

# Traction control of an electric formula student racing car

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## TRACTION CONTROL OF AN ELECTRIC FORMULA STUDENT RACING CAR

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**KEYWORDS** – Traction-control, Longitudinal vehicle control, Powertrain control, Slip-ratio control, Driver-support

**ABSTRACT** – This article describes the design of a traction control system in an electric Formula Student vehicle. In many race applications the accelerator pedal is difficult to control for an in-experienced driver, especially in the case of electric vehicles, where a large torque is available from standstill. A 3-DOF driveline model is used in combination with a 7-DOF vehicle model and a non-linear tyre model based on 10 parameters. The driveline and tyre model are validated by measurements. These models are used to design a suitable traction control system. This system consists of an open-loop part, which uses the longitudinal and lateral acceleration to calculate a torque limit via a driver provided friction estimation. The feed-back part of the controller regulates the slip-ratio of the rear-wheels. The traction control system is first implemented in the vehicle model and later in an electric Formula Student vehicle. A comparison is made between the vehicle with and without traction control. The vehicle with traction control performs significantly better in terms of longitudinal acceleration and shows a better driveability in terms of lateral acceleration.

## INTRODUCTION

Electric vehicles are becoming more popular, since fuel consumption and emissions are more important issues these days. Many electric vehicle use inboard motors connected to the wheels via drive-shafts and a fixed reduction. Electric motors are able to deliver a high torque from standstill and often have the possibility to deliver very high peak torques for a short amount of time. This can lead to traction problems and dangerous situations, especially at higher speeds.

The objective of this research is to design a traction control system to enhance the traction of an electric rear-wheel driven Formula Student vehicle with one central motor as in Figure 1. The empty car weight is approximately 245 kg and the motor is able to produce 85 kW which results in 1200 Nm on the rear axle. The system should be able to:

- Operate at all speeds, even at standstill.
- Enhance traction during cornering.
- Deal with the flexible driveline and possible resonances caused by this flexibility.



Figure 1: URE07

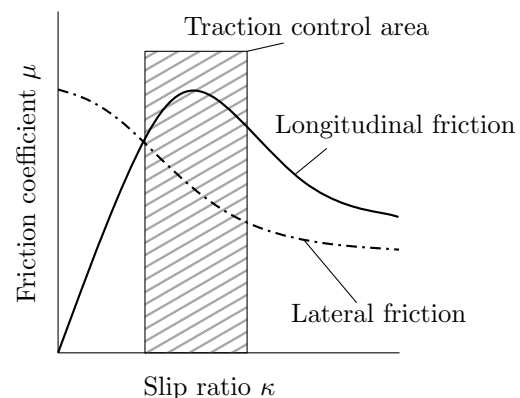


Figure 2: Typical tyre behaviour [1]

Traction control is defined as a system to maximise the utilisation of the tyres by controlling the drive torque applied by the motor. Since the lateral tyre force strongly depends on the longitudinal force applied to the tyres, traction control can improve the stability of the vehicle [1]. Electric motors are able to deliver high torques at low speeds and therefore the need for traction control is even more present. Figure 2, which shows the friction

coefficient  $\mu$  as a function of the longitudinal slip-ratio  $\kappa$ , also shows that equilibrium operating point for optimal friction is unstable.

Different forms of traction control for electric vehicle have been researched by Hori from 1998 until 2002. He used Model Following Control (MFC) to regulate the slip ratio to small values [2]. Slip ratio control regulates the slip ratio of the driven wheel to a pre-set value. For this the vehicle speed and the driven wheel speed has to be known. Furthermore prior knowledge of the tyre is required and more computation power is necessary. Tests show that this performs better than slip ratio control in terms of grip but the road condition estimator poses a challenge [2]. Maximum Transmissible Torque Estimation (MTTE) is a control method based on the acceleration of the driven wheels. The controller calculates a maximum acceleration of the wheel if adequate friction where to be present. When the measured acceleration exceeds this threshold, the torque to this wheel is reduced [3].

All of the above control methods use wheel-speed sensors to regulate the torque. For low vehicle speeds, the wheels-speed measurement can be unreliable because of a low sampling rate. In this paper the behaviour at low vehicle speeds is improved by making use of the acceleration sensors, the geometric properties of the vehicle and an estimated peak friction coefficient. Furthermore the driveline dynamics such as flexibility, free-play and friction have been taken into account to obtain a higher bandwidth and thus a better tracking performance of the desired slip-ratio.

This paper is organized as follows; first a vehicle model is created which consists of a 10 parameter non-linear tyre model, a 3-DOF driveline model and a 7-DOF vehicle body model. The tyre model is validated by measurements performed on the TNO test-trailer and the driveline model is validated by dynamic measurements. This vehicle model is used to design a controller which consists of an open and closed loop part. The open loop part uses the longitudinal and lateral acceleration and a user provided friction estimate, the closed loop part regulates the slip-ratio between the front and rear wheels. The designed traction control system is implemented in the vehicle model and later on in the URE07. Outside tests are performed to compare the situation with and without a traction control system.

## VEHICLE MODEL AND VALIDATION

To do offline testing and to design the traction control system, a vehicle model is required. This vehicle model is divided in three subparts; the tyres, the driveline and the vehicle body which connects the parts. Some general parameters of the vehicle are given in Table 1.

Parameter	Value
Location of the centre of gravity (x/y/z)	0.7675/0/0.24 m
Peak power	85 kW
Peak torque after reduction	1200 Nm
Track width (front/rear)	1.18/1.14 m
Unloaded wheel radius	0.27 m
Weight	247 kg
Wheelbase	1.535 m

Table 1: Vehicle specifications

### Tyres

The tyres used on this vehicle are custom Vredestein tyres with a 195/40R15 size. The tyre model is based on the work of van Rijk [4]. The model uses the adaptions by Svendius [5], to incorporate a-symmetric behaviour in longitudinal and lateral direction in the brush tyre model of Pacejka. Furthermore pressure dependency is included as well as slip speed velocity dependency as presented by Grosch.

The tyre model has been validated by pure longitudinal and lateral slip measurements as well as combined slip measurements. Those have been executed by the TNO tyre test trailer. Parameter estimation has been done by numerical optimization where the Sum of Squared Errors is minimized. Figure 3 shows the pure slip conditions and Figure 4 shows the combined slip conditions. The figures have been made dimension-less and are scaled for confidentiality reasons.

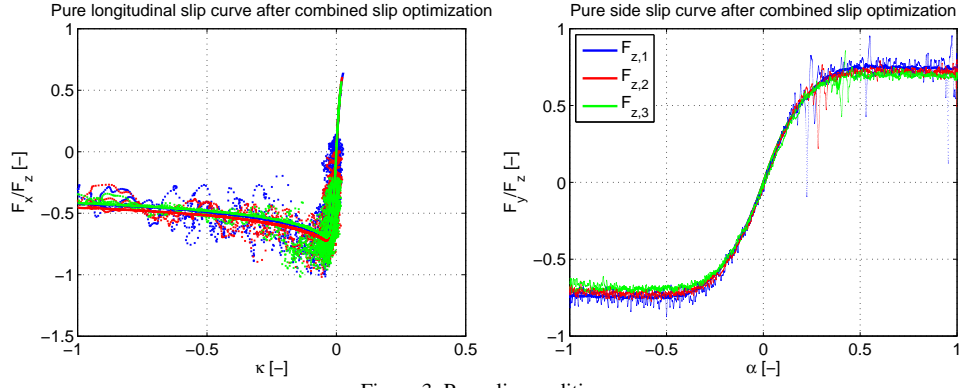


Figure 3: Pure slip conditions

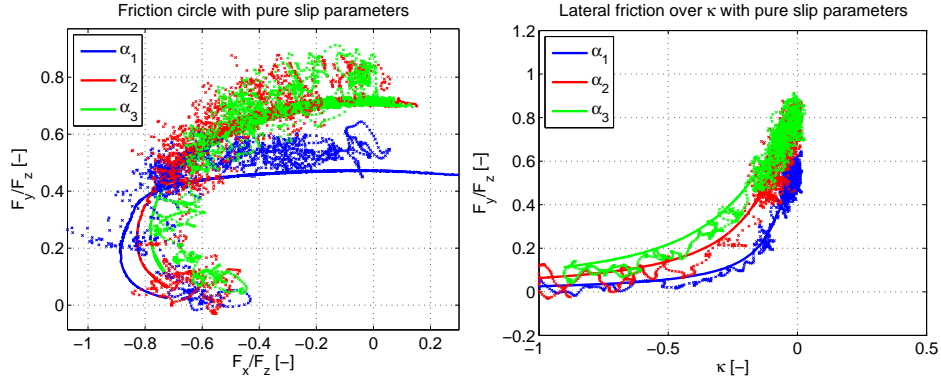


Figure 4: Combined slip conditions

### Driveline

The driveline has been modelled as a mechanical system with one inertia to represent the motor ( $J_p$ ), two spring-damper systems to describe the drive-shaft flexibility and two additional inertias ( $J_s$ ) to represent the wheels. The differential is not taken into account. A schematic overview is given in Figure 5. The stiffness and damping of the driveshaft is referred to as  $k_d$  and  $d_d$  respectively. The velocity at the motor side is indicated with  $\omega_p$  and the wheels with  $\omega_{left}$  and  $\omega_{right}$ . The angles in the system are referred to as  $\theta_p$ ,  $\theta_{left}$  and  $\theta_{right}$ . The system is actuated by the motor torque  $T_{drive}$  and the torques  $T_{left}$  and  $T_{right}$  represent the torques produced by the tyre-road interface. The equations of motion are given by:

$$J_p \dot{\omega}_p = T_{drive} - k_d(2\theta_p - \theta_{left} - \theta_{right}) \quad (1)$$

$$J_s \dot{\omega}_{left} = k_d(\theta_p - \theta_{left}) + d_d(\omega_p - \omega_{left}) - T_{left} \quad (2)$$

$$J_s \dot{\omega}_{right} = k_d(\theta_p - \theta_{right}) + d_d(\omega_p - \omega_{right}) - T_{right} \quad (3)$$

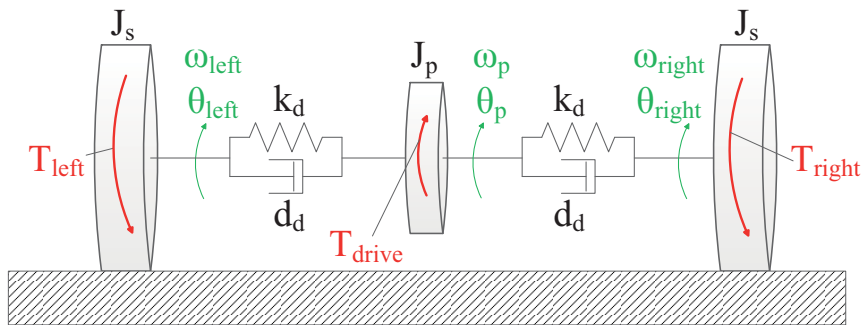


Figure 5: Driveline model

The parameters for this model have been identified by dynamic frequency response measurements (FRF). Two types of measurements have been performed, one without tyre contact and one with tyre contact. Figure 6 shows the transfer function from motor torque to motor speed without and with tyre contact. The measurements have been performed at 40 km/h. The measurement without contact has been used mainly to identify the inertia related properties. These determine the low frequency behaviour (lower than 1 Hz). The location of the resonance and anti-resonance is mainly determined by the spring stiffness. The height is an indicator for the damping present in the system. A significant difference is seen between the situation with and without road-contact where the resonance peak shifts from 10 to 12.5 Hz. This can be an indicator of free-play in the system or flexibility in the tyre itself.

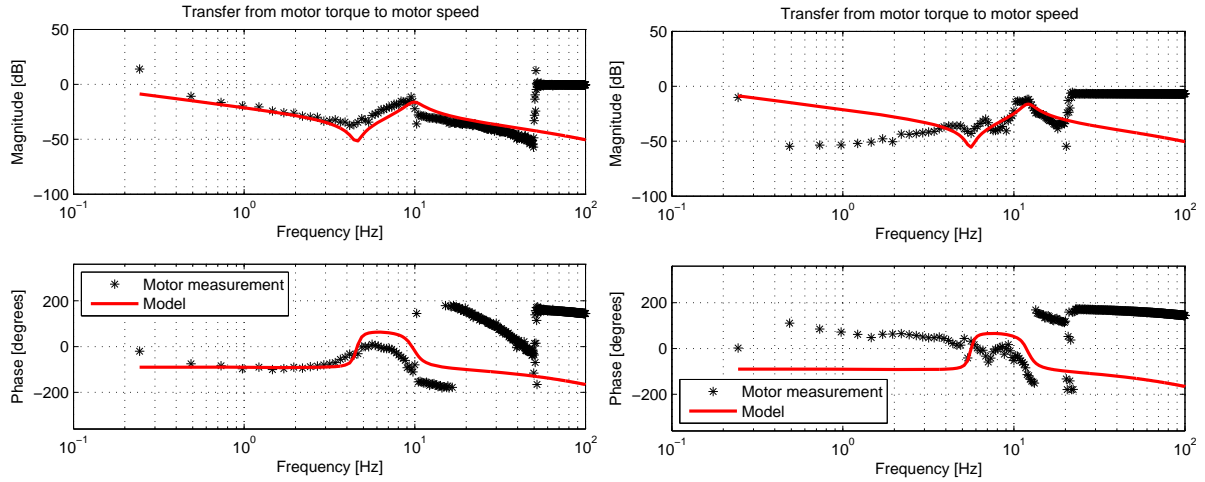


Figure 6: FRF measurement without (left) and with (right) tyre contact

### Vehicle body

The vehicle body model used is a two-track model as described by Pacejka [4]. The parameters for this model come from static measurements, except for the yaw inertia, this has been estimated. This body model is combined with the custom tyre model and the driveline model to represent the whole vehicle.

### Sensors

Important parts in the control-loop are the wheel-speed sensors. Proximity switches are used in combination with steel rings with 22 holes per revolution. Especially at lower speeds this resolution poses a challenge since this limits the update rate for the information to the traction control system.

## CONTROLLER DESIGN

Figure 7 gives an overview of the controller. A torque request is send by the driver, according to the open loop controller and the tyre model, the torque is immediately reduced before the tyre starts slipping. The friction coefficient can be adjusted by the driver to adjust for different weather and road conditions. When the vehicle starts driving, the speed of the driven wheels is determined by the average of the rear wheels and the vehicle speed is estimated by making use of the front wheels. The tyre model produces a set-point for the slip-ratio controller which results in an error that is fed back to the feed-back controller.

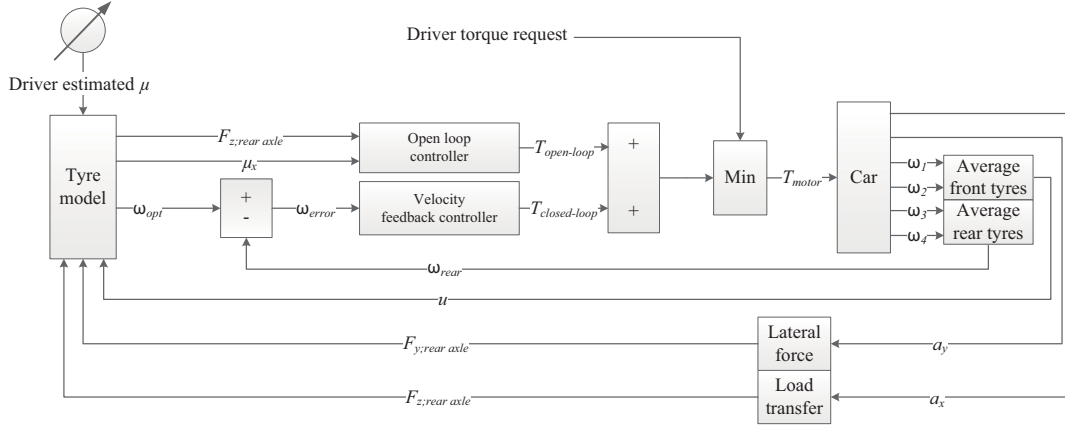


Figure 7: Controller design

### Open loop controller

The open loop controller is based on the longitudinal and lateral acceleration. By means of the longitudinal acceleration  $a_x$  the vertical force on the rear axle  $F_{z;rear axle}$  is calculated by making use of:

$$F_{z;rear axle} = \frac{2ag + h_{CG}a_x}{2L}m \quad (4)$$

where  $a$  is the distance from the centre of gravity to the front axle,  $g$  is the gravity,  $h_{CG}$  is the height of the centre of gravity,  $m$  the vehicle mass and  $L$  is the wheelbase. The lateral acceleration  $a_y$  is used to calculate the lateral rear axle force:

$$F_{y;rear axle} = \frac{a}{L}ma_y \quad (5)$$

This lateral component of the force determines the position in the friction ellipse as shown in Figure 8. The wheel radius and the planetary gear reduction are used to calculate the required torque. A notch filter has been used to suppress the driveline resonance as shown in Figure 9.

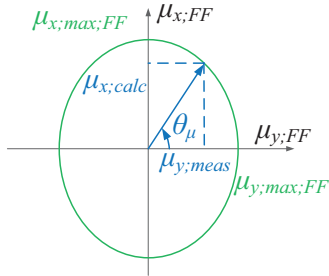


Figure 8: Friction ellipse

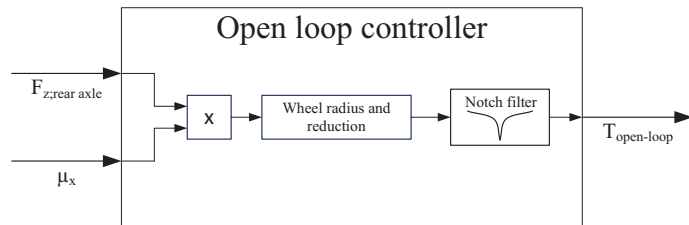


Figure 9: Open-loop controller

## Feed-back controller

The feed-back controller has been designed, based on the driveline model. The transfer function from motor torque to wheel-speed is shown in Figure 10 where the driveline model is compared with measurements without road contact.

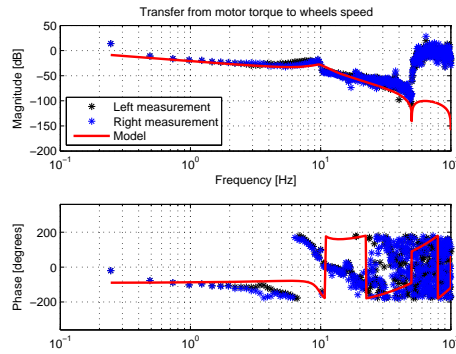


Figure 10: Measured transfer function

A PI controller is used to increase the bandwidth and a notch filter is set in series to deal with the resonance of the driveline. Figure 11 shows the PI controller in series with the notch filter is at 12.5 Hz. A bandwidth of 3 Hz is obtained. Higher bandwidths are difficult to achieve while remaining stable due to phase lag.

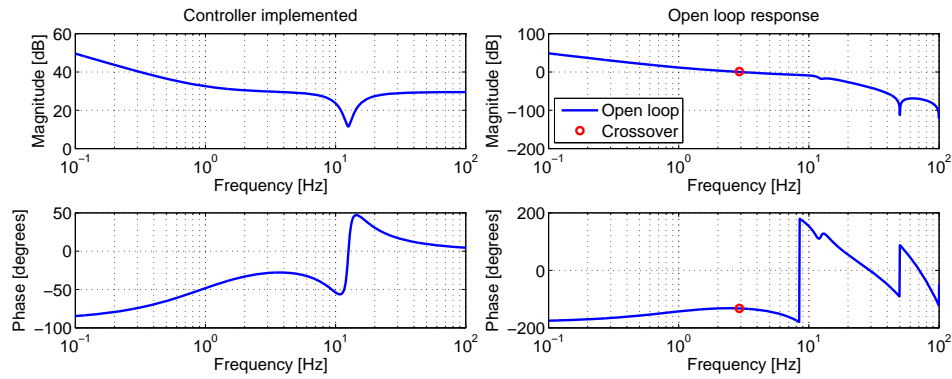


Figure 11: Feed-back controller design and open loop response

## TEST RESULTS

### Implementation in the vehicle model

The traction controller is implemented in the vehicle model. A test is performed by starting from standstill and applying a step signal on the throttle, the throttle is released after 4 seconds. Figure 12 shows the results of this simulation. The top-left figure shows the vehicle speed in red, the reference wheel speed in green and the achieved wheel speed in blue. A 0-100 km/h time of 3.30 seconds is achieved. The top-right figure shows the error between the reference and the achieved wheelspeed. The bottom left shows the controller signals. The black line is the maximum torque available, the red line is the open loop part, the blue line the feed-back part and the green line is the sum of the red and blue line. At 2.8 seconds the green line crosses the black line which means that there is not enough torque present anymore to maintain the desired slipratio. The bottom-

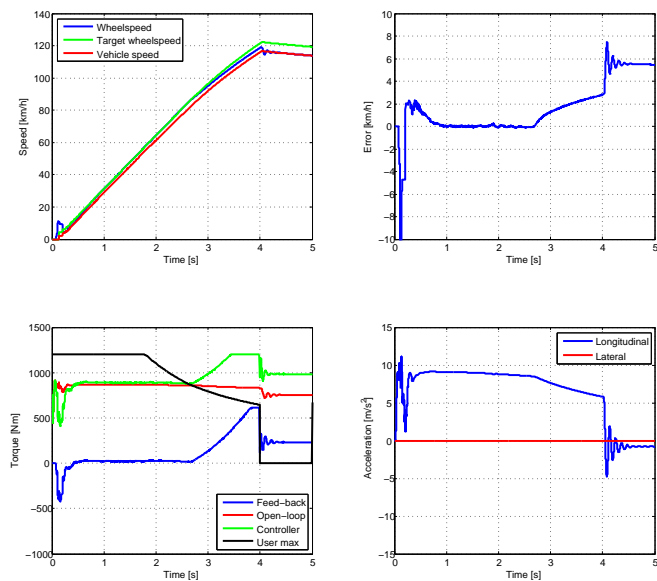


Figure 12: Simulation results with adequate friction estimate

right graph shows the longitudinal acceleration in blue. Some resonance is seen at close to standstill, this is the result of the limited resolution of the sensors at low speeds combined with the driveline resonance. At 2.8 seconds the acceleration decreases strongly which is result of the torque limit of the motor. When the throttle is released, the driveline resonates again.

Tests have been done with a friction estimation which is to high or to low. In case of a low friction estimation the error is small and the feed-back controller needs time to build up due to the integral action. This results in a 0-100 km/h time of 3.42 seconds. In the case of a friction estimation which is to high, the error is larger and the feed-back controller is able to decrease the torque fast. However, due to limited resolution of the wheel-speed sensors at low speeds this correction shows some overshoot. Overall the result is better with a 0-100 km/h time of 3.32 seconds, which is approximately the same as with an adequate friction estimate.

### Implementation in the vehicle

The controller is implemented in the URE07 by making use of a dSPACE Micro-AutoBox II (MABX). The MABX receives four wheelspeed signals and two throttle position signals. A host-pc can be connected to control the MABX via ethernet, but stand-alone operation is also possible. The MABX sends an analog throttle signal to the motor controller.

Step responses are used to estimate the limit of the feed-back proportional gain ( $K$ ) for the controller. Figure 13 shows the results of the step responses for different proportional gains. For gains greater than 50 the system tends to oscillate a lot upon application of the step. For a gain of 80 the system is unstable. Hence, a gain of 50 is used as a maximum.

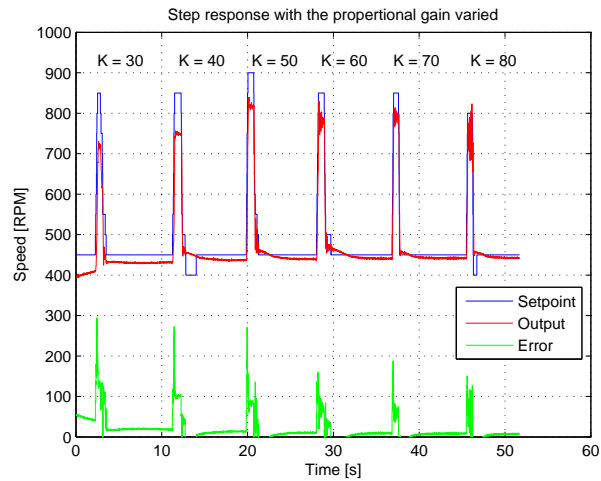


Figure 13: Step responses

On-track performance is the next step. The performance criteria are the time to reach a certain speed, the build-up of longitudinal acceleration and the amount of lateral acceleration during a straight-line acceleration of approximately 75 meters. Hereto, tests on a wet surface are performed. 4 runs without traction control are compared with 7 runs with traction control. Table 2 shows an overview of the results, the acceleration from standstill until 30, 50 and 80 km/h are compared. The bottom part of the table shows the mean times and the variance in the times. Until 30 km/h the gain in times is the highest with a 26.3% improvement. Until 80 km/h this gain reduces to a 17.3% improvement.

	Traction control off [s]			Traction control on [s]		
	0-30 km/h	0-50 km/h	0-80 km/h	0-30 km/h	0-50 km/h	0-80 km/h
	2.03	3.06	4.73	1.28	2.13	3.42
	1.82	2.67	4.09	1.39	2.20	3.49
	1.90	2.66	4.15	1.32	2.09	3.34
	3.28	4.40	6.14	1.32	2.06	3.30
				1.35	2.10	3.30
				1.15	1.91	3.21
				1.42	2.20	3.53
Mean	2.26	3.20	4.78	1.32	2.10	3.37
Variance	0.47	0.68	0.91	0.01	0.01	0.01
Improvement on mean time				26.3%	20.7%	17.3%

Table 2: Results without and with traction control



Figure 14 shows the throttle signals without and with traction control. In the right-hand graph the user (blue) applies full throttle and the output (red) is regulated by the system. The left hand figure shows that the driver takes approximately 1 second to apply the throttle and has to reduce the throttle again after  $t = 1$  second. From  $t = 2$  seconds the throttle remains somewhat constant. The right-hand figure shows a fast application of the throttle, again a reduction of the throttle, but a much more constant signal. Also, as shown in blue, the driver applies full throttle the entire time.

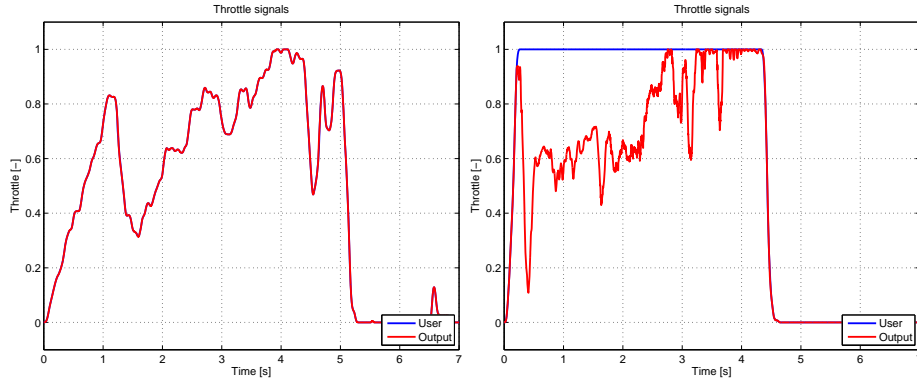


Figure 14: Throttle signals without and with traction control

Observation of the acceleration sensor data shows a faster build-up of the longitudinal acceleration is seen together with a higher RMS acceleration (0.59g vs 0.69g) over the whole run. Less lateral acceleration is seen (RMS 0.19g vs 0.10g) which indicates a more stable vehicle at higher speeds.

The measured wheel-speed signals are shown in Figure 15. A significant amount of wheel-spin is seen without traction control, where in the situation with traction control this is much more limited. From  $t = 0$  seconds the left-hand figure shows no wheel-slip until  $t = 1$  seconds. Suddenly the tyre enters the unstable region due to a high slip-ratio. The driver is able to compensate for this but this takes almost 1 second. Further along the rear wheel-slip increases again and the driveline resonance can be seen. This time the driver does not compensate for it and the tyre remains in the unstable region. In the right-hand figure the run starts with some wheel-slip which is corrected for immediately. Along the run some wheel-slip occurs but this is kept within boundaries.

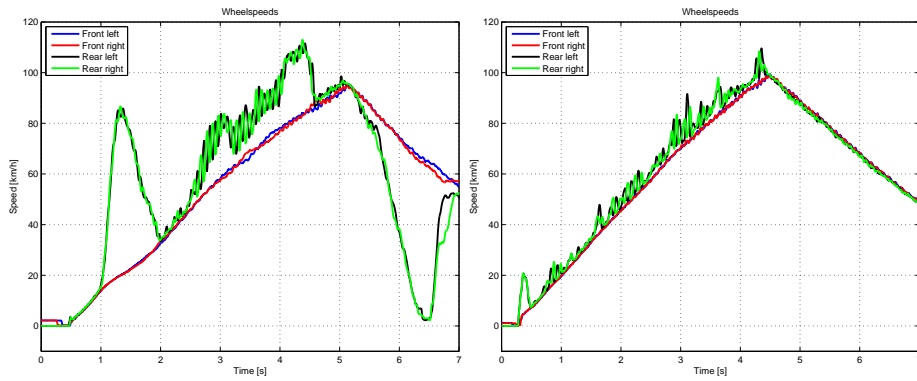


Figure 15: Wheel-speeds without and with traction control

## CONCLUSIONS AND RECOMMENDATIONS

A traction control system has been designed for a single-motor full electric Formula Student vehicle. This system is a combination of a user provided friction estimate and slip ratio control. Tyre measurements have been performed to obtain a tyre model and a driveline model has been validated to take account for the flexibility of the drive shafts and resulting resonance present.

Offline testing has been done by making use of a two-track vehicle model in combination with a non-linear tyre model and a flexible driveline. The wheel-speed sensors have been incorporated in this model as well. Simulations show that the controller improves the longitudinal acceleration of the vehicle. Roller bench testing has been done to verify the controller in the vehicle. These tests have been used as an indicator to estimate the maximum amount of feed-back gain. The found gain has been used on-track as well where no instability occurred. Higher feed-back gains resulted in instability, lower feed-back gains in less performance. An improvement of 26.3% was seen in the 0-30 km/h time and 17.3% in the 0-80 km/h time. This large performance gain indicates how important software in a vehicle these days is.

The proposed controller has not yet been tested in cornering situations, simulations show promising results here as well. Some instability for low vehicle speeds occurs; this can be solved by reducing the proportional feed-back gain for lower vehicle speeds, simulations show promising results here as well. Since electric vehicles have the ability to use regenerative braking, a similar approach could be used to regulate the wheel-slip while braking. In this way the maximum amount of energy can be recuperated, while keeping the vehicle stable.

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