A Generalized Approach to Virtual Driveline Systems for E-Vehicle Operation Improvements

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Abstract. Electric vehicles with the wheels individually driven by e-motors have promising potential for improving performance through finer control over the power distribution among the wheels. Due to the absence of a mechanical driveline to connect the wheels to the transmission and engine, the virtual driveline system (VDS) is proposed as a conceptual framework to connect virtually the individual electric motors and, thus, to optimize and analyze the dynamics and performance of vehicles. Conceptually, the VDS is based on vehicle-generalized parameters (VGP), which are used in the VDS principle to establish relationships between VGPs and, thus, to manage the wheel power split and set up interactive/coordinated controls of the e-motors to optimize and improve energy efficiency, terrain mobility performance, maneuver, etc.

Keywords: Vehicle Dynamics Theory, Modeling

1 Introduction

The mechanical driveline system of a vehicle determines the power split among the driving wheels and, thus, the driveline, with different configurations of power-dividing units (e.g., differentials, etc.) and gear ratios, has a strong contribution to the vehicle's overall energy efficiency, mobility performance, maneuver, etc. Electric vehicles can include multiple motors driving individual axles or wheels. Torque vectoring controls are used to manage the distribution of motor torques [1]. Different formulations of objective functions have been studied to enhance yaw stability or improve energy efficiency [2]. Controls based on tire slip have been used to improve performance [1]. For these electric vehicles, where the mechanical driveline is replaced with in-wheel e-motors, the wheels are not joined by axles and a transfer case, yet their dynamics influence each other, which may be understood as a virtual driveline system influencing the wheel power distribution by "connecting" the wheels through a computer code. A general method is detailed here for a vehicle with multiple individually-driven wheels.

1.1 Driveline and Power Dividing

Vehicle drivelines comprise a number of power dividing units (PDUs) which distributes power to the wheels. PDUs are classified on the basis of their effect on the distribution of power between their output shafts. The number of PDUs is one less than the number of the driving wheels: a 4x4 vehicle will have 2 interwheel and 1 interaxle PDU, a 6x6 vehicle will require 5 PDUs, etc.

For an electric vehicle, individual motors may be applied to separately drive each wheel. This removes the need for a mechanical driveline and PDUs. These motors do not have any mechanical connection, however, this does not mean that the motors do not influence each other through their contributions to the tire-road traction. The said can be explained using a diagram of one wheel shown in Fig. 1 for a steady motion (similar can be done for a non-steady motion).

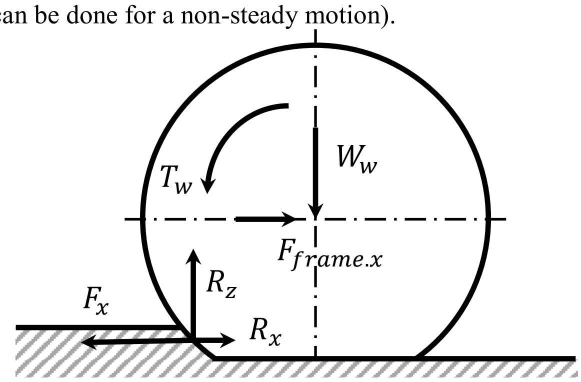

Fig. 1. Diagram of one wheel

When having several electric motors, different torques are easily applied to each wheel. With different torques applied to the wheel in Fig. 1, the magnitude of the wheel circumferential force F_x changes. In steady motion, the wheel torque and F_x are related by

$$T_w = F_x r_w^0 \tag{1}$$

where r_w^0 is the tire rolling radius in the driven mode (a wheel is pulled with zero torque applied). The rest of force F_x left after compensating the rolling resistance R_x is a the net tractive force, F_w:

$$F_w = F_x - R_x \tag{2}$$

When the tractive forces and terrain conditions at each wheel are different, the discrepancy between tire slippages impacts vehicle dynamics parameters such as slip efficiency. A system of coordination between the motors is necessary. In this regard, the principle of electric virtual driveline systems is proposed. The shafts of the motors are connected virtually as shown in Fig. 2. The electric motors are controlled to produce a desired driveline characteristic. A general approach to model driveline characteristics is proposed in the next section.

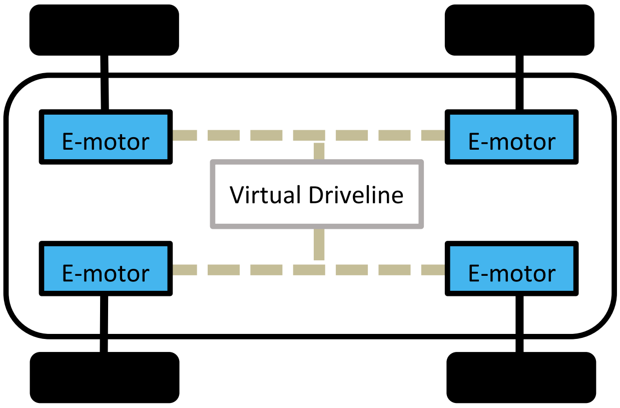

Fig. 2. Virtual driveline of 4x4 vehicle with 4 individual motors

2 Concept and Principle of Virtual Driveline System

The virtual driveline system concept is based on a system of generalized parameters. The generalizing of parameters involves taking a group of wheels and calculating a reduced set of parameters, including tire rolling radii in the driven mode r_w^0 and tire longitudinal slippage s_δ. Mathematically, the generalized parameter represents an equivalent single wheel replacing the group (Fig. 3).

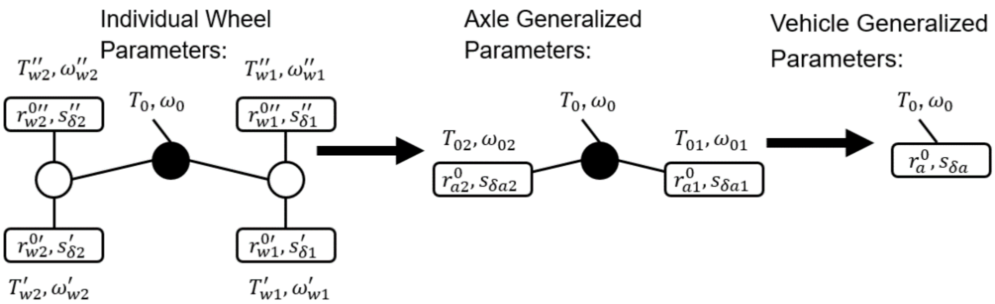

Fig. 3. Vehicle generalized parameters from individual wheel parameters of 4x4 vehicle

Notation $'$ and $''$ index the right and left wheels of an axle, $i = 1, n$. Equations have been derived to link single-wheel parameters to the generalized parameters for different configurations of mechanical drivelines of vehicles with n-number of the drive axles [3]. Through the use of these sets of equations, the dynamics of any number of driving wheels are linked, even when driven by separate e-motors with no mechanical connection, mimicking a mechanical driveline with variable gear ratios controlling the wheel power split. Generalized vehicle parameters provide a research foundation for any type of virtual driveline system. The tire slippage, s_δ, is defined using the reduction in velocity from the wheel's theoretical velocity V_t to the actual velocity V_x when torque is applied to the wheel as shown by Eq. (3).

$$s_{\delta i}^{\prime(\prime\prime)} = \frac{V_{ti}^{\prime(\prime\prime)} - V_x}{V_{ti}^{\prime(\prime\prime)}}, i = 1, n \tag{3}$$

A generalized slippage of a drive axle, $s_{\delta ai}$, can be computed using the slippages of its wheels. For an axle with an interwheel open differential Eq. (4) is in use.

$$s_{\delta ai} = 1 - \frac{R_t\left(r_{wi}^{0\prime\prime} + r_{wi}^{0\prime}\right)(1 - s_{\delta i}^{\prime\prime})(1 - s_{\delta i}^{\prime})}{R_t^{\prime\prime} r_{wi}^{0\prime}(1 - s_{\delta i}^{\prime}) + R_t^{\prime} r_{wi}^{0\prime\prime}(1 - s_{\delta i}^{\prime\prime})}, i = 1, n \tag{4}$$

where R_t terms are turn radii of the axle's wheels (neglected from the equation in straight line motion).The generalized slippage of the vehicle,$s_{\delta a}$, represents the drop in velocity to V_x from the theoretical velocity of the vehicle without slip, V_a.

$$s_{\delta a} = \frac{V_a - V_x}{V_a} \tag{5}$$

The axle actual linear velocities have the same projections on the vehicle's longitudinal axis and are equal to the velocity represented by the generalized parameters:

$$V_{x1} \cos \delta_1 = V_{x2} \cos \delta_2 = \omega_0 r_a^0 (1 - s_{\delta a}) \tag{6}$$

The actual velocity of the vehicle is

$$V_x = \omega_{0i} r_{ai}^0 (1 - s_{\delta ai}) \tag{7}$$

Between the outputs of the interaxle differential and the inputs to the interwheel differentials are final drive gear ratios u_i.

$$\omega_0 = \frac{\omega_{01} u_1 + \omega_{02} u_2 u_d}{1 + u_d} \tag{8}$$

Through manipulation of Eqs. (5-7), the generalized slippage of the vehicle with an open differential in the transfer case is obtained as

$$s_{\delta a} = 1 - \frac{(1 - s_{\delta a2})(r_{a2}^0 u_1 + r_{a1}^0 u_2 u_d) \cos \delta_2}{r_{a1}^0} \left[u_2 u_d + \frac{r_{a2}^0 u_1}{r_{a1}^0} \frac{(1 - s_{\delta a2}) \cos \delta_2}{(1 - s_{\delta a1}) \cos \delta_1}\right]^{-1} \tag{9}$$

A second example is considered for a positive engagement (i.e., the transfer case is locked or a full time locking differential is used in the transfer case).
The theoretical velocities of the wheels change from $V_{ti}^{\prime(\prime\prime)}$ to V_a because all the wheels are linked by the vehicle frame. On the way of this change, the theoretical velocities of the left and right wheels of an axle can be converted in the theoretical velocity of the axle, V_{ti}. Since all the axles are connected to the vehicle frame and, hence, all together have the same velocity of V_a, the velocity differences creates a kinematic discrepancy factor, m_H, for each axle

$$m_{Hi} = \frac{V_{ti} - V_a}{V_{ti}}, i = 1, n \tag{10}$$

When Eqs. (3, 5, 10) are combined through substitution to remove velocity terms, the result is Eq. (11), which shows that the kinematic discrepancy factor may be used to link each axle generalized slippage to the vehicle generalized slippage.

$$s_{\delta ai} = m_{Hi} + (1 - m_{Hi}) s_{\delta a}, i = 1, n \tag{11}$$

Therefore, the vehicle generalized slippage $s_{\delta a}$, which characterizes velocity loss of the vehicle, may be related to individual tire slippages (see Eqs. (4) and (11)), which result from the distribution of torques among the wheels. The physical meaning of the kinematic discrepancy of the ith axle is that the factor is the generalized slippage of the ith axle when traction load is zero, i.e., $s_{\delta a} = 0$ [3]. The rolling radius of a wheel in the driven mode, r_w^0, can be used as the reference value for zero torque for calculating theoretical velocity of each wheel [3]:

$$V_{ti}^{\prime(\prime\prime)} = \omega_{wi}^{\prime(\prime\prime)} r_{wi}^{0\prime(\prime\prime)}, i = 1, n \tag{12}$$

However, two wheels of the same axle have the same theoretical velocity when pulled together since they are connected to the same frame.

$$V_{ai} = \omega_{0i} r_{ai}^0, i = 1, n \tag{13}$$

here, ω_{0i} is the angular velocity of the interwheel differential, r_{ai}^0 is the axle generalized rolling radius in the driven mode, which is introduced as the rolling radius of a hypothetical wheel that runs with the velocity of ω_{0i}.

Using the relationship between the angular velocities and the torques of two wheels linked by an open differential, the axle generalized rolling radius in the driven mode can be derived as follows

$$r_{ai}^0 = \frac{2 r_{wi}^{0\prime\prime} r_{wi}^{0\prime}}{(r_{wi}^{0\prime\prime} + r_{wi}^{0\prime}) u_{ki}} \tag{14}$$

here, u_{ki} is the gear ratio of the wheel-hub gear set that is usually the same for the left and right wheels of an axle.

Finally, the rolling radius in the driven mode may then be generalized to the vehicle as radius r_a^0 using

$$V_a = \omega_0 r_a^0 \tag{15}$$

r_a^0 is the rolling radius of a hypothetical wheel that rotates with the angular velocity of the input shaft of the transfer case, ω_0, and has the same theoretical velocity as the vehicle. When the vehicle is pulled in the driven mode, the total torque at the input shaft of the transfer case is equal to zero

$$T_0 = \sum_{i=1}^{n} \frac{T_{wi}^{\prime(\prime\prime)}}{u_{ki} u_i^{\prime(\prime\prime)}} = 0 \tag{16}$$

However, the torques at some wheels in Eq. (16) can be either positive or negative due to the sign of the kinematic discrepancy factor at $s_{\delta a} = 0$ in Eq. (11).

In the driven mode, torque and slippage are low and the function between the wheel circumferential force F_x and tire slippage may be approximated as a linear relation

$$F_{xi} = K_{ai} s_{\delta ai} \tag{17}$$

in which K_{ai} is the equivalent longitudinal stiffness of two wheels of an ith axle [3]. Using this linear relationship and Eqs. (10, 12), an equation for r_a^0 is derived:

$$r_a^0 = \left(\sum_{i=1}^{n} K_{xi}^{\prime(\prime\prime)} r_{wi}^{0\prime(\prime\prime)} / u_{ki} u_i^{\prime(\prime\prime)}\right)\left(\sum_{i=1}^{n} K_{xi}^{\prime(\prime\prime)}\right)^{-1} \tag{18}$$

As the wheel gains a torque in the driving mode of operation, it also develops slippage and the rolling radius drops to r_w from r_w^0. The generalized rolling radius of the vehicle in the driving mode, r_a, is introduced as the rolling radius of a hypothetical wheel which would make the linear velocity of the vehicle as follows

$$V_x = \omega_0 r_a \tag{19}$$

The generalized rolling radii and slippage provide an analytical foundation for controlling the power split among the wheels as explained below.

3 VDS Principle for Wheel Power Management

When all tires have the same longitudinal stiffness, Eqs. (10,12,13) lead to the kinematic discrepancy given by Eq. (20)

$$m_{Hi}^{\prime(\prime\prime)} = 1 - \frac{u_{ki} u_i^{\prime(\prime\prime)}}{r_{wi}^{0\prime(\prime\prime)}} r_a^0, i = 1,2 \tag{20}$$

As seen from Eq. (20), for a vehicle with the mechanical driveline system, the kinematic discrepancy can be controlled by changing gear ratios, u_{ki}, $u_i^{\prime(\prime\prime)}$ and radius r_a^0, leading to different tire slippages as follows from Eq. (11). Tire slippage is closely linked to the wheel circumferential force; the following form is used [4]:

$$F_{xi}^{\prime(\prime\prime)} = \mu_{pxi}^{\prime(\prime\prime)} R_{zi}^{\prime(\prime\prime)} \left(1 - \frac{s_{\delta c}^{\prime(\prime\prime)}}{2 s_{\delta}^{\prime(\prime\prime)}} \left(1 - \exp\left(-\frac{2 s_{\delta}^{\prime(\prime\prime)}}{s_{\delta c}^{\prime(\prime\prime)}}\right)\right)\right) i = 1, n \tag{21}$$

where μ_{px} is the peak friction coefficient, R_z is the normal reaction, and $s_{\delta c}$ is the tire characteristic slippage. Characteristic slippage is the point at which the $F_x - s_\delta$ curve becomes strictly nonlinear. The circumferential forces at all four wheels must add up to the total force, $F_{x\Sigma}$, to overcome the resistance to motion ($F_{x\Sigma} = \sum_{i=1}^{n} F_{xi}^{\prime(\prime\prime)}$). Different combinations of the individual F_x forces however, may add up to the same result but give different slippages. These combinations may give radically different results in terms of vehicle performance.

The usage of the VGPs is now extended to e-vehicles by introducing the VDS principle for the wheel power management. The principle states that the gear ratios u_{ki} and $u_i^{\prime(\prime\prime)}$ are interpreted as individual voltage inputs to control the e-motors and, thus, to provide sets of VGPs, including wheel circumferential forces and tire slippages, which correspond to required/reference assessment indices of vehicle operation. Hence, in the VDS, gear ratios u_{ki} and $u_i^{\prime(\prime\prime)}$do not physically exist but instead become control signals determining VGPs and tire traction and slippage at each wheel. The above analysis demonstrates how generalized parameters may be employed to manage the power split of an e-vehicle with multiple wheel motors through a mechanical analogy, treating the motor control signals as the manipulation of the gear ratios of a virtual driveline.

For a more generalized case, the power distribution of individual drives can be represented with a circumferential force distribution factor ν, and slippage proportional factor γ. The circumferential force factor for a wheel $\nu_i^{\prime(\prime\prime)}$ expresses the proportion of the total circumferential force $F_{x\Sigma}$ carried by that wheel.

$$F_{xi}^{\prime(\prime\prime)} = \nu_i^{\prime(\prime\prime)} F_{x\Sigma} = \nu_i^{\prime(\prime\prime)} R_{m\Sigma} \tag{22}$$

where $R_{m\Sigma}$ is total resistance to motion. The vehicle generalized rolling radius r_a^0 is derived using by relating the sum of wheel torques to $R_{m\Sigma}$ multiplied by r_a^0.

$$\sum F_{xi}^{\prime(\prime\prime)} r_{wi}^{0\prime(\prime\prime)} = R_{m\Sigma} r_a^0 \tag{23}$$

$$\sum \nu_i^{\prime(\prime\prime)} R_{m\Sigma} r_{wi}^{0\prime(\prime\prime)} = R_{m\Sigma} r_a^0 \tag{24}$$

$$r_a^0 = \sum \nu_i^{\prime(\prime\prime)} r_{wi}^{0\prime(\prime\prime)} \tag{25}$$

The slippage proportional factor $\gamma_i^{\prime(\prime\prime)}$ for a wheel relates the tire slippage $s_{\delta i}^{\prime(\prime\prime)}$ to that of the generalized slippage, $s_{\delta a}$.

$$s_{\delta i}^{\prime(\prime\prime)} = \gamma_i^{\prime(\prime\prime)} s_{\delta a} \tag{26}$$

Using the slippage proportional factors and circumferential force distribution factors, each wheel's individual traction-slip curve is linked to $R_{m\Sigma}$ and $s_{\delta a}$.

$$\nu_i^{\prime(\prime\prime)} R_{m\Sigma} = \mu_{pxi}^{\prime(\prime\prime)} R_{zi}^{\prime(\prime\prime)} (1 - \frac{s_{\delta ci}^{\prime(\prime\prime)}}{2\gamma_i^{\prime(\prime\prime)} s_{\delta a}} (1 - \exp(-\frac{2\gamma_i^{\prime(\prime\prime)} s_{\delta a}}{s_{\delta ci}^{\prime(\prime\prime)}}))) \tag{27}$$

ν and γ therefore are the factors which characterize the power split of the individual drives (in an unrestricted fashion), modeling how total circumferential force is distributed among the wheels and how each contributes to slippage. The power split among the wheels may be reconfigured in real time to match determined ideal values. Specifically, mobility performance, energy efficiency, and vehicle maneuver may be improved. Different combinations $\left\{F_{xi}^{\prime(\prime\prime)}, s_{\delta i}^{\prime(\prime\prime)}\right\}, i = 1, n$ may give radically different results in vehicle performance measurements. This becomes an optimization problem where the combination of wheel slippages is determined to meet the performance needs of a vehicle. Examples are shown here for a 4x4 e-vehicle with gross weight 85 kN moving at 10 mph on off-road meadow terrain.

Slippages results in a reduction of efficiency, measured by the vehicle slip efficiency given in Eq. (28) [3].

$$\eta_\delta = \frac{F_{x\Sigma}}{F_{x\Sigma}+\sum_{i=1}^{2} F_{xi} s_{\delta i}/(1-s_{\delta i})} \tag{28}$$

Slip efficiency becomes maximum when slippages at all wheels are equal. This is achieved by zeroing the kinematic discrepancy and, thus, determining the control voltages, which are the virtual gear ratios in Eq. (20). Figure 4 shows generalized slippages for a conventional driveline with three open differentials and the generalized slippages of an electric driveline with individual motors with slippages of all wheels made equal. The vehicle was simulated with these driveline configurations on off-road stochastically-changing terrain with a varying distribution of the peak friction coefficient.

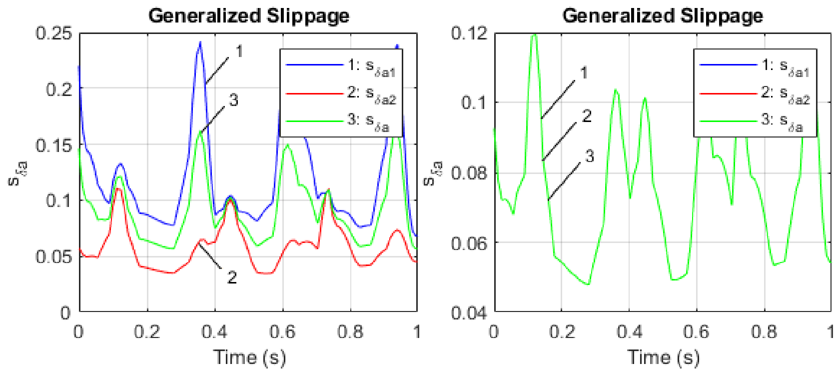

Fig. 4. Generalized slippages of mechanical driveline with open differentials (left) and optimal generalized slippages for energy efficiency (right): 1 – front wheels' generalized slippage, 2 – rear wheels' generalized slippage, 3 – vehicle generalized slippage

Resulting slip efficiency calculated from Eq. (28) is shown in Fig. 5; with slippages equal, the e-driveline achieves higher energy efficiency at all times of the simulation.

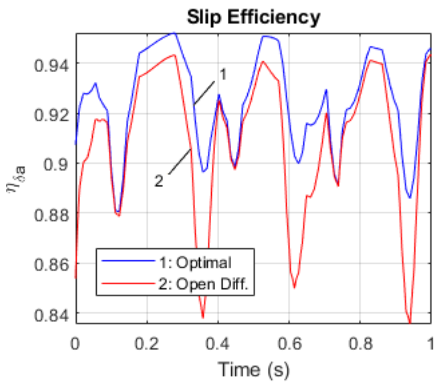

Fig. 5. Slip efficiency resulting from generalized slippages in Fig. 4

Alternate combinations of the optimal control voltages correspond to maximum mobility. The Vehicle Mobility Index (VMI) given by Eq. (29) allows assessing the contribution and establishing boundaries that each individual driving wheel provides and imposes on the vehicle mobility [5].

$$VMI_{\mu} = 1 - \frac{1}{2n}\sum_{i=1}^{n}\frac{F_{xi}^{\prime(\prime\prime)}}{F_{xi}^{max\prime(\prime\prime)}} = 1 - \frac{1}{2n}\sum_{i=1}^{n}\frac{\mu_{xi}^{\prime(\prime\prime)}}{\mu_{pxi}^{\prime(\prime\prime)}} \tag{29}$$

here, $F_{xi}^{max\,\prime(\prime\prime)}$ is the maximum circumferential force determined by the gripping properties of tires, $F_{xi}^{max\,\prime(\prime\prime)} = \mu_{pxi}^{\prime(\prime\prime)} R_{zi}^{\prime(\prime\prime)}$. As shown in Fig. 6, employing individual motors may improve the vehicle's mobility index in terrain conditions.

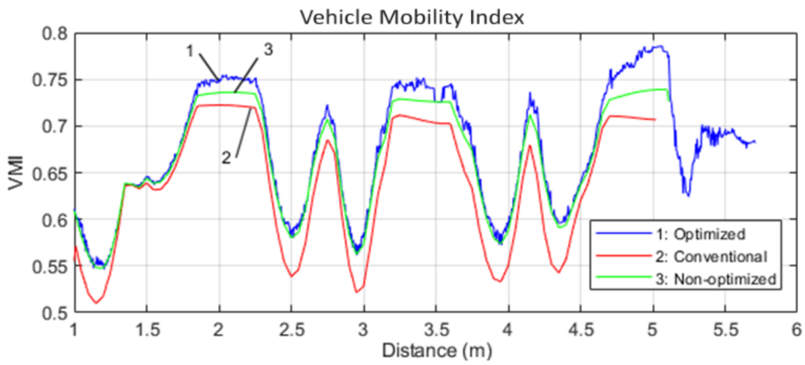

Fig. 6. VMI of conventional driveline compared with non-optimized and optimized for mobility virtual driveline

Additionally, vehicle maneuver may also be improved by controlling the kinematic discrepancies. In Fig. 7, a hybrid-electric power transmitting unit was modeled to control the kinematic discrepancies between the front and rear axles of a single-motor electric vehicle [6]. Individually changing the angular velocities of the wheels provides a steering effect on the turning moment.

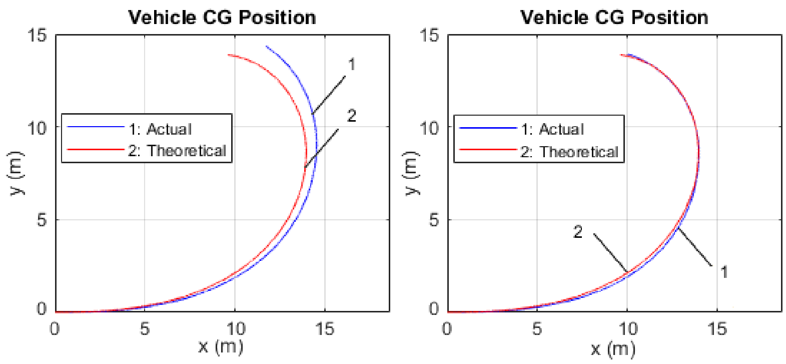

Fig. 7. Trajectory path of 4x4 vehicle with open differentials (left) and controllable front/rear power split (right)

4 Conclusion

The VDS principle for managing multiple driving wheels not connected by a mechanical driveline and driven by individual e-motors is proposed. Wheel parameters are linked to each other and to the vehicle generalized parameters, allowing optimal power split to be determined through virtual gear ratios. The method shows possibilities for the maximization of mobility performance and energy efficiency for terrain vehicles, while maneuver of vehicles is improved by controlling the yaw moment. The most general approach of the method is being prepared as a journal paper covering the building of a virtual driveline from templates of generalized parameters for different types of differentials and individual drives. Current work based on this study is on a control design to control wheel power distributions in different modes to optimize for energy efficiency, terrain mobility, and maneuver.

ACKNOWLEDGMENTS

This research supported by NSF award S&AS-1849264 and partially by Automotive Research Center Phase V - Cooperative Agreement #1.A85

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