

Expanded Human Choice based on Duty/Demand Cycles for In-Wheel Motors in Electric Vehicles

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ABSTRACT

In order to design the in-wheel motor (IWM) for Electric Vehicles (EV), it is necessary to analyze the desired (expected) duty cycle at a higher performance level in order that the IWM becomes commercially relevant. The duty cycle may be representative of different segments of the customer base. Or, the individual customer may wish to have a set of IWMs that uniquely meet his/her measured "demand" cycle for a balance of drivability and efficiency. Questions then arise: How to measure the demand cycle of an individual? What 2 or 3 standard duty cycles should be offered as customer choices for their vehicle? Should the IWM represent multiple speed domains to enhance efficiency and drivability? Can the vehicle be updated rapidly 2 to 3 years after purchase? Etc. In this paper, we lay the groundwork to answer these types of customer questions for an EV with four independent IWMs. In addition, we propose the customer-oriented duty cycle analysis in order to obtain not only how to maximize efficiency (efficiency-priority driver) but also how to maximize drivability (aggressive driver), based on the existing driving cycles: Urban Dynamometer Driving Schedule (UDDS) and Aggressive Driver (US06). Given the individual demand cycle, the EV can be tailored to meet the particular customer, which leads to expanded human choice which can be characterized in terms of two basic operating regimes: efficiency and drivability.

INDEX TERMS: Duty Cycle Analysis, In-Wheel Motors, Efficiency, Drivability, Demand Cycle

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I. INTRODUCTION

Optimal sizing of IWMs, choice of battery and capacity, development of controllers, and realistic charging scenarios requires a discussion of vehicle duty cycles. In this paper, the demand cycle refers to the individual's driving history and can be described by a speed versus time curve. The duty cycle refers to a vehicle's history of power usage and the manufacturer can use it to design the drive wheel actuator. As a result, the components of the IWM will be sized to meet the selected duty cycle [1, 2].

For instance, an aggressive driver might want 0-60 mph acceleration time in 5 s, but an efficiency-priority driver will want high efficiency instead of the quick 0-60 mph acceleration time. The demand cycle, depending on the customer, will be determined by the driver history's speed versus time curve. Consequently, manufacturers can configure specific driving cycles for each particular customer. Based on their

driving cycles, the IWM will be customized to that particular customer. This could lead to a more optimized IWM so that the customer can be best satisfied with their purchase. It is discussed in detail how to evaluate, classify, and satisfy these individual customers. In addition, the expanded choice for the customer can be characterized in terms of two basic operating regimes: drivability and efficiency. It is analytically demonstrated how the selection of the design components of IWMs differs for different types of customers such as an aggressive driver vs. an efficiency-priority driver, and describe design specifications such as different g levels, gear ratio, power rating, weight of the IWM, clutch shift point, efficiency, 0-60 acceleration time etc.

In this paper, the longitudinal vehicle dynamics are presented. Based on that, the customer-oriented duty cycle analysis is developed. After that, simulation results and conclusions are presented.

II. LONGITUDINAL VEHICLE DYNAMICS

A passenger vehicle can be analyzed as a system consisting of one vehicle body (sprung mass) and four wheels (unsprung mass), which are IWMs in this case. The vehicle can be modeled as interactions among five bodies comprising sprung and unsprung masses [3].

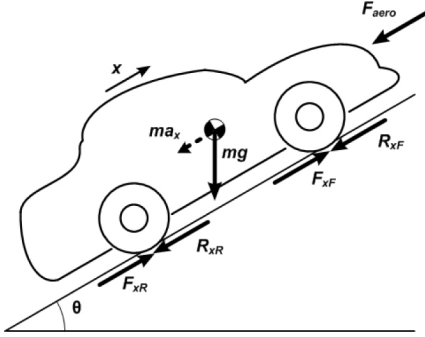


Figure 1. Forces acting on a two-axle vehicle

As shown in Figure 1, the forces act on a vehicle moving on an inclined road. The equation of motion can be expressed by the following [4]:

$$\sum F_{longitudinal} = m_{eg} a_x; \quad F_t - \left(F_{aero} + F_{rolling} + F_{grade} \right) = m_{eg} a_x \quad (1)$$

$$\text{where } F_t = F_{xR} + F_{xF}, \quad F_{aero} = \frac{1}{2} \rho_a A_f C_d (v_v + v_w)^2$$

$$F_{rolling} = R_{xR} + R_{xF} = C_r mg \cos \theta, \quad F_{grade} = mg \sin \theta$$

$$m_{eg} = m_v + N_w \left(\frac{I_w}{r_w^2} + g_r^2 \frac{I_m}{\eta_g r_w^2} \right)$$

The dynamic equation of motion along a vehicle's longitudinal direction is influenced by traction forces, aerodynamic drag resistance, rolling resistance, and grade resistance. The sum of aerodynamic drag resistance (F_{aero}), rolling resistance ($F_{rolling}$), and grade resistance (F_{grade}) is referred to as "road load," which is a minimal force on a vehicle moving on a road. For example, during cruising (i.e., acceleration $a_x = 0$), traction force (F_t) given in terms of the slip between the tire and the road is equal to the road load, which consists of front traction force (F_{xF}) and rear traction force (F_{xR}). Similarly, rolling resistance ($F_{rolling}$) consists of front rolling resistance (R_{xF}) and rear rolling resistance (R_{xR}). The difference between road loads and the traction force make it possible to accelerate or decelerate the vehicle [4]. The total equivalent mass (m_{eg}) is the sum of vehicle mass (m_v) and equivalent mass of the rotating parts of each wheel [5].

From Equation(1), the traction force can be expressed by:

$$F_t = m_{eg} a_{car} + F_{aero} + F_{rolling} + F_{grade} \quad (2)$$

Given a Electric Vehicle (EV) equipped with four-independent IWMs, the required motor torque can be obtained by [6, 7]:

$$\tau_m = \frac{r_w F_t}{N_w g_r \eta_g} = \frac{r_w}{N_w g_r \eta_g} (m_{eg} a_{car} + F_{aero} + F_{rolling} + F_{grade}) \quad (3)$$

Where r_w is the radius of the wheel, N_w is the number of wheels, g_r is the gear ratio, and η_g is the gear efficiency. With respect to each IWM, mechanical power from the motor is transmitted through the gear train to the tire. The motor power required to accelerate a vehicle is given by:

$$P_m = \left(\frac{2\pi n_w}{60} \right) \times \frac{r_w}{N_w g_r \eta_g} (m_{eg} a_{car} + F_{aero} + F_{rolling} + F_{grade}) \quad (4)$$

The Multi-speed hub Drive Wheel (MDW) suggested by the Robotics Research Group at the University of Texas at Austin (UTA) is designed to improve human choice by achieving the desired drivability and efficiency so that customer requirements can be met at the time of purchase for an open architecture EV which would then be assembled on demand. Since the MDW has 4 distinct speeds (two electrical and two mechanical), the g (acceleration) level will have 4 different levels. Four distinct acceleration (g) levels associated with four distinct speed ranges will be g_1 (0 - 10 mph), g_2 (11 - 20 mph), g_3 (21 - 40 mph), and g_4 (41 mph - 70 mph). The clutch shift occurs at 20 mph from 49-to-1 (for low speed and high torque) to 14-to-1 (for high speed and low torque) for two mechanical speeds [8].

III. CUSTOMER-ORIENTED DUTY CYCLE ANALYSIS

It is demonstrated how the selection of IWM matches to the driver whose driving cycles are UDDS and US06. These driving cycles would be based on an average driving style. The individual demand cycle can be defined as the driving cycle associated with a particular customer[9]. Customer's demand cycle is critical for their desired drivability and efficiency. The question arises: how to measure a customer? how to classify a customer?, and how to satisfy a customer?.

To address these problems, an automobile company might provide a customer with smart phone application to obtain the individual demand cycles. The customer runs the application on his phone when he drives his current vehicle for a period of day for two weeks. Whenever he drives, the app records the velocity and acceleration of the car. After that, a third-party "software application" analyzes specific duty cycle for that driver. Consequently, the resulting customer's duty cycle can be used to predict what the customer wants, and thus the EV will be customized to suit the customer's demand style, so that the customer can be best satisfied with their purchase [8].

Since, at this time, the individual demand cycles are not available, assumption is that the standard duty cycle is given by: Urban Dynamometer Driving Schedule (UDDS) and Aggressive Driver (US06). In this paper, the maximum vehicle speed is determined by the maximum speed of the traction motor. Therefore, in this design the maximum speed of the IWM is constrained by 70 mph (i.e., Switched Reluctance Motor (SRM) motor speed 13000 ~ 15000 RPM). According to this constraint [10], if the motor operates above 15000 RPM (which can be regarded as high speed), it is important to evaluate the effects of hoop stress, inertia, and critical speed.

A. Standard Duty Cycles

The duty cycles are used to assess vehicle fuel economy and emissions. Moreover, in order to develop the automotive power trains, they are used as a valuable design evaluation tool. EPA dynamometer driving schedules list 17 types of road driving cycles [11]. Two driving cycles are selected to simulate the vehicle performance such as UDDS and US06, assuming that each cycle corresponds to efficiency-driver and aggressive-driver, as shown in Figure 2 (a) and (b), respectively

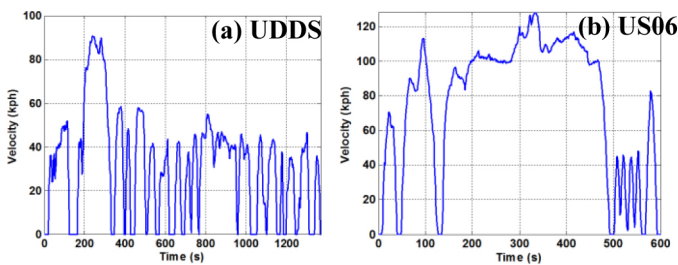


Figure 2. Standard duty cycles

Figure 2 shows the existing duty cycles. The UDDS is called “LA4” or the “city test”, and represents a city duty cycle with continuous stop-start operation. The US06 represents a high acceleration driving schedule which can be used for an aggressive driver. Assuming

B. Total Wheel Torque and Power

The total wheel power required to accelerate a vehicle is the product of the wheel speed (ω_w) and total wheel torque (τ_{tw}) given by:

$$P_{tw} = \omega_w \times \tau_{tw} = \left(\frac{2\pi n}{60} \right) \times (F_t r_w) \quad (5)$$

After substituting traction force (F_t) from Equation (1) into (5), the above equation becomes:

$$P_{tw} = \left(\frac{2\pi n}{60} \right) \times (m_{eg} a_{car} + F_{aero} + F_{rolling} + F_{grade}) r_w \quad (6)$$

The total wheel power required is simulated given Equation (6)

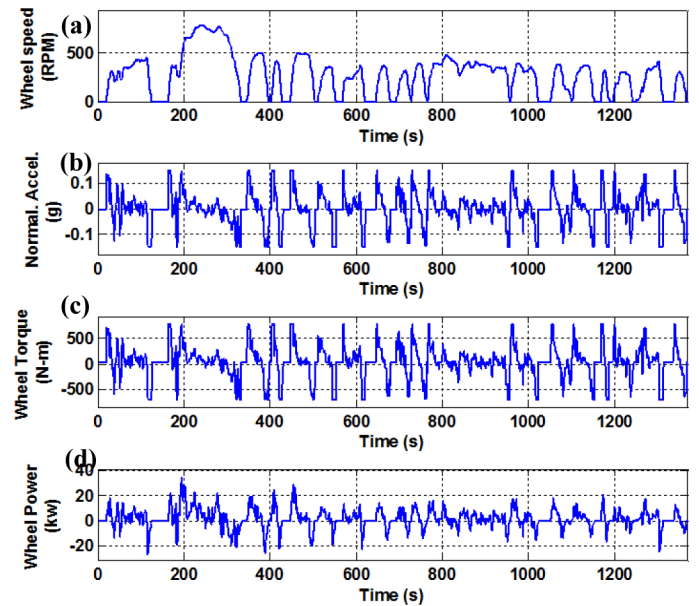


Figure 3. Total wheel torque and power requirement derived from a duty cycle (UDDS)

Figure 3 (a) - (d) shows wheel speed, normalized acceleration, total wheel torque, and total wheel power, respectively. From the relation between normalized acceleration and corresponding total wheel torque/power, it is concluded that the peak torque and power are strongly associated with acceleration events.

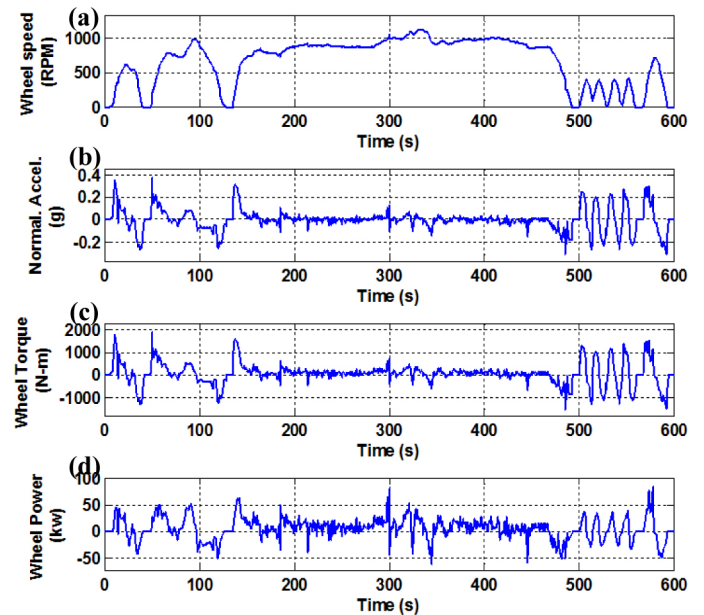


Figure 4. Total wheel torque and power requirement derived from a duty cycle (US06)

A US06 duty cycle is translated into wheel speed, normalized acceleration, total wheel torque, and total wheel power requirement, as shown in Figure 4 (a) - (d), respectively. The maximum normalized acceleration is around 0.35 g. Clearly, it is evident that

the instantaneous torque and power are strongly dependent on acceleration events. The variation of total wheel torque and power shows how the aggressive driver behaves and what he / she expects (wants).

C. Speed-Acceleration Frequency Distribution

The required information about the time proportions of individual duty cycles can be obtained from the Speed-Acceleration Frequency Distribution (SAFD) [1]. In addition, this plot provides a visual picture of each duty cycle, and hence, this 3D map can illustrate whether the duty cycle is biased in terms of speed or acceleration regions.

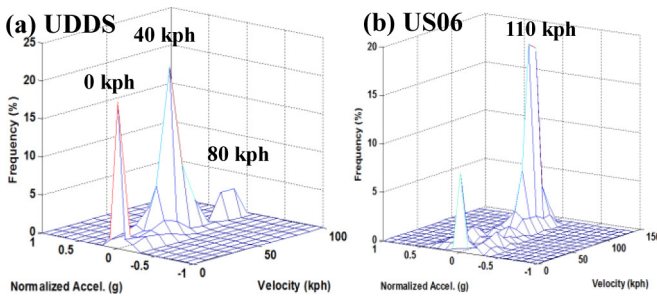


Figure 5. Speed-acceleration frequency distribution plot

Figure 5 (a) and (b) show 3D maps of speed-acceleration frequency distribution with respect to normalized acceleration level and velocity in terms of UDDS and US06. This map helps us to understand how the duty cycle is biased. In other words, this map describes the range and distribution of time spent at different speeds and acceleration. For instance, duty cycles regarding urban driving conditions represent a city cycle with continuous stop-go operation. With regard to UDDS, it is apparent that there is a bias around 0 kph due to a city cycle with continuous stop-start operation, around 40 and 80 kph. The peak acceleration and deceleration are around 0.15 g, as shown in Figure 5 (a). Compared to UDDS, Figure 5 (b) US06 has much higher rates of acceleration and deceleration. Peak acceleration and deceleration is around 0.35 g and a peak deceleration of 0.3 g, respectively. This is a characteristic of an aggressive driver. In addition, each operating points are shown in Figure 6 (a) and (c) in detail.

D. Wheel Torque-Speed

Torque-speed characteristics play an important role in designing the proper motor power rating, which is the primary consideration with respect to performance specifications. Figure 6 (a) and (b) show the total wheel torque vs. speed scatter plot and the number of occurrence vs. total wheel torque histogram, respectively, for the UDDS driving cycle. As can be seen in Figure 6 (a), each circle is the operating point associated with the UDDS duty cycle. As seen at symbol 'A' and 'B' in Figure 6 (a), two significant findings are highlighted. First, the symbol 'A' (40 kph (25 mph)) represents characteristics of downtown driving (i.e., stop and go operation). The second, symbol 'B' (80 kph (50 mph)) indicates that the operating points move away from the center due to the quadratic aerodynamic drag force as wheel

speed increases. In order to find the continuous torque and peak torque specifications, it is necessary to obtain a histogram in terms of the number of occurrences vs. total wheel torque, as shown in Figure 6 (b). The majority of the energy consumed corresponds to those occurrences that occur at low torque levels between 0 and 100 N-m. Given four-independent drive wheels, the total continuous wheel torque is 400 N-m (i.e., 100 N-m per wheel) and total peak wheel torque is 1000 N-m (i.e., 250 N-m per wheel) [12, 13].

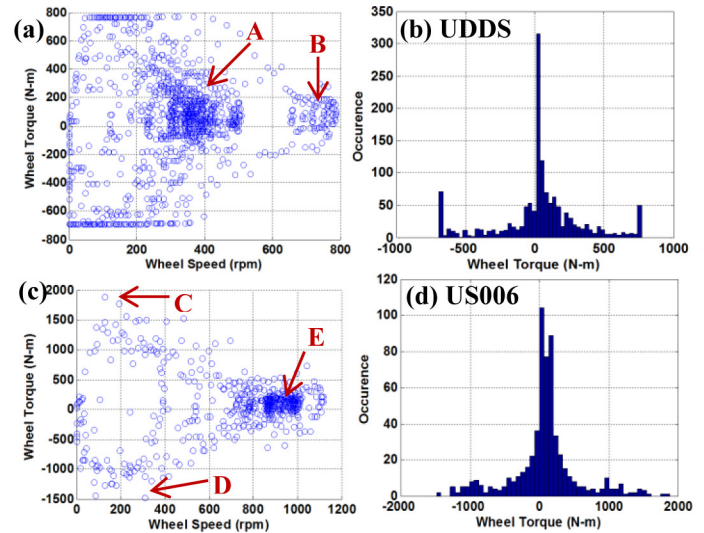


Figure 6. Total wheel torque-speed scatter plots and histogram

Regarding the US06 driving cycle, Figure 6 (c) and (d) show the total wheel torque-speed scatter plot and the number of occurrences vs. total wheel torque histogram, respectively. As seen at symbols 'C', 'D', and 'E' in Figure 6 (c), three significant findings are highlighted. First, symbol 'C' indicates that high acceleration levels for the aggressive driver occur at low speed, and are relatively short in duration. Second, symbol 'D' indicates that high deceleration levels for the aggressive driver also occur at low speeds. Third, the symbol 'E' indicates that the operating points move away from the center due to the aerodynamic drag force. In other words, during steady state speed, it required the minimum wheel torque. As shown in Figure 6 (d), the majority of the energy consumed occurs at low torque levels between 0 and 400 N-m wheel torque. Given four independent drive wheels, the total continuous wheel torque is 1600 N-m (i.e., 400 N-m per wheel) and total peak wheel torque is 2400 N-m (i.e., 600 N-m per wheel). The peak wheel torque is limited by the power rating of the IWM.

E. Motor Output Torque and Power

The IWM designed by UTA has four distinct speeds (two electrical and two mechanical) in order to improve efficiency and enhance drivability such as acceleration, braking, and climbing a hill on the command of the operator. One controller configuration operates the g_1 and g_3 regimes at low power while the other controller configuration operates the g_2 and g_4 regimes at higher power for the two electrical speeds. Hence, the IWM has four choices regarding efficiency and drivability. [8].