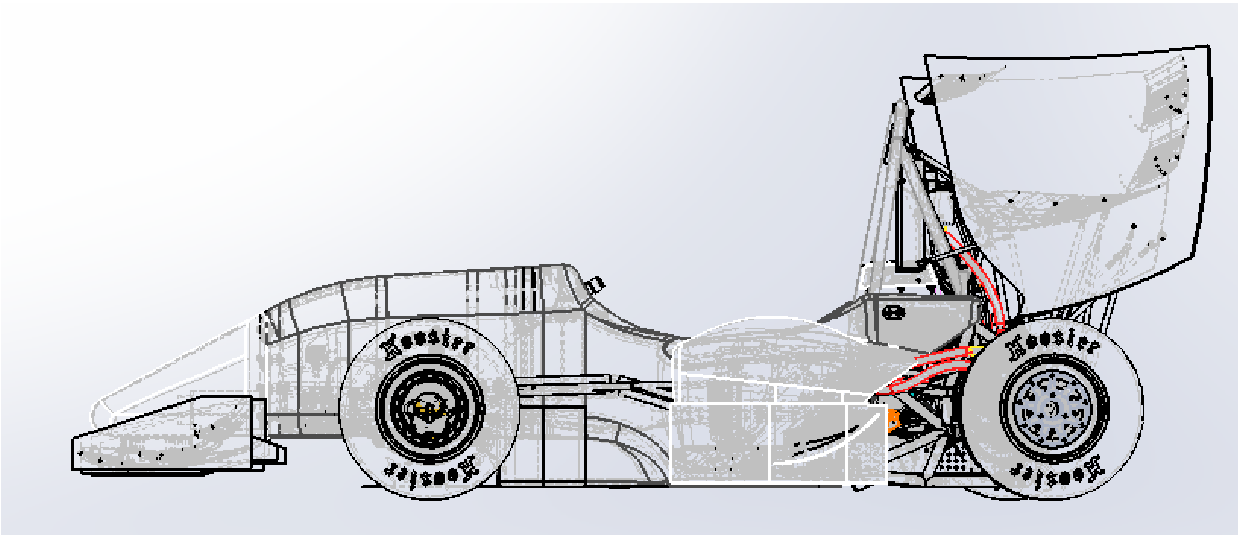
WR-217e



Powertrain Architecture, Simulation, and Controls

WISCONSIN

UNIVERSITY OF WISCONSIN–MADISON

WR-217e Architecture Design

The first step of architecture design, especially when given a more-or-less blank slate, is to create an architecture development model. It is better to make *one* architecture development model that only uses high level car parameters (e.g. a lap time simulator) and then make a few more specific models (e.g. step steer torque vectoring response, quarter-car suspension model, etc), rather than to try to incorporate all of the models together. When starting to make a model, ask yourself: “What am I trying to get out of this model?” or “What will I be using this model to optimize?” From there, start simple and add complexity to the model as needed in order to achieve your goals.

These are the goals for the architecture development model that will be outlined in this document:

Goals

- Determine the optimal power source characteristics required to achieve high level performance targets* (e.g. peak torque, peak power, torque curve shapes)
- Determine optimal transmission ratio(s) to achieve high level performance targets
- Determine required onboard energy to achieve the desired vehicle range
- Determine aerodynamic properties that result in the optimal tradeoff between performance and efficiency
- Determine optimal high level chassis parameters (e.g. vehicle CG location)

*note: High level performance targets can include metrics such as peak acceleration and top speed. These are chosen at the architecture engineer’s discretion, and may need to be revised later as physical constraints are realized (e.g. packaging, vehicle mass, etc).

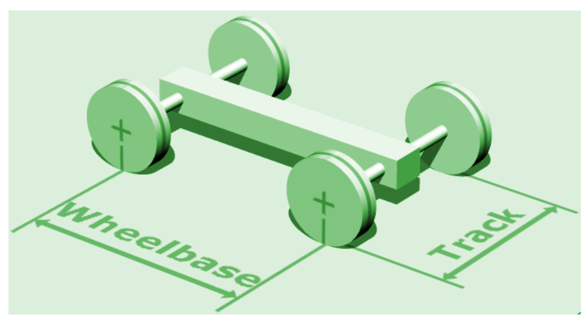
Now, we can identify the vehicle parameters we will need to include in our model:

Propulsion

- Torque-speed curve for the vehicle power source
- Number of gears and corresponding transmission ratios (N_1 , N_2 … N_n)
- Torque-speed efficiency map for power source (optional)

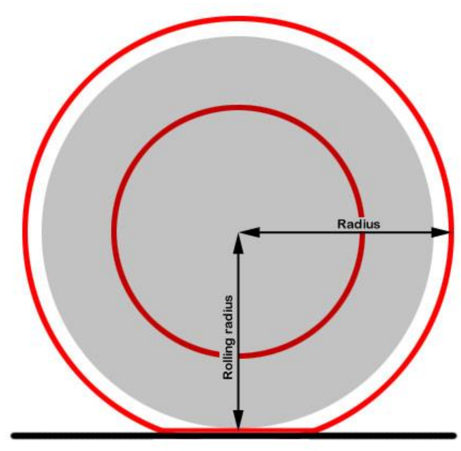
Chassis

- Total vehicle mass w/ driver (m)
- Height of the center of gravity (CG_z)
- Wheelbase and track width(s) (wb, tw_f, tw_r)
- Static weight distribution (swd) (from 0 to 1, higher is more rearward)



Tires

- Estimate of tire rolling radius (r_{tire})
- Estimate of maximum longitudinal coefficient of friction (μ_{long})
- Estimate of maximum lateral coefficient of friction (μ_{lat})
- Sensitivity of coefficient of friction to normal load (optional)



Aerodynamics

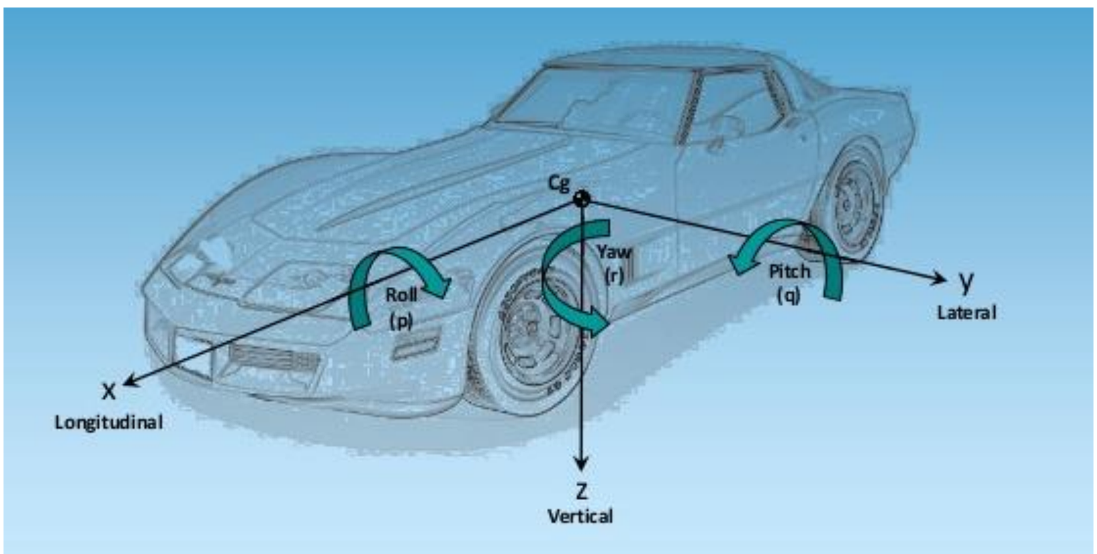
- Coefficient of downforce (C_{df})
- Coefficient of drag (C_d)
- Frontal area (A_f) (area of the vehicle from a front view)
- Center of pressure (cp) (from 0 to 1, higher is more rearward)
- Air density (ρ)

Notice what is missing:

- Suspension geometry
- Spring rates, damping rates
- 26 DOF Pacejka tire model
- KNC compliance data

This is on purpose! We don't need any of that information to achieve our goals. Notice that all of the parameters of the car that we are trying to optimize are included as input parameters for our model. Because of this, we will need to make some educated guesses for our initial inputs. These guesses will be driven by a combination of data from older vehicles and first principles.

The vehicle coordinate system



Steps to build your model:

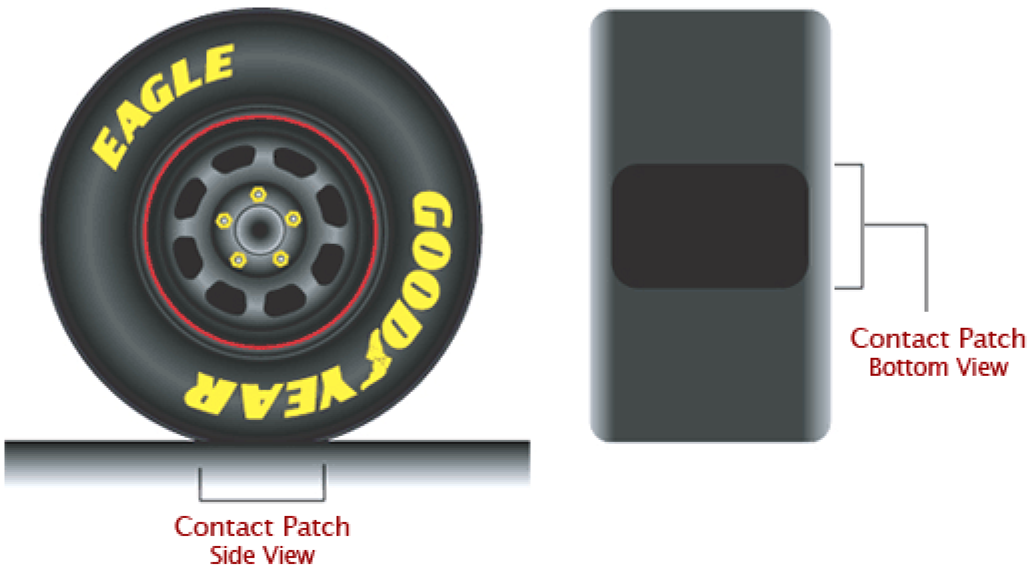
Step 1: Point-mass acceleration event

Step 2: Bicycle model acceleration event

Step 3: Two-track model autocross event

Step 1: Part A – The Point Mass

To build your model, start with a point mass that accelerates in one dimension. We will call this type of model a 1-DOF traction limit acceleration model, because we will calculate the "traction limit" based on the given longitudinal coefficient of friction between the point mass and the ground. In this model, it is assumed some propulsion system is capable of fully utilizing the available friction force at the contact patch to accelerate the vehicle in all conditions.



Inputs:

- Mass of the point [kg]
- Coefficient of friction [-]

Equations:

1) Kinematic relationships using the *symplectic (semi-implicit) Euler approximation:*
    - $v_i = v_{i-1} + a_{i-1}dt$
    - $x_i = x_{i-1} + v_i dt$
2) $F_{net} = ma$
3) $F_{friction} = \mu F_N$

Calculations:

$F_{net} = F_{friction}$

$ma = \mu_{long} F_N$

$F_N = mg$

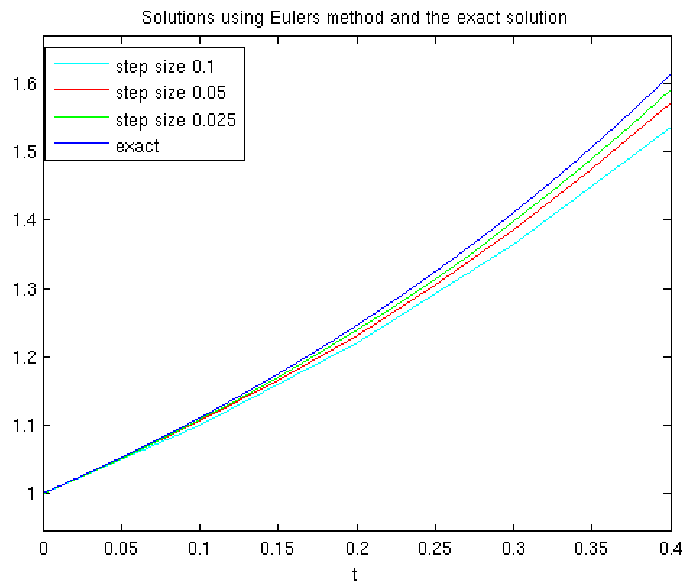
$a = \mu_{long} g$

Mass cancels out in this first case, but is included in the model as we will need it later on. When you run your simulation, you should be able to plot acceleration, velocity, and position vs. time. You should be able to see a linearly increasing velocity and a quadratically increasing position.

For this first model and as we move forward, I will walk you through how to calculate the accelerations of the vehicle. It is up to you to create a numerical (discrete time-step) model to calculate vehicle velocity and position.

Q: How do I decide on what increment to make dt?

A: It entirely depends on the time constants of different phenomena in your dynamic system. For this application, I recommend $dt = 0.001$. As $dt \to 0$, the behavior of your numerical model will asymptotically approach the exact solution, as shown in the plot of an arbitrary function below.



Step 1: Part B – Aerodynamic Effects

We will now add some aerodynamic parameters to our point-mass acceleration event.

Additional Inputs:

- Coefficient of downforce [-]
- Coefficient of drag [-]
- Air density [kg/m^3]

Additional Equations:

1) $F_{downforce} = \frac{1}{2}\rho A_f C_{df} v^2$
2) $F_{drag} = \frac{1}{2}\rho A_f C_d v^2$

Calculations

$$F_{friction} - F_{drag} = F_{net}$$

$$F_N = mg + F_{downforce}$$

$$\mu_{long}\left(mg + \frac{1}{2}\rho A_f C_{df} v_{i-1}^2\right) - \left(\frac{1}{2}\rho A_f C_d v_{i-1}^2\right) = ma$$

$$a = \mu_{long} g + \frac{\frac{1}{2}\rho A_f\left(\mu_{long} C_{df} - C_d\right) v_{i-1}^2}{m}$$

Now, mass no longer cancels out. Acceleration still has the $\mu_{long} g$ component from Part A, but now also has an additional component due to combined aerodynamic effects. Even from this simple model, it can be seen that the ***aerodynamic effects will impact vehicle acceleration more as the vehicle mass is decreased.*** *We also find that when $\mu_{long} C_{df} > C_d$, the net effect of aerodynamics will increase acceleration. This is not necessarily true once a more complex model is developed, but still interesting to note here. This is because in reality vehicles usually do not ride the traction limit all the way to top speed due to the incredible power required to do so. When the vehicle become power limited (no longer traction limited), downforce is no longer increasing the vehicle's ability to accelerate, but drag is still decreasing the vehicle's ability to accelerate. From this, we can conclude that given the above condition is satisfied, the net effect of aerodynamics is positive when the vehicle is traction limited, and negative when the vehicle is power limited.*

Note: since acceleration is now velocity-dependent, we use the velocity from the previous time step as an approximation for the velocity at the current time step.

Step 1: Part C – The Propulsion System

In this part, we will add a defined propulsion system to our model. For example purposes, we will keep our propulsion system simple: an electric motor that can be characterized by a peak torque, peak power and a fixed gear ratio. Now, instead of our point-mass vehicle accelerating at the limit of traction, we will compare the capability of the motor to the traction limit and use the smaller of these forces to accelerate the point mass. This is, in effect, a rudimentary "traction control" system.

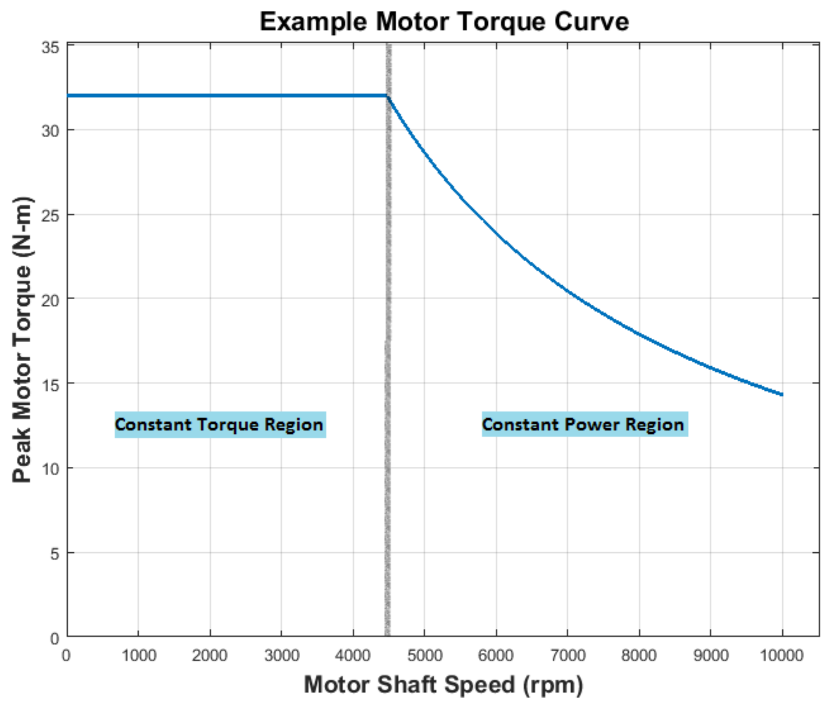
Additional Inputs:

- Motor peak torque (N-m) (T_{max})
- Motor peak power (W) (P_{max})
- Transmission ratio [-] (N)

Additional Equations:

1) $F_{motor} = \min\left(\frac{T_{\max * N}}{r_{tire}}, \frac{P_{max}}{v_{i-1}}\right)$
2) $F_{cp} = \min\left(F_{friction}, F_{motor}\right)$

(force at the contact patch of the tire)



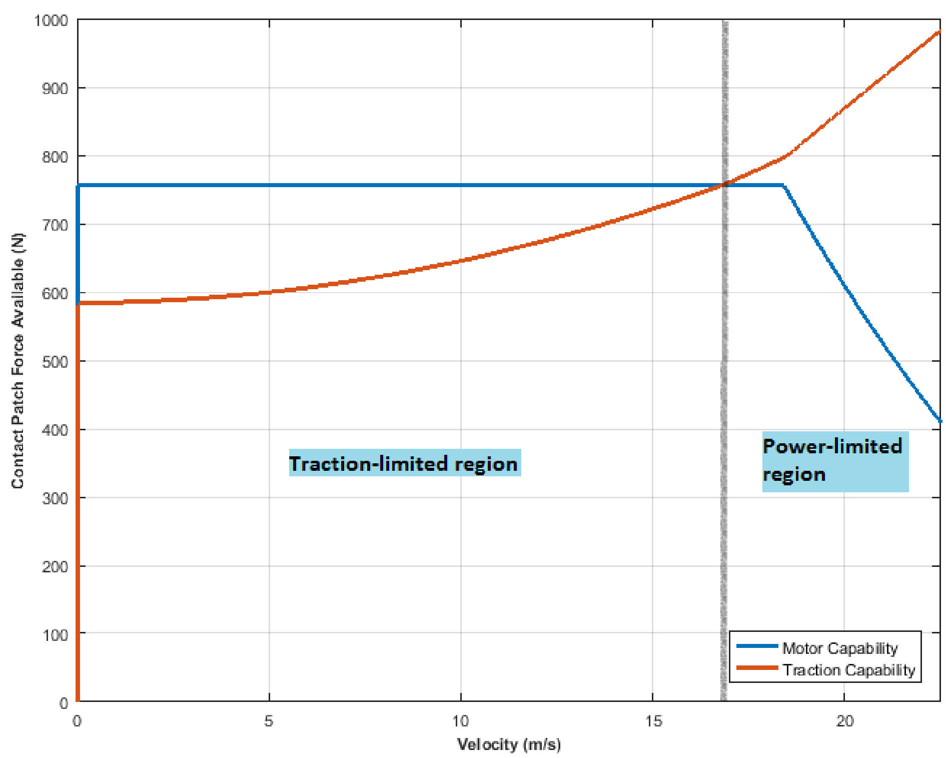
Calculations:

$$F_{cp} - F_{drag} = F_{net}$$

$$F_N = mg + F_{downforce}$$

$$F_{cp} - F_{drag} = ma$$

$$a = \frac{F_{cp} - F_{drag}}{m}$$



Step 2: The Bicycle Model

In this part, we turn our point mass into a two-wheeled bicycle. We will need to make a couple adjustments to our equations to accommodate this, mainly by introducing the concept of weight transfer (or, more correctly and specifically, *longitudinal load transfer*).

When a force is applied at the bottom of an object with a center of gravity some distance above the ground, a moment is applied about the CG. The result of this moment on an object like a bicycle (with two points in contact with the ground and therefore two separate normal forces) is that some of the normal load from the front tire is transferred to the rear tire during acceleration (and vice versa during braking). By doing a moment balance about one of the tires, this load transfer can be described:

$$WT_{long} = ma_{long} * \frac{CG_Z}{(wb)}$$

The magnitude of this load transfer will be added or removed from each tire depending on the direction of acceleration. We will now begin distinguishing accelerations and weight transfers as longitudinal (in the direction of the bicycle's travel), and eventually when we get to the two track model as lateral (in the direction pointing from the vehicle to the center of a turn). We will now also have separate equations for the front tire and the rear tire. For this example, we will assume there is a motor for each tire as well.

Additional Inputs:

- Static weight distribution (swd)
- Wheelbase of the bicycle (wb)
- Height of the center of gravity of the bicycle (CG_z)
- Center of pressure (cp)

Additional Equations:

1) $WT_{long} = ma_{long} * \frac{CG_Z}{(wb)}$
2) $F_{N,front} = mg * (1 - swd) + F_{downforce} * (1 - cp) - WT_{long}$
3) $F_{N,rear} = mg * (swd) + F_{downforce} * (cp) + WT_{long}$

Calculations

$F_{cp,front} + F_{cp,rear} - F_{drag} = F_{net}$

For the sake of space, we solve more generally:

$F_{cp,front} + F_{cp,rear} - F_{drag} = ma$

$a = \frac{F_{cp,front} + F_{cp,rear} - F_{drag}}{m}$

If the front and rear wheels are both traction limited, acceleration actually reduces to the same formula in Step 1 – Part B. However, when a wheel is power limited (i.e. the motor cannot fully utilize the available grip), non-linearities are introduced into our model. Now, we have longitudinal acceleration that is self-affecting.

$$a = f(F_{net}) \rightarrow F_{net} = f\left(WT_{long}\right) \rightarrow WT_{long} = f(a)$$

For example, let us say that we start accelerating from a standstill. Initially, we assume there is no longitudinal weight transfer, so the dynamic normal load distribution is such that both wheels are traction-limited. Now, we have the resulting acceleration that we got in Step 1 – Part B. However, now that the bicycle is accelerating, there is longitudinal weight transfer. If we go back and recalculate this, we find that the new dynamic normal load distribution is such that now the front wheel is traction-limited and the rear wheel is power-limited due to the additional normal load on the rear wheel. As such, if we subsequently recalculate the contact patch forces, the net force on the bicycle will be different than our first calculation. So what is actually going on?

This is a good time to bring up that the model we have made is a quasi-steady state *model. This is because the weight transfer equations that we are using ignore the transient effects of the spring/dampers on the vehicle. As such, in reality weight transfer has a transient component from the suspension and does not reach a steady state value instantaneously. Since we are neglecting this, we must implement an* iterative solver *to determine the steady-state weight transfer for each time step. For more information on how to implement a simple iterative solver, see Appendix X.*

Using the steady state value of weight transfer at each time step is perfectly fine given our goals for our architecture design simulation. While the transient effects of load transfer through a suspension system have a massive and very important impact on the real-life handling (and subsequently performance) of a vehicle, <u>they do not significantly affect the optimization of the high level parameters</u> we have set out to investigate. For a more accurate model of our suspension, we could make a smaller and more specific quarter-car model.

Step 3: The Two-Track Model

Congratulations! We've arrived at the part where we begin modeling a four-wheeled vehicle. In this section, we will introduce a couple new ideas and turn our acceleration event simulator into an autocross event simulator. I recommend saving a copy of the acceleration event simulator at this point because it will come in handy later on.

Lateral weight transfer occurs when a vehicle experiences a centripetal acceleration towards the center of a turn. This type of weight transfer can only occur for a vehicle that has two tracks (i.e. not a bicycle). This transfer of normal load is also a result of a force being applied to an object at the ground, below its CG. By doing a moment balance about one side of the vehicle, this load transfer can be described:

$$WT_{lat} = ma_{lat} * \frac{CG_Z}{(tw)}$$

It should be noted that vehicles can have different track widths for the front and rear axles. In that case, later weight transfer must be calculated separately for each axle. In our example, we will assume equal track widths: $tw = tw_f = tw_r$

We will use a simple law of circular motion to calculate lateral acceleration:

$$F_{centripetal} = ma_{centripetal} = m * \frac{v^2}{R_c}$$

$$a_{lat} = a_c = \frac{v^2}{R_c}$$

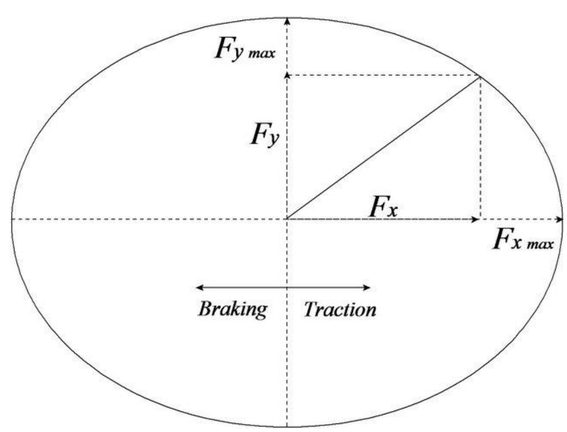
Tires have what we call a "traction ellipse". A tire cannot produce its maximum longitudinal and lateral forces simultaneously (otherwise it would be referred to as a traction rectangle). Therefore, for a given lateral tire force, there is a finite amount of grip remaining for longitudinal force. Race tires usually have higher peak lateral capability than peak longitudinal capability. Therefore, we need to be able to calculate the available longitudinal force for a given lateral force. In our ellipse calculations, we will refer to longitudinal force as F_x and lateral force as F_y.

$$\frac{F_x^2}{F_{x,\max}^2} + \frac{F_y^2}{F_{y,\max}^2} = 1$$

Additional inputs:

- Track width (tw)
- Estimate of maximum lateral coefficient of friction (μ_{lat})
- Corner radius: (R_c)

[lap simulator introductory guide is unfinished and will be expanded upon. To be continued…]



Example Case: WR-217e Architecture

For the Formula SAE Electric competition, vehicles are restricted to drawing a maximum of 80 kW from the tractive battery. This rule is actually very useful in helping constrain our architecture design. First, let us run a two-track (four-wheeled) acceleration event simulator given the following constraints:

1. The FSAE Electric Acceleration Event is 75m long
2. The maximum usable power is 80 kW
3. The vehicle cannot propel itself beyond the capabilities of its tires.

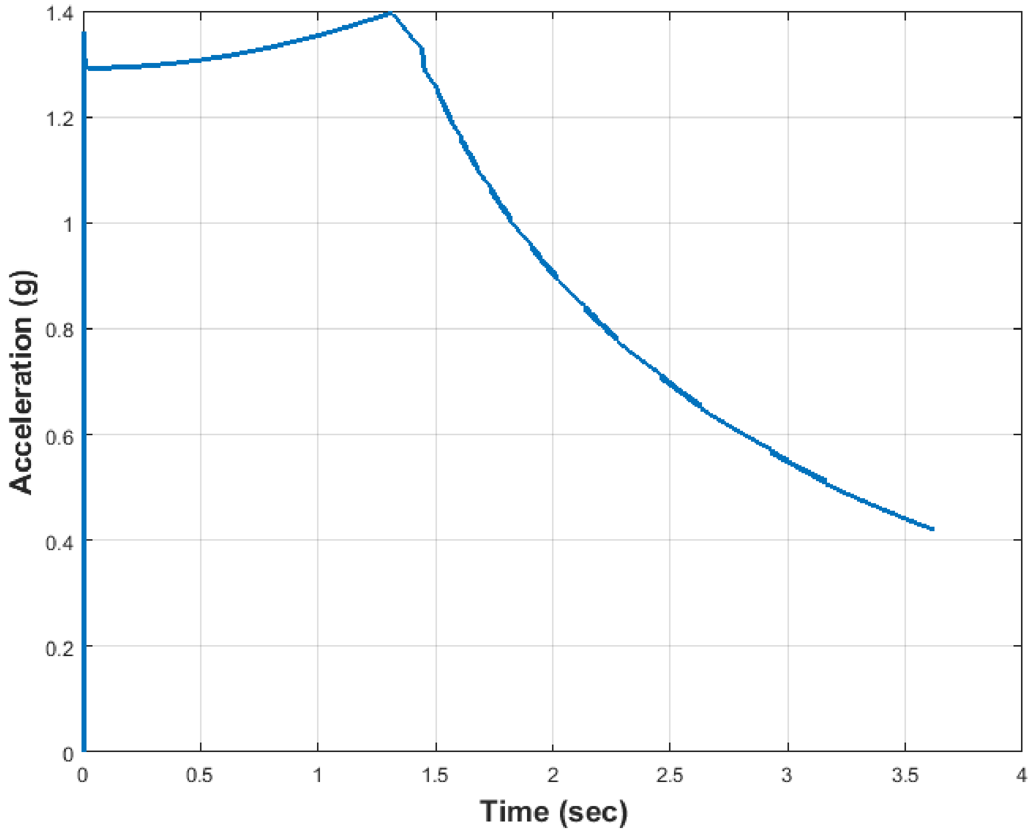
In addition to these three constraints, we will need to determine some initial values for our vehicle model. Luckily for us, we have data from our combustion car (cCar) to use as a jumping off point.

Parameter	Variable	Value [units]	Reasoning
Total mass of vehicle w/ driver	m	320 [kg]	(cCar mass)*1.3 + heavy driver, conservative
Center of gravity height	CG_Z	0.25 [m]	(cCar CG height)*0.85
Vehicle wheelbase	wb	1.6 [m]	Same as cCar
Static weight distribution	swd	0.55 [-]	2% more rearward than cCar
Estimate of tire rolling radius	r_{tire}	0.22 [m]	Same as cCar (at std. pressure & load)
Estimate of maximum longitudinal coefficient of friction	μ_{long}	1.4 [-]	Correlated to cCar track data
Estimate of maximum lateral coefficient of friction	μ_{lat}	1.7 [-]	Correlated to cCar track data
Coefficient of downforce	C_{df}	3 [-]	Same as cCar
Coefficient of drag	C_d	1.5 [-]	Same as cCar
Frontal area	A_f	1.21 [m^2]	Same as cCar
Center of pressure	cp	0.55 [-]	Same as cCar
Air density	ρ	1.15 [kg/m^3]	In Lincoln, NE (where comp. is held)

We now have enough information to get this car model rolling! But first, some notes:

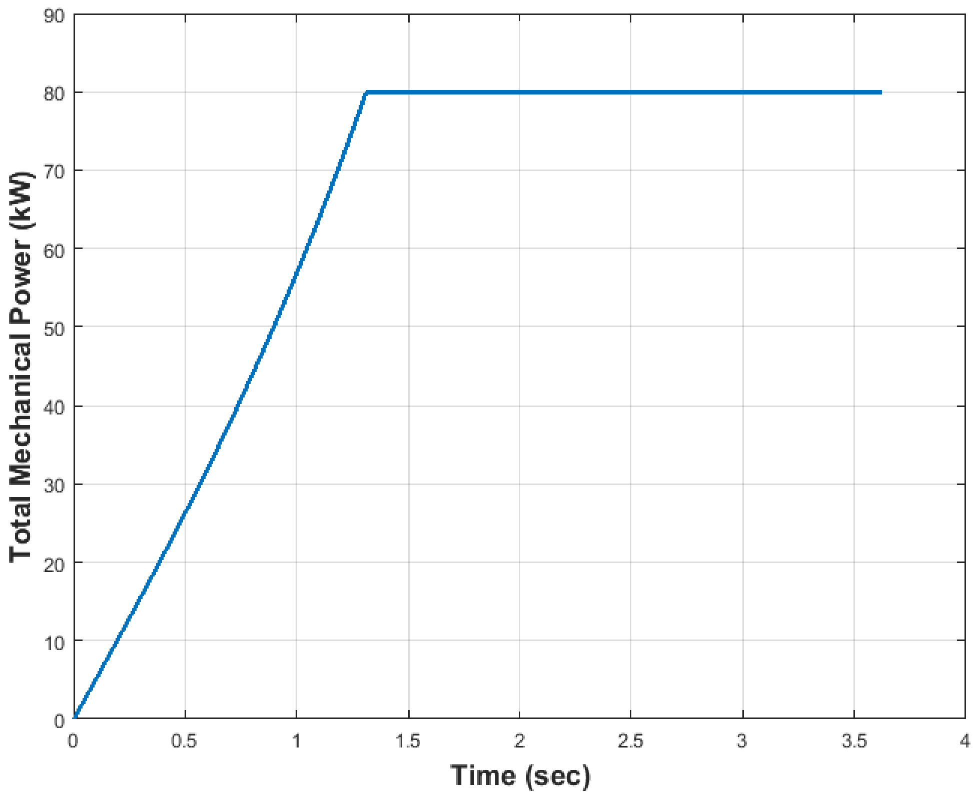
1. We will be using the metric of 0 – 75m time to gauge the vehicle's performance.
2. We will keep the aerodynamics package at the standard angle of attack: this is more conservative than trimming out wings since our goal is properly sizing our propulsion system. This is because the additional downforce will result in more available grip that the propulsion system will have to utilize.
3. When the amount of total available tractive power ($\sum F_{x,tires} * v_{vehicle}$) exceeds 80 kW, force will be removed from each contact patch proportional to where the grip is available. At the power limit, force can be removed from the contact patches in any combination- doing so in this particular way will ensure that neither the front nor the rear propulsion systems will have to work disproportionately hard. This mechanical 80 kW power limit will be an approximation of the rules-imposed 80 kW electrical power limit (since we have no way of quantifying electrical system losses yet). In the real world, these electrical losses would result in some percentage (80-95%) of the electrical power to be converted to mechanical power at the wheels of the vehicle.

Accel @ 80 kW Plots



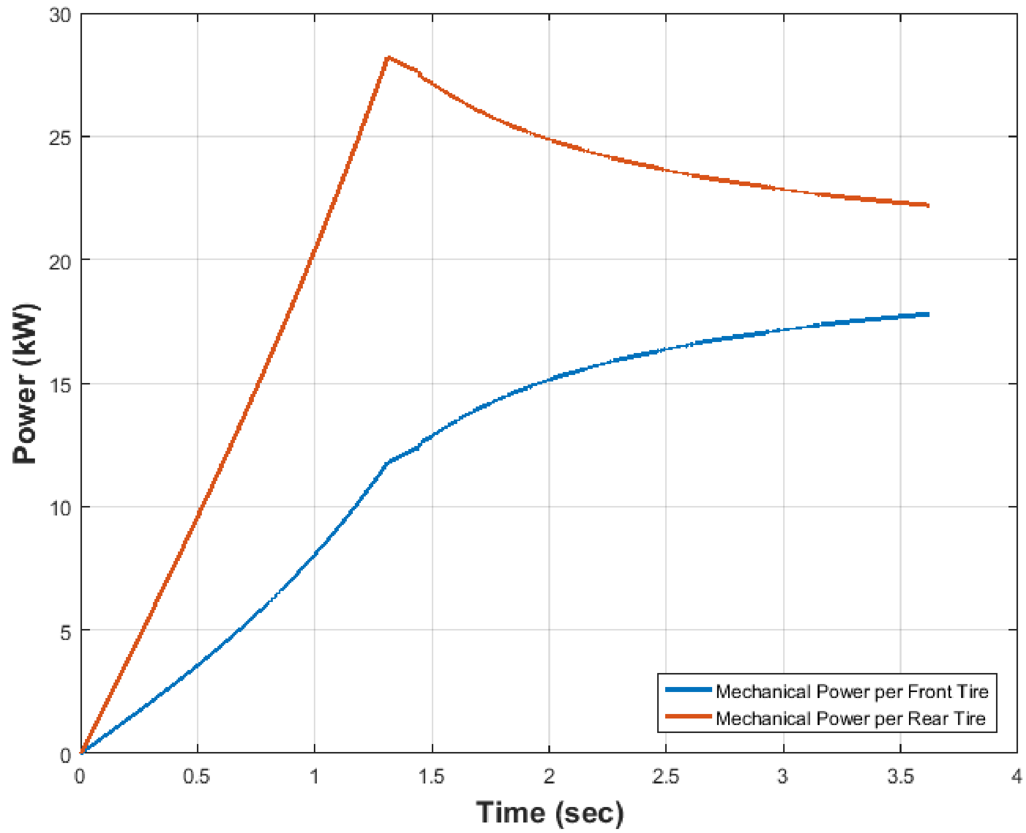
Plot 1: Acceleration

This plot is useful mostly for sanity checking the simulation. Initial acceleration appears to be around 1.3G, which is to be expected with $\mu = 1.4$ and some tire load sensitivity built into the model. Acceleration increases quadratically in the first second or so, which is to be expected due to increasing tire capability (as a function of downforce, which itself is a function of v^2). At around $t = 1.35\ sec$, the mechanical power limit kicks in and acceleration starts to fall off.



Plot 2: Total Mechanical Power

This plot is also mostly useful for sanity checking the simulation. Power starts at zero (which makes sense given $v = 0$), and increases slightly more than linearly (which makes sense given slightly increasing available traction due to downforce), and saturates at $P_{mech} = 80\ [kW]$ as expected.



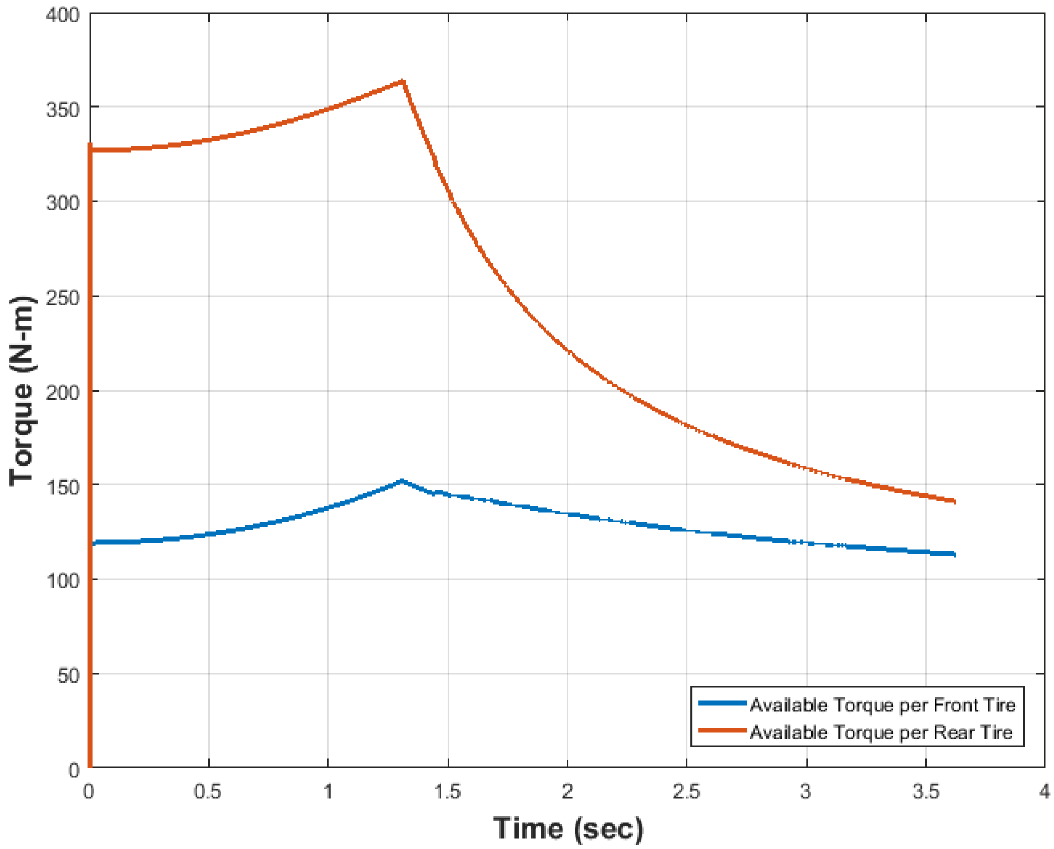
Plot 3: Individual Tire Powers

This plot gives us a good idea of the peak motor power our propulsion system should be capable of producing for both the front and rear wheels. From the plots, we can generally say:

$$P_{peak,F} = 18\ [kW]$$

$$P_{peak,R} = 28\ [kW]$$

This is true if each wheel is individually driven by a motor. If, for example, one wanted to implement a single inboard rear motor, the peak power requirement would be $2 * P_{peak,R}$.



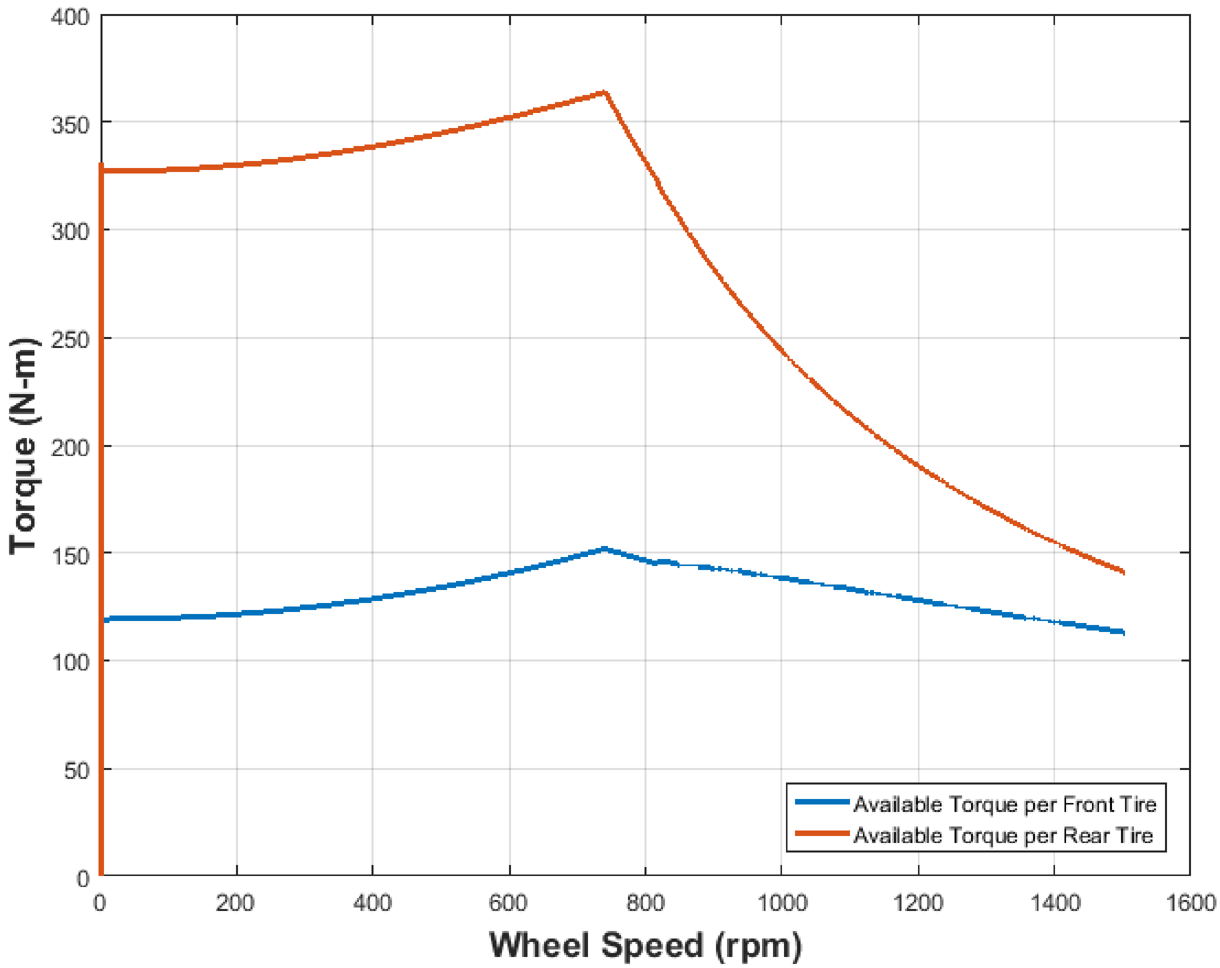
Plot 4: Individual Tire Torques

This plot gives us a good idea of the peak tire torque (post-transmission) our propulsion system should be capable of achieving for both the front and rear wheels. From the plots, we can generally say:

$$T_{peak,F} = 150\ [Nm]$$

$$T_{peak,R} = 360\ [Nm]$$

This is true if each wheel is individually driven by a motor. If, for example, one wanted to implement a single inboard rear motor, the peak torque requirement would be $2 * T_{peak,R}$.



<u>Plot 5: Torque-Speed Curves</u>

This plot gives us an idea of the torque-speed relationship we want for our electric motors.

Justifying All-Wheel Drive Architecture

The heaviest component in an electric vehicle is the high voltage tractive battery. As such, the electric motors and inverters account for a significantly smaller portion of the vehicle mass. Upon inspecting the above plots, it is clear that having tractive force at the front tires (contact patches) is preferable from an architecture standpoint, as it allows for ~30% higher longitudinal acceleration capability for a small increase in mass. We can ground this idea in first principles:

$$F = ma$$

In a simplified sense, as long as the addition of a front wheel propulsion system increases the accelerating force on the vehicle F more than it increases m, the net effect will be positive on longitudinal vehicle performance.

Electric Motor Selection

In addition to hitting our peak torque and power specifications, the *shape* of the torque-speed curve is important in the motor selection process. The general shape of the desired torque curves for our motors include a relatively flat peak torque region followed by a downward sloping relatively constant power region (due to the nature of the overall power limit on the car). Motors that have this characteristic have what is called *flux-weakening* capability. Ideally, we would like motors with this capability.

That being said, there are many other factors that can influence the selection of a propulsion system, chiefly packaging constraints and other electrical rules (such as maximum tractive voltage). During our selection process we must meet a series of requirements before even considering performance targets.

1) Motors and motor controllers must obey all FSAE rules including tractive voltage limit.
2) Motors, motor controllers, and required transmissions must be able to package inside the geometric constraints of the vehicle.
3) Motors and motor controllers must have a lead time that is compatible with the project timeline.
4) Motors and motor controllers must be affordable by the team.

If a propulsion system meets all of these requirements, we can move onto achieving performance targets, such as peak power, peak torque, and torque curve shape.

Case Study: WR-217e Motor Selection

Plettenberg Nova 15 Specifications (Front in-hub motors):

- Peak torque: 28 [Nm]
- Peak power: 20 [kW]
- Linear peak power curve, constant peak torque curve (no flux-weakening)
- Required gear reduction to achieve target tire torque: 5.3:1
- Mass: 3 [kg]
- Water jacket

Plettenberg Nova 30 Specifications (Rear inboard motors):

- Peak torque: 61 [Nm]
- Peak power: 30 [kW]
- Linear peak power curve, constant peak torque curve (no flux-weakening)
- Required gear reduction to achieve target tire torque: 5.9:1
- Mass: 6 [kg]
- Water jacket

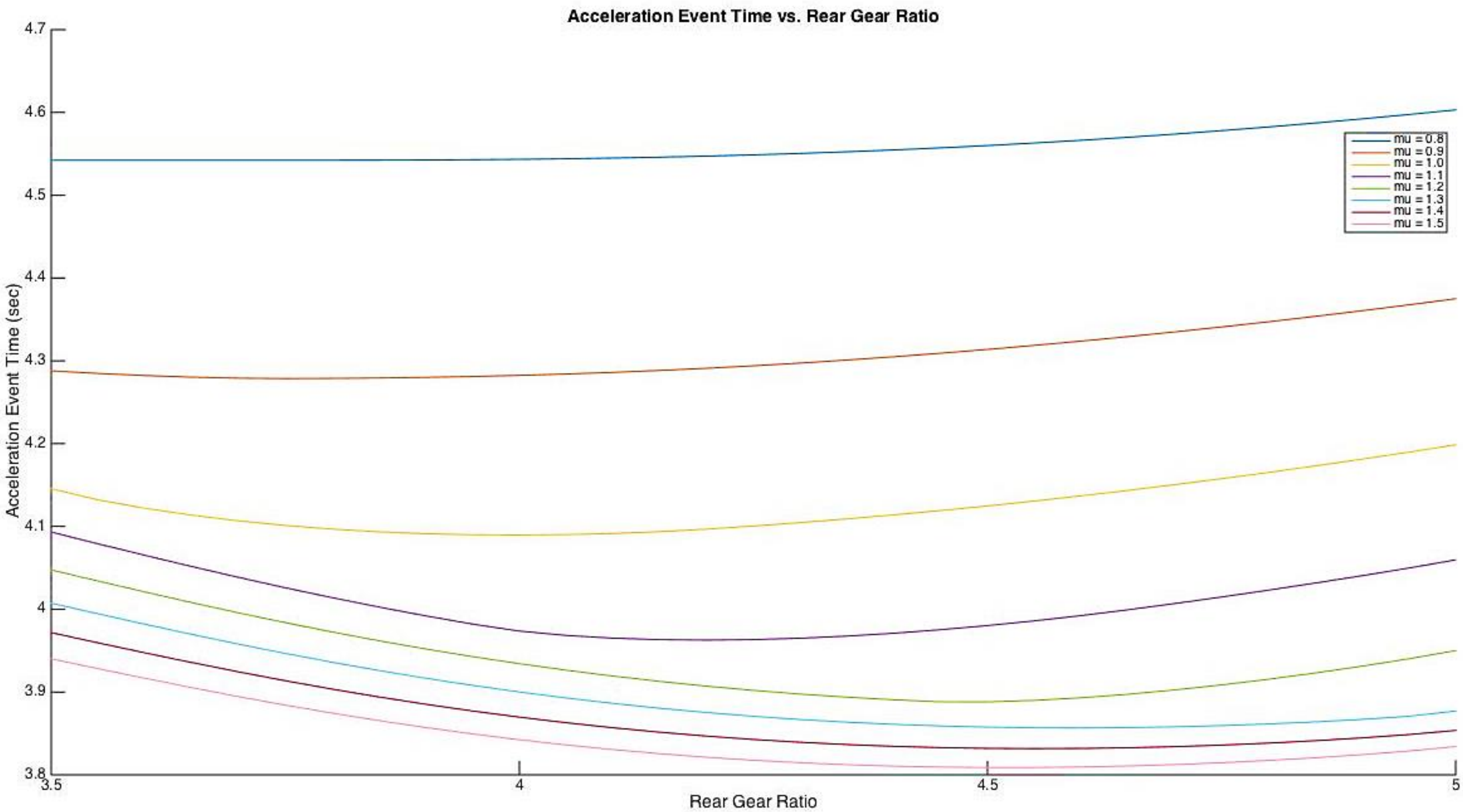
Parameter Optimization

No matter what range of values are swept for vehicle mass, acceleration event time will be minimized as mass gets smaller. On the other hand, the gear reduction ratio from a motor shaft to the tire will have a global minimum in acceleration time. This is because as gear reduction increases, the top speed of the vehicle decreases, and as gear reduction decreases, peak acceleration decreases. Because high acceleration and high top speed are both desirable, there is a tradeoff involved in determining the optimal gear ratio. Vehicle mass and gear reduction are examples of two kinds of parameters- parameters that produce a global extrema of the output metric at some value within their defined domain, and parameters that do so at a boundary of the domain. It is useful to sweep both types of parameters around a nominal value in order to determine the sensitivity of acceleration event time to that parameter. For example, by sweeping vehicle mass -20 kg to + 20 kg of the nominal value, we can determine the change in acceleration event time per unit mass ($\frac{dt}{dm}$). Besides sensitivity analysis, parameters with corner solutions are less interesting from an architecture optimization standpoint.

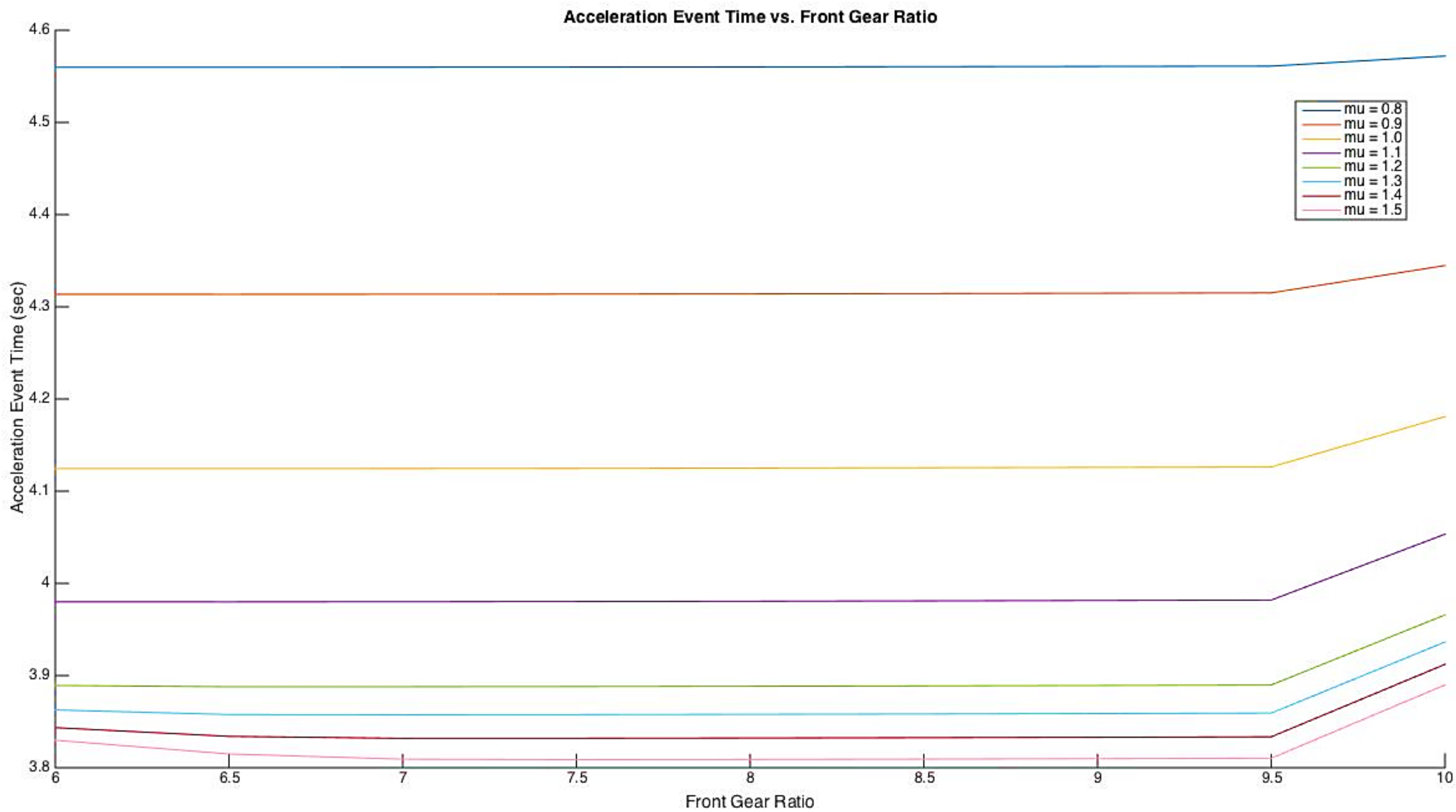
Gear Ratio Optimization

Now that we have selected the electric motors for our case study, we must determine the optimal gear ratios to minimize acceleration event time. Previous lap time simulations have shown that optimizing gear ratios for an acceleration event are sufficient for optimal performance in autocross events as well. We will use an acceleration event due to less computational overhead.

It should be noted that the optimal gear ratios are not necessarily just the ratio required to achieve the target peak tire torque. This is because that particular ratio may reduce the top speed of the car in such a way that the overall acceleration event time goes up as a result.



First, we will sweep the rear gear ratio. It is imperative that we perform this analysis for varying coefficients of friction- it is obvious from the result that the coefficient of friction changes the optimal ratio drastically. This is because at low coefficients of friction, the car's ability to accelerate is limited by traction and the fastest acceleration event time will be achieved by increasing top speed (lower gear ratio). On the other hand, at higher coefficients of friction the car has more grip it can utilize so the optimal gear ratio will trend higher. As the coefficient of friction increases, the optimal ratio shifts from 3.5:1 to 4.5:1. Since higher coefficients of friction better represent the track surface at competition in Lincoln, NE, a rear gear ratio of 4.5:1 was selected.



Next, we sweep the front gear ratio. This plot looks different from the previous plot due to the fact that the front motors are significantly traction limited. Therefore, in the range of 6:1 to 9.5:1, lap time simulation shows that there is no significant difference in performance. For a given output torque, the motor itself will be more efficient with a higher gear ratio (since more torque is a result of the speed reduction instead of phase current). That being said, packaging constraints limit our gear reduction to a maximum of 6:1 in a single stage planetary gear set. Since packaging constraints must take priority (and a compound gear reduction is a challenge to be tackled in a future year), a 6:1 reduction was selected.

Accumulator Energy Determination

Next, a full endurance lap simulator is utilized in order to determine the required accumulator energy to finish our 22 km race. Using some relatively basic loss models for the electric motors/motor controllers and the battery, we can include the effect of electrical losses in our analysis.

The electric motors have losses that are loosely categorized as torque-based losses (joule losses, I^2R) and speed-based losses (hysteresis losses, eddy current losses). Torque-based losses are easily accounted for, and a linear speed-based loss model was developed for the motors in absence of dyno data.

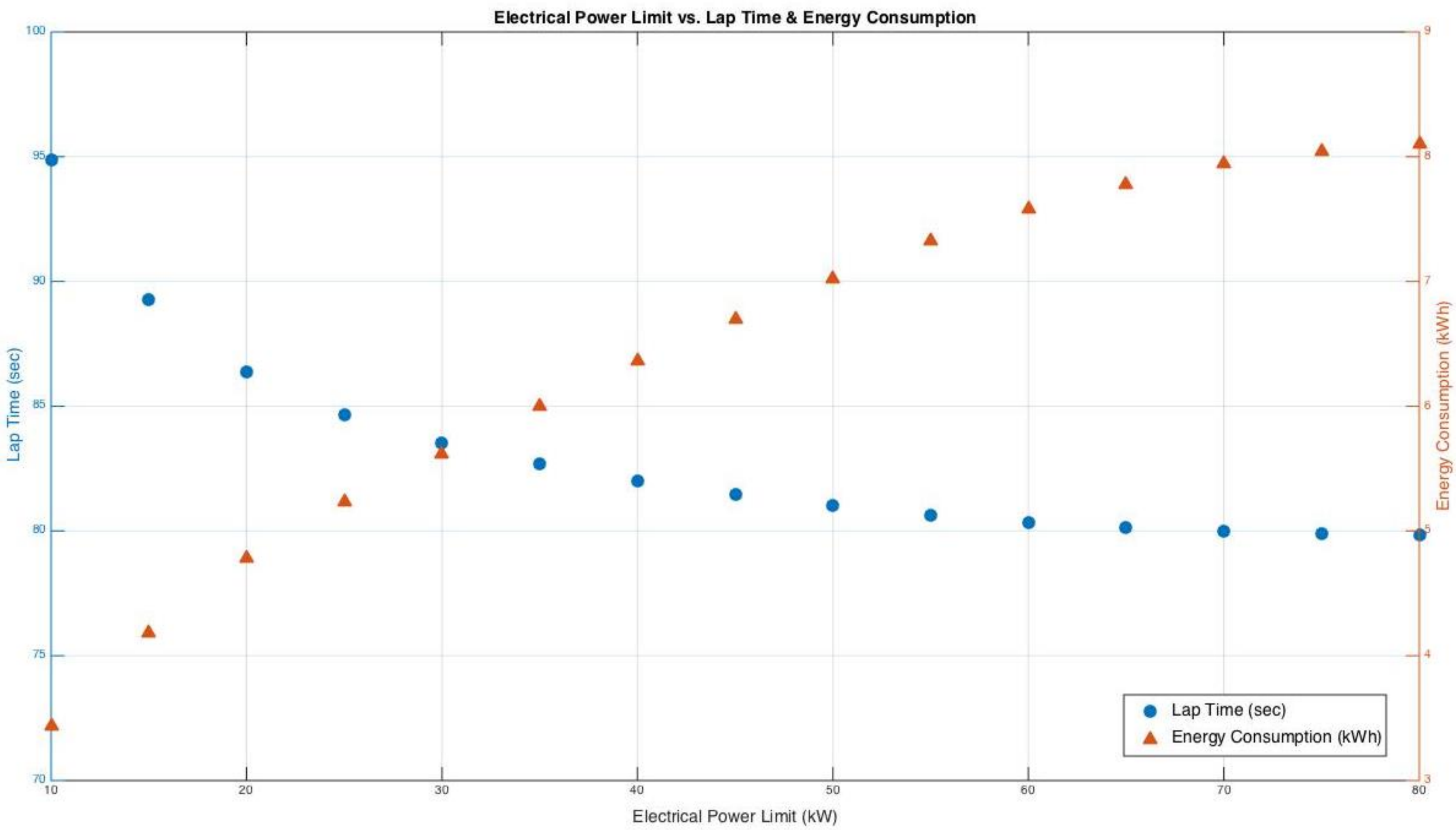
$$Loss_{motor} = I_{phase}^2 * R_{phase} + V_{backEMF} * I_o \ (where\ I_o = no\ load\ current)$$

The battery losses are tracked based on the battery's internal resistance. The losses from the battery are both electrical and chemical in nature, but can be roughly modeled as joule losses based on

$$Loss_{acc}[W] = I_{acc,instantaneous}^2 * R_{acc,nominal}$$

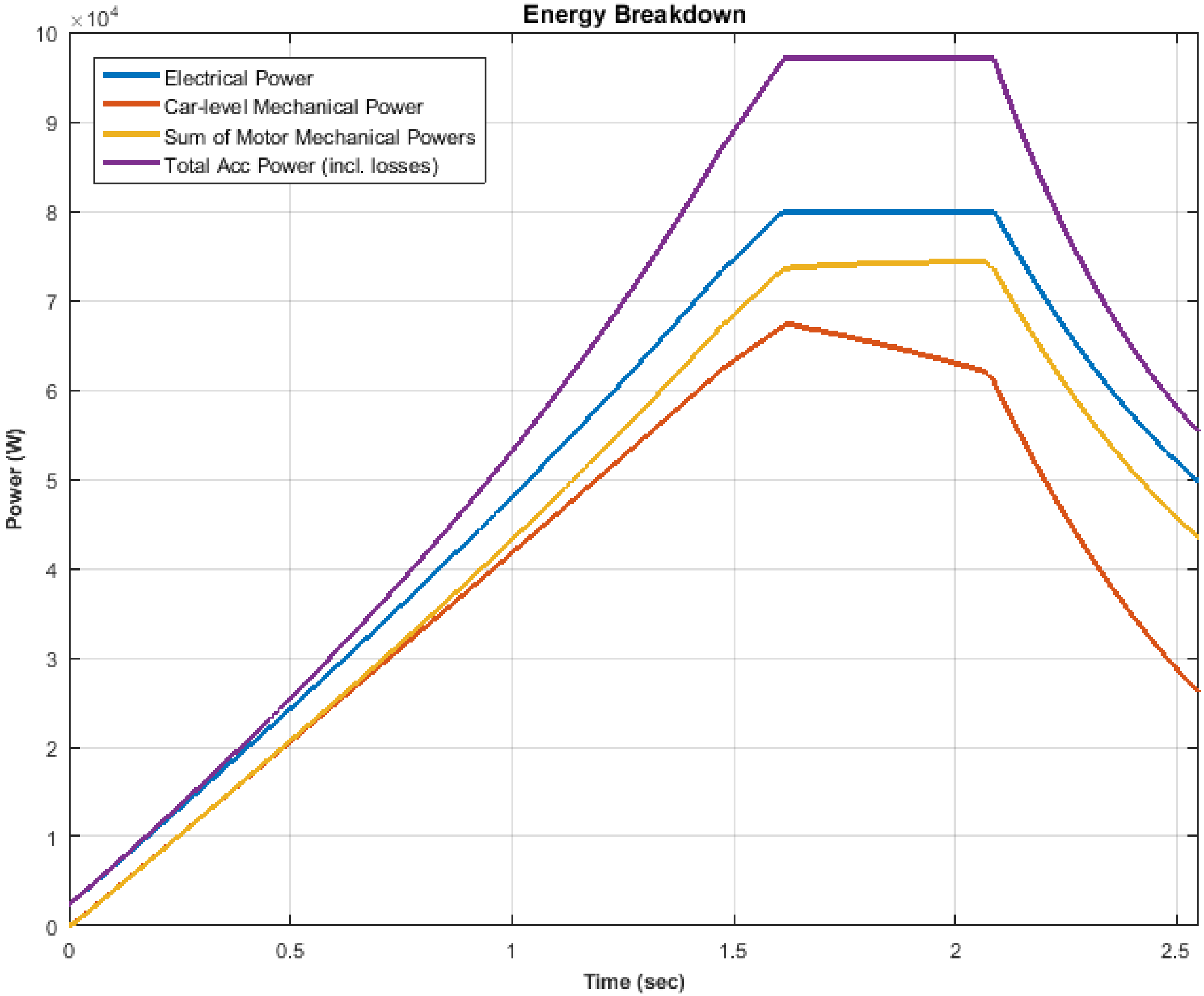
The tricky part of this is that the accumulator losses depend on the resistance of the accumulator, which depends on the number of cells in the accumulator. Since we are trying to determine the number of cells to implement, an educated guess for accumulator resistance must be made at the beginning of optimization.

One tool we will be implementing in this analysis is the idea of a software power limit. By rules, the vehicle may not draw more than 80 [kW] from the accumulator. It is, however, within the ability of the team to voluntarily lower the power limit on the vehicle in the control software. This has the potential to significantly decrease energy consumption due to the fact that losses go up quadratically (or more!) at higher power draws. A sweep of software power limit vs. endurance lap time and energy consumption was performed.



It can be seen from this sweep that as the power limit is reduced from 80 [kW] to 40 [kW], lap time increases from 80 [sec] to 82 [sec], or a 2.5% increase. At the same time, energy consumption goes from 8.1 [kWh] to 6.5 [kWh], or a ***20% decrease!*** For the endurance event where the main goal is to finish at all, this tradeoff between performance and probability of event completion is well worth it. From the plot above, it can be seen that the lap time and energy consumption curves begin to change slope significantly below 40 [kW], and so the endurance software power limit we selected was 40 [kW]. At this power level, the required energy is **6.5 [kWh].** Our next question becomes: why is there such a large drop in energy consumption between 80 [kW] and 40 [kW]?

For this, we will have to dig a bit deeper into the simulation to determine where the energy is going. Simulation shows that the majority of accumulator losses occur in straightaways at peak accumulator power draw.



The above plot was generated from an acceleration event. From t = 0 to t = 1.6, power increases more or less linearly as the car accelerates at a relatively constant rate. From t = 1.6 to t = 2.1, the power limiter saturates the usable electrical power at 80 [kW]. After t = 2.1, the car approaches top speed and power decreases once more. The purple line represents the amount of power produced by the cells in the accumulator, which is a combination of the usable electrical power measured by the energy meter and the power lost to heat. It can be seen that the purple line diverges quickly from the blue (usable electrical power at the battery terminals) line when the usable electrical power exceeds 40 [kW]. The delta between the purple and blue lines is the amount of power lost to heat in the battery, and at peak power the battery is ~80% efficient with losses close to 20 [kW]. The delta between the blue and yellow lines is the power lost to electrical losses in the motors/motor controllers. The yellow line is the mechanical power produced by the vehicle. The delta between the yellow and red lines is the power lost to drag. The takeaways from this plot are a) the battery is very inefficient at high powers (in upcoming years it would be worth developing **a lower internal resistance battery**), and b) the aggressive aerodynamics package requires a significant amount of power to overcome drag at high speeds, in excess of 10 [kW], indicating it would be worth designing **an active aerodynamic drag reduction system** in future years. This is especially true given that the energy density of a lithium battery is an order of magnitude worse than that of a chemical racing fuel, and as a result the weight savings achievable by implementing a smaller battery due to the reduction in energy consumption could be substantial.

Optimal Split: Maximizing Efficiency

You may have noticed the delta between the blue and yellow lines decreasing as time progresses at the 80 [kW] power limit. This is because the simulation employs an optimal splitting strategy once the electrical power limit is engaged in order to maximize the amount of those 80 [kW] being converted into mechanical power (or, alternatively, minimize the amount of electrical losses). This is accomplished by minimizing the torque-based joule losses in the system. Because the front motors are different than the rears, the [W] of loss per [N] at the contact patch is different for each motor.

$$P_{elec,total} = \sum F_{cp} * v + \sum (joule\ losses) + \sum (speed\ losses)$$

At the electrical power limit, the total electrical power is a constant, and the control system has no way of directly controlling the speed-based losses. Therefore, we focus on the joule losses. We want to find a way to split the tractive force between the front motors and rear motors to minimize these losses.

$$Joule\ losses = 2(I^2R)_{front} + 2(I^2R)_{rear}$$

We will now rewrite the joule losses equation in terms of contact patch forces.

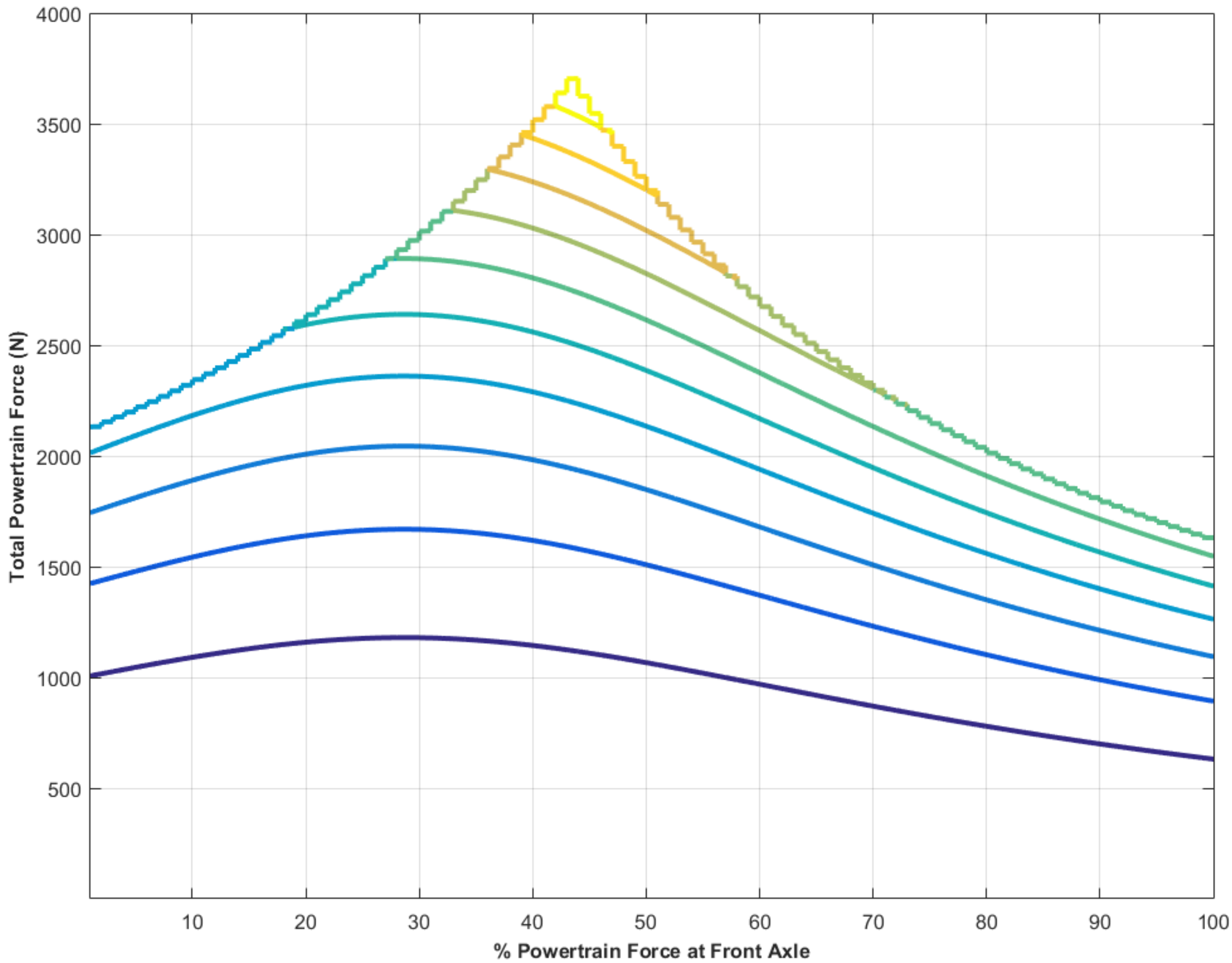
$$Joule\ losses = 2\left(\frac{r_{tire} * F_{cp,front}}{N_{front} * K_{t,front}}\right)^2 * \Omega_{front} + 2\left(\frac{r_{tire} * F_{cp,rear}}{N_{rear} * K_{t,rear}}\right)^2 * \Omega_{rear}$$

where K_t $is\ the\ motor\ torque\ constant\ in \frac{Nm}{A}$

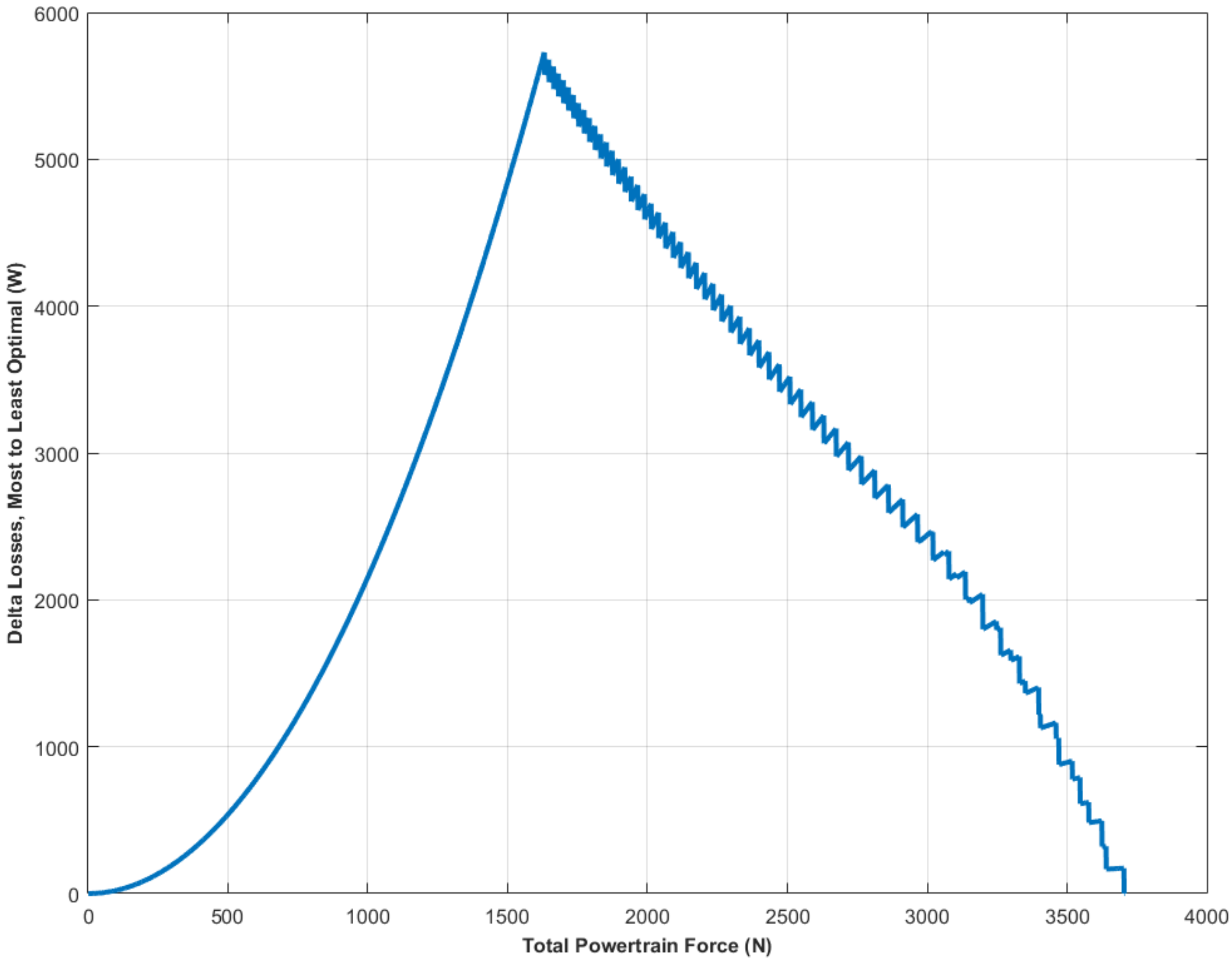
Note that the dimension of the product inside the parentheses is equal to amps. To find where losses are minimized, we want to take the partial derivative of losses with respect to front contact patch force and rear contact patch force, and set both of them equal to zero. Since both are set equal to zero, we can set the partial derivatives equal to each other and solve for a force ratio.

$$\frac{F_{cp,front}}{F_{cp,rear}} = \frac{\left(\frac{1}{N_{rear} * K_{t,rear}}\right)^2 * \Omega_{rear}}{\left(\frac{1}{N_{front} * K_{t,front}}\right)^2 * \Omega_{front}}$$

A quick sanity check is to think about what would happen if the front motors and rear motors were the same. If this were the case, the optimal force ratio would be 1, or 50% of the force in the front and 50% of the force in the rear. This makes sense because it would be less efficient to have either the fronts or the rears do the majority of the work. In the case of our electric racecar, optimal split is around 25% front. This also intuitively makes sense- a smaller motor will be less efficient at producing a force at the contact patch than a larger one.



The above plot is one way to visualize optimal split. The contours are ***constant loss lines***. It can be seen that for constant losses, one can achieve higher total powertrain force by using a split of around 25% front. This is why all of the contours have a peak around that percentage. The peak in the middle is a result of the diminishing number of ways you can split tractive force as it increases. At the extreme, around 3600 [N], there is only one way to split the force: around 45% front. There is only one solution because this is the point where all motors are producing peak torque simultaneously. The lower the total powertrain force is, the more flexibility the control system has in terms of splitting it up.



This is a different way to visualize optimal split. For a given amount of tractive force requested by the driver, there is a finite power savings between the most and least optimal split percentages. Again, as force approaches 3600 [N], the number of split possibilities approaches zero and therefore the power savings between the best and worst split percentage also approaches zero. Around 1500 [N], the maximum power savings are possible, with a delta of over ***5.5 [kW]!***

Note: the jagged line is just an artifact of numerical calculation error. The true curve is smooth except for the peak.

Sensitivity Report

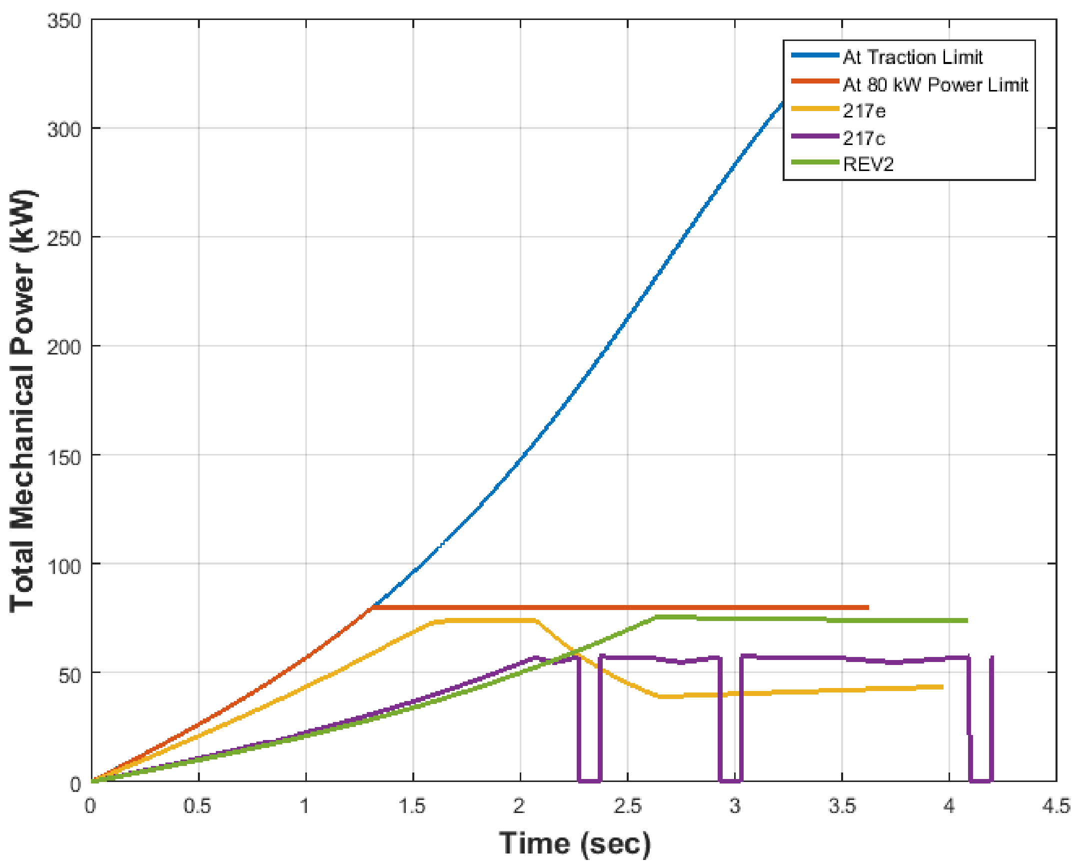
Variable	Nominal Value	Delta	Delta Accel Time (ms)	Sensitivity	Sensitivity Units
m	320 [kg]	-20 [kg]	-45.4	2.3	[ms/kg]
μ_{long}	1.4 [-]	+0.1 [-]	-18.7	-186.5	[ms]
CG_z	25 [cm]	-1 [cm]	-6.8	6.8	[ms/cm]
wb	160 [cm]	+10 [cm]	-9.9	-9.9	[ms/cm]
swd	55 [%]	-5 [%]	-41.0	8.2	[ms/%]
r_{tire}	22 [cm]	+1 [cm]	-35.2	-35.2	[ms/cm]
C_{df}	3 [-]	+0.1 [-]	-0.5	-5.5	[ms]
C_d	1.5 [-]	-0.1 [-]	-2.7	27.0	[ms]
A_f	1.21 [m^2]	-0.1 [m^2]	-1.9	19.2	[ms/m^2]
cp	55 [%]	-5 [%]	-2.1	4.2	[ms/%]
Ω_{cell}	23 [mOhm]	-5 [mOhm]	-35.8	7.2	[ms/mOhm]
$N_{cells,series}$	30 [cells]	+1 [cell]	-50.2	-50.2	[ms/cell]
$N_{cells,parallel}$	24 [cells]	+1 [cell]	-6.8	-6.8	[ms/cell]
$K_{t,front}$	0.1 [N-m/A]	-0.01 [N-m/A]	-6.6	662.9	[ms/(N-m/A)]
$K_{t,rear}$	0.217 [N-m/A]	-0.02 [N-m/A]	-56.8	2840.8	[ms/(N-m/A)]
$T_{max,front}$	28 [N-m]	+1 [N-m]	4.6	4.6	[ms/N-m]
$T_{max,rear}$	61 [N-m]	+2 [N-m]	-20.7	-2.1	[ms/N-m]
N_{front}	6 [-]	+0.5 [-]	10.6	21.2	[ms]
N_{rear}	4.5 [-]	-0.5 [-]	-61.9	123.8	[ms]

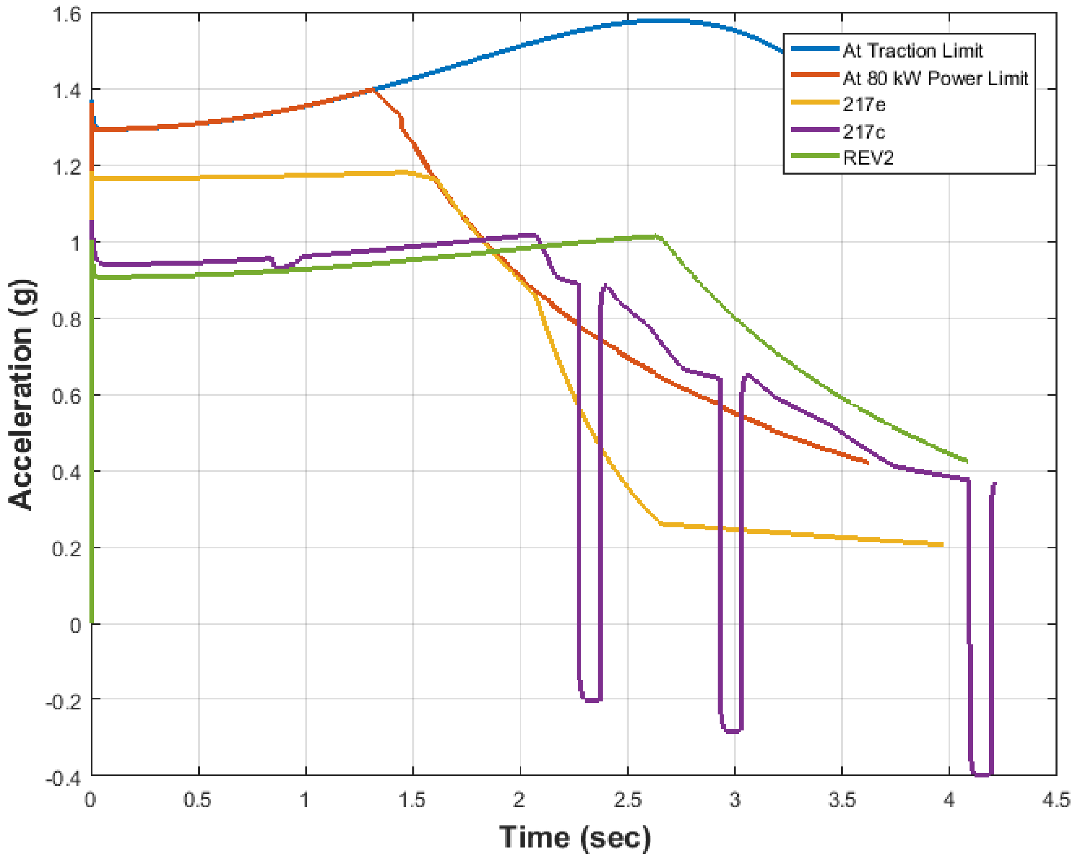
**highlighted in green are the areas in which the architecture has significant room for improvement. The car could and should certainly be lighter. The static weight distribution should be closer to 50/50 in order to better utilize the front powertrain. The rolling radius of the tire should be carefully identified in order to ensure it is correct due to the high sensitivity of the accel event to it. Lower internal resistance cells should be investigated in order to reduce losses at the power limit (and maximize mechanical power). A higher voltage should be investigated, in this case in order to increase the top speed of the car. The rear motors should either have a slightly lower gear ratio or lower Kt, as the rear motors are the limiting factor for the top speed of the car, and a slightly higher top speed would further reduce accel times.*

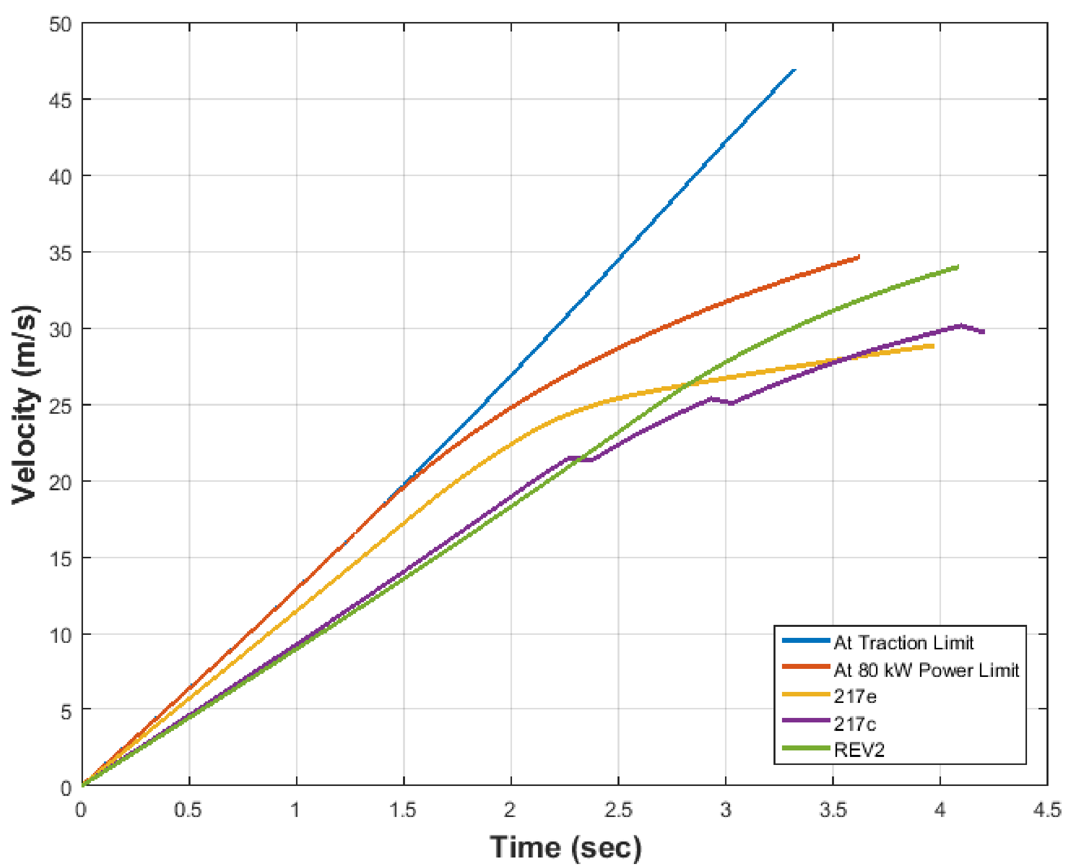
Overall Architecture Comparison

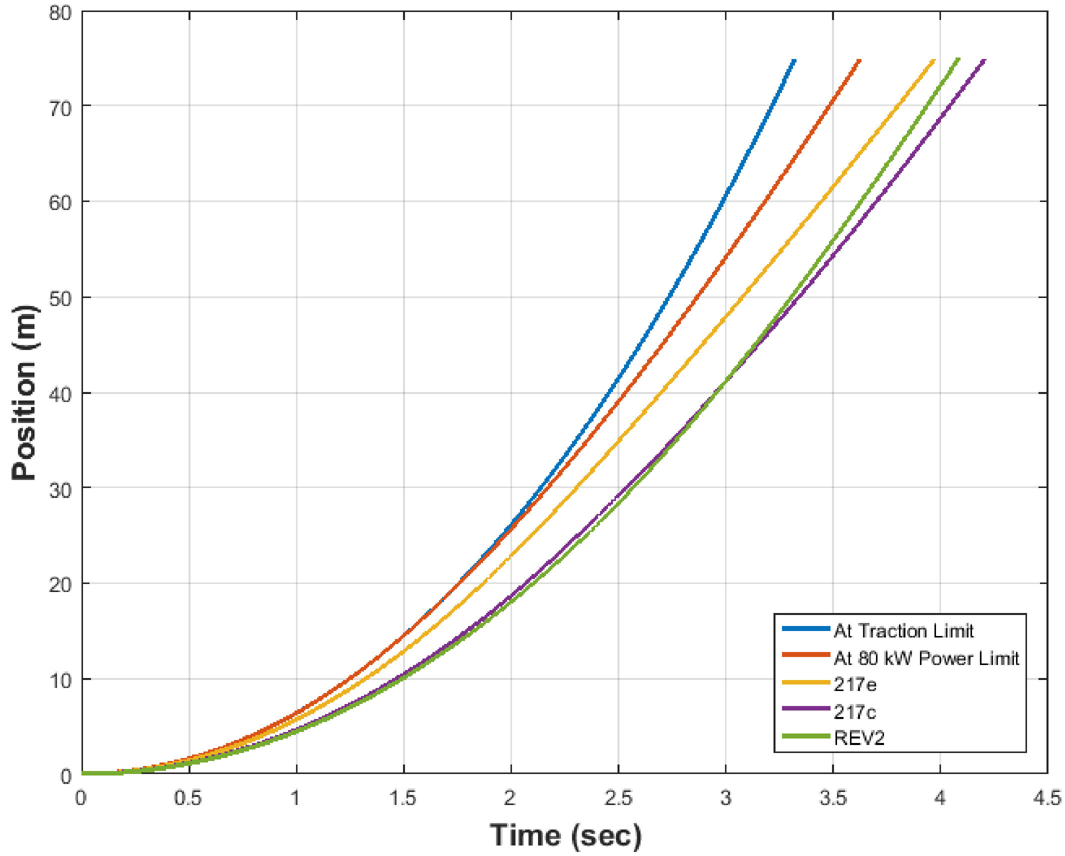
To wrap up, we should evaluate the architecture we settled on. We will benchmark our architecture in an acceleration event against: a) riding the traction limit to 75 m, b) riding the traction limit and then limiting to 80 kW mechanical, c) one of our competitor's architecture, and d) Wisconsin Racing's own combustion car.

For these simulations, our architecture will be benchmarked against a vehicle of the same mass for a) and b), and will use real car parameters for c) and d).




1.6
1.4
1.2
1
0.8
0.6
0.4
0.2
0
-0.2
-0.4
Acceleration (g)
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
Time (sec)
At Traction Limit
At 80 kW Power Limit
217e
217c
REV2


50
45
40
35
30
25
20
15
10
5
0
Velocity (m/s)
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
Time (sec)
At Traction Limit
At 80 kW Power Limit
217e
217c
REV2



Takeaways:

1) Delta between yellow and red is our room to improve in terms of powertrain architecture alone (chassis mass would shift blue, red, and yellow leftwards).
2) 217e succeeds at besting competitor UPenn Electric, whose car is significantly lighter but RWD only.
3) Wisconsin Racing's first ever electric car should be the quickest vehicle ever made by the university!

Model Validation: Reality vs. Expectation and Model Correction

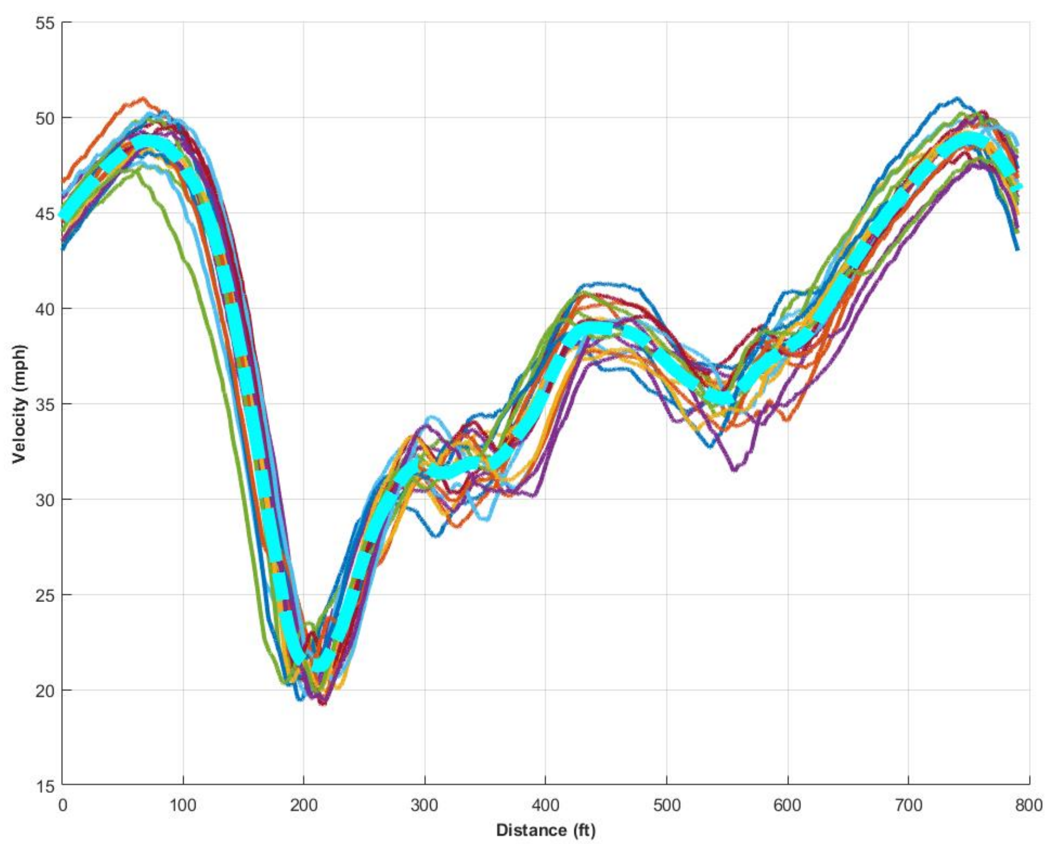
After dyno testing electric motors and motor controllers, we learned:

- Motor controller maximum continuous phase current: 200 A reality vs. 280 A expectation
- Rear motor torque constant: 0.175 Nm/A reality vs. 0.217 Nm/A expectation
- Transmission losses greater than expected: 15% reality vs. 5% expectation
- No regeneration capability from powertrain

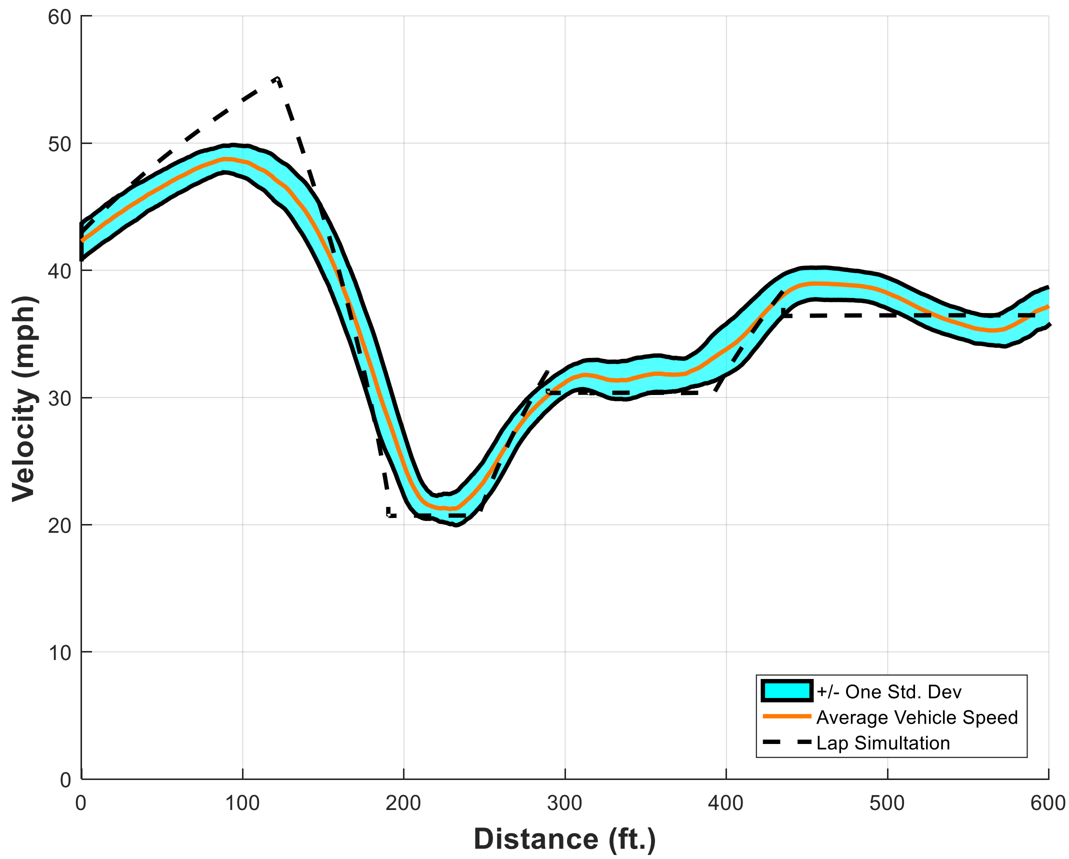
Effects on the vehicle:

- Accel times increased by 40%
- Peak tire torque decreased by 50%
- Autocross times increased by 5%
- Peak electrical power draw from battery decreased by 20% during accel and autocross (80 kW -> 65 kW)
- Endurance energy consumption had negligible change
    - 40 kW software power limit was selected for endurance to balance tradeoff of energy consumption and laptime. This limiter remains the dominating factor in endurance energy consumption- even with the loss in performance from expectation vs. reality, average accumulator power draw during endurance was mostly unaffected
    - Accumulator energy was designed for the contingency that regen would be nonfunctional. As such, the car still would have been able to finish an endurance.

Processing track data: Twenty laps processed, dashed line is the mean lap velocity

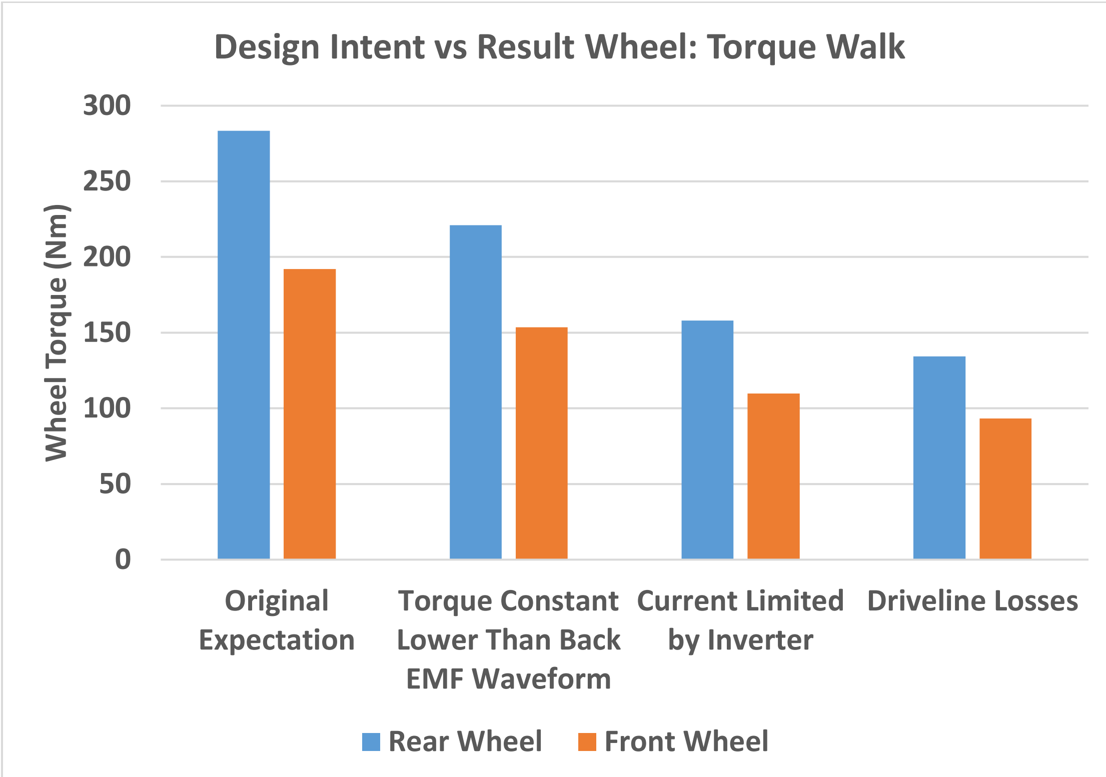


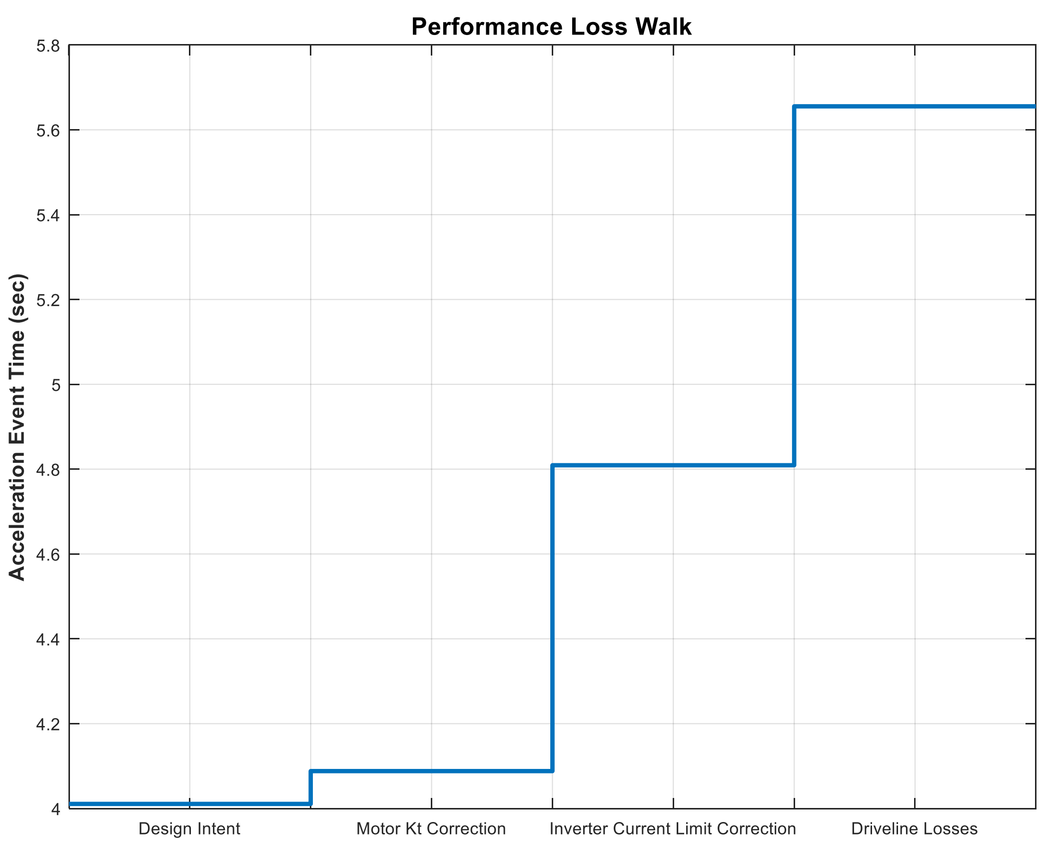
Statistical analysis:

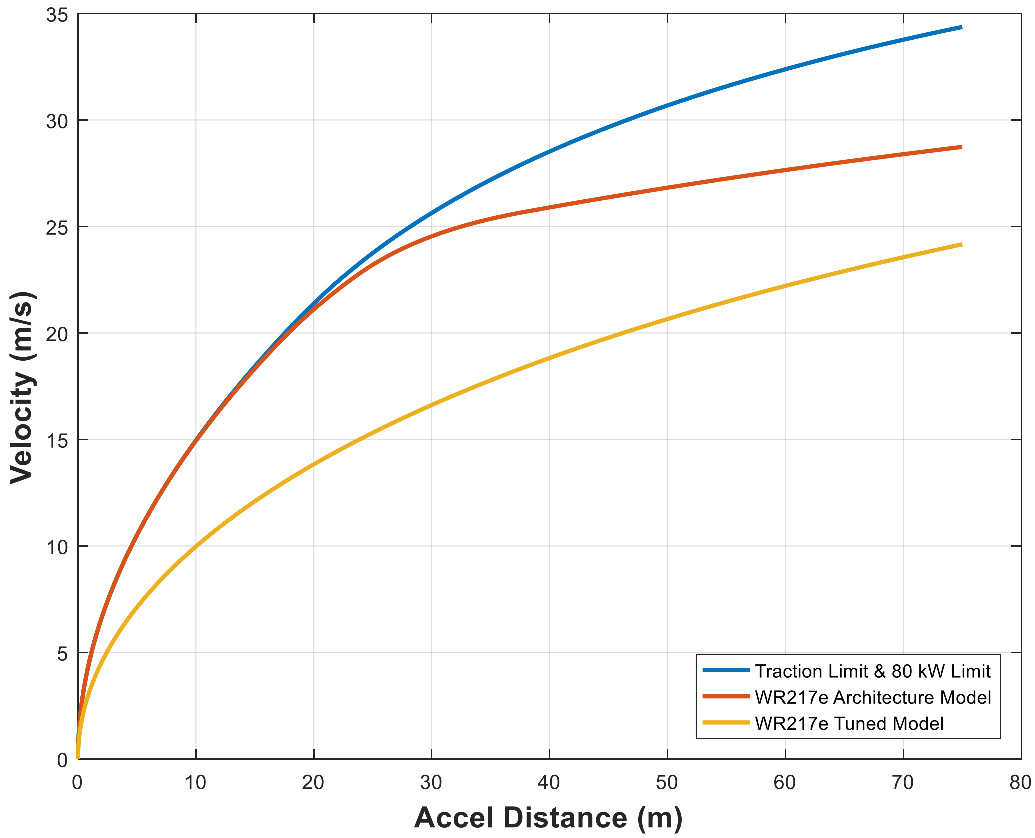


- Orange line is mean lap velocity from track data
- Blue region is the result of statistical analysis
- Dashed line is the LapSim velocity after tuning the model to track data

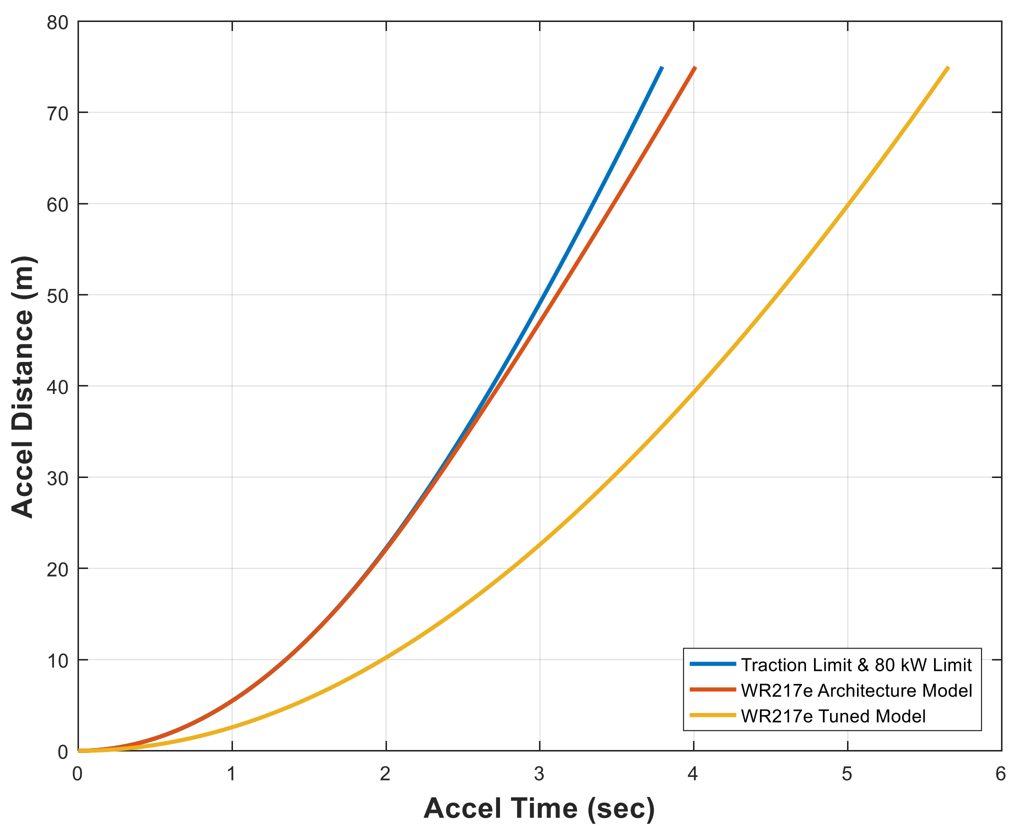
Lost Torque from Original Design Intent:



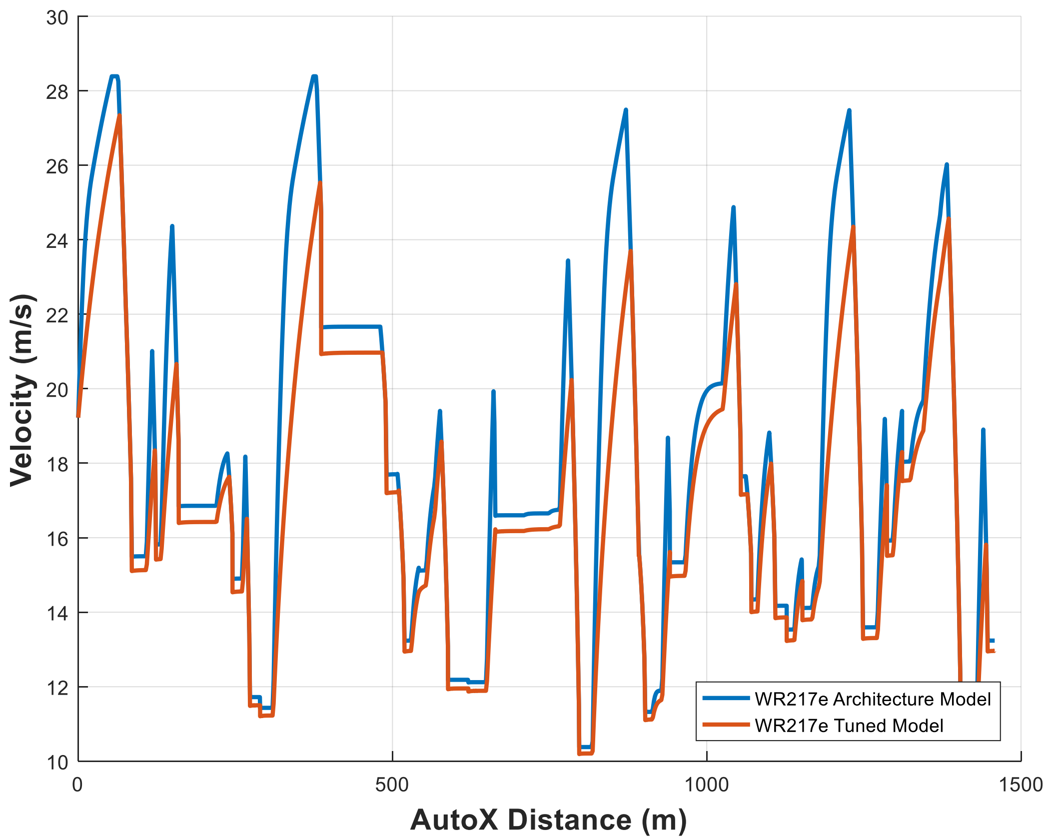


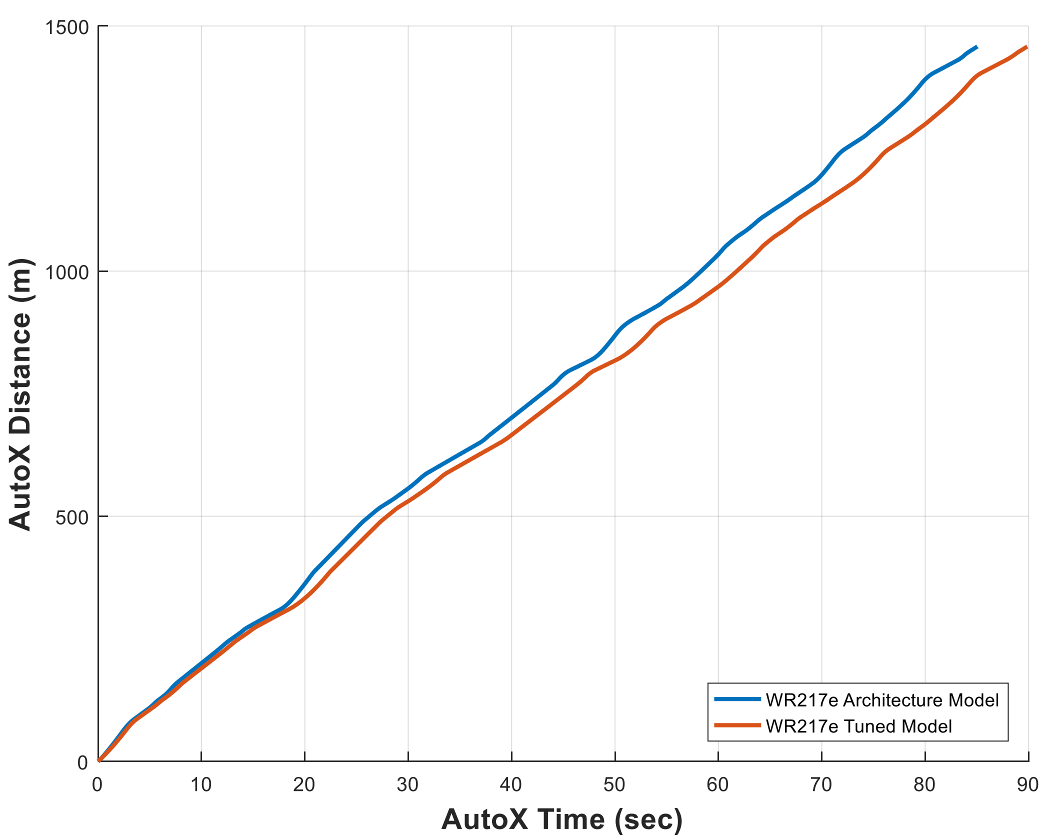


Velocity (m/s)
Accel Distance (m)
Traction Limit & 80 kW Limit
WR217e Architecture Model
WR217e Tuned Model
0
5
10
15
20
25
30
35
0
10
20
30
40
50
60
70
80



Accel Distance (m)
Accel Time (sec)
Traction Limit & 80 kW Limit
WR217e Architecture Model
WR217e Tuned Model
0
10
20
30
40
50
60
70
80
0
1
2
3
4
5
6


30
28
26
24
22
20
18
16
14
12
10
0
500
1000
1500
Velocity (m/s)
AutoX Distance (m)
WR217e Architecture Model
WR217e Tuned Model


1500
1000
500
0
0
10
20
30
40
50
60
70
80
90
AutoX Distance (m)
AutoX Time (sec)
WR217e Architecture Model
WR217e Tuned Model

Takeaways:

1) Be very skeptical of “peak” specifications. Peak can mean anything from “maximum continuous capability” to “highest possible value for 1 ms before damage”. Be fastidious and insistent- press the manufacturer to be as clear as possible about their peak specifications. You should always be able to make a plot of “pulse performance” vs. “pulse time”
    a. For example, a battery or inverter could supply 250 A for 1 sec, 200 A for 20 sec, 150 A for 20 min, and 100 A continuously, etc
2) Always ensure the correct safety measures are implemented- either by yourself or by the manufacturer. Lack of standard safety mechanisms such as overcurrent protection, overvoltage protection, reverse voltage protection, etc should be a red flag that the product has corners being cut. This is important to ensure the safety of your team members, but also important to ensure that in a fault condition your hardware does not become damaged.
3) If no data is available, be conservative in modeling loss mechanisms such that powertrain performance has a reasonable factor of safety (ie motor losses, inverter losses, transmission losses).
4) Take the “make vs. buy” decisions very seriously. We’re here to learn, and sometimes the best way to do that is to make things yourself. Get out there and push yourself to design new things and learn new concepts!

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