Modeling and Simulation of Launch Control System for Formula Student Electric Vehicle

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***Abstract*—To reduce excessive wheel spin which results in a loss of traction different types of control systems are used in an electric car, one of which is- Launch Control System (LCS). These control systems prevent excessive spinning of wheels by keeping them in optimal slip range to produce maximum tractive force. While developing such systems attention is given to identify proper slip estimates, deriving the required torque value, and finally, a control strategy to keep these in the optimal range. In this paper we propose a constraint optimal slip control to maintain the maximum possible tire force possible, the control strategy is applied to a Formula Student vehicle, which is constrained to certain rules as specified by the competitions. MATLAB/Simulink software was used to develop a full vehicle model including a tire model having a tire slip control system. The vehicle model was then simulated using IPG CarMaker software to estimate the difference in lap time with and without the slip control, this was done for a straight-line acceleration event.**

***Index Terms*—Acceleration, electric vehicle, formula student electric, launch control, powertrain control, traction control, slip-ratio control, vehicle dynamics control.**

I. Introduction

Formula Student Electric is an international design competition where students build single-seat electric race car, constrained by the competition rules [1]. Being an electric vehicle competition, electric motors are the primary powertrain component, which can deliver very high peak torques for a short amount of time. This high standstill torque when exceeds the tire traction limit can cause excessive wheel spin resulting in loss of traction and can lead to vehicle instability. This paper discusses the design, and simulation of a launch control system that can maximize vehicle acceleration performance as well as improve stability. The launch control system is defined as a system that maximizes the tire traction force by controlling the drive torque applied by the motor. This paper focuses on longitudinal force control, which depends on the wheel slip, and road conditions, where the output torque command from the throttle pedal can be controlled by regulating the wheel slip at the desired value to produce the desired amount of longitudinal traction force (slip control) [2].

For maximum longitudinal acceleration, wheel slip must be limited to an optimal value. Although the driver can control the torque command by decreasing the throttle response and prevent tires from slipping, the reaction time to control the throttle response might not be ideal. Hence, the traction control system can be used to limit the torque command of the motor when the tires slip above the optimal slip value. There are different traction control methods available. One is the model following control and the other is the optimal slip ratio control researched by Hori. [3].

To get an idea of the amount of torque required to achieve the optimum slip we need to model a tire-road interaction. There are many ways that can be used to model a given tire these mainly depend on the amount of complexity and the intended purpose of the model. The pacejka magic formula model is one such example that includes the camber angle as a parameter in modeling the tire, the plot of longitudinal force against the slip ratio can be obtained by using this semi-empirical magic formula model [4]. Other models include brush model, TMeasy model and non steady-state models [5] [4]. The raw testing data of the tire is obtained from the Tire Testing Consortium (TTC) and then a curve is fitted using the magic formula to get the value of coefficients which were then used for dynamic modeling. Further, a full vehicle model is required to represent the vehicle in a simulation environment. The vehicle model included aerodynamic forces, longitudinal and lateral load transfer.

This paper focuses on the slip ratio control model, and pacejka magic formula to estimate the torque required by the wheels. The main hardware components of traction control are similar to Anti-lock Braking System (ABS) [6]. The vehicle utilizes wheel speed sensors to monitor slip. The measured wheel speed data is sent to the Electronic Control Unit (ECU), which then processes the data and according to the coded algorithm reduces torque command. It is started automatically when sensors detect excessive wheel spin. A 3 axis accelerometer is used to measure longitudinal and lateral force and calculates the weight transfer force.

The launch control also acts as a safety feature that prevents wheel slip and increases vehicle stability, especially during acceleration in slippery road conditions. In racing, the launch control is rather used as a performance enhancement system, allowing maximum traction during acceleration, by keeping the tires at the optimal grip level, thereby reducing wheel slip, and increasing the performance of the car. If an excessive amount of throttle is applied throughout the cornering, the wheels can lose traction and slide sideways resulting in oversteer.

II. Tire Model

The interaction between the tire and the road is the source of all forces and moments that are experienced by the vehicle body. This makes tires one of the most important factors which determine the performance, control and stability of the vehicle. The freely rolling wheel, that is without a driving torque over a plane surface will result in zero slipping condition. During this condition, the tire is not capable of producing driving or traction force. On the other hand, when a tire is subjected to a torque the tire motion deviates from this definition of zero slip. The contact patch of the tire travels with a greater velocity than the vehicle or the driven wheel which causes the tire to slip and produce traction force. The slip ratio is the difference between the angular velocity of the deformed tire and the free-rolling of the car that is not subjected to torque [7]. The traction force is supposed to increase with the slip ratio up to a certain range exceeding those results in a decrease of tractive force which is the result of excessive slip at the tires. This behavior results in an optimal slip condition for a specific vertical load keeping every other factor the same.

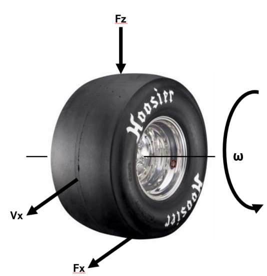


Fig. 1: Tire road interaction forces.

TABLE I: Tire road contact variables.

Symbol	Description and Unit
ω	Wheel angular velocity
r_w	Wheel radius
V_x	Wheel hub longitudinal velocity
$r_w\omega$	tire tread longitudinal velocity
$V_{sx} = r_w\omega - V_x$	Wheel slip velocity, defined as the difference between the longitudinal velocities of the wheel hub and the tire tread
$k = \frac{V_{sx}}{\lvert V_x \rvert}$	Wheel slip
F_z	Vertical load on tire
F_{z0}	Nominal vertical load on tire
$F_x = f(\kappa, F_z)$	Longitudinal force exerted on the tire at the contact point. Also a characteristic function f of the tire.

Equation:

$$F_x = D_x sin(C_x\ arctan\{B_x\kappa_x - E_x[B_x\kappa_x - arctan(B_x\kappa_x)]\}) + S_{Vx} \tag{1}$$

where, $df_z = \frac{F_z - F_{z0}}{F_{z0}}$
$\kappa_x = \kappa + S_{Hx}$
$C_x = p\_Cx1$
$D_x = \mu_x \cdot F_z$
$\mu_x = p\_Dx1 + p\_Dx2 \cdot df_z$
$E_x = (p\_Ex1 + p\_Ex2 \cdot df_z + p\_Ex3 \cdot df_z^2)[1 - p\_Ex4 \cdot sgn(\kappa_x)]$
$K_{x\kappa} = F_z \cdot (p\_Kx1 + p\_Kx2 \cdot df_z) \cdot e^{(p\_Kx3 \cdot df_z)}$
$B_x = \frac{K_{x\kappa}}{C_x D_x + \epsilon_x}$
$S_{Hx} = p\_Hx1 + p\_Hx2 \cdot df_z$
$S_{Vx} = F_z \cdot (p\_Vx1 + p\_Vx2 \cdot df_z)$

S_{Hx} and S_{Vx} represent offsets to the slip and longitudinal force in the force-slip function, or horizontal and vertical offsets if the function is plotted as a curve. μ_x is the longitudinal load-dependent friction coefficient. ϵ_x is a small number inserted to prevent division by zero as F_z approaches zero [4].

The plot was obtained in MATLAB for different loads and the variation of maximum longitudinal with the vertical load was studied. The plot inferred that as the vertical load is increased the value of slip required to produce maximum force decreases, this is an important topic that is to be considered as the vertical load on an actual car keeps on changing, so does its capability to produce maximum longitudinal force. The following graph represents the Fx – SR characteristics of the selected tire.

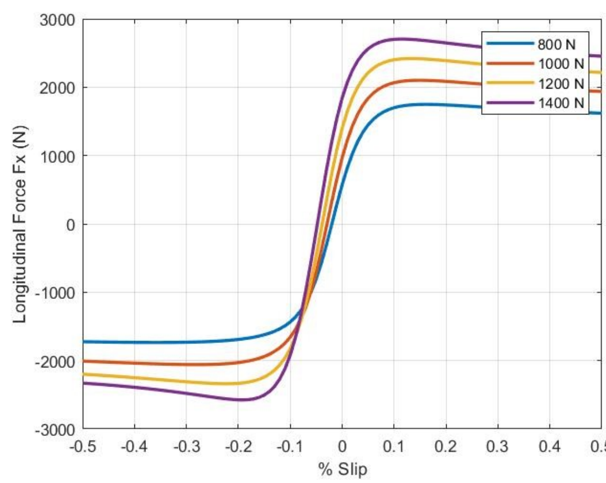


Fig. 2: Fx Vs SR plot generated from the MATLAB using the equation (1).

III. Vehicle Model

The dynamic model of the vehicle in longitudinal motion depends on two major subsystems- Vehicle dynamics and powertrain dynamics [8]. The vehicle dynamics domain is influenced by longitudinal tire forces, aerodynamic forces, rolling resistance, and gravitational forces. The powertrain dynamics domain is governed by motor, drivetrain, and wheel assembly. Consider a vehicle moving in the longitudinal direction, the driving force or the tractive force coming from the tire tends to accelerate the vehicle whereas the resistive forces such as the aerodynamic drag force, rolling resistance force, losses in driveline assemblies reduce the total tractive effort [9].

$$F_x = F_t - (F_d + F_r + F_l) \tag{2}$$

where, $F_x = nettractiveforce$
$F_t = tractiveforcefromtire$
$F_d = aerodynamicdrag$
$F_r = tirerollingresistance$
$F_l = lossesindriveline$

A. Longitudinal Force from tire

The longitudinal force for the tire is identified with the help of the tire model discussed earlier. The results of the tire model depend on
a) The slip ratio
b) Normal force on the tire
c) The friction coefficient of the tire-road surface
The normal force on the tire in a dynamic condition considering the whole vehicle as the system changes at every instant given that the vehicle is accelerating or decelerating.

B. Aerodynamic Force

Aerodynamic forces become dominant as the speed of the car increases. The goal of the aerodynamic design of every car is to minimize this drag, at the same maximizing the amount of downforce generated by the car at a particular speed. The aerodynamic force can be represented by using non-dimensional coefficients CL and CD indicating the lift and drag force respectively [10].

$$F_d = \frac{1.\rho.C_d.A.V^2}{2} \tag{3}$$

where, $Fd = aerodynamicdrag$
$\rho = massdensityofair$
$Cd = coefficientofdrag$
$A = frontalarea$
$V = longitudinalvelocityofcar$

C. Normal Force Calculation

A car under acceleration or deceleration transfers some fraction of load from one axle to another. Along with the weight of the car, the normal force is influenced by the following factors: -
a) Centre of gravity location
b) Longitudinal acceleration of the car
c) Aerodynamic forces
Consider the car in positive acceleration i.e. under driving traction, the amount of load transferred is given by:

$$\delta W = \frac{W_t.ax.h}{l} \tag{4}$$

Therefore the total force on axle is: -

$$W_{F/R} = \frac{W_t}{2} + \delta W \tag{5}$$

where, $W_t = totalweight$
$ax = longitudinalacceleration$
$h = CGheight$
$l = wheelbase$
The effect of the aerodynamic forces is significant at higher speed and is included in the mathematical model.

IV. Model Design

Figure 3 shows the overall flowchart of the model. The model consists of an open-loop estimator model and a closed-loop corrector model. During acceleration, the driver-requested torque is immediately estimated based on the open-loop model using the weight transfer block-set and tire model. The reference slip value can be set by the driver, which accordingly sets the sensitivity of the model. The dynamic slip is measured by taking the average of driven wheel speed, and the vehicle speed is measured using the front wheel speed. The closed-loop model estimates the error slip between reference and dynamic slip values, which is fed back to the tire model as correction torque.

A. Open Loop Estimator Model

The open-loop estimator model is based on the lateral, and longitudinal accelerations. The open-loop model estimates the output torque using the vehicle model and tire model. The longitudinal and lateral weight transfer block set adds or subtracts the torque due to weight transfer based on acceleration forces. By means of the longitudinal acceleration ax, the vertical force on the rear axle Fz, the rear axle is calculated by making use of (4). The lateral acceleration a_y is used to calculate the lateral rear axle force:

$$F_y = \frac{a.m.a_y}{l} \tag{6}$$

This lateral component of the force determines the position in the friction ellipse. The wheel radius and the gear ratio are used to calculate the required torque.

B. Closed Loop Correction Model

The closed-loop is based on reference slip value which can be set by the driver and accordingly the ECU calculates the dynamic slip value of the vehicle from wheel speed data. A PID controller is used to correct the dynamic response of the vehicle by decreasing or increasing the torque command to match the reference slip value. The PID controller calculates the error which will then be used to calculate the output signal from the controller u(t) with the equation below (parallel PID loop).

$$u(t) = K_p.e(t) + K_i \int_{t_o}^{t_i} e(t').d(t') + K_d.\frac{de(t)}{d(t)} \tag{7}$$

V. Motor Power Map

Also, the competition rules state that the power output from the battery should not cross above 80kW power [1]. Therefore, a motor map is coded on the ECU which maintains the motor power below 80kW including all efficiency factors using the mechanical rotational power equation given by equations below.
$MotorPower(watt) = \tau.\omega.efficiency$
or
$MotorPower(watt) = \frac{\tau.RPM.efficiency}{9.550}$

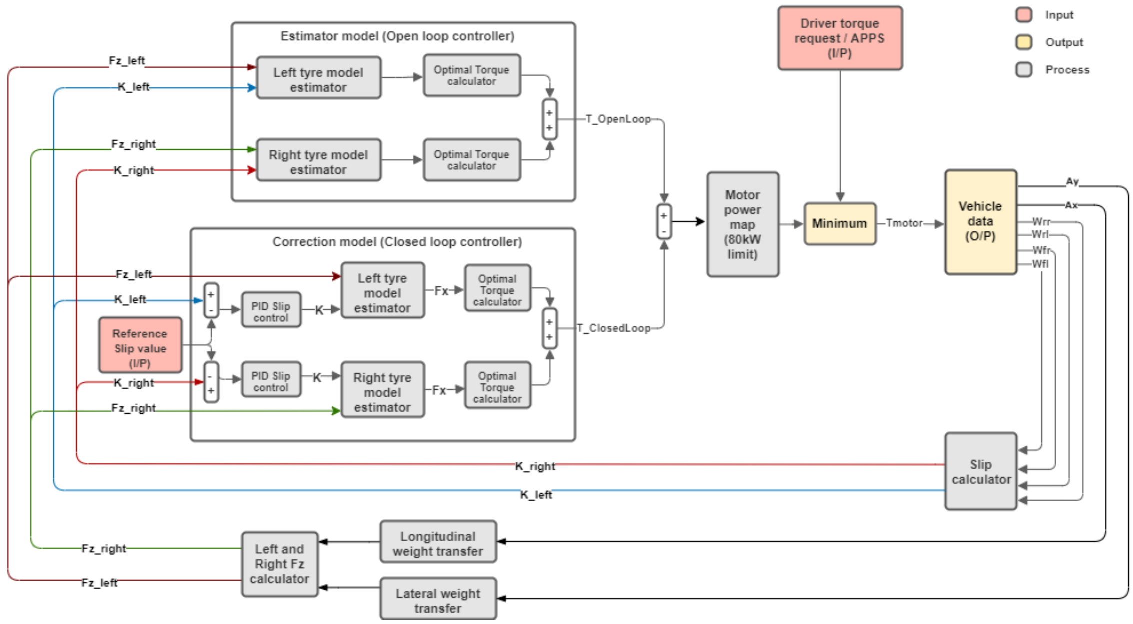


Fig. 3: Launch control system flowchart depicting the interconnection of open loop, closed loop and motor power limit blocks.

VI. Simulations

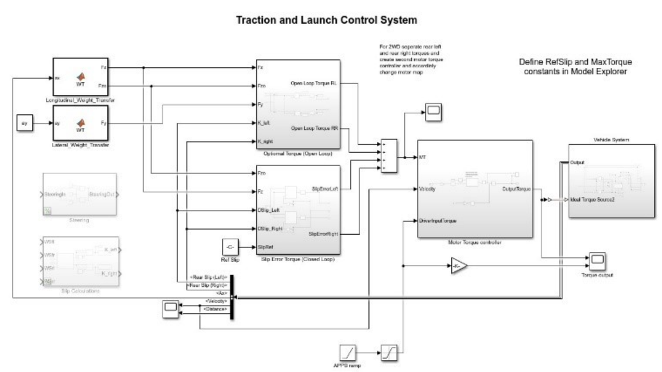


Fig. 4: Simulink Simulation Model.

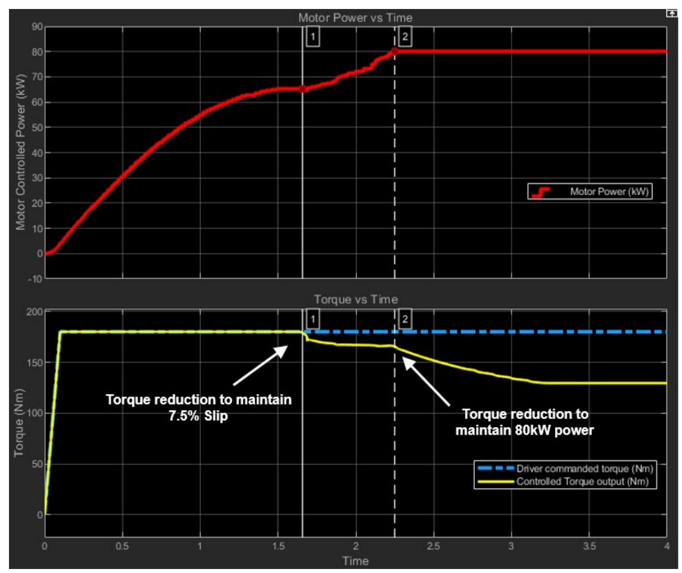


Fig. 5: Output (Power = 80KW, Torque = 180Nm, Slip reference = 7.5%).

A closed-loop simulation was done on MATLAB/Simulink where Vehicle Dynamics block set and Tire-Road Interaction (Magic Formula) block models were utilized to generate vehicle output data. The PID parameters defined the aggressiveness of the model concerning the convergence of dynamic slip and reference slip. See Fig. 4.

A. Simulink simulation results

Fig. 5 shows a simulation setup of acceleration event (completion rule: Max power limit = 80kW, Track length = 75m) on Torque limit: 180Nm, and Slip reference = 7.5%.See Fig. 5.From the simulation output, it can be seen that to maintain a 7.5% reference slip value the controlled output torque reduces after 1.5 seconds and after the vehicle has reached 80kW power limit again the output torque reduces after 2.3 seconds to maintain the power threshold value according to motor mechanical power equation.

Similarly, the simulation output for 200Nm maximum torque undergoes torque reduction after 0.5 seconds to maintain a 7.5% reference slip value. See Fig. 6 and after the vehicle has reached 80kW power limit again the output torque reduces after 2.3 seconds (similar to 180Nm simulation) to maintain power threshold value according to motor mechanical power equation.

The output from the MATLAB Simulink model for different vehicle setups is shown in Table.II These simulations were not conclusive at showing the best-simulated results as they do not include any powertrain, drivetrain, or external vehicle

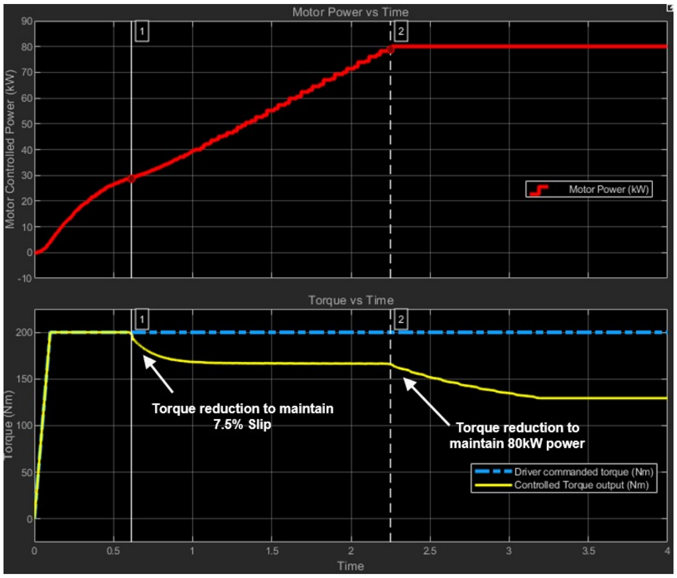


Fig. 6: Output (Power = 80KW, Torque = 200Nm, Slip reference = 7.5%).

parameters. Hence, IPG CarMaker software was used which includes internal and external vehicle parameters.

TABLE II: LCS Simulink results.

Power limit	Torque limit	Slip%	Time to reach 75meter
80kW	180Nm	7.5%	3.931s
80kW	180Nm	9%	3.869s
80kW	200Nm	7.5%	3.834s
80kW	200Nm	9%	3.745s
60kW	200Nm	7%	4.121s
60kW	180Nm	9%	4.037s

B. IPG CarMaker simulation results

To test and simulate the launch control logic, full vehicle simulations were performed in IPG CarMaker. In IPG CarMaker a full vehicle model can be simulated without having to model the whole car and instead focus on the model that needs to be evaluate [11]. The parameters of the vehicle model were updated and the powertrain block was modified to implement the traction control model. Referring back to Fig.4 model only the vehicle dynamics block set is removed and the rest of the model is integrated with the IPG CarMaker Simulink environment to replace the gas (throttle) command in IPG CarMaker. Fig.7 shows the complete model of IPG CarMaker and Simulink environment including the same open-loop model, closed-loop model, and motor power map model.

Table. III shows all the variables that are taken as input from the IPG CarMaker software and the calculated output DMGasout variable taken as input to the IPG CarMaker software to control the torque command of the vehicle.

The data from the IPG CarMaker Fig.10a and Fig.10b show the controlling nature of the throttle and torque response set by the PID controlled closed-loop system. The virtual driver

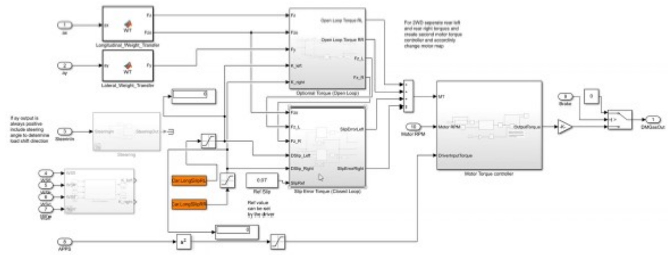


Fig. 7: Simulink model integrated with IPG CarMaker.

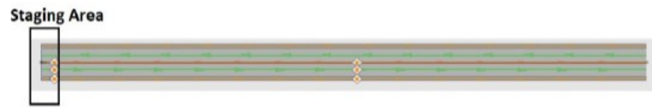


Fig. 8: Acceleration track of 75m length.

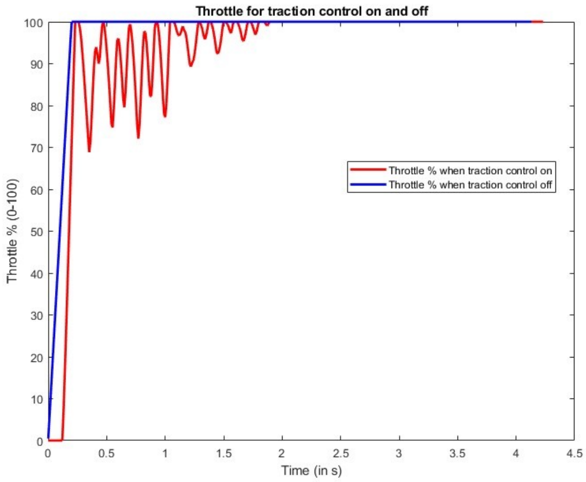
Fig. 9: Virtual environment in CarMaker.

TABLE III: IPG CarMaker parameters.

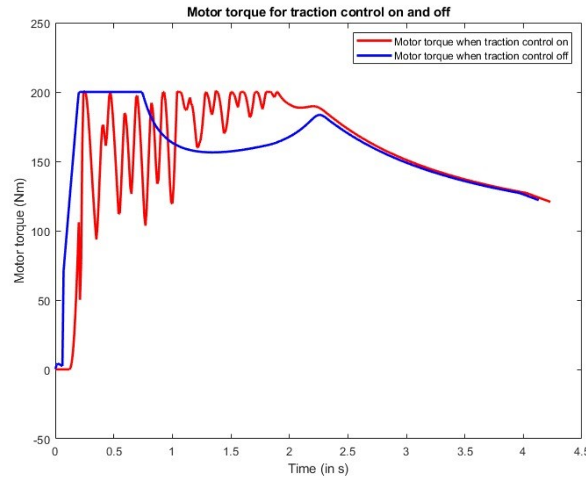
CarMaker variables	Details	Input/Output
Car.ax	acceleration in x (longitudinal acceleration)	Input
Car.ay	acceleration in y (lateral acceleration)	Input
Car.LongSlipRR	longitudinal slip of rear right tire	Input
Car.LongSlipRL	longitudinal slip of rear left tire	Input
Car.WheelSpd FL	wheel speed of front left tire	Input
Car.WheelSpd FR	wheel speed of front right tire	Input
Car.WheelSpd RL	wheel speed of rear left tire	Input
Car.WheelSpd RR	wheel speed of rear right tire	Input
PT.Motor.rotv	motor RPM	Input
DMBrake brake	pedal position (0-1)	Input
DMGasout	throttle/Gas pedal position (0-1)	Output

slams the throttle during the acceleration run in the simulation during LCS off the command is a ramp whereas when LCS is on we can see the controlled throttle from Fig.10d shows how the PID closed-loop system tries to minimize the error and stabilize the dynamic slip value closed to the reference slip value. The overall peak overshoot is significantly reduced which again helps in reducing that overshoot in the rear wheel speed.

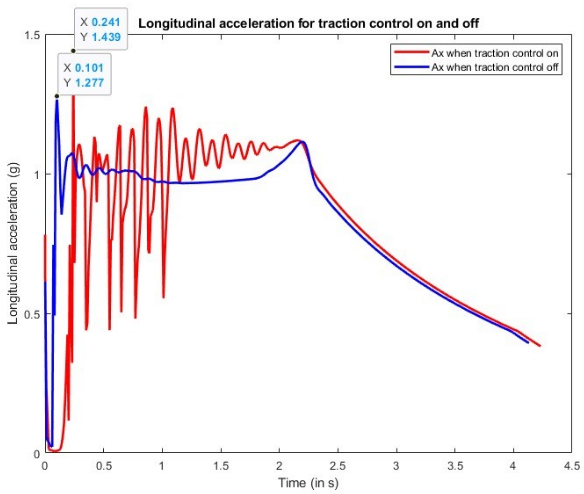
It can be inferred from the data that whenever the slip overshoots beyond the target slip value the throttle or torque



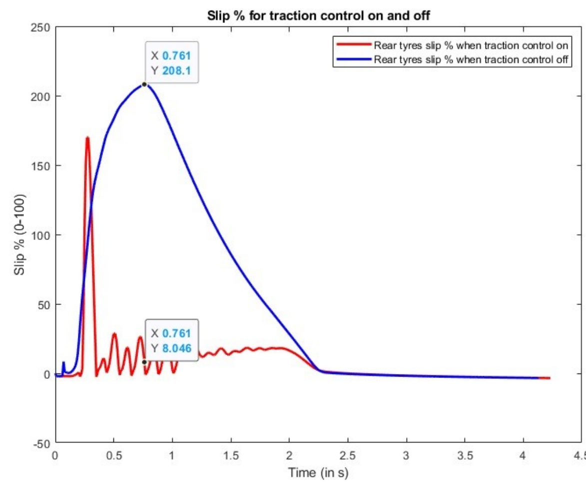
(a) Motor Torque



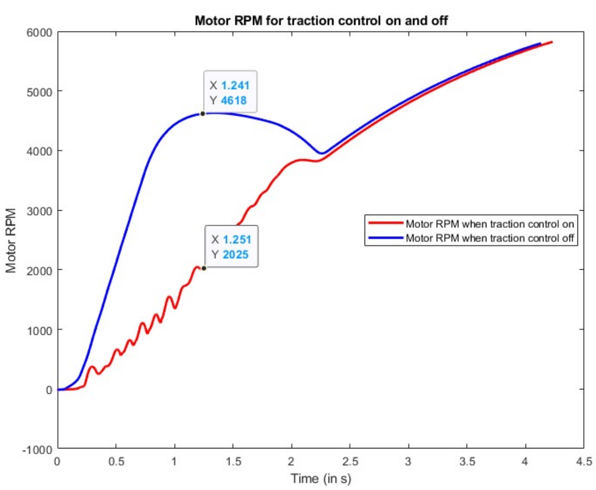
(b) Throttle



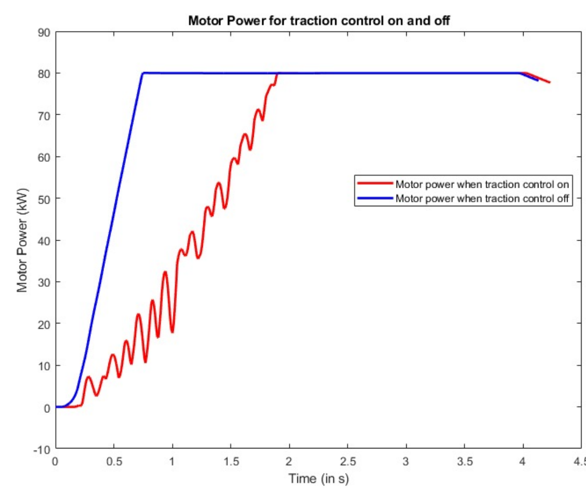
(c) Longitudinal force (Fx)



(d) Slip



(e) Motor RPM



(f) Motor Power

Fig. 10: LCS on and off analysis.

command from the motor is reduced using a closed-loop feedback system till the slip matches the target value. Hence from the above data, it can be concluded that the closed-loop system is working as expected dependent on the PID tuning parameters which set the transient and steady-state response of the system.

TABLE IV: Lap times with LCS on and off.

Power limit (kW)	Torque limit (Nm)	Slip %	LCS off (s)	LCS on (s)
60	200	7	3.935	3.921
		8		3.901
		10		3.913
80	160	7	3.916	3.935
		8		3.883
		10		3.902
	180	7	3.825	3.833
		8		3.801
		10		3.814
	200	7	3.792	3.764
		8		3.703
		10		3.728

Table. IV shows the simulation results on different vehicle powertrain parameters and target slip value. At around 8% target slip value the vehicle performs better which is ideal.

VII. Conclusion

The launch control system showed promising results in simulation. The Simulink model did not provide a detailed evaluation of the vehicle as a whole, so IPG CarMaker was used to simulate the entire vehicle in a virtual environment of an acceleration track as specified by the FS rules. The model gave the lap time delta of about 100 milliseconds and the car achieved a maximum longitudinal acceleration of 1.439g as compared to 1.277g, Figure 10c. Virtual simulation models always give a great estimate of what will happen on the track but the real-world scenario would only give the real numbers to validate the simulations.

References

[1] F. S. Germany. (2020) FS Rules 2020 V1.0.pdf. [Online]. Available: https://www.formulastudent.de/fsg/rules/

[2] H. Lee and M. Tomizuka, "Adaptive vehicle traction control," *Institute of Transportation Studies, UC Berkeley, Institute of Transportation Studies, Research Reports, Working Papers, Proceedings*, 01 1995.

[3] Y. Hori, Y. Toyoda, and Y. Tsuruoka, "Traction control of electric vehicle: basic experimental results using the test ev" uot electric march"," *IEEE transactions on Industry Applications*, vol. 34, no. 5, pp. 1131–1138, 1998.

[4] H. Pacejka, *Tyre and Vehicle Dynamics*, 2nd ed. SAE International, 2005.

[5] F. Conte, "Expanding the brush model for energy studies," Ph.D. dissertation, Stockholm, Sweden, 2014.

[6] V. Ivanov, D. Savitski, and B. Shyrokau, "A survey of traction control and anti-lock braking systems of full electric vehicles with individually-controlled electric motors," *IEEE Transactions on Vehicular Technology*, pp. 2013–2026, 2014.

[7] T. D. Gillespie, *Fundamentals of Vehicle Dynamics*. Warrendale, PA: Society of Automotive Engineers, 1192.

[8] B. Janarthanan, C. Sujatha, and C. Padmanabhan, "Longitudinal dynamics of a tracked vehicle: Simulation and experiment," *Journal of Terramechanics*, vol. 49, pp. 63–72, 04 2012.

[9] R. Rajamani, *Vehicle Dynamics and Control*, ser. Mechanical Engineering Series. New York NY: Springer, 2006.

[10] J. Katz, *Race Car Aerodynamics*. Cambridge MA: Bentley Publishers, 1995.

[11] IPG Automotive. (2021) CarMaker for simulink. [Online]. Available: https://ipg-automotive.com/products-services/simulation-software/carmaker/