i r - v_i \cos \alpha_i = \omega_r - \Omega_k \rho_i \cos \alpha_i.
\]

Calculate

\[
s_1 - s_2 = \omega_1 r - \omega_2 r - \Omega_k (\rho_1 \cos \alpha_1 - \rho_2 \cos \alpha_2) = (\omega_1 - \omega_2) r + 2 \epsilon_1 \Omega_k
\]

where we used the geometry in Fig. 3 to find \( \rho_1 \cos \alpha_1 - \rho_2 \cos \alpha_2 = -2 \epsilon_1 \). A similar equation holds for the trailer

\[
s_3 - s_4 = (\omega_3 - \omega_4) r + 2 \epsilon_3 \Omega_{k+1}.
\]

Combine the equations for the tractor and trailer unit to obtain

\[
s_1 - s_2 + s_3 + s_4 = (\omega_1 - \omega_2 - \omega_3 + \omega_4) r + 2 \epsilon \dot{\gamma} \tag{16}
\]

where we used \( \Omega_2 = \Omega_1 - \dot{\gamma} \) and \( c_1 = c_3 = c \) [17]. We are able to remove the wheel slip angles \( \alpha_i \) without setting them
to zero as in Section III-D and III-E. Equation (16) is on the forms (3) on the form $g(y, z)|_{z=0}$ is given by (13), their boundary (blue), and convex hull (red). The data are obtained from the ADAMS model on the road in Fig. 5 on gears 1, 2, and 3 with empty, half, and full load.

To find all combinations of wheel slip that cannot be detected, we write (16) on the matrix form

$$\begin{bmatrix} 1 & -1 & -1 & 1 \\ s_1 \\ s_2 \\ s_3 \\ s_4 \end{bmatrix} = 0$$

and calculate the null space as

$$\text{ker} \begin{bmatrix} 1 & -1 & -1 & 1 \end{bmatrix} = \text{span} \left\{ \begin{bmatrix} 1 \\ 0 \\ 0 \\ 0 \\ -1 \end{bmatrix}, \begin{bmatrix} 0 \\ 1 \\ 0 \\ 0 \\ -1 \end{bmatrix}, \begin{bmatrix} 0 \\ 0 \\ 1 \\ 0 \\ -1 \end{bmatrix}, \begin{bmatrix} 0 \\ 0 \\ 0 \\ 1 \\ -1 \end{bmatrix} \right\}. \quad (17)$$

Slip is undetectable to the control strategy whenever

$$s = [s_1 \ s_2 \ s_3 \ s_4]^T \in \text{ker} \begin{bmatrix} 1 & -1 & -1 & 1 \end{bmatrix}.$$

Note that the nullspace (17) is the span of vectors where multiple wheels slip. Hence, (16), such as our other criteria, is the most useful for detecting the slip of a single wheel.

G. Tuning

The functions $u(y)$ and $l(y)$ in (4) should be tuned so that no slip is mistakenly detected during normal driving without delaying the detection of slip. This tuning is done based on data from the simulation model in ADAMS.

Consider first the basic sensor network. Fig. 7 shows a point cloud of pairs $(\gamma, g(y, z)|_{z=0})$ where $g(y, z)|_{z=0}$ is given by the kinematic relation between the tractor and the trailer unit (13). The road in Fig. 5 on which the data are collective is such that only tolerable amounts of wheel slip are present. The upper boundary of the convex hull of the point cloud is $u(\gamma)$, and the lower boundary is $l(\gamma)$. Both functions are piecewise linear. The relation is not symmetric, i.e., $l(\gamma) \neq -u(\gamma)$.

Consider the ground speed sensor algorithm, where $g(y, z)$ is given by (14) or (15). We only consider (14) here since the case of (15) is similar. Fig. 8 shows that the relation between $\gamma$ and (14) is that of a quadratic function. Hence, it is possible to fit two functions $u(\gamma) = u_0 + u_1 \gamma + u_2 \gamma^2$ and $l(\gamma) = u_0 + u_1 \gamma + u_2 \gamma^2$ to describe the upper and lower bounds. Unlike for the basic sensor network-based algorithm, here, we need to use $u(0) = u_0 \approx 0.2$ and $l(0) = l_0 \approx -0.2$. It may be possible to lower the $l_0$ value by using a filter since the value of $l(0) = l_0 \approx -0.2$ appears to be an outlier. Still, $l_0$ is unlikely to be a problem since it would indicate that the vehicle is braking due to $v > |\omega|$.

Consider the individual wheel tachometers algorithm, where $g(y)$ is given by (16). The $\dot{\gamma}$ signal in ADAMS changes rapidly. To remove some sharp peaks from the curve, we run a filter that averages the kinematic relation to equal the mean rapidly. To remove some sharp peaks from the curve, we run a filter that averages the kinematic relation to equal the mean. In practice, we see that $l(\gamma)$ and $u(\gamma)$ given in Fig. 9 is irregular. In theory, the condition (16) is very attractive since it excludes the slip angles. In practice, we see that $l(\gamma)$ and $u(\gamma)$ has to be chosen quite large. In particular, we have $l(0), u(0) \neq 0$, which was not the case for the relation (13) based on the basic sensor network. Moreover, the boundary is very irregular in some places, suggesting that more data are required. Suppose that the error $g(y)$ is equal to the slip of wheel 1. Then, we have $s_1 = g(y)$. The RPM difference over the front transversal differential is $6g(y)$ where 6 stems from the hub reduction gear conversion ratio. We have to tolerate a maximum difference of $u(\pi/4) = 6 \cdot 0.35 \times 60/(2\pi r) \approx 21$ RPM. The tolerable difference for $\gamma = 0$ is at least $u(0) = 6 \cdot 0.1 \times 60/(2\pi r) \approx 6$ RPM and $l(0) = 6 \cdot 0.05 \times 60/(2\pi r) \approx 3$ RPM.

IV. BENCHMARK

To benchmark the performance of the three proposed algorithms, we compare the convex hulls of their point clouds to one generated by the algorithm in [10], which is given by

$$g(y, z)|_{z=0} = \frac{\omega_3 + \omega_4 r - q(0, 0, \gamma)}{2}$$

$$\approx \frac{\omega_{bg, in} r}{i} - q(0, 0, \gamma) \omega_{\Delta bx, ou} r / i. \quad (18)$$

The algorithm [10] is the basic sensor network with the angular velocity of the steering angle set to zero for the case of steady-state turning. Fig. 10 shows the convex hulls of the point clouds for the four algorithms. By inspection of Fig. 10,
we find that the algorithm [10] is preferable to the ground speed algorithm and the individual wheel tachometer algorithm but not to the basic sensor network algorithm.

V. CONCLUSION

Three traction control algorithms are presented together with their relative strengths and weaknesses. The basic sensor network suffices for a feasible ATC algorithm since one is implemented on Volvo CE haulers [3]. It may be possible to improve on this algorithm using additional sensor data. In particular, ground speed measurements based on GPS receiver data are expected to be available in the future. We speculate on the benefits of individual wheel tachometers. In theory, individual wheel tachometers present the most accurate data, allowing us to remove the influence on wheel slip angles. In practice, however, we find that the tachometer data are plagued by too much variation and many outliers. The ground speed sensor data are well behaved but offer no significant advantage over the basic sensor network-based algorithm.

In conclusion, the results of this study favor the basic sensor network-based algorithm.

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