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Analysis of Series Hybrid Electric Vehicle with Auxiliary Power Unit

Avizit Basak ^α, M. M. Israfil Shahin Seddique ^σ, Md Sabbir Alam Chowdhury ^ρ & Md. Khairul Bashar ^ω

Abstract- Hybrid electric vehicles represent a promising future direction for the transportation sector in terms of decreasing the reliance on fossil fuels while simultaneously decreasing emissions. Energy used for driving is fully or partially shifted to electricity leading to lower emission rates. In this regard, the use of a power generator and motor as the primary power source for an automobile is very attractive. This thesis develops model of a series hybrid vehicle with a power generator as its Auxiliary Power Unit (APU). Simulation using the model helps to provide an understanding of the interaction and flow of power. Moreover, electrical power sharing between the APU and the Electrical Storage System (ESS) is shown. Conclusion is also discussed.

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operational missions is available from two or more kinds or types of energy stores, sources or converters with at least one store or converter onboard. A more specific definition of a hybrid electric vehicle is given as a hybrid vehicle in which at least one of the energy stores, sources or converters can deliver electric energy.

a) Hybrid Vehicle Architectures

The parallel configuration illustrated in figure 1.1 also represents a typical design, where both the ICE and the electric motor can provide torque to drive the wheels.

I. INTRODUCTION

A Hybrid Vehicle, as defined by Technical Committee 69 (Electric Road Vehicles) of International Electro Technical Commission, is one in which propulsion energy during specified

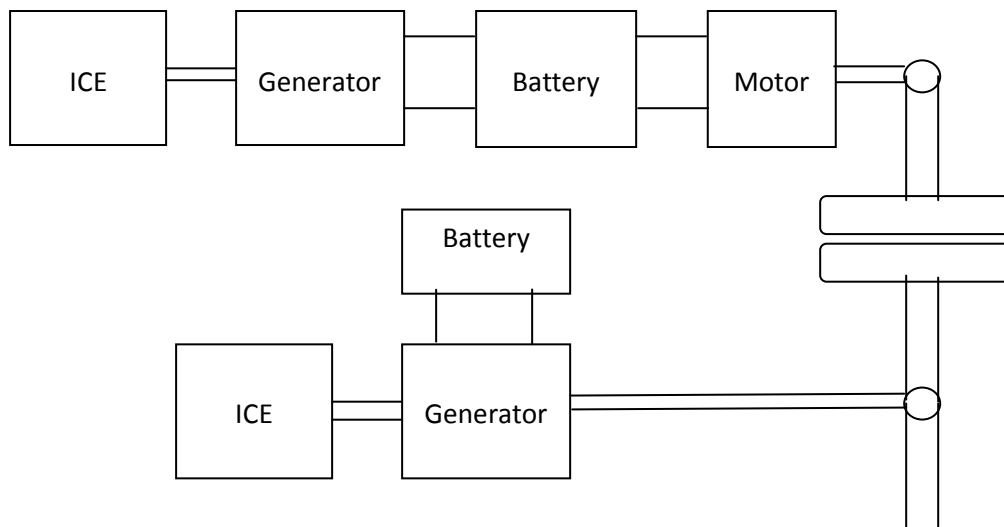


Figure 1.1 : Typical series and parallel hybrid vehicle architectures

II. SERIES HYBRID DRIVE TRAIN

Although there are many possible configurations for a series hybrid drive train, a traction motor propels the vehicle. The motor delivers high torque at low speeds and low torque at high speeds. The motor gives a constant torque for variable speed up to the

'base speed' of the motor; beyond the base speed, the torque of the motor decreases with increase in the speed. The traction motor is powered by a battery pack and/or an engine generator unit. The engine/generator unit either helps the batteries to power the traction motor when load power demand is large, or charges the batteries when the load power demand is small. The motor controller is used to control the traction motor to produce the power required by the vehicle.

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a) Maximum State of Charge of Peaking Power Source Control Strategy

Consider Point A. The demanded power is more than what the engine/generator alone can meet ($P_{e/g}$). Hence the motor supplies the extra power to meet the load demand. (P_{pps}). At point B, the power required is less than the power produced by the engine /generator at its optimal operating point. In this case, this surplus power that is being produced by the ICE can either be used to charge the peaking power source else the

engine is operated in its non- optimal region, to supply just the traction power. Point C has negative (braking power) which is more than the braking power that the motor can alone produce, hence hybrid braking is used. Here, the electric motor produces its maximum braking power and mechanical brakes are also applied in addition. Point D represents the commanded braking power that is less than the maximum braking power that the motor can produce.

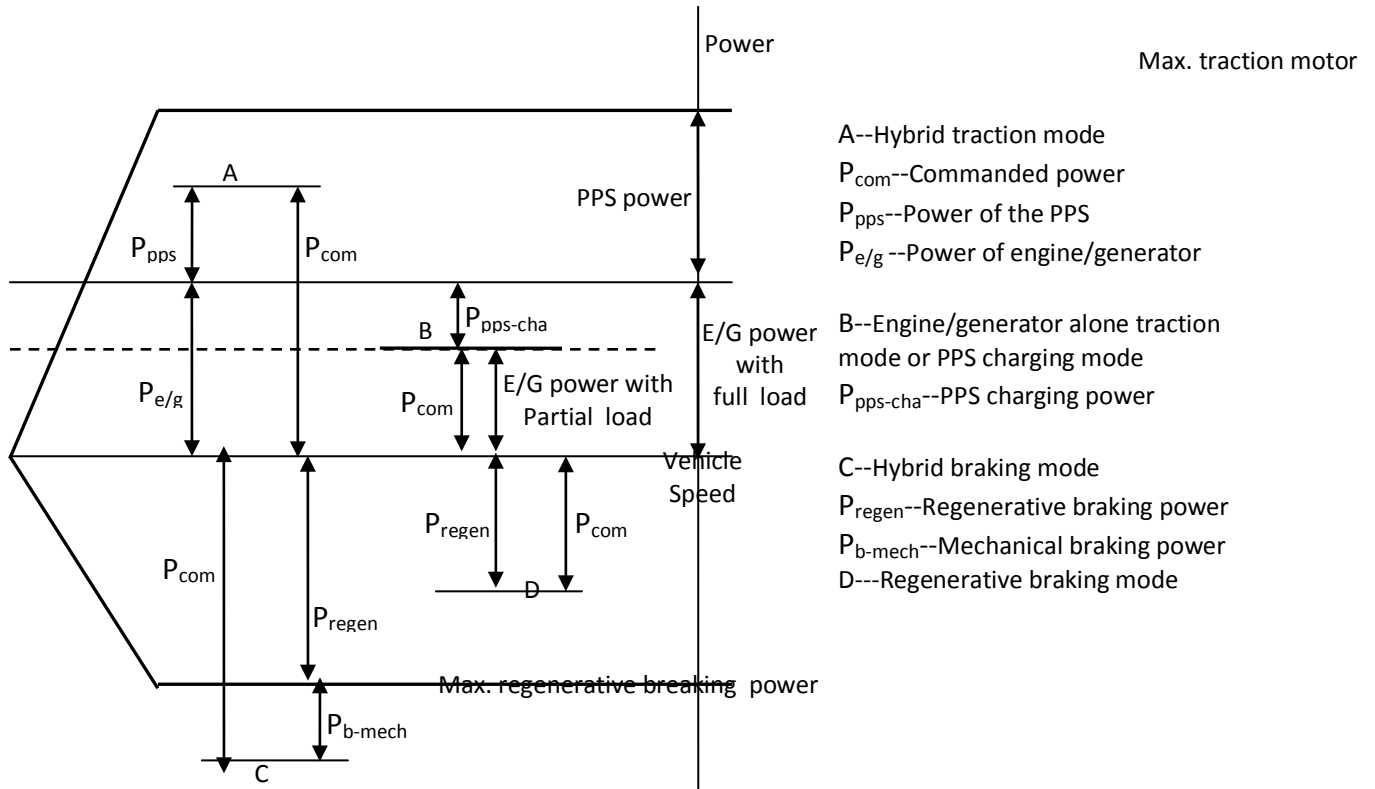


Figure 2.1 : Maximum SOC control strategy

III. DESIGN

For designing the hybrid electric vehicle we need to consider vehicle maximum mass, front area, maximum velocity, acceleration time, transmission efficiency, traction motor efficiency, generator efficiency, aerodynamic drag coefficient, rolling resistance coefficient, tire radius and air density. For a four passenger carry able vehicle, normally the maximum mass is 1200-1500Kg. When a vehicle runs, then a resistance is created by air. This resistive force act against car speed is proportional to vehicle front area, aerodynamic drag co-efficient, air density and vehicle speed. To overcome from this resistive force, traction motor has to consume certain amount of its total consumed power.

Table 3.1 : Drag coefficients for some passenger vehicles[20]

Vehicle (class)	C_D	$C_D \times A_f$ (m ²)
VW Polo (class A)	0.37	0.636
Ford Escort (class B)	0.36	0.662
Open Vectra (class C)	0.29	0.547
BMW 520i (class D)	0.31	0.649
Mercedes 300SE (class E)	0.36	0.785

From this table, we can consider our vehicle's front area is 2m² and hence drag co- efficient is approximately 0.3.

Table 3.2 : Air density at various temperatures[20]

T in °C	ρ in kg/m ³ (at 1 atm)	T in °C	ρ in kg/m ³ (at 1 atm)
-25	1.423	5	1.269
-20	1.395	10	1.247
-15	1.368	15	1.225
-10	1.342	20	1.204
-5	1.316	25	1.184
0	1.293	30	1.164

We take temperature of air is normally 25°C and so air density 1.184 kg/m³ (at 1 atm).

Table 3.3 : Co efficient of rolling friction of tire[20]

Tire Type	Coefficient of Rolling Friction
Low rolling resistance car tire	0.006 - 0.01
Ordinary car tire	0.015
Truck tire	0.006 - 0.01
Train wheel	0.001

We consider our desired rolling co efficient is 0.01.As motors & generators efficiency are normally 80% to 90%, we consider motor efficiency as 85% and generator efficiency as 90%. Table 3.4: Various types of gear with their efficiency[20]

Name of gear	Efficiency	Name of gear	Efficiency
Spur Gears	90%	Helical Gears	80%
Sprocket Gears	80%	Bevel Gears	70%
Rack and Pinion	90%	Worm Gears	70%

Weight of the vehicle, M_v =1500Kg(with 4 passenger),Front area of the vehicle, A_f= 2.0 square meters, Transmission efficiency (η_t)=90%, Traction

motor drive efficiency (η_m)= 85%,Generator efficiency (η_g)= 90%,Aerodynamic drag coefficient (C_d) = 0.3,Rolling resistance coefficient (f_r) = 0.01,Tire radius = 0.3 meters,Air density (ρ_a) = 1.184 kg/cubic meters,Acceleration time (from 0 to 100 km/h) = 10 Seconds,Grad ability more than 5% at 100 km/h.,Maximum speed = 160 km/h.

a) Design of Traction Motor

The power rating of the traction motor is determined by using the equation:

$$P_t = \frac{M_v}{2t_a} (V_f^2 + V_b^2) + \frac{2}{3}M_v g f_r V_f + \frac{1}{5}\rho_a C_D A_f v_f^3$$

In this case, we know that,M_v = Vehicle mass = 1500 kg.,t_a = Acceleration time in seconds = 10 seconds.V_f = Final Speed of the Vehicle in m/s = 160 km/h.V_b = Final speed corresponding to motor base speed.

The motor base speed and the maximum speed of the motor are related by a factor 'x'.

$$X = \frac{\text{Motor max.Velocity}}{\text{Motor base velocity}}$$

The factor 'x' is arbitrarily chosen to have a value of 4, the ratio between maximum vehicle velocity (V_f) and V_b will be 'x' = 4 The Motor power thus calculated will be 82.5 KW. Hence the value of V_b can be calculated. The values of C_D, A_f, f_r,ρ_a have already been specified. The motor maximum speed is chosen to be 5000 RPM. Correspondingly the motor base speed is 5000/4 that is 1250 RPM .Motor power = π × Motor base speed × Ratedmotor torque.The motor torque thus calculated is found out to be 630 Nm. Thus the specifications of the traction motor are as follows: Motor power = 82.5 KW. Motor rated torque = 630 Nm. Motor base Speed = 1250 RPM. Motor maximum Speed = 5000 RPM.

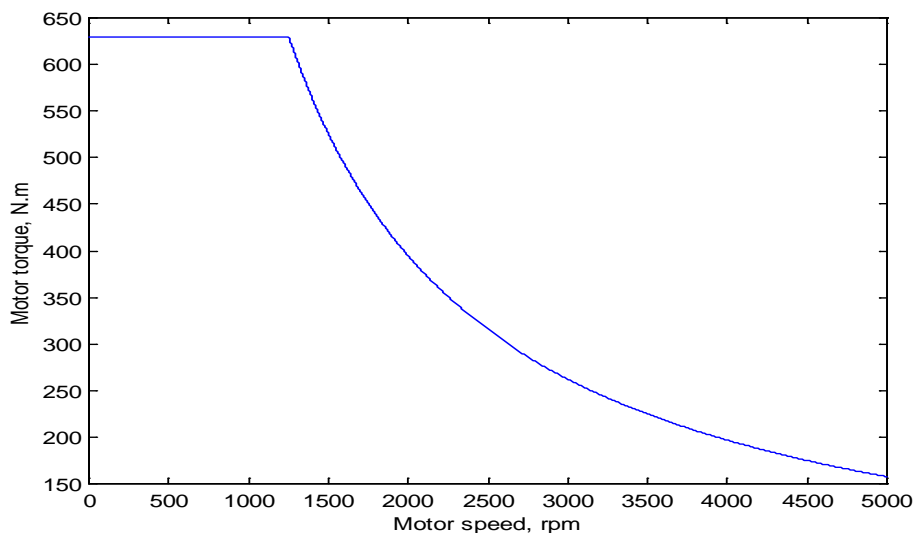


Figure 3.1 : Speed vs. Torque curve for the designed traction motor

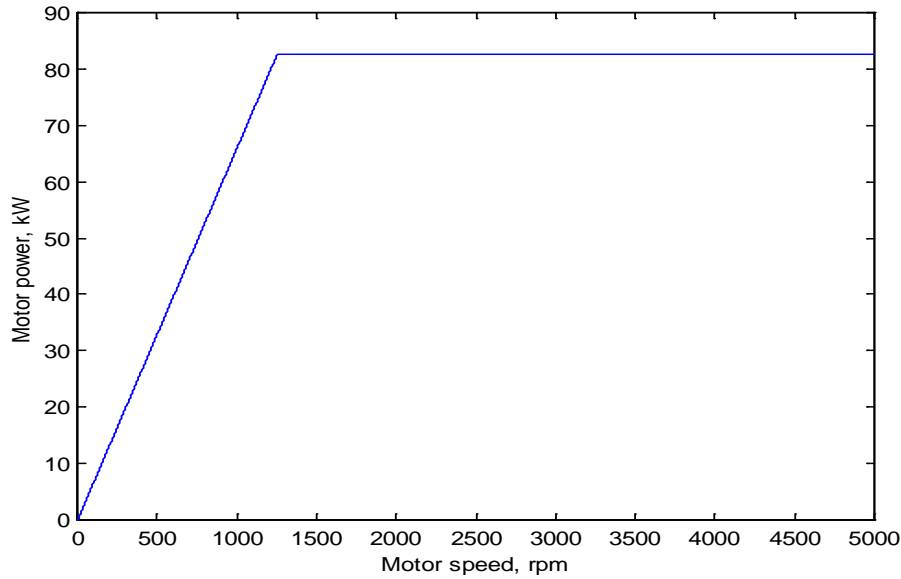


Figure 3.2 : Speed vs. Power curve for the designed traction motor

b) Design of Gear Ratio

Hence the gear ratio is designed such that the vehicle maximum speed corresponds to the motor maximum speed

$$i_g = \frac{2 \pi n_{m,max} r}{V_{max}}$$

where $n_{m,max}$ = Maximum motor speed = 5000 RPM. r = tire radius = 0.3 meters. V_{max} = 160 km/hr. The gear ratio thus calculated is 3.53.

$$F_t = \frac{T_p i_g \eta_t}{r}, \text{ Here, motor rated torque, } T_p = 630\text{N-m}$$

Gear ratio, $i_g = 3.53$, Transmission efficiency, $\eta_t = 90\%$, Tire radius, $r = 0.3\text{m}$ So, rated tractive effort will be 6.2 kN. To check if the motor is able to meet the gradability condition, the tractive effort of the vehicle is plotted with the vehicle resistance (aerodynamic drag + rolling resistance + the hill climbing).

c) Calculation of Tractive Effort

Tractive effort of the designed vehicle is calculated by following equation.

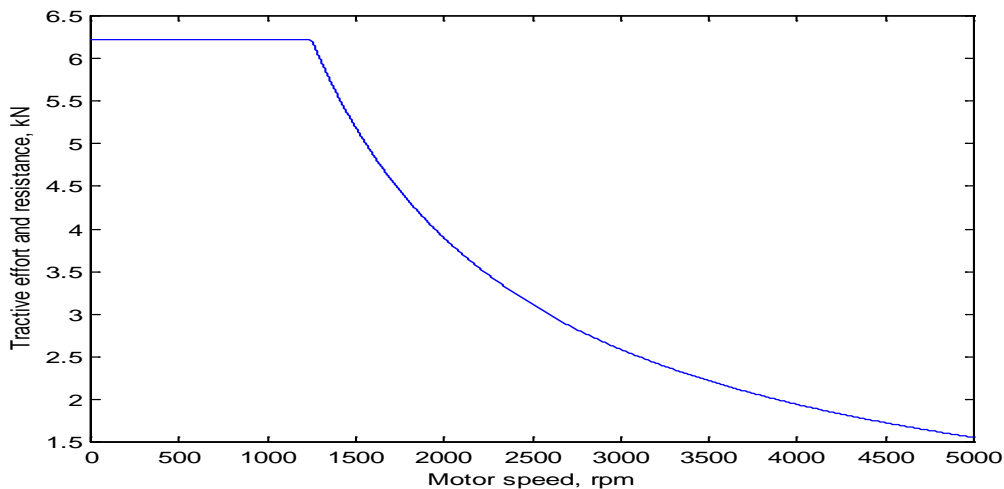


Figure 3.3 : Traction effort and resistance of vehicle versus vehicle speed

d) *Design of Generator*

Considering a 5% grade, the engine power needed to support the vehicle at this speed, considering transmission efficiency to be 90%, motor drive efficiency to be 85% and the

generatorefficiency to be 90%. $P_g = \frac{V}{1000 \eta_t \eta_m} (M_v g f_r + 0.5 \rho_a C_D A_f V^2)$ kW. Using this eqⁿ our desired generators power rating is 29KW.

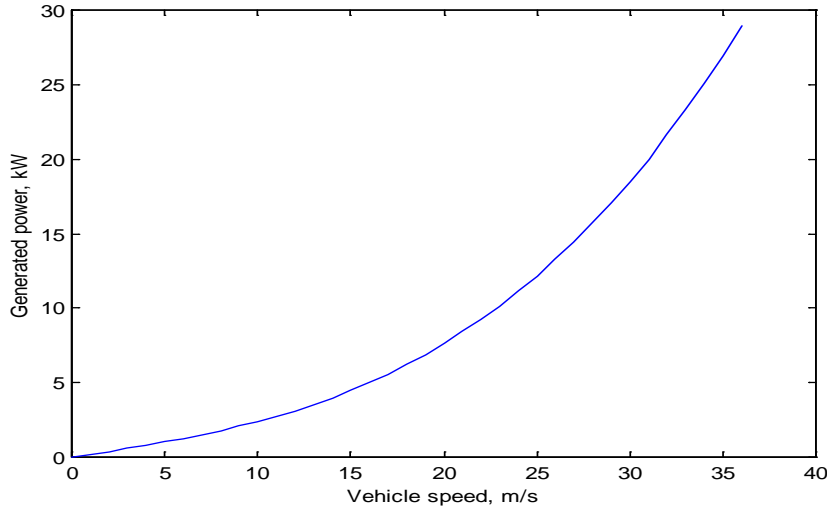


Figure 3.4 : Generated power versus vehicle speed

The load can be given by

$$P_{ave} = \frac{1}{T} \int_0^T (M_v g f_r + 0.5 \rho_a C_D A_f V^2) V dt + \frac{1}{T} \int_0^T M_V \frac{dV}{dt} dt$$

Consider that drive cycle is about 16 hour. At this condition P_{ave} will be approximately 26.45kW.

e) *Power Rating of Peaking Power Source*

Since the power rating of the generator and the traction motor has been decided, the rating of the peaking power source can be easily calculated.

$$P_{PPS} = \frac{P_{motor}}{\eta_{motor}} - P_g = \frac{82.5}{0.85} - 29 = 68 \text{ kW}$$

f) *Design of the Peaking Power Source Energy Capacity*

The S.O.C is decided to vary from 0.6 to 0.4. Hence $\Delta SOC = SOC_{top} - SOC_{bott} = 0.2$. Also, the energy capacity of the PPS is decided to vary from 1.5 kWh to 1 kWh. Hence $\Delta E_{max} = 0.5 \text{ kWh}$. Having decided how much the state of charge as well as the energy rating of the peaking power source, its energy capacity can be calculated.

$$E_{PPS} = \frac{\Delta E_{max}}{\Delta SOC} = \frac{0.5}{0.2} = 2.5 \text{ kWh}$$

g) *Design of the Vehicle Analytical Tool*

The equations are listed below.

$$P_t = \frac{M_v}{2t_a} (V_f^2 + V_b^2) + \frac{2}{3} M_V g f_r V_f + \frac{1}{5} \rho_a C_D A_f V_f^3,$$

Motor power = $2\pi \times$ Motor base speed \times Rated motor torque $3. P_g = \frac{V}{1000 \eta_t \eta_m} (M_v g f_r + 0.5 \rho_a C_D A_f$

$V^2)$ kW 4. $P_{ave} = \frac{1}{T} \int_0^T (M_v g f_r + 0.5 \rho_a C_D A_f V^2) V dt + \frac{1}{T} \int_0^T M_V \frac{dV}{dt} dt$, 5. $F_t = \frac{T_p i_g \eta_t}{r}$ 6. $i_g = \frac{2 \pi n_{m,max} r}{V_{max}}$, 7. $P_{PPS} = \frac{P_{motor}}{\eta_{motor}} - P_g$

Now text box of all input parameters are filled with a specified value. Here base factor is taken as $X=4$.

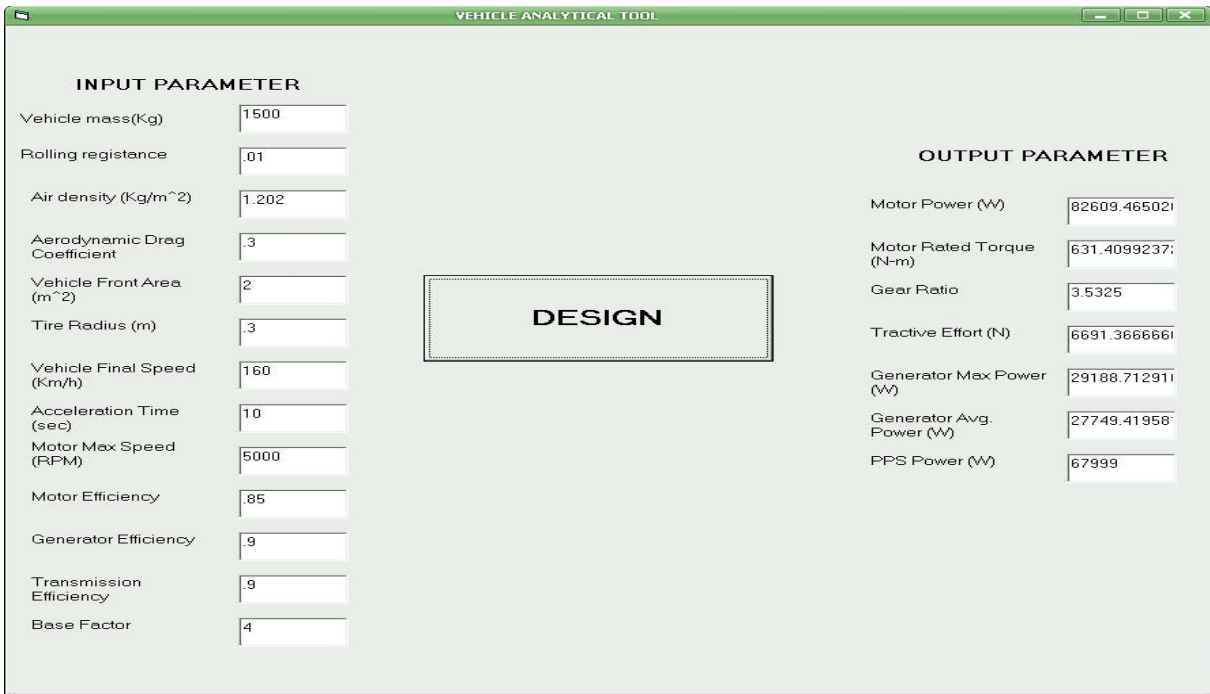


Figure 3.5 : Window of the design tool with base actor $X=4$

By clicking DESIGN button we can get all our desired output parameters. Any one input parameters can be changed easily for various design condition.

Here for example we change the base factor $X=6$ and again design tool window is shown with all the input-output parameters.



Figure 3.6 : Window of the design tool with base factor $X=6$

IV. SUMMARY

In this chapter, a series hybrid electric vehicle was designed. By calculation we find that, our designed

vehicle require a traction motor which takes a maximum power of 82.5 kW and the rated torque produced by it is 630Nm. Rated tractive effort of the vehicle that is the traction motor is approximately 6.2 kN. Maximum

generation power by the generator is 29 kW is required in flat road and the average power generated in urban road is approximately 26.5 kW. Rest power needed for the traction motor is supplied by the PPS i.e, battery. Maximum power rating of the PPS is 68 kW and energy capacity of PPS is 2.5 kWh.

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