

Electric Vehicle Energy Consumption Simulation by Modeling the Efficiency of Driveline Components

Mohammad Gerami Tehrani
Lappeenranta University of Technology

Juuso Kelkka
Valmet Automotive Inc

Jussi Sopanen, Aki Mikkola, and Kimmo Kerkkänen
Lappeenranta University of Technology

ABSTRACT

The feasibility of improving the energy efficiency of Electric Vehicles (EV) by manipulating operation points by means of a variable transmission is investigated with an efficient mathematical model of power losses in all driveline components. Introduced model can be solved in real-time making it possible to embed it to a control scheme of EV. Empirical test results are employed to derive the efficiency of the power electronics and electric motor at operation points while the mechanical power losses are predicted by a comprehensive and generic formulation for efficiency analysis. The simulation model used comprises electrical component efficiency, drivetrain inertias, gearbox efficiency, regenerative braking, and gear ratio selection. Three different transmission types are studied in this work; a single reduction gear, a five-step gearbox and an Infinitely Variable Transmission.

CITATION: Gerami Tehrani, M., Kelkka, J., Sopanen, J., Mikkola, A. et al., "Electric Vehicle Energy Consumption Simulation by Modeling the Efficiency of Driveline Components," *SAE Int. J. Commer. Veh.* 9(1):2016, doi:10.4271/2016-01-9016.

INTRODUCTION

Power consumption in Electric Vehicles (EV) is a major concern in driveline architecture design as it affects the life cycle and trip range of the EV. Permanent Magnet Synchronous Motors (PMSM) are commonly applied in electric driveline design because of their ability to deliver high torque and their compact size as well as constant power at high speeds. Furthermore, since the traction control method in EVs is Direct Torque Control (DTC), PMSM can be considered an optimal option because they can be controlled fast and efficiently [1]. Unlike Internal Combustion Engines (ICE), where maximum torque and power are produced at high rotation speed, in PMSM maximum torque is available even at the lowest rotation speeds allowing the power production to be constant.

In most EVs currently available, variable transmission is omitted from the powertrain because, from the performance point of view, the output power of PMSM is constant while the output torque is fixed. However, EV powertrain efficiency is a significant factor that cannot be neglected. A technical and financial comparison of conventional and equivalent hybrid and fully electric vehicles in different classes is presented for the Australian market in [2][3].

In parallel hybrid EVs, use of a combination of two or more propulsion sources permits management of the operation of the electric motor and ICE at optimal efficiency [4]. By contrast, in series

hybrid and all-electric drivelines, the electric motor has to spin at the corresponding road speed regardless of the demanded torque. In view of this characteristic of PMSM, conventional transmission is not required because the electric motor can provide similar power at different speeds. However, electric motor efficiency is not homogeneous over equivalent power points, which means that although the same power can be achieved by different torque-speed combinations, the efficiency can vary by up to 30% [5]. Improving the electric motor efficiency (without consideration of auxiliary components) by shifting the operation point along constant power curves by means of a variable transmission is studied in [6][7][8].

Definitive conclusions on the overall efficiency of the driveline cannot be drawn if power losses from the transmission, e.g. the effect of friction, are not taken into account. Thus, a precise model of the gearbox is needed to be able to predict power dissipation.

With this goal in mind, the study by [9] can be taken as a platform for development of a gearbox model and the friction coefficient formulation proposed by [10] applied. In the power loss prediction model developed by [9], the friction coefficient is considered constant while in more recent studies, the friction coefficient value varies according to the applied torque and operation speed. To be able to monitor how power losses due to downstream components, the driveline compromise the total efficiency and trip range of an EV

requires an agile mathematical model that predicts both electrical and mechanical efficiency instantaneously [11]. Such a model would enable assessment of the feasibility of applying a variable transmission in an EV driveline [12].

To overcome power dissipation in mechanical linkages and electrical components, the amount of power required in each driveline component should be somewhat higher relative to the previous stage. In other words, every component in the driveline flow chart has a value for its efficiency which compensates the losses while transmitting the power. The total efficiency of the driveline is a multiplication of the efficiency of the mechanical and electrical components.

The efficiency drop occurs principally in the electric machines, e.g. in the electric motors in which the electric power is converted to mechanical power. In electric machines, where the electric power - a product of current I and voltage U - is converted to the mechanical power - torque τ times angular velocity ω - some portion of power dissipates via both electrical and mechanical components.

In order to calculate the total efficiency, power extracted from the batteries compared to power exerted on the wheels should be derived, for both mechanical and electrical efficiency, at any arbitrary operation point. Any proposed modification to improve total efficiency should consider all factors in parallel [13]. For example, minimizing the power loss in the power electronics, including both switching and conduction losses, while modifying electric drive efficiency according to the operation point may improve the driving range [14]. However, optimizing the driving range parameter may compromise the efficiency of other stages, as is clear from discontinuous efficiency waterfall modeling.

Another component that has a determinant effect on driveline efficiency is the gearbox through which the power is transmitted while the torque and speed vary. In the context of this study, unlimited combinations of torque and angular velocity are possible in selection of a specific amount of power, whereas the efficiency of the electrical or mechanical systems fluctuates, with one or the other system showing greater efficiency. Thus, the operation point at which total efficiency of the driveline results in the highest value determines the optimum operational point from an efficiency perspective.

The objective of this study is to assess the feasibility of implementing a variable transmission in EV driveline architecture, from the efficiency point of view, by developing an efficiency model for the gearbox that is a function of torque and angular speed to predict power losses in accordance with the operating point of the electric motor. The efficiency maps of electric components such as the power electronics and the electric motor are taken from manufacturers' empirical test results and the power losses in the gearbox are predicted by formulation in [9]. The power losses resulting from components that function in the same way in the driveline whether a variable gearbox or a constant reduction gear is included, are considered as constant.

The rest of the paper is constructed as follows. The next section considers theoretical aspects, in particular mechanical losses, which are divided into load-dependent losses and spin losses. Section 3 presents the simulation model, which is a comprehensive EV driveline model built by applying a novel real-time efficiency calculator for the gearbox. The proposed generic model is not only compatible with all kinds of geartrain but can also be used for Continuously Variable Transmission (CVT) and Infinitely Variable Transmission (IVT), thus enabling power consumption comparisons between different transmission configurations. The studied structure is presented in section 4. Section 5 presents the simulation results, which are analyzed with the aim of finding optimal performance points and assessing the feasibility of adding a variable transmission to the EV driveline. The paper ends with a short section giving conclusions of the study.

GEARBOX MODELING

In mechanical power transmission by means of a geartrain, gear teeth deliver the power by rolling and sliding over each other. Although tooth surfaces look plain and fully burnished in macro view, they are ragged surfaces whose asperities resist sliding. As a result, a portion of power is lost as heat, wear and noise.

Losses due to friction in support bearings and the gear mesh are called load-dependent losses, and losses that come from air resistance and the lubricant used are termed spin losses. The challenge in formulation of load-dependent losses comes from the need to derive a friction coefficient that varies during the mesh cycle. In Coulomb's law, the friction coefficient (μ) is a constant value and the resistive force is dependent on normal force variation. In gear tooth pairing, the friction coefficient varies according to the mesh cycle sequence. For this reason, the time dependent method for calculating the friction coefficient proposed by [10] is applied in this work in a modified version of Coulomb's law for friction by [15] that was developed for estimating the resistive force. The methodology developed by [16] is applied in calculation of spin losses, which are divided into oil churning and windage power losses, which represent power losses due to the interaction of individual gears with lubrication fluid and pumping of oil at the gear mesh.

In order to have a model that considers both rolling and sliding interaction between the gear teeth, a modification of Coulomb's law is applied to obtain the equivalent kinetic friction coefficient. Resistive frictional torque in the supporting bearings is also considered based on the construction of load carrying shafts and gears in the gearbox. The load-dependent power losses are defined as a function of rotating speed and applied torque, while spin losses vary by rotational speed.

For smoother and more efficient operation, a gear box needs to be lubricated. The thickness of the formed lubricant film that eases the rolling and sliding movements of parts is dependent on the applied pressure between the solid surfaces. At excessive pressures and point contacts, the lubricant film can become ruptured, so in order to select a suitable lubricant, the mean operational pressure needs to be calculated. There are three types of solid-to-solid sliding condition: dry sliding, fully lubricated sliding and semi-lubricated sliding.

Although using lubricant fluid with high viscosity results in thicker films and decreases the friction coefficient, it increases the rolling (pumping) resistance [17].

By considering the lubricant as a Newtonian fluid, the rolling losses (P_R) due to hydrodynamic resistance can be categorized as oil drag and oil pumping losses, both of which are dependent on the lubrication type, i.e. whether an oil jet or oil bath is used.

Rolling (or pumping) loss is the power required to draw and compress the lubricant to form a pressurized oil film that separates the gear teeth in order to make the contact surface smoother and, thus, reduce friction. At light loads, the rolling loss is a major portion of the system losses.

A number of friction models have been proposed for calculation of the friction coefficient, such as the Coulomb Model, the Benedict and Kelley Model, Xu's Full Model and the Smoothened Coulomb Model, based on Anderson and Loewenthal, it is clear that the friction coefficient is crucial in calculation of sliding power losses (P_S). In this work, the formulation suggested by Xu [10] is utilized for calculation of the friction coefficient, and the friction type is assumed to be fully lubricated in all cases.

The load dependent power losses are mainly divided into sliding losses and rolling losses. The sliding power losses can be derived as:

$$P_S = C_{th} V_S F_S \quad (1)$$

where C_{th} is the thermal correction coefficient, V_S is tooth surface sliding speed and F_S is the sliding force.

The Coulomb's law of friction can be used to define the resistive force between two involute spur gear teeth:

$$F_S = \mu W \quad (2)$$

where μ is the friction coefficient, and W is the normal load on the sliding surfaces.

As mentioned earlier, μ is calculated according to Xu's method [10]:

$$\mu = e^f P_h^{b_2} |SR|^{b_3} V_e^{b_6} v^{b_7} R^{b_8} \quad (3)$$

where f is a function of the gear tooth's relative speed, P_h is the relative maximum Hertzian pressure, SR is the slide-to-roll ratio around the gear tooth mesh point, V_e is the linear speed of the entering tooth, v is dynamic viscosity, R is the gear tooth equivalent radii of curvature, and b_i are constant coefficients that can be defined manually according to the lubrication condition or based on running an elastohydrodynamic lubrication (EHL) simulation [10].

Simultaneous rolling and sliding movements take place between the

tooth flanks of two mating gears, except at the pitch point, where pure rolling takes place. As explained in [10], three different regions can be roughly defined on a μ versus SR curve. In this study, in order to simplify the friction coefficient formulation and hasten the simulation process the mesh cycle is discretized into seven points which are critical points and friction coefficient is calculated at those points only. The trend of discretized mesh cycle conforms to continuous models in [10].

As it can be seen in Figure 1, when the sliding velocity is zero, there is no sliding friction, and only rolling friction (though very small) exists. Thus, the value of the friction coefficient should be almost zero at the pitch point. When the slide-to-roll ratio, SR , increases from zero, the friction coefficient, first increases linearly with small values of SR . This region is defined as the linear or isothermal region. When the SR increases further, μ reaches a maximum value and then decreases as the SR value increases beyond that point. This region is referred to as the nonlinear or non-Newtonian region. As SR increases still further, the friction decreases in an almost linear fashion; the thermal region [18].

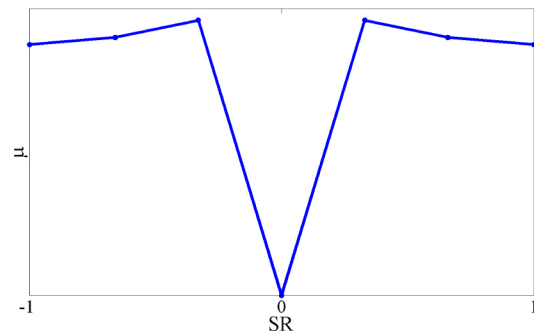


Figure 1. Friction coefficient variation over mesh cycle

Power losses due to rolling can be obtained as follows:

$$P_R = C_{th} V_R F_R \quad (4)$$

where V_R is gear tooth rolling speed and F_R is the rolling load at the gear tooth [19].

Compression from the pressure of one tooth on another tries to eliminate the oil film between the teeth. Although the lubricant film stretches because of molecular cohesion (viscosity), it also resists, thus behaving like a spring. A portion of this stress is converted to heat and the remainder breaks the molecular bonds of the oil [20]. Consequently, film thickness is the main factor determining the behavior of the lubricant and losses and wear in gear meshing.

In order to derive average power losses during the mesh cycle for both sliding and rolling losses, discretized integration is done in each sequence. As illustrated in Figure 2, the mesh cycle is split into three sequences: before the mesh point, at the mesh point and after the mesh point.