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INDICES AND COMPUTATIONAL STRATEGY FOR UNMANNED GROUND WHEELED VEHICLE MOBILITY ESTIMATION AND ENHANCEMENT

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ABSTRACT

The United States Army began developing Unmanned Ground Vehicles (UGV) in the early 1900's. Concurrently, researchers developed and enhanced passenger and commercial ground vehicles. Although significant progress has been made for improving vehicle mobility for all ground vehicles throughout the past century, mobility has lacked a concise mutually agreed definition and analytical standardized criteria. The implementations of improved technologies, such as vehicle traction control, stability control, and torque vectoring systems require researchers to take a step back and reevaluate mobility criteria. UGVs require additional enhancement to include on-line mobility estimation since the vehicle cannot predict nor anticipate terrain conditions on their own prior to the vehicle traversing those conditions.

This paper analyzes methodologies researchers have employed for defining and improving vehicle mobility of wheeled vehicles. The analysis is done from a view point of concurrent mobility methodologies' enhancement and applicability to wheeled UGVs.

This analysis is then used to develop off-line and on-line analytical criterion for mobility estimation, and to derive a strategy which can be applied to wheeled vehicles, both manned and unmanned. The on-line mobility estimation enables the UGV to make control changes as the events occur rather than after the event, causing the vehicle to then optimize its reaction to regain control.

NOMENCLATURE

a_d tire off-set distance

D_a aerodynamic drag force

F_a inertia force

F_x circumferential force at the wheel

$F_{x\Sigma}$ total circumferential force of the vehicle

$F_{x\Sigma}^{\mu}$ total circumferential force of the vehicle determined by gripping conditions to the terrain

F_w wheel traction force

F_{wmax} max traction force at the wheel

F_ψ total force of resistance to motion

i subscript for axle: 1, 2, or 3

R_x rolling resistance force of the wheel

R_z normal reaction on the wheel

t_b wheel tread (width between wheels of an axle)

T_w wheel driving torque

$T_{w\mu}$ wheel torque which value is limited by the tire-soil gripping condition

W_a weight of the vehicle

θ_l lateral inclination

θ_n longitudinal inclination

μ_{px} peak (max) friction coefficient

$\mu_{px\Sigma}$ total vehicle peak (max) friction coefficient

$'$ superscript symbol for the right wheel(s)

$''$ superscript symbol for the left wheels(s)

PDU   power dividing unit

UGV   unmanned ground vehicle

VMI   vehicle mobility index

WMI   wheel mobility index

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INTRODUCTION

For more than 100 years, researchers have been analyzing methods to improve vehicle mobility of wheeled vehicles. The United States Army began developing Unmanned Ground Vehicles (UGV) in the early 1900's. Through the years, mobility was defined differently and different approaches were developed to estimate mobility of conventional (with a driver) vehicles [1-21].

It should be mentioned that, additionally to the above-listed publications, numerous research papers were published (especially in the Journal of Terramechanics); they facilitated research on vehicle mobility estimation by discovering and clarifying tire-soil interaction and thus providing conditions for better defining mobility and its quantitative estimation.

Here are some of mobility definitions. "Mobility of military vehicles is concerned with the maximum feasible speed between two points in a given region" [22]. In [22], "mobility relates to the performance of the vehicle in relation to soft terrain, obstacle negotiation and avoidance, ride quality over rough terrain, and water crossing". In work [23], three levels of mobility are introduced: strategic, operational, and tactical. Strategic mobility is the ability to move, operational is the ability to move on their own; tactical is the ability to move over various terrains and obstacles. In addition to the aforementioned works in the area of mobility, efforts that were collectively undertaken by professional societies should also be mentioned. In [24], mobility is defined as the overall capability of a vehicle to move from place to place while retaining its ability to perform its primary mission.

A detailed analysis of the above-listed and many other works in the vehicle mobility field revealed that, despite of different approaches, two main features of mobility as a vehicle's operational properties are commonly included in its definition. Mobility is an ability of vehicles to (i) move under road-less terrain conditions while (ii) performing their functions. A distinction is usually made between the topographical and support-surface mobility. The topographical (geometric, profile) mobility represents the ability of a vehicle to surmount various obstacles including ditches and trenches, hillsides, man-made obstacles (this applies to combat and tactical vehicles), etc. The support-surface mobility is defined by the ability to move over deformable surfaces and over snow. This paper deals primarily with support-surface mobility.

Researches proposed and used various indices and criteria to estimate support-surface mobility of vehicles, which were based on inconsistent mobility definitions. Usually, vehicles are evaluated on conditions of tractive force, tractive effort, drawbar-pull, sinkage, safe weight pressures, ground pressures, and slippage [7, 10, 11, 23, and 25-27], etc. Significant research was done in the Waterways Experiment Station (WES) and a mobility index was proposed to link vehicle design parameters with its mobility [28, 29]. The bevameter technique was originated in [7] and enhanced in [22]. These and other research efforts resulted in NATO Reference Mobility Modeling software and its applications [30-32, etc.]. Today's research in mobility takes in consideration stochastic characteristics of terrain mechanical and geometric characteristics [10, 20, 33-35, etc.].

As an analysis of the above-presented research work, mobility of a terrain vehicle is traditionally estimated as a vehicle's capability of "to go through" or "not to go through" the given terrain conditions using original data that has been obtained in mathematical modeling or experimental tests. For example, the WES penetrometer and Vehicle Cone Index can be mentioned here which is usually assigned for 1 through 50 passages of the vehicle [24].

Vehicle mobility strongly depends on the driveline system characteristics that distribute power between the driving wheels [14]. Non-efficient wheel power split and distribution can immobilize a vehicle in terrain conditions when tire gripping conditions change from one to another wheel. This is caused due to a change in physical properties of local terrain when the vehicle's wheels move in the same track(s). Thus, this simple approach that was designed for "go/not go" mobility estimation, does not take in to consideration the impact the driveline has on the power distribution between the driving wheels. The driveline can deteriorate vehicle mobility and immobilize the vehicle in terrain conditions. The problem here is that non-appropriate wheel power distribution can cause significant differences in tire slippages, and the numerical values that are usually nonlinear functions of the circumferential forces of wheels on pliable surfaces. These forces can exceed the maximum values permissible by the gripping conditions and the vehicle then becomes immobilized.

In some research work, an attempt to include the wheel power split factor in mobility indices was undertaken. In [37] several integral criteria to estimate mobility are considered. One of them is:

$$P_\varphi = 1 - \frac{F_{w\Sigma}}{\sum_{i=1}^n F'^{(\prime\prime)}_{wmaxi}} \qquad (1)$$

here, $F'^{(\prime\prime)}_{wmaxi}$ is the max traction force of a wheel (the difference between the max circumferential force and the rolling resistance of the wheel); $F_{w\Sigma}$ is the vehicle draw bar equal to the external resistance to motion (includes the inertia force, the weight component on slopes, load from trailers, etc.). Still, an additional step of calculation is needed when Eqn. (1) is in use; namely, it is needed to determine the traction forces as a difference between the circumferential wheel force (that comes from the wheel torque) and the rolling resistance force that is not easy to measure.

In book [12], vehicle mobility is proposed to be estimated using the actual wheel torques and the max possible wheel torques due to the tire-soil gripping:

$$P_\varphi = 1 - \frac{\sum T_{wi}}{\sum T_{w\mu maxi}} \qquad (2)$$

here, T_{wi} is the current wheel torque, and $T_{w\mu maxi}$ is the maximal wheel torque which value is limited by the tire-soil

gripping condition. If the actual wheel torques can be easily measured, this cannot be said with regard to the max torques.

As a further development, in reference [14] criterion in Eqn. (2) was presented using the circumferential wheel forces:

$$p_\mu = 1 - \left(\frac{F_\psi}{F_{x\Sigma}^\mu}\right) \qquad (3)$$

, and

$$F_{x\Sigma}^\mu = \sum_{i=1}^n \left(\mu'_{pxi} R'_{zi} + \mu''_{pxi} R''_{zi}\right) \qquad (4)$$

where F_ψ is the total resistance to motion that is equal to the sum of the wheel circumferential forces, $F_\psi = F_{x\Sigma}$. Therefore, (4) can be re-written as:

$$VMI_F = 1 - \left(\frac{F_{x\Sigma}}{F_{x\,max\Sigma}}\right) \qquad (5)$$

Formula (5) can be further expressed in terms of the current and pick friction coefficients:

$$VMI_\mu = 1 - \left(\frac{\mu_{x\Sigma}}{\mu_{px\Sigma}}\right) \qquad (6)$$

Here, $\mu_{px\Sigma}$ is the sum of the pick friction coefficients, $\mu_{px\Sigma} = \sum_{i=1}^n \left(\mu'_{pxi} + \mu''_{pxi}\right)$ and $\mu_{x\Sigma}$ is the sum of current friction coefficients, $\mu_{x\Sigma} = \sum_{i=1}^n \left(\mu'_{xi} + \mu''_{xi}\right)$, $\mu_x = F_x/R_z$. Equation (6) makes a basis for the developments presented in this paper.

An analysis of unmanned ground vehicle research showed, that UGV topography mobility has been developed in many studies as the ability to avoid obstacles. However, UGV support-surface mobility is still in its infancy and needs to be intensively researched. The problem here is that research work in this field was done not from an actual vehicle dynamics point of view, but from a control design point of view. For control purposes, UGVs are considered as non-holonomic systems [38-41], and this was the rationale since UGVs were initially considered for either indoor applications or moving on improved (paved) roads. In fact, for terrain UGVs, there is always a partial lateral (side) sliding of the tires relative to the ground even if the active lateral forces are close to zero; additionally, intensive tire longitudinal slippage can occur in rough terrain conditions. Math fundamentals for modeling UGVs as holonomic rheonomic systems have been presented in [42]. Pneumatic tire UGV support-surface mobility is awaiting its in-depth research since available studies were mostly conducted for planetary rover applications or vehicles with rigid wheels [43, 44, etc.].

This paper is concerned with one, but important, feature of UGV support-surface mobility. This feature is namely the influence of power distribution between the driving wheels, which is determined by configurations of the driveline system. New indices for mobility estimation are proposed and limitations that the driveline exposes on the wheel power distribution are formulated.

WHEEL MOBILITY AND WHEEL-BASED VEHICLE MOBILITY INDICES

Figure 1 represents a two dimensional drawing of a 6 wheeled Unmanned Ground Vehicle (UGV) heavy truck's geometry on a horizontal surface, while Fig. 2 gives the generalized three-dimensional case of motion of the truck with a downhill pitch $(-\theta_n)$ and lateral roll to the left $(-\theta_l)$. These are essential to make it possible to effectively derive typical use cases of motion for the UGV during its mobility operations.
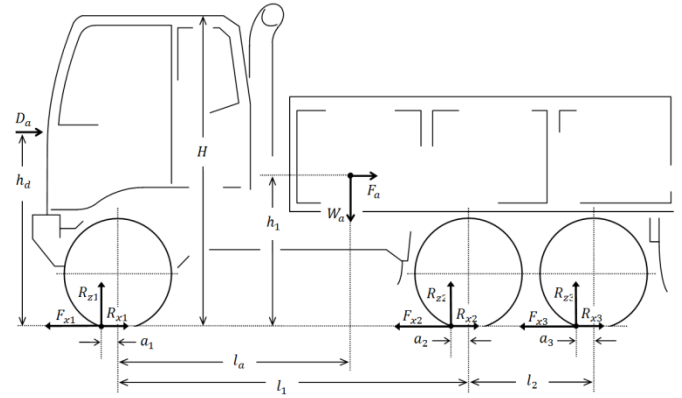

Fig. 1. 2D DRAWING OF A 6X6 UGV TRUCK AND GEOMETRY ON A HORIZONTAL SURFACE
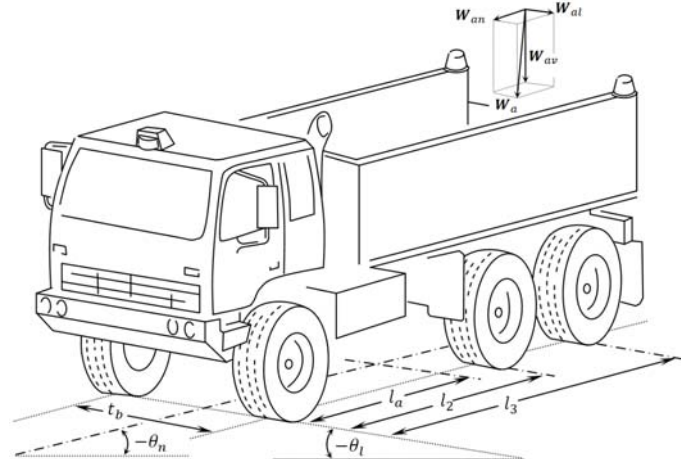

Fig. 2. 3D DRAWING OF A 6X6 UGV TRUCK'S GENERAL CASE OF MOTION WITH LATERAL AND LONGITUDINAL INCLINATIONS

For both off-line and on-line mobility analysis, it is required to evaluate the truck's traction during specified operations. A wheel's traction is affected by the normal forces applied to the tire. A six wheeled UGV that is considered in the paper contains three axles with two possible configurations of the wheel formula, 6x4 and 6x6 where the rear 4 wheels or all six are driving the vehicle. Figure 3 show the acting forces and reactions that are exerted by a wheel in the driving mode.

Whereas, Fig. 4 shows a wheel in the driven mode (zero torque). For simplicity, both figures illustrate the forces in the steady motion.
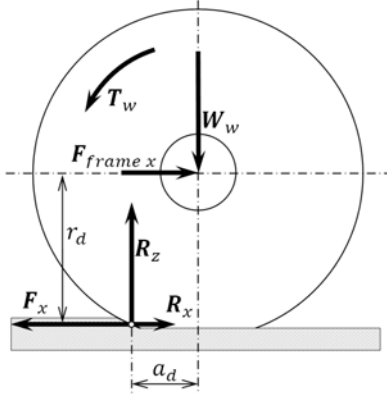
Fig. 3. FREE-BODY DIAGRAM OF A WHEEL IN THE DRIVING MODE
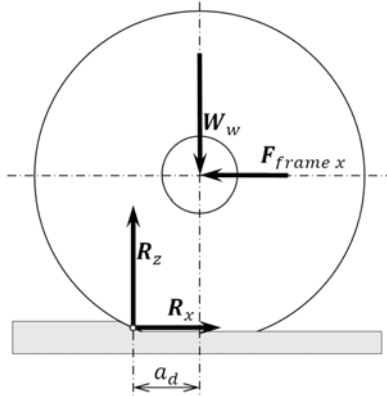
Fig. 4. FREE-BODY DIAGRAM OF A WHEEL IN THE DRIVEN MODE

The circumferential force (F_x) is developed by the wheel torque. In the driving mode, the wheel torque T_w and is larger than the rolling resistance of the magnitude of the force $F_{frame\,x}$ (see Fig. 3). In the driven mode, the force $F_{frame\,x}$ must overcome the rolling resistance (R_x).

As it was shown before in this paper, traditional mobility analysis does not take into account each wheels independent forces when calculating and analyzing the vehicle's mobility. Traditionally the vehicle is viewed as a single body of motion and utilizes the torque applied from the engine and the overall resistive force affecting the forward motion of the UGV. This paper focuses on the individual affect each wheel has on the UGV's mobility. The maximum circumferential force $(F_{x\,max})$ at each wheel can be expressed as:

$$F'^{('')}_{x\,maxi} = \mu'^{('')}_{pxi} R'^{('')}_{zi} \tag{7}$$

where μ_{px} is the maximum friction coefficient, and R_z is the normal reactionary force for the given wheel. Here the single apostrophe ($'$) and double ($''$) apostrophe refer to the right and left wheels respectively, and i is the i-th axle as illustrated

in Fig. 1. To obtain the total maximum circumferential force for the UGV, a summation of each wheel's maximum force evaluated from Eqn. (7);

$$F_{x\,max\Sigma} = \sum_{i=1}^{n} \mu'^{('')}_{pxi} R'^{('')}_{zi} \tag{8}$$

where n represents the total number of driving axles. For example, in a 6 wheeled vehicle configuration, i would equal 2 or 3 for a 6x4 or 6x6 vehicle respectively. The maximum friction coefficient for all wheels can be expressed as a summation of each wheel;

$$\mu_{px\Sigma} = \sum_{i=1}^{n} \mu'^{('')}_{pxi} \tag{9}$$

and the current friction coefficient at each wheel is derived by the relationship between the current values of the circumferential forces at the surface interaction of the tire

$$\mu'^{('')}_{xi} = \frac{F'^{('')}_{xi}}{R'^{('')}_{zi}} \tag{10}$$

To calculate the total friction coefficient for the UGV, a summation of the current friction coefficient for all wheels is determined

$$\mu_{x\Sigma} = \sum_{i=1}^{n} \mu'^{('')}_{xi} \tag{11}$$

The difference between the maximum and current friction coefficients $(\Delta\mu_x)$ is introduced in this paper as an important parameter to estimate mobility of an UGV based on individual wheel estimation. It can be derived by subtracting the current friction coefficient (μ_x) from the wheels maximum friction coefficient for each wheel (μ_{px});

$$\Delta\mu'^{('')}_{xi} = \mu'^{('')}_{pxi} - \mu'^{('')}_{xi} \tag{12}$$

It is clear that the bigger the delta, the higher mobility capacity of a driving wheel. Based on this approach, the Wheel Mobility Index (WMI) for an individual wheel (either side of the vehicle $'$ or $''$) of the i-th axle is presented as follows:

$$WMI'^{('')}_{\mu_i} = 1 - \left(\frac{\mu'^{('')}_{xi}}{\mu'^{('')}_{pxi}} \right) \tag{13}$$

Likewise, the difference between the maximum and current circumferential forces (ΔF_x) can be derived by subtracting the current circumferential force (F_x) from the wheels maximum circumferential force for each wheel $(F_{x\,max})$.

$$\Delta F'^{('')}_{xi} = F'^{('')}_{x\,maxi} - F'^{('')}_{xi} \tag{14}$$

In addition to the friction coefficient, the circumferential force-based WMI can be derived by using

$$WMI'^{(\prime\prime)}_{F_i} = 1 - \left(\frac{F'^{(\prime\prime)}_{xi}}{F'^{(\prime\prime)}_{x\,maxi}}\right) \qquad (15)$$

The larger the value of the WMI in Eqn. (13) and (15), the higher the mobility capacity of this particular wheel becomes.

Sensor-based information related to the components in Eqn. (13) and (15) about the vehicle and the given terrain parameters and characteristics can be gathered and analyzed to serve as on-line (real-time during the UGV's motion) vehicle mobility analysis. Through this on-line collection and analysis, it becomes possible to study mobility capacity of each and all of the wheels, and thus estimate and preemptively enhance vehicle mobility. To simplify this technical approach: if one or several wheels are experiencing poor terrain conditions and have low values of indices from Eqn. (13) and (15), the power can be re-distributed to the other wheels that are traversing through better terrain conditions (wheels with higher WMI values). This will reduce the current friction coefficients, $\mu'^{(\prime\prime)}_{xi}$, of the wheels in poor conditions, increase their wheel mobility indices, and thus enhance mobility of the entire UGV.

To illustrate this, a new Vehicle Mobility Index (VMI) is introduced in this paper:

$$VMI_{\mu_{Imp}} = \frac{\sum_{i=1}^{n} WMI'^{(\prime\prime)}_{\mu_i}}{2n} = 1 - \frac{1}{2n}\sum_{i=1}^{n}\frac{\mu'^{(\prime\prime)}_{xi}}{\mu'^{(\prime\prime)}_{pxi}} \qquad (16)$$

or, written in terms of forces

$$VMI_{F_{Imp}} = \frac{\sum_{i=1}^{n} WMI'^{(\prime\prime)}_{F_i}}{2n} = 1 - \frac{1}{2n}\sum_{i=1}^{n}\frac{F'^{(\prime\prime)}_{xi}}{F'^{(\prime\prime)}_{x\,maxi}} \qquad (17)$$

Comparing Eqn. (16) and (17) to the more traditional equivalent, Eqn. (5) and (6), one can see the principle difference in mobility estimation which comes out from the use of these two methods. In fact, Eqn. (5) and (6) give an average mobility index based on the sum of the current total vehicle circumferential force and the maximum total vehicle circumferential force. Whereas in the new approach, Eqn. (16) and (17) present a more precise estimation of vehicle mobility which counts the mobility indices of all wheels relative to the number driving wheels. Wheels in the driven mode (non-driving wheels) negatively affect the vehicle's mobility indices.

The advantage of Eqn. (16) and (17) can be proved by the following illustrative example taken from computations that will be presented in detail later in the paper. Assume that the normal reactions of the wheels are the same and the peak friction coefficients of the four driving wheels of a 6x4 terrain track are 0.5, 0.6, 0.4, and 0.35. The current friction coefficients are 0.30, 0.35, 0.38, and 0.34. Using this data, the VMI from Eqn. (6) is 0.2595. However, if the mobility situation of every wheel is taken in consideration by using Eqn. (13), the mobility index of the vehicle from Eqn. (16) goes as 0.2238. As shown in this example, actual mobility estimated using the proposed indices Eqn. (13) and (16) is 16% worse than the mobility situation estimated on the average index Eqn. (6). This means that the vehicle is in much worse mobility-wise situation than one thought when using Eqn. (6), which falsely represented the UGV's mobility.

Still another conclusion can be drawn from the above-example, which the whole point of this paper. If the vehicle is equipped with an autonomous wheel power-vectoring system, the system can further enhance mobility of the vehicle by re-distributing power between the wheels. To further express this, assume that the autonomous wheel power-vectoring system made a random change (random change, not an optimal change since this is beyond the scope of this paper) of power distribution towards the two wheels with larger pick friction coefficients, which then resulted in a new set of the current friction coefficients of the wheels: 0.33, 0.39, 0.35, and 0.30 (one can notice that the sum of the current friction coefficients ($\mu_{x\Sigma}$) is kept constant at 1.37, meaning the UGV can overcome the resistance to motion). This power re-distribution leads to the UGV mobility index from Eqn. (16) from 0.2595 to 0.2395. Mobility here is improved by 8.22%. In the following section, this paper will walk through computational simulations supporting the proposed mobility indices.

COMPUTATIONAL RESULTS AND MOBILITY STRATEGY

An Unmanned Ground Vehicle's capability to move in terrain conditions can be hindered or improved based on the combination of different Power Dividing Units (PDU) in the driveline system [45]. In many applications, an advanced driveline system for improved mobility with a stability control system can utilize differentials with torque biasing capabilities. Table 1 represents symbols for the various PDU's that were analyzed for a particular UGV.

Table 1. SYMBOLS OF RELATIVE POWER DIVIDING UNITS (PDU)

Mechanism in PDU	Designation
Symmetrical Open (free-running) PDU	○
Asymmetrical Open (free-running) PDU	▢
Constant (100%) locking engagement of output shafts	⊠
Non-constant engagement with autonomously controlled disengagement of one of the output shafts	◁

This paper considers position numbers 1 through 4 in Tab. 2, which represent combinations of the driveline PDU's configurations that are employed in the large six wheel UGV in this project.
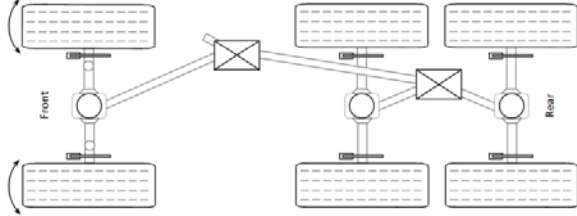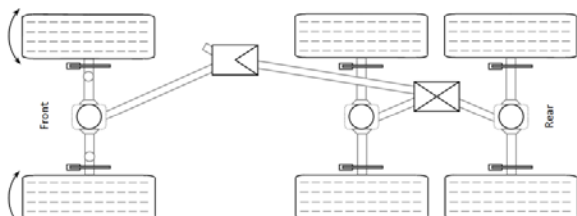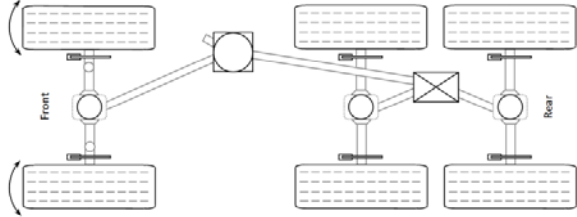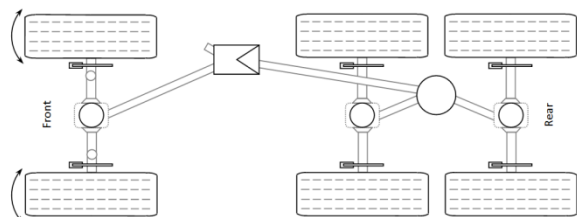
A mathematical and computer model of a 6 wheeled UGV (that is beyond the scope of this paper) was developed to simulate and demonstrate the indices under various terrain conditions, vehicle accelerations and speeds, and longitudinal inclinations. Computer simulations that were run on the UGV and the UGV's condition can be seen in Tab. 3.

Table 2. DRIVELINE SYSTEMS FOR A 6X6 TERRAIN/OFF-ROAD UGV

No.	Chassis and Driveline System Layout	No.	Chassis and Driveline System Layout
1	Positively locked PDU in transfer case Positively locked inter-axle PDU in tandem	2	Positively locked PDU in tandem Front axle disconnected from tandem inter-axle PDU
3	Open asymmetrical differential PDU in transfer case Positively locked inter-axle PDU in tandem	4	Front axle disconnected from tandem inter-axle PDU Open symmetrical differential PDU in tandem

Table 3. SETS OF COMPUTATIONAL RUNS

| Run	θ_l	Max Speed	Max Acceleration	Right Wheels Conditions	Left Wheels Conditions
1	0°	10 m/s	2.20 m/s²	Dirt Road $\mu_{px} = 0.55$	Haul Road $\mu_{px} = 0.7$
2	0°	10 m/s	1.10 m/s²	Dirt Road $\mu_{px} = 0.6$	Dirt Road $\mu_{px} = 0.6$
3	0°	5 m/s	1.10 m/s²	Slippery Road $\mu_{px} = 0.3$	Dirt Road $\mu_{px} = 0.55$
4	0°	5 m/s	0.55 m/s²	Slippery Road $\mu_{px} = 0.3$	Slippery Road $\mu_{px} = 0.3$
5	10°	3 m/s	1.10 m/s²	Dirt Road $\mu_{px} = 0.55$	Dirt Road $\mu_{px} = 0.55$

The computer simulations identified in Tab. 3 at the time of motion of 20 sec were completed for each of the four vehicle driveline configurations identified in Table 2. The results of these simulations can be seen in Tab. 4. As one can see from the results, VMI_μ is steadily a higher value than $VMI_{\mu_{Imp}}$, which supports the analytical analysis described earlier in the paper. Vehicle configurations 1 and 3, maintained a higher VMI over configurations 2 and 4, markedly due to the vehicle being driven by all six wheels (6x6) rather than four of the six (6x4). The difference between the two calculated VMI approaches (ΔVMI), VMI_μ and $VMI_{\mu_{Imp}}$, poses similar values towards like driveline configurations between the front axle and the rear tandem. Although one configuration contains one less driving axle, depending on the terrain conditions the UGV's mobility isn't hindered as much. More slippery terrain caused more of a delta (ΔVMI) between the two technical approaches.

Table 4. RESULTS OF COMPUTATIONAL RUNS

Run	Configuration	Time	VMI_μ	$VMI_{\mu_{Imp}}$	ΔVMI	Run	Configuration	Time	VMI_μ	$VMI_{\mu_{Imp}}$	ΔVMI
1	1	@ 20 s	0.940	0.934	0.006	3	3	@ 20 s	0.866	0.851	0.0147
1	2	@ 20 s	0.907	0.901	0.006	3	4	@ 20 s	0.794	0.755	0.0189
1	3	@ 20 s	0.958	0.9572	0.0008	4	1	@ 20 s	0.724	0.723	0.001
1	4	@ 20 s	0.944	0.9436	0.0004	4	2	@ 20 s	0.556	0.555	0.001
2	1	@ 20 s	0.928	0.922	0.006	4	3	@ 20 s	0.722	0.719	0.003
2	2	@ 20 s	0.890	0.884	0.006	4	4	@ 20 s	0.577	0.579	-0.002
2	3	@ 20 s	0.946	0.9454	0.0006	5	1	@ 20 s	0.591	0.581	0.010
2	4	@ 20 s	0.918	0.9185	-0.0005	5	2	@ 20 s	0.352	0.346	0.006
3	1	@ 20 s	0.861	0.844	0.0168	5	3	@ 20 s	0.591	0.582	0.009

3	2	@ 20 s	0.861	0.844	0.0168	5	4	@ 20 s	0.387	0.384	0.003

For simulations in driveline configurations 2 and 4, the autonomously controlled engagement of the forward axle was turned off, and the simulations contest to four driving wheels vs. all six when required due to good gripping conditions. As illustrated in Fig. 5, for the same run under the two different driveline configurations, 3 and 4, the 6x6 UGV results in mobility indices improvement of 9.5% for both friction coefficient and circumferential wheel force.


b.)
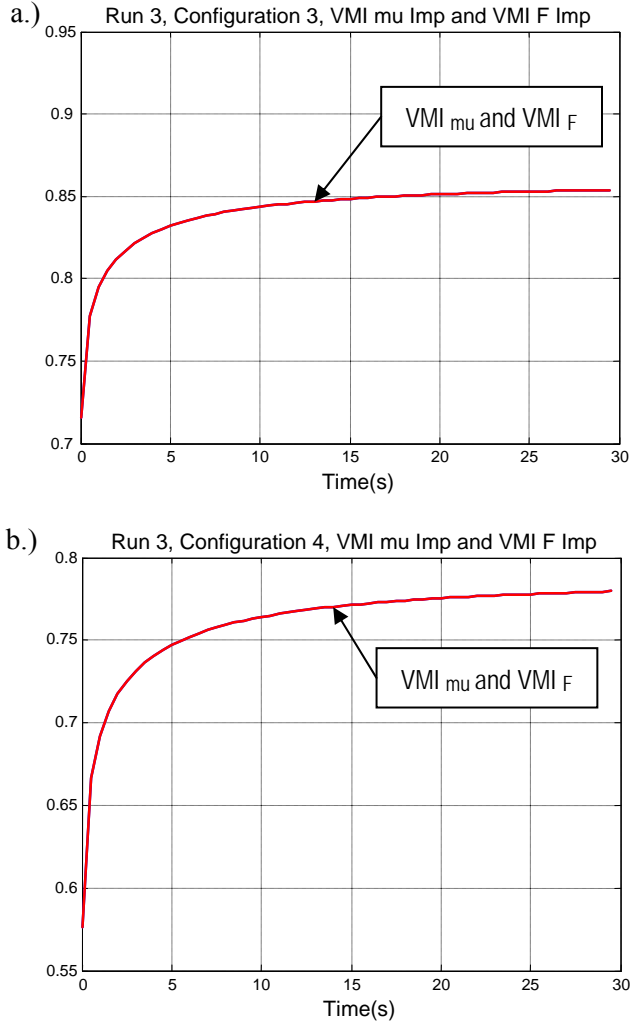
Fig. 5. FRICTION COEFFICIENT AND FORCE VMI IMPROVED, Run 3, (a.) CONFIGURATION 3 6X6, AND (b.) CONFIGURATION 4 6X4

As one can see from Tab. 4, there are various differences not only between the technical approaches for determining the UGV's VMI, but the driveline configuration also demonstrated impact. The driveline configurations offer advantages and disadvantages over one another. Driveline configurations 1 and 2 contain a constant 100% locking PDU, whereas configurations 3 and 4 contain an asymmetrical and symmetrical open free-running PDUs. The constant locking PDU applies a torque from the transmission towards the

forward and rearward axles that depends on the kinematic discrepancy between the driving axles [14]. When the vehicle is experiencing poor terrain conditions between various wheels, a wheel may be under supplied with torque that could benefit and improve the UGV's mobility, whereas the other wheel could be over supplied with torque causing the wheels to slip and lose mobility. During the mathematical simulations of runs 1 and 3, the UGV is accelerating over two different friction coefficients on the left and right wheels.

Figure 6 illustrates the WMI_μ for the front axle in configurations 1 and 3. Using Eqn. (10) and (13), the wheels with a higher current friction coefficient would contain a lesser WMI. In these simulations, as a computational analysis revealed, the wheels in driveline configuration 1 are maintaining lower mobility capability, which is due to the wheels being over supplied by the driveline system with torque for the given terrain conditions.

a.)

b.)
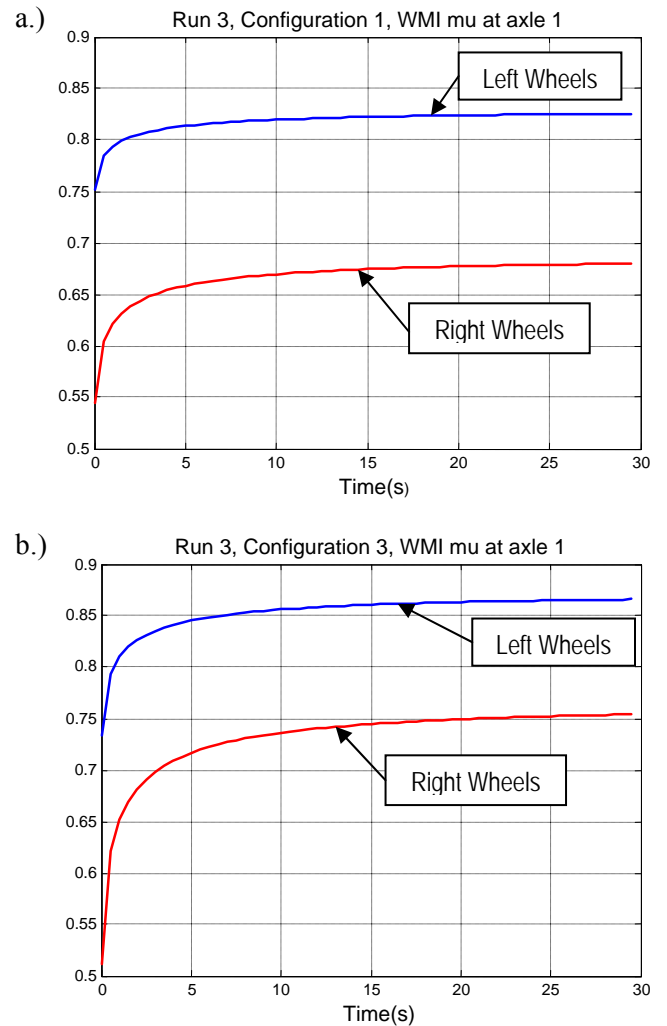
**Fig. 6**. FRICTION COEFFICIENT WMI, RUN 3, (a.) CONFIGURATION 1, AND (b.) CONFIGURATION 3, AT AXLE i = 1

Figure 7 illustrates the WMI_μ for configurations 1 and 2. Here, Fig. 7a represents the 6x6 with positive engagement of the axles and Fig. 7b represents the 6x4 driveline configuration with a positive engagement in the rear tandem. Positive engagement of the axles (transfer case and rear tandem) in configuration 1 provides high mobility as illustrated by the WMI of the front wheels (see higher values of WMI in Fig. 7a). For positive engagement PDU's, disconnecting the front axle results in higher current friction coefficient of the forward axle of the rear tandem and thus lower WMI (Fig. 7b). .

a.)

b.)
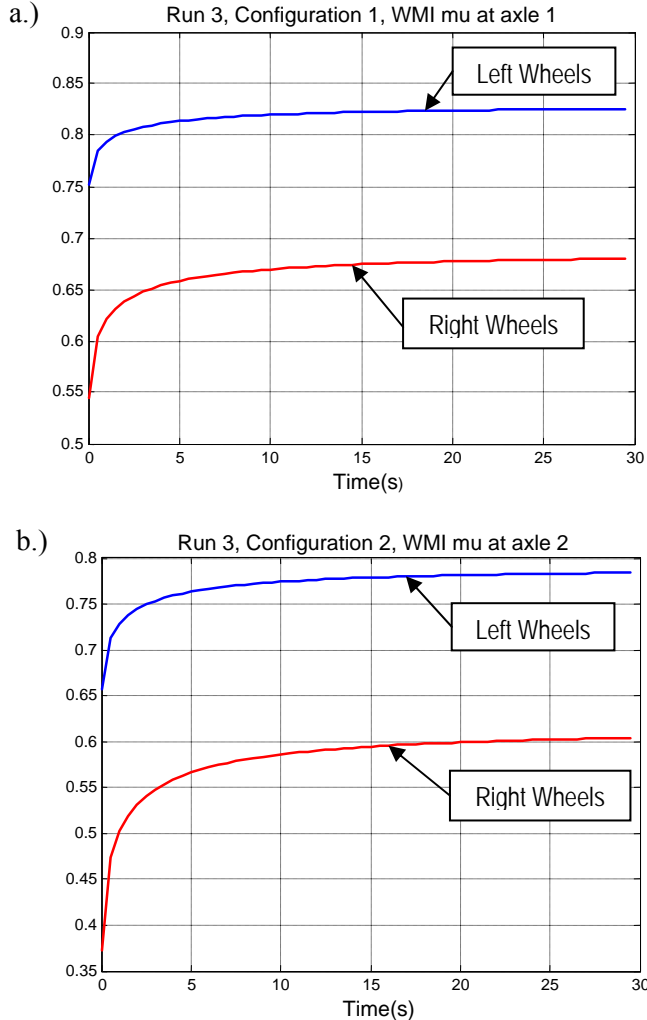
Fig. 7. FRICTION COEFFICIENT WMI, RUN 3,
(a.) CONFIGURATION 1, AND (b.) CONFIGURATION 2,
AT AXLE i = 1, 2

For illustration purposes, the considered computational simulations were presented using the average values of the peak friction coefficients. A consideration of tire gripping conditions as a stochastic process and similar analysis of the UGV with four driveline system configurations undertaken in this study resulted in a generalized strategy for computational mobility estimation. Figure 8 illustrates the strategy. The

circumferential forces F_x, $i = 1, 3$ and tire slippages are computed first. At the same moment of time the three check-cells check conditions $F_{xi} < \mu_{px} R_z$ (for the purpose of this paper, tire slippage values at $F_{x\,max}$ are not included) and then compute wheel mobility and vehicle mobility indices. The point here is that this computation process and analysis is conducted during j –moment of time but using terrain data that will be under the wheels in $(j + 1)$ – moment of time. The Engine/Transmission vs. Driveline Control check-cell gives instructions on changing the vehicle acceleration/speed or wheel torques (i.e., change the driveline configuration) that will be implemented in $(j + 1)$ – moment of time. Therefore, the proposed strategy is pre-emptive by its nature allowing for pro-active control of every wheel mobility and mobility of the vehicle as a whole.
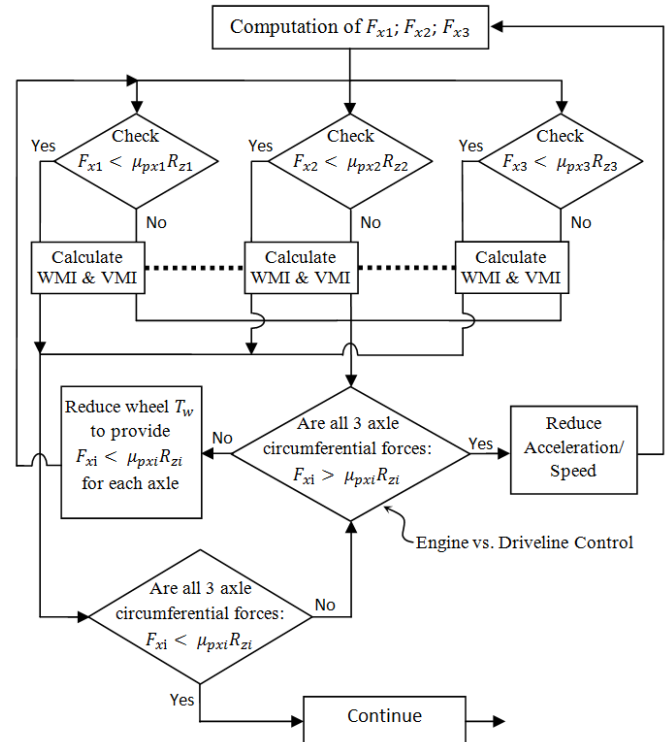
Fig. 8. ON-LINE MOBILITY ESTIMATION STRATEGY
SOFTWARE BLOCK DIAGRAM

CONCLUSION

In this paper, new support-surface mobility evaluation indices and on-line strategy were established for multi-wheel UGV applications as holonomic systems moving in unprepared terrain conditions. The novelty of these developments is that the contribution of each and every driving wheel has towards vehicle mobility can be precisely evaluated, and vehicle mobility can be improved by re-distributing power between the wheels via the driveline system of the vehicle. Thus, an optimal configuration of the driveline system as a set of power-dividing units can be found with taking in consideration surface-wheel-vehicle vertical and longitudinal dynamics, stochastic terrain characteristics,

and physical design of the power-dividing units. With this strategic approach, UGVs will be enabled to virtually eliminate general immobility situations by redesigning the driveline system towards correcting power distribution on-line and hence maintaining proper tire to terrain interactions.

The current work applies this strategic approach to both multi-wheel UGVs with mechanical/mechatronic driveline systems and individual wheel control (electric vehicle with e-motors in each wheel). As expected, this will enable the electric vehicle to maintain the optimal power for each wheel compared to mechanical drivelines that are not much flexible to broadly adapt their characteristics to verities of terrain situations.

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