Dynamic performances simulation of an electric Formula Student vehicle

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Abstract. The development of a new model of a vehicle is a long and tedious process. To reduce the costs, simulation models can be used to validate design decisions. The complexity of the models which allow for the analysis of dynamic performances, impact the adaptability to this kind of studies. A MATLAB model has been used to simplify the process of simulating a Formula Student racing vehicle on a racing track, so the impact of changes done to the main parameters can be easily monitored. After implementing the model, the influence of different types of tires (slick tires and wet tires), on the dynamic performances of the vehicle, was analysed. This is done by comparing the acceleration limits diagram and speed profile calculated for both vehicle configurations.

Keywords: vehicle dynamics, Formula Student, dynamic performances

1. Introduction

One of the main goals in the development of a Formula Student electric vehicle is to obtain satisfactory dynamic performance, especially regarding the vehicle speed and time needed to finish a lap. To reduce the costs and time of developing any type of vehicle, computer simulations can be used to estimate dynamic performance before the vehicle is even built. In order to obtain a reduced time for each lap, several parameters are of interest, such as the tractive force, maximum speed and acceleration and braking capabilities of the vehicle. To evaluate the dynamic performances, it is necessary to take into consideration different constructive parameters of the vehicle such as the mass, center of mass position, minimum and maximum speed, the torque and power curves of the vehicle, and the tires used [1,2].

In recent years, different racing teams have developed their simulation tools that can estimate the vehicle's dynamic performances. These models are used for both internal combustion engines vehicles and electric vehicles to reduce the costs to test design decisions. The usage of multibody dynamics, chassis motion, suspension/sprung mass displacement during cornering and wheel spin dynamics increase the precision of the simulation bringing the results closer to reality. Domenighini et al. [3] implemented a full articulate-body model that has a more realistic dynamic approach than a baselined double track model by making no restrictive assumptions to check the dynamic behaviour of a vehicle on a racing track. Dal Bianco et al. [4] used a 14 degree-of-freedom (DOF) with full chassis motion to simulate the lap time of a GP2 vehicle. Imani Masouleh et al. [5] analysed the impact of multilink suspension on aerodynamic performances. The main disadvantage of these models is that they cannot to be used for other types of vehicles without substantial calibrations to the model. If a full multibody is used, adapting the model to another chassis is not an easy process making the multibody model not feasible for massive parametric sweeps, and 14-DOF multibody models are hard to generalize [3–6].

Simplifications can be done to reduce the complexity of the simulation models. The multibody simulation can be reduced to a point-mass quasi-steady state model. This simplifies the computational demand and the parametrization process. A quasi-steady state model would allow for the provision of valuable data about the main dynamic performances such as a speed profile on a racing track, but with a lower accuracy than a multibody model. Biniewicz et al [7] used experimental data gathered from a racing motorcycle to optimize the driver behaviour of a quasi-steady state model. Heilmeier et al. [8] implemented a quasi-state model for a hybrid racing car using publicly available data emphasising the limitations of the tire model and gear changes.

In this paper, a single mass point steady state simulation model has been implemented in MATLAB for an electric vehicle to analyse the impact of tires and road conditions on the performances of the vehicle.

2. Methodology

The first step was to define the vehicle to be simulated. In this case, a Formula Student vehicle equipped with two electric motors having 27 kW power and a maximum 60 Nm torque at 4200 rpm [9], connected to the wheels on the rear axle, was considered. The main parameters of the vehicle are presented in Table 1, and the electric motor torque/power curve is shown in Figure 1.

Table 1. The main input parameters of the considered vehicle.

Parameter	Value	Measuring unit
Total mass	320	kg
Front mass distribution	55	%
Wheelbase	1525	mm
Lift coefficient	-0.88	-
Drag coefficient	0.81	-
Frontal aerodynamic distribution	50	%
Frontal area	1.34	m^2
Tire type	182.88/41.667R13	-
Final gear efficiency	0.98	-
Final gear ratio	5	-

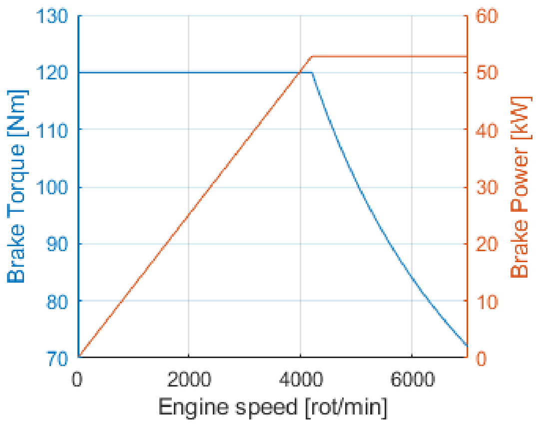


Figure 1. Engine map (torque/power curve) of the electric motors equipping the considered vehicle [9].

In the simulation model, after defining the main parameters, the powertrain energy transmission process is defined from the electric motors to the wheels. It was considered that both motors provide the same amount of torque. The tire's tractive force F_{Tr} was obtained using the relation [10]:

$$F_{Tr}=\frac{M_m \cdot i_t \cdot \eta_t}{r_r} \tag{1}$$

where: M_m is engine torque [Nm]; r_r – wheel rolling radius [m]; i_t – total transmission ratio; η_t – final gear efficiency. In this case, the engine torque takes into consideration both motors, so F_{Tr} will be the sum of both tires tractive forces.
After knowing the vehicle available tractive force, it is important to define the road load forces, which are the rolling resistance (2) - the rolling coefficient is considered constant for this application, the aerodynamic resistance (aerodynamic drag and lift, 3) and the road slope resistance (4). One important aspect is that the tire tractive force needs to be higher or at least equal to the road load forces. If the tire traction force is equal to the road load forces, then the maximum speed is reached for that type of road, and the vehicle can no longer accelerate. The relations needed used to calculate the road load forces are [6,10]:

$$R_r=f \cdot Z \tag{2}$$

$$R_a=0.5 \cdot \rho_{air} \cdot C_{l,d} \cdot A_f \cdot v^2 \tag{3}$$

$$R_p=m \cdot g \cdot \sin(\alpha) \tag{4}$$

where: R_r is the rolling resistance [N]; f – rolling coefficient [-]; Z – normal load on all wheels [N]; $C_{l,d}$ – lift and drag aerodynamic coefficients [-]; A_f – frontal area of the vehicle [m^2]; α – longitudinal inclination of the road [°].

In a Formula Student type competition, the main goal is to use the vehicle at its limits. In this case, the study of the vehicle at its grip limit is relevant for the maximum speed and the best theoretical time a driver can obtain. The maximum force applied on a tire is obtained when the tractive force is equal to the tire normal load multiplied with the grip factor. The maximum acceleration is solved along two perpendicular directions. The dependency between the longitudinal and lateral accelerations is given by [2,10,11]:

$$1=\frac{a_x^2}{a_{xmax}^2}+\frac{a_y^2}{a_{ymax}^2} \tag{5}$$

where: a_x is the acceleration component on longitudinal axis [m/s^2]; a_y – acceleration component on the lateral axis [m/s^2].

In the field of motorsport, the acceleration limits diagram is used to evaluate the grip limits of a vehicle, independent of the road profile. It can be used to find both acceleration and deceleration limits. To calculate the diagram, it is important to define the longitudinal and lateral limits for the acceleration and deceleration sequence. When accelerating, the maximum lateral acceleration (6) is given by the lateral grip coefficient and the normal load on the drive wheels (it also takes into consideration aerodynamic lift and downforce, respectively). The maximum longitudinal acceleration (7) is given by the longitudinal grip coefficient and the normal load on the drive wheels. For the maximum longitudinal deceleration (8), the normal load on all the vehicle wheels is considered. The following relations were used to calculate these accelerations and decelerations [2,10]:

$$a_{ymax}=\frac{\varphi_y\cdot(Z-R_{az})}{m} \tag{6}$$

$$a_{xmax\_ac}=\frac{\varphi_x\cdot Z_d}{m} \tag{7}$$

$$a_{xmax\_dec}=\frac{\varphi_x\cdot(Z-R_{az})}{m} \tag{8}$$

where: Z_d is the normal load on the driving wheels [N]; R_{az} – aerodynamic lift/downforce [N]; φ_x,φ_y – grip coefficient for longitudinal and lateral directions.

Considering that the lateral acceleration varies from 0 to a_{ymax}, the actual longitudinal acceleration and deceleration can be obtained [2,10]:

$$a_{x\_acc}=\min\left(a_x,\ a_{en\_lim}\right)-a_R \tag{9}$$

$$a_{x\_dec}=a_{xmax\_dec}\cdot\sqrt{1-\frac{a_y^2}{a_{ymax}^2}}-a_R \tag{10}$$

where: a_{en_lim} is the tractive wheel acceleration (obtained from the wheel's maximum tractive force at different motor speeds) [m/s^2]; a_R – acceleration of the road load forces [m/s^2]. By calculating the accelerations at different speeds, the acceleration limit diagrams at different speeds diagram can be generated. This map-type diagram provides a more detailed look at the vehicle's constructive limits, independent of the track form.

To simulate the vehicle on track, the maximum speed at any point needs to be calculated. While the vehicle is on a straight road, speed will be limited only by the motor design and road resistances. In a cornering situation, the lateral force will be the case of pure lateral condition (the longitudinal acceleration is equal to 0, 11). This pure lateral force (when the longitudinal acceleration is 0) cannot be equal to the maximum value from the friction ellipse because the vehicle has driving resistance forces. Solving relation 11 will give the theoretical maximum speeds for every point on the track, this speed needs to be adapted in case it is higher than the vehicle's maximum constructive speed. In a cornering situation, the tire lateral force is given by the relation [2,10,11]:

$$F_y=m\cdot g\cdot\sin(\beta)+\frac{v^2 m}{R} \tag{11}$$

where: R is the radius of the track [m]; v – is the maximum speed in the calculated point [m/s]; β – the lateral inclination angle of the track [°].

To define the real speed profile, a combined model that takes into consideration both the maximum lateral speed and the vehicle possibility to accelerate and decelerate, is needed. To reach a peak speed point, the vehicle must first brake and, after reaching the peak speed, starts to accelerate. Between any two peak speed points, there is an acceleration and a deceleration process. To control the duration and determine the real speed at any point, the speed needs to be the highest speed in order not to lose traction during the next braking.

Getting the speed profile requires a double calculation between two speed peaks, one for acceleration and one for braking. In this case, for any given position, there were two speeds calculated, the real value is the lower one between them, because in the other case, the vehicle would not be able to accelerate or decelerate without losing grip. To calculate the speeds for the acceleration and deceleration events the following relation was used [2,11]:

$$v_2=\sqrt{v_1^2+k\cdot 2\cdot a_x\cdot dx} \tag{12}$$

where: k is a sign coefficient that takes the value of 1 for acceleration and -1 for braking; dx – the distance between two points.

One of the main limitations of this model is that it does not consider suspensions, so a weight transfer analysis cannot be carried out. Another weakness is that the tire model is not detailed − in this model, the tire alignment angles and slip, are not impacting the results.

3. Simulation test cases

Depending on the weather conditions, different types of tires have been used in motorsport competitions. In the case of a dry road surface, the tires used are slick tires. This type of tires has a higher grip factor than regular tires available for purchase. Slick tires for Formula student applications can have a grip factor around 1.4-1.8 based on the tire data provided by the "*Formula SAE Tire Test Consortium (FSAE TCC)*" while average road vehicle tires have a grip factor of 0.7-0.8 [10]. For raining conditions, slick tires having completely smooth surfaces cannot channel the water away from the contact patch of the tire, causing hydroplaning to appear [12]. In this situation, wet tires are used. They are more like road vehicle tires and have a lower grip factor [13]. This is because they have threaded grooves to channel the water away, but by doing so they decrease the contact patch surface area [13].

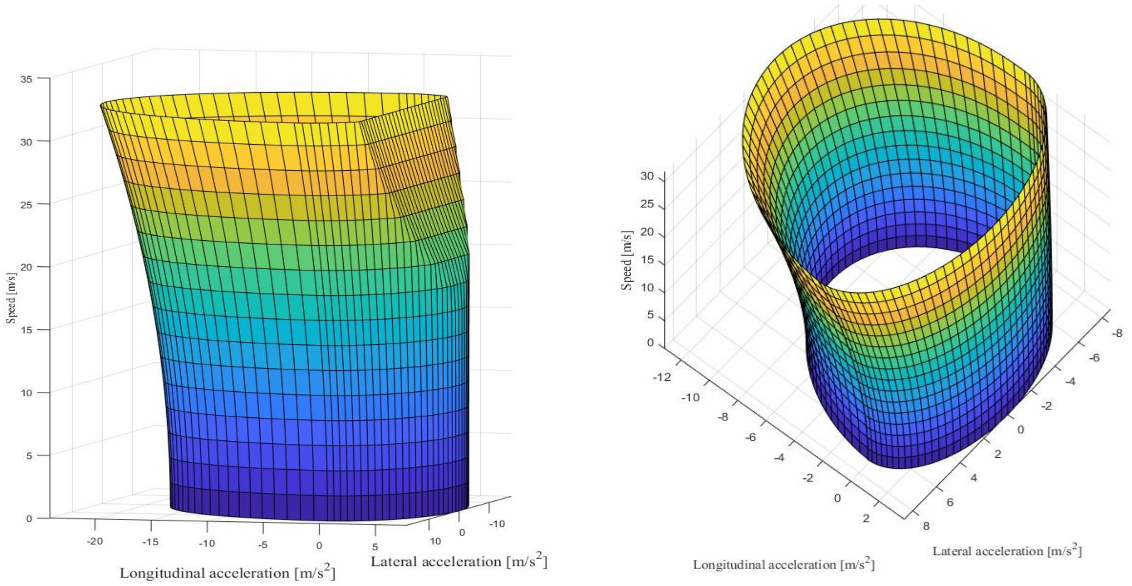
In this simulation test case, two types of tires (one slick and one wet) were simulated. First, the acceleration limit diagrams at different speeds were generated for both configurations, after the vehicle was simulated on the Most racing track (Czech Republic).

The relations (1-12) were implemented in MATLAB, using the main parameters of the vehicle, to first calculate the acceleration capabilities. Using the acceleration capabilities, the vehicle's maximum theoretical speed on the track is calculated when the lateral acceleration is maximum. To not lose grip between track sectors, the real speed profile was determined based on the lower speed of the vehicle considering a braking and acceleration sequence between two turns.

4. Results and discussions

The acceleration limit diagrams at different speeds have been generated and displayed in Figure 2. For the slick tires test case, the maximum longitudinal acceleration available (when the lateral acceleration is null), decreases as the speed rises, from 7.78 m/s^2 to 2.81 m/s^2. For the longitudinal deceleration, the deceleration capabilities increase with speed, from -17.77 m/s^2 to -23.98 m/s^2. For the acceleration, the decrease of available acceleration is the result of the electric motors limitation. Therefore, with the increase of the vehicle speed, the available tractive force is decreasing. For the considered electric motors, after they reach 4200 rpm, the available brake torque begins to decrease [9]. For the deceleration, the increase of the limit with speed is due to the downforce that increases with the increasing speed of the vehicle.

For the wet tires, maximum longitudinal acceleration available (when the lateral acceleration is null), decreases as the speed rises, from 3.81 m/s^2 to 2.73 m/s^2. For the longitudinal deceleration, the deceleration capabilities increase with speed, from -8.96 m/s^2 to -13.12 m/s^2. While the reasons for the tendencies of the acceleration and deceleration limits are the same as for slick tires, wet tires having a lower grip factor lead the vehicle to a lower grip limit [2,10]. If the electric motor delivers the entire tractive force to the wet tires, the vehicle will lose its grip. In this case, the vehicle is grip limited instead of engine limited.



a. b.

Figure 2. Acceleration limit diagrams at different speeds for different tyres: a. slick tires; b. wet tires.

After the vehicle acceleration limits have been determined, the vehicle can be simulated on the considered track (Most circuit, Czech Republic). When the vehicle is equipped with the slick tires, it has the speed profile presented in Figure 3, and when equipped with wet tires, the speed profile is depicted in figure 4. For this track structure, the average speed was 45.54 km/h, and the vehicle completed an entire lap in 88.52 s if it was equipped with the slick tires. With the wet tires configuration (Figure 4), the vehicle finished the lap in 126 s, with an average speed of 31.89 km/h. The lower performance of the vehicle equipped with wet tires was a result of lower acceleration possibilities because the engine cannot use its entire tractive force without the vehicle losing its grip.

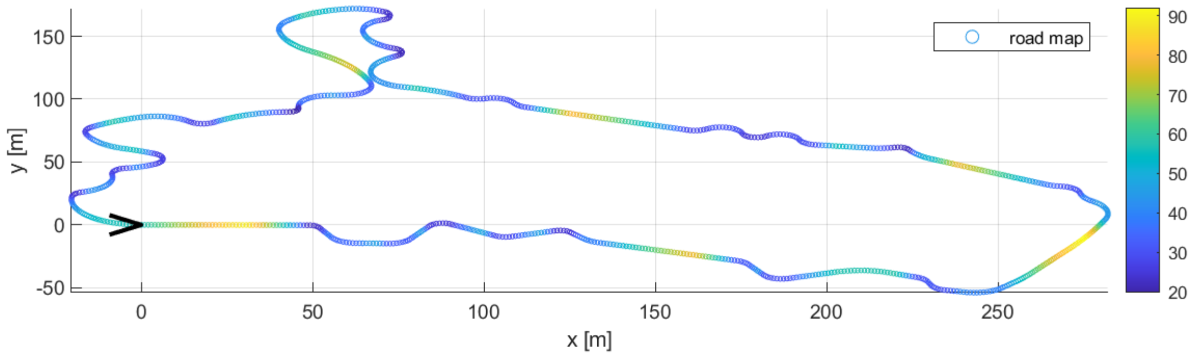


Figure 3. Track speed profile for slick tires configuration.

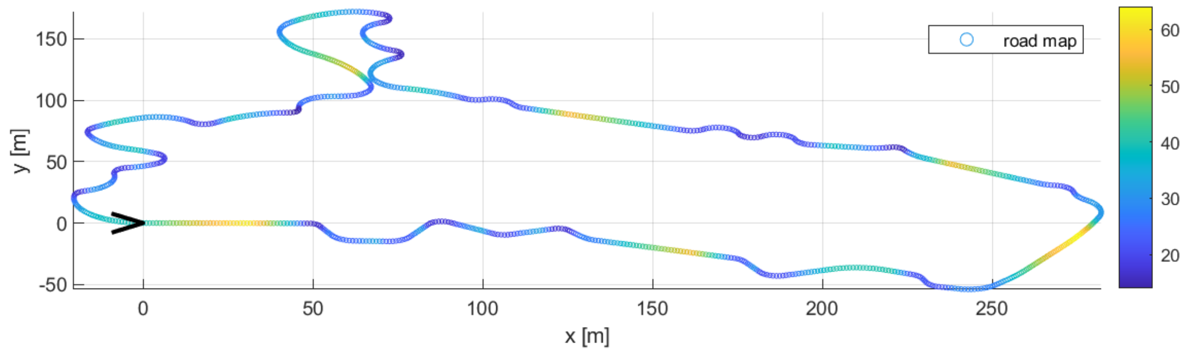


Figure 4. Track speed profile for wet tires configuration.

5. Conclusions

The Formula Student electric vehicle behaviour on a racing track was estimated with a mass point quasi-state model. When the tires designated for the raining conditions are used, the time needed to finish the lap increased by 40% and the average speed decreased by 30%. The main reason for the loss in performances is that the vehicle cannot use its entire tractive force of the motor. This could impact the drivability of the vehicle, since the driver could lose grip when trying to use the entire tractive force in any driving scenario. A solution could be to limit the tractive force of the engines, so the acceleration would be limited by the electric motors, and not by the grip coefficient.

To further develop the simulation model, an energy analysis of the vehicle on the track could be performed and asses its energy efficiency and consumption in different driving scenarios.

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