Electric Power Train design for FSAE Electric Car and Study of that Performance Parameters

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Abstract
The aim of this work is to develop the power train of an FSAE electric car by modelling the vehicle on MATLAB/Simulink and then estimating the performance parameters such as acceleration (longitudinal and lateral), velocity (longitudinal and lateral), force (longitudinal and lateral), energy consumption (mechanical and electrical), power (mechanical and electrical), voltage drop and many more. These parameters are used to define endurance and acceleration capabilities of the vehicle as well as cornering properties. The future goal of this work is to estimate the different vehicle parameters and use these parameters to generate algorithms and design Electronic Differentials for the upcoming generations of FSAE vehicles.

Keywords: modelling, velocity, force, power

1. Introduction
Electronic differential is a technology used to drive a car with four individual motors on each wheel and then distribute the power electronically using driver inputs such as acceleration, braking, steering, etc. and then taking feedback from the suspensions system. This technology is very popular and effective in electric cars at FSAE and Formula E level but are not commercially acceptable in the light of high costs. Romani et al. explained in a vehicle, the powertrain consists of the main power generating and delivery components that deliver powers to the tires of the vehicle. The powertrain is a major component in all vehicles and can be found in cars, trucks, and buses irrespective of the kind of drive used.[1] The only difference in a power train is the use of electric components or a combustion engine. Vehicles running on fuels such as petrol, diesel, or CNG consist of a combustion engine that supplies power to drive the vehicle, whereas, in an electric vehicle, a motor is used to provide the required power. Zhang Y et al. stated that to reduce the dependence on fossil fuels and thereby lessen pollution, the automobile world has been looking for alternative ways to run vehicles for personal and commercial use.[2] Electric vehicles have gained visibility ever since a trend towards alternative energy has taken off. These vehicles reduce pollution and currently are the most eco-friendly option. Tuljapurk et al. explained power trains of Electric vehicles consist of a drive system which is powered by a battery pack. Typically, an AC motor is employed that is connected to a single-speed transmission. The energy is stored in chemical form in the battery pack of the vehicle. Control systems and power converters are used to perform functions such as DC/AC, AC/DC conversion, and voltage amplification. [3] Short et al. stated that in the modern era, an electric vehicle is expected to have characteristics such as high range, and high-performance motors that can provide an ample amount of power when needed.[4] However, the cost factor often outweighs the expectations of buyers, especially in the mass market.
Therefore, for a brand to gain acceptance, the vehicle must have all of the above features and should be competitively priced. Power trains of electric cars can be one of the most expensive components in the vehicle. Xiong et al explained that the power train not only delivers power to all wheels of the vehicle but also holds components such as the controllers and converters, motors, batteries, and cooling systems.[6] This often results in increased pricing and more complex systems. The need to reduce the complexity of power trains is ever rising. There have been major improvements in areas of design of the battery pack, motor, and power trains of electric vehicles as they are high-cost components. Kalmakov et al mentioned massive cost reductions have been made but are still not sufficient for mass-market acceptance. Other developments in electric vehicles have been the use of independent motors on each wheel known as an electronic differential. Modern Formula SAE(FSAE) and Formula E level cars employ this technology as it helps put power on the road more efficiently and effectively.[7] Such differentials take inputs from the driver such as acceleration, braking, steering, and feedback from the suspension system. All these inputs help in the distribution of power to each wheel depending on the amount of grip available. This technology has not been adopted by mass-market electric vehicle manufacturers due to its high costs and system complexity. Tey et al. explained improvements in acceleration, range, and endurance have also been a result of better and more efficient power train components. The battery pack of electric vehicles adds maximum weight due to the dense chemical energy stored in the battery.

2. Problems identified based on literature review

I. Motor efficiency.
II. Two motors used and coupled together caused trouble due to different efficiencies and misalignment with the gear box. Excessive bending stress on the shaft of the motor and gear box resulted in shaft failure even before it reached 5% of its lifecycle. Wobbling was another problem in one of the motor shafts.
III. Low torque constant, high rpm constant and lack of liquid cooling made the motors incompatible with our new cars.
IV. Improper gearbox mounting resulted in vibrations of high amplitude and excessive stress was induced due to vibrations on the rotating parts of the transmission system.
V. Power to weight ratio of the current motors was found to be very low. The two motors are capable of producing 64kW of power under proper cooling conditions, only for 5 seconds. A nominal power of 32kW can be generated with the two motors weighing 23kg without the mounts.
VI. The gearbox and differential together weigh around 20kgs.
VII. Theoretical top speed was found to be 90kph.

All the above reasons make the current design reliable only for short and long runs at constant speed without sudden acceleration and braking. These runs always require proper cooling.
3. Overview of Vehicle Model

The model mentioned above is designed using reverse engineering. It means that we have to provide the velocity profile of the track and radius of curvature corresponding to same time steps. The model will then take input and provide information to different blocks of vehicle dynamics equations which will then generate the required parameters and this model will run for the duration provided in the time slot on Simulink. The parameters are there after displayed in the graphical scope.

Note: The running time should be less than or equal to the input data time. For example- if we provide the velocity profile data for 60 seconds then the running time of model should be 60 seconds or less. Solver used in the model is ODE45 (Dormant Prince) and steps are variable steps (auto generated).
4. Longitudinal dynamics

The governing equation of the longitudinal dynamics is: \( F_w = F_a + F_g + F_r + F_i \)

Here, \( F_w \) = force at wheels (N)
\( F_a \) = Aerodynamic drag (N)
\( F_g \) = Gradient force (N)
\( F_r \) = Rolling resistance (N)
\( F_i \) = Inertial force (N)

In the above equation rolling resistance force is divided into four parts. This explains that each wheel has different rolling resistances due to variation in load and tire parameters.

![Figure 2. Schematic representation of longitudinal dynamics](image)

5. Load Transfer (Longitudinal)

The governing equation of the load transfer is for front wheels: \( N_f = \frac{M_f}{2} g - \frac{h}{2} (F_a + F_i) \)

The above equation is for front left and right wheel. Also gradient force considered 0 assuming 0° inclination in road surface will result in sin0=0.

The equation is then reframed and stated as: \( F_w = \frac{1}{2} A p C_d u^2 + M g \sin \theta + M g \mu r + \frac{M d u}{d t} \)

Here, Frontal area of the vehicle (m^2)
\( C_d \) - drag coefficient
\( M \) - Mass of the vehicle (Kg)
\( \rho \) - density of the air (Kg/m^3)
\( u \) - longitudinal velocity (m/s)

The governing equation for rear wheel is: \( N_r = \frac{M_r}{2} g + \frac{h}{L(F_a + F_i)} \)
The above equation is for rear left and right wheel.

Here, Msf- sprung mass front (Kg)
Msr- sprung mass rear (Kg)

h- C.G height from the ground (m)

L- Wheelbase (m)

Nf- Normal reaction front (N)
Nr- Normal reaction rear(N)

![Figure 3. Longitudinal load transfer profile](image)

The governing equations for this model are:

\[ F_{zs} = \frac{M_{sf} d^2 x}{dt^2} + C_s \left( \frac{dx}{dt} - \frac{dy}{dt} \right) + K_s (x - y) \]

\[ F_{zt} = \frac{M_u d^2 y}{dt^2} + C_s \left( \frac{dy}{dt} - \frac{dx}{dt} \right) + K_s (y - x) + K_t (y - z) \]

Here, Ms- sprung mass (Kg)

Mu- unsprung mass (Kg)

Cs- damping coefficient of spring (Ns/m)

Ks- Stiffness constant of spring (N/m)

Kt-stiffness of tire (variable) (N/m)

x- Sprung mass displacement (m)

y- Unsprung mass displacement (m)

z- Road profile (m)
Figure 4. Bicycle model of lateral dynamics

The variables Cs and Ks are replaced by Cf and Kf respectively for front suspensions. Similarly, Cr and Kr are for rear suspensions. The tire stiffness has been taken from the lookup table consisting of Load Vs vertical stiffness data of the Continental C16 tire. Note: The profile is assumed constant. This means no deviation and therefore value of z is not considered in the equation.

6. Lateral dynamics

Let the longitudinal velocity at some arbitrary time be constant, and assume lateral velocity of the vehicle is v and the vehicle is turning with steering angle \( \delta \) with the Wheelbase L and turning radius R.

When we talk about a car, the relation gets divided into parts due to different steering angles while cornering.

Therefore, the outer steering angle is given by \( \delta o = \frac{L}{R + \frac{r}{2}} \)

And the inner steering angle is \( \delta i = \frac{L}{R - \frac{r}{2}} \)

While turning, the vehicle produces yaw which means it has a rate due to variability w.r.t t, assume it to be r.

Therefore, yaw rate(r) is defined by: \( r = \frac{u}{R} \)

If we integrate this w.r.t time we will get an angle known as body slip/sideslip angle represented by \( \beta \):

\[ \beta = \int r \, dt \]

From trigonometry we can say that: \( tan \beta = \frac{v}{u} \)

And yaw acceleration and lateral acceleration are the 1st derivatives of yaw rate and lateral velocity. The governing equations of lateral dynamics derived from Bicycle model are:
\[ F_y = \left( -\frac{a}{u} Ca_f + \frac{b}{u} Ca_r \right) r - (Ca_f + Ca_r) \tan \beta + Ca_f \cdot \gamma \]
\[ M_z = \left( -\frac{a^2}{u} Ca_f - \frac{b^2}{u} Ca_r \right) r - (aCa_f - bCa_r) \beta + aCa_f \cdot \gamma \]

Here, \( F_y \)- Lateral force acting at C.G (N)
\( Ca_f \)- Lateral stiffness of front tire (N/°)
\( Ca_r \)- Lateral stiffness of rear tire (N/°)
\( r \)- Yaw rate (rad/s)
\( \beta \)-body slip angle (rad)
\( a \)-C.G distance from front axle (m)
\( b \)-C.G distance from rear axle (m)
\( \gamma \)-steering angle (rad)
\( u \)-longitudinal velocity (m/s)

\[ F_y = M \left( \frac{dv}{dt} + ru \right) \]

The above lateral force equation will be equated with the Bicycle model equation. From this we get \( \gamma \) (steering angle). The steering angle value has been used in the following equations:
Front slip angle, \( \alpha_f = \tan^{-1} \left( \frac{ar}{u} + \tan \beta - \gamma \right) \)

Rear slip angle, \( \alpha_r = \tan^{-1} \left( \tan \beta - \frac{br}{u} \right) \)

\[ k = \frac{M(bCa_r - aCa_f)u^2}{L^2Ca_f Ca_r} \]

Along with calculation of Yaw rate, Steering angle and other parameters

7. Tires

Continental C16 tires are used by FSAE teams. The Formula used in modelling of tire is Magic formula 5.2.
The modelling is divided into four major parts: Pure lateral slip, Pure longitudinal slip, combined longitudinal slip, Combined lateral slip.

8. Powertrain modelling

The power train is modelled as the output of longitudinal dynamics block. Tractive force is then used as the input in the equations:
\[ Tm = \frac{Fw \cdot \eta_t}{i} \]

From this, we get torque required by the motor. This torque is then converted into current by dividing it by torque constant.
\[ I = Tm/Tc \]
Here, \( Tm \)- torque from motor (Nm)
\( Fw \)- Tractive force (Nm)
\( \eta_t \)- total efficiency of the drive line
\( r \)- radius of the wheel (m)
i-gear ratio
9. Results and discussion

Results have been generated to design a battery pack. The parameters determined are: Cell configuration-72s9p Max torque at wheels- 600Nm Max speed- 120kph Nominal Energy- 6.318kWh at 3.9V across each cell Endurance run- 23.7kms, if operated between 45 to 70 kph on track. 57kms, if operated at 60 kph constantly on a straight track. Peak Energy- 6.795 kWh at 4.2 V across each cell. Endurance run- 25.5km at variable speed if operated between 45 to 70 kph on track.

10. Conclusions

An electric power train was successfully designed as per FSAE and Formula E parameters and designs using MATLAB Simulink. The power train was tested to study the required parameters, and the following conclusions were drawn:
1. 5% increase in overall acceleration was noted
2. An increase of 15% in the endurance capacity of the power train was noted as compared to the ones previously used.

Other vehicle parameters tested provided promising results and showed a greater scope for improvement in the field of electric vehicle powertrains.
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