DEVELOPMENT OF TRANSMISSION SPECIFICATIONS FOR AN ELECTRIC VEHICLE

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At present, a great deal of efforts are on to develop personal modes of transportation such as an electric vehicle (EV) with potentially a higher degree of sustainability as compared to a conventional IC (internal combustion) engine powered vehicle. A transmission system with torque amplification capabilities is essential for an IC engine-based vehicle as, at low rpm, such engines develop extremely low torques which are not sufficient to move a vehicle from rest. However this is not the case for a DC (direct current) motor powered electric vehicle in which the motor intrinsically generates a high torque at rest conditions and delivers maximum power over a wide range of speed which can be well within the operational requirement of a vehicle. Thus, by varying the armature speed of a DC motor with an electronic controller and maintaining a uniform gear ratio, its drivability can be achieved. As most small EVs using DC traction motors are equipped with a single gear transmission system, it may appear that multi-speed gears are redundant for such vehicles. The current study dispels this myth by showing that a two-speed gear box can actually improve the functionality of a DC traction motor-based EV by increasing the rated top speed, initial acceleration and gradeability. In the process, a systematic methodology is developed for estimating traction requirements when a vehicle is propelled into motion from rest and remains in accelerated or steady state conditions subsequently. Together with this, a step-by-step approach has been devised for arriving at the specifications for a two-speed gear box taking into account speed-dependent motor torque and power characteristics.

Keywords: Electric Vehicle, Drivetrain.

1. INTRODUCTION

Currently, the development of more sustainable transportation systems is receiving a lot of attention. Due to the problems of global warming and limited fossil fuels, electric vehicles (EVs) have the potential to replace conventional internal combustion (IC) engine powered vehicle as a viable mode of personal transportation, as EVs do not consume fossil fuel and do not release greenhouse gases during operation. Basically, an electric vehicle (EV) differs in energy storage and power-train from an IC engine powered vehicle.

Since an IC engine is not capable of propelling a vehicle from rest on its own on account of generating low torques at low rpm, a torque amplifying transmission system is very much required for an IC engine driven vehicle. For such vehicles, transmission is also required to allow a vehicle to stop by disconnecting the drive and change the speed ratio between engine and wheels whenever required [1]. On the other hand, this is not the case for an electric vehicle powered by a DC (direct current) motor, since the motor inherently generates a high torque at rest conditions and low rpms, and also delivers maximum power over a wide range of speed. Thus, by varying the speed of motor using an electronic controller and maintaining a constant transmission ratio, drivability can be achieved, which leads to high peak currents [2] or requires heavy electric motor and inefficient usage of the energy stored in

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battery to achieve the target gradeability and acceleration. As most small EVs (such as Honda EV [1], Reva [3], etc.) using DC traction motors are equipped with single speed ratio transmission [2–4], it may appear that transmissions with multi-speed ratios are unnecessary for EVs.

The current study shows that a two-speed transmission system for a DC motor powered EV can improve the functionality by increasing the top speed, initial acceleration and gradeability, where a low transmission ratio for low-speed/high-traction conditions, and a high transmission ratio for high-speed/low-traction conditions is used. In the process of arriving at the specifications of the two-speed transmission system, a methodology is developed to estimate the traction requirements for starting the vehicle from rest and accelerating or moving in steady state conditions afterwards. These estimated torques are subsequently used as in a step-by-step method, which is formulated for arriving at the specifications of transmission system considering the speed-torque-power characteristics of the motor.

2. VEHICLE REQUIREMENT

In this section, the methodology for arriving at gearbox reduction ratios is discussed. In this methodology, first required tractive effort is estimated and then it is used for deciding reduction ratios and the number of speeds in transmission system.

2.1. Tractive Effort

The tractive effort can be estimated with the aid of Eq. (3) using information from Eq. (1) and Eq. (2). Eq. (2) accounts for the friction between the driven tires and pavement. The use of Eq. (3) ensures that the tractive effort does not approach infinity at low vehicle speeds.

$$F_t = 3.6\eta P/V \tag{1}$$

$$F_{\max} = \mu M_{ta} g \tag{2}$$

$$F = min(F_t, F_{\max}) \tag{3}$$

where

F_t = tractive effort (N);
P = motor power (W);
V = vehicle vehicle speed (km/h);
η = transmission efficiency;
$F_{\max}$ = maximum tractive force (N);
M_{ta} = vehicle mass shared by tractive axle (kg);
μ = coefficient of friction between tires and pavement; and
F = effective tractive effort acting on vehicle (N).

The total resistance force or road load is computed as the sum of the three resistance components, as summarized below in Eq. (4).

$$R = R_a + R_r + R_g \tag{4}$$

where

R = total resistance (N);
R_a = air drag or aerodynamic resistance (N);
R_r = rolling resistance (N); and
R_g = grade or gradient resistance (N).

2.1.1. Aerodynamic Resistance (R_a)

The aerodynamic resistance (R_a), or air drag, is a function of the vehicle frontal area, altitude, and square of the relative speed of vehicle and wind, as indicated in Eqn (5).

$$R_a = c_1 C_d C_h A(V + V_h)^2 \tag{5}$$

$$C_h = 1 - 8.5 * 10^{-5}H \tag{6}$$

where

$$A = \text{vehicle frontal area (m}^2\text{);}$$
$$V = \text{vehicle speed (km/h);}$$
$$V_h = \text{headwind speed (km/h);}$$
$$C_h = \text{altitude coefficient;}$$
$$C_d = \text{vehicle drag coefficient;}$$
$$c_1 = \text{constant that equals to 0.047285; and}$$
$$H = \text{altitude (m).}$$

2.1.2. Rolling Resistance (R_r)

The rolling resistance (R_r) is a linear function of the vehicle speed and mass, as given in Eq. (7). The rolling coefficient (C_r) depends on the road surface type and condition. In addition, the rolling resistance coefficients (c_2 and c_3) vary as a function of the vehicle's tire type. Generally, radial tires provide a resistance that is 25% less than that for bias ply tires [6].

$$R_r = C_r(c_2V + c_3)MgCos(\alpha)/1000 \tag{7}$$

where

$$M = \text{vehicle total mass (kg);}$$
$$g = \text{acceleration due to gravity (m/sec}^2\text{);}$$
$$\alpha = \text{gradient, as shown in Figure 1(°);}$$
$$C_r = \text{rolling coefficient; and}$$
$$c_2, c_3 = \text{rolling resistance coefficients.}$$

2.2. Grade Resistance (R_g)

The grade resistance varies as a function of the total weight of vehicle and gradient negotiated by it, as indicated in Eq. (8) and schematically shown in Figure 1. The grade resistance accounts for the proportion of the vehicle weight that resists the movement of the vehicle.

$$R_g = MgSin(\alpha) \tag{8}$$

where

$$M = \text{vehicle total mass (kg);}$$
$$g = \text{acceleration due to gravity (m/sec}^2\text{);}$$
$$\alpha = \text{gradient, as shown in Figure 1(°);}$$

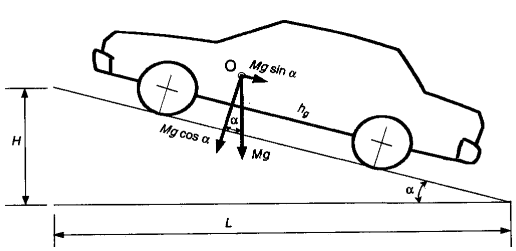

Figure 1. Grade Resistance.

2.3. Maximum Vehicle Acceleration

The maximum acceleration is a function of the forces acting on the vehicle and can be computed as given in Eqn (9).

$$a = (F - R)/M \tag{9}$$

where

a = maximum vehicle acceleration (m/s^2);
F = tractive effort (N);
R = total resistance force (N) (Eq. (4)); and
M = vehicle total mass (kg).

2.4. Required Reduction Ratio (RR)

Required tractive effort (F_r) for accelerating the vehicle with an acceleration of a is given by Eq. (10) i.e. another form of Eq. (9).

$$F_r = Ma + R \tag{10}$$

If the torque generated by the prime mover of a vehicle is T (Nm), required reduction ratio (RR) for accelerating the vehicle with an acceleration of a is given by Eq. (11) or (12).

$$RR = F_r * r_w/T \tag{11}$$

or

$$RR = F_r/F_p \tag{12}$$

where

r_w = wheel radius of vehicle (m);
F_r = required tractive effort (N); and
F_p = available tractive effort at prime mover (N).

For a given transmission ratio or reduction ratio (RR) and motor torque characteristic, the acceleration and traction variations with respect to speed can be plotted.

3. TRANSMISSION SPECIFICATIONS

This section covers the application of the quintessential relations given above for suggesting an better transmission specifications for an EV which is currently based on a single speed transmission system. It may be noted that a single transmission ratio is a workable system for an EV employing a DC motor as the prime mover for its nearly uniform torque-speed characteristic including high torque at low speeds. The basic specifications for the vehicle are given in Table 1. Table 2 contains the values of various constants required for tractive force calculation. These values are adapted from [6].

Technical targets for transmission system of vehicle are assumed as:

- Initial acceleration of 1.85 m/sec^2
- Grade-ability for a slope of 10°

Table 1. Basic specifications of the vehicle considered.

Parameter	Value
Wheel Base (mm)	2500
Gross Vehicle Weight (kg)	1000
Rated Motor Power (W)	5850
Base Motor Torque (kgm)	3

Table 2. Values of constants [6].

Constants	Values
C_d	0.70 (Single Unit Vehicle)
C_h	$1 - 8.5 \times 10^{-5}$ H
C_r	1 (Concrete Pavement)
c_1	0.047285
c_2, c_3	0.0328, 4.575 (Radial Tyres)
μ	0.80 (Concrete Pavement)

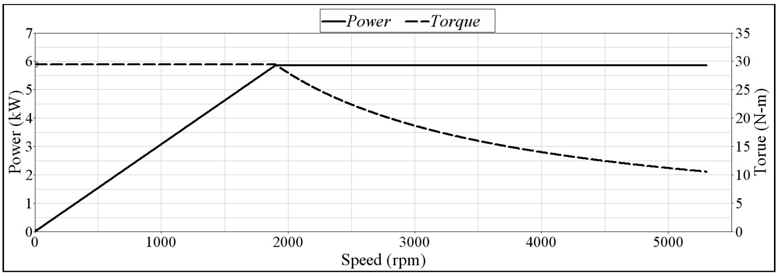

Figure 2. Characteristics of prime mover (DC Motor).

The characteristic curves of the prime mover i.e. DC motor are shown in Figure 2, where in constant torque region power is proportional to the speed of motor and constant power region torque is inversely proportional to the speed.

Now required tractive effort is calculated as the function of speed as aerodynamic drag (R_g) and rolling resistance (R_r) are dependent on it. Required Tractive Effort, Rolling Resistatance, Aerodynamic Resistance, Grade Resistance, etc are plotted in Figure 3 with legends F_r, R_r, R_a, R_g, etc respectively. This explains the characteristics of required tractive effort with respect to speed. Total resistive load (R in Figure 3) or road load is total of grade resistance (R_g), aerodynamic resistance (R_a) and rolling resistance (R_r). Here grade resistance (R_g) depends only on slope (α) according to Eq. (8); if vehicle has to go uphill then it is positive and negative for downhill which will reduce the required traction. Whereas, aerodynamic resistance (R_a) is proportional to the square of the relative velocity of wind and vehicle, as given in Eq. (5), so it has insignificant contribution at initial stage but it increases rapidly with speed. On the other hand, rolling resistance (R_r) depends on velocity according to Eqn (7).

Finally, required tractive effort (F_r in Figure 3) is calculated by adding total resistance (road load R) and effort required for acceleration i.e. mass of vehicle multiplied by acceleration. Total required tractive effort, total resistance and effort for accelerating are given as F_r, R and R_i respectively in Figure 3.

The required reduction ratios (given by Eq. 12) for transmission are calculated using the traction available at prime mover (i.e. a DC motor in current case) and required tractive effort (shown as F_r in Figure 3) for specified initial acceleration and gradient. These calculated reduction ratios for initial acceleration of 1.85 m/sec^2 at different gradients are given in Table 3. And Table 4 has the required reduction ratios for the gradient of 10° and initial acceleration varying from 0.67 m/sec^2 to 1.85 m/sec^2.

The reduction ratio of 24.78 is required for starting the vehicle at the gradient of 10° with the initial acceleration of 1.11 m/sec^2 (i.e. average acceleration for reaching from 0 to 60 kmph in 10 seconds). And the required reduction ratio is 23.85 for initial acceleration of 1.85 m/sec^2 at 5° gradient. Apart from the performance of the vehicle, the selection of reduction ratios also depends on the packaging (size), weight and inertia and manufacturing costs (if it requires upgradation in production line or not), etc. In the current study, reduction ratios are chosen and decided to just show improvement in the performance of the vehicle. Therefore, the reductions ratios of 24.78, 17.57, 12.46, 8.83 and 6.26 are

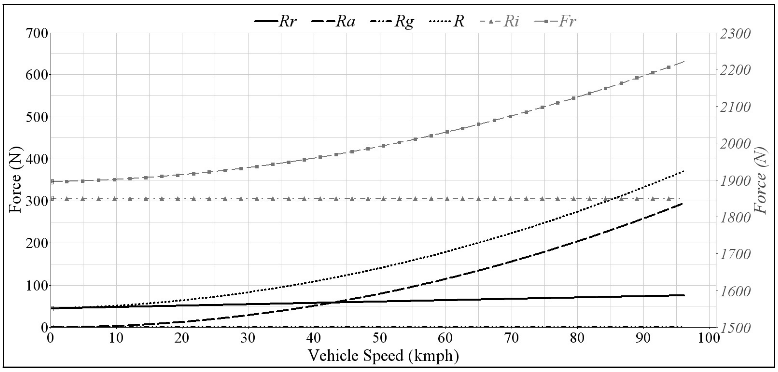

Figure 3. Required traction for 0° gradient and constant acceleration of 1.85 m/sec^2 with respect to speed of vehicle.

Table 3. Required speed reduction ratio (RR) of gear box for different gradient and initial acceleration of 1.85 m/sec^2.

Initial Acceleration (m/sec^2)	Gradient (°)	Reduction Ratios
1.85	0	16.43
1.85	5	23.85
1.85	10	31.21
1.85	15	38.44
1.85	20	45.51

Table 4. Required speed reduction ratio (RR) of gear box for 10° gradient and different initial accelerations.

Initial Acceleration (m/sec^2)	Gradient (°)	Reduction Ratios
0.67	10	20.97
0.84	10	22.44
1.11	10	24.78
1.67	10	29.64
1.85	10	31.21

considered for study and available tractive efforts at the wheel are plotted for each of them in Figure 4. Along with available tractive effort, total resistance ($R = R_a + R_r + R_g$ i.e. Eq. 4) for the gradient of 0°, 10°, 15° and 20° is also plotted in Figure 4. It is clear from Figure 4 that vehicle cannot start at a gradient of 20°, as road load is more than available tractive effort with the reduction ratio of 24.78. It is similar case for the reduction ratio of 12.46 and 10° gradient. It is clear that higher reduction ratio is required to negotiate and start at steeper gradients but this reduces the highest possible speed of the vehicle due to the speed limit of the prime mover (DC motor in current case), i.e. 20.56 kmph for reduction ratio of 24.78 and for reduction ratio of 17.57 it is 28.99 kmph. On the other hand with reduction ratio of 6.26, the vehicle can achieve a speed upto 81.26 kmph on cost of acceleration and gradeability, as the accelerating capability of a vehicle is determined by available tractive effort for accelerating after overcoming the resistance i.e. Available Tractive Effort - Total Resistance. Hence, to have a higher accelerating capability and gradeability, higher reduction ratio is required and a smaller reduction ratio will improve the highest speed of the vehicle.

As it is mentioned in last paragraph, a reduction ratio of 24.78 is required for vehicle to start with an acceleration 1.11 m/sec^2 at an uphill gradient of 10°, but this reduction ratio limits the speed of vehicle to 20.56 kmph and it can achieve a speed upto 81.26 kmph with the reduction ratio of 6.26.

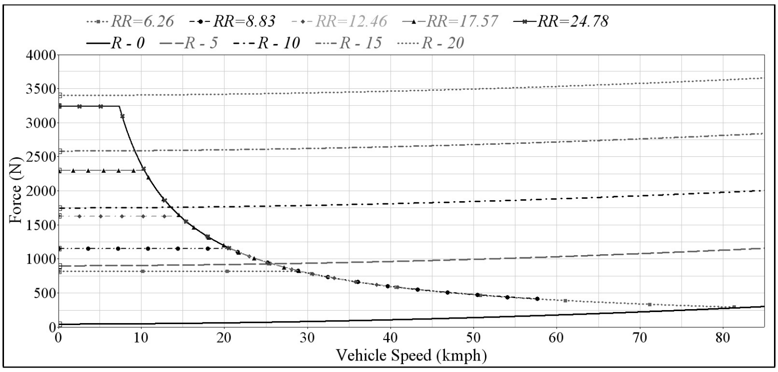

Figure 4. Resistive loads and available tractive effort for different gradient and reduction ratios.

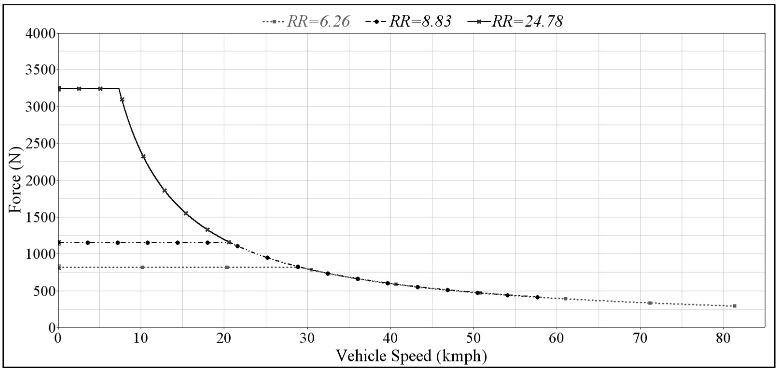

Figure 5. Tractive effort vs vehicle speed for different reduction ratios.

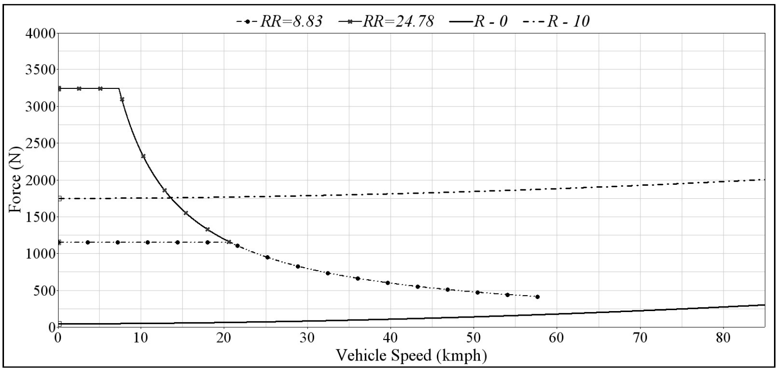

Figure 6. Road loads for 10° and 0° gradient, and available tractive effort for RR=24.78 and 8.83.

Hence, it is proposed here that using two speed gear box in the considered electric vehicle will widen its performance in terms of gradeability, accelerating capability and top speed. It can be seen in Figure 5 that the maximum tractive effort for the reduction ratio of 6.26 is less than the minimum tractive effort for the reduction ratio of 24.78, in other words, these two reduction ratios do not have a common range of vehicle speed and available tractive effort, which will result in a jerk during gear shift from the reduction ratio of 24.78 to 6.26. Therefore, these two cannot be used in combination. On the other hand, the reduction ratios 24.78 and 8.83 have a common combination of tractive effort and vehicle speed. Hence, this combination is chosen in the current study, even though it might not be optimum combination. Figure 6, which contains the plots of available tractive effort for the reduction ratios of 24.78 and 8.83 and the total resistance for 0° and 10° gradients, shows that if the gear box with the mentioned reduction ratios is used in an electric vehicle, it will have enough traction to accelerate on a gradient of 10° and can reach upto 57 kmph speed at flat road. Furthermore the acceleration and velocity profile can be plotted to see the effect of single and double speed gear boxes. And a study about the efficient use of the batteries for standard drive cycle is to be performed for optimizing the reduction ratios.

4. CONCLUSION

Most of the state-of-art EVs use fixed speed transmission system, but the current study shows that it will be beneficial to use a transmission system with multiple speeds in attaining high initial torque as well as high speed with a limited speed and torque of a DC Motor. In particular, a two-speed system studied here, significantly widens the operation range of the considered electrical vehicle. Further studies are required by considering typical driving cycles, power variations and cost aspects before arriving at a optimum gear ratios for attempting a physical prototype two-speed gear box.

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