

Chapter 20

Devising Product Design Architecture Strategies: Case of HEV Powertrain



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Abstract This article presents a comparative analysis of possible combinations of three transmission systems, namely, Continuously Variable Transmission (CVT), Planetary Gear Transmission (PGT), and Dual Clutch Transmission (DCT), to carry out the comparative study of fuel consumption to deduce new gearbox architecture. Modern automotive industry is moving towards greener technology that leads to the introduction of the Hybrid Electric Vehicle (HEV) as electric motor has higher efficiency compared to an IC engine. Different combinations of available transmission systems, viz., CVT-DCT, CVT-PGS, DCT-PGS, and CVT-DCT-PGS are tested in a MATLAB/Simulink environment. From this experimentation, it has been observed that the topology of transmission architecture also influences the fuel consumption apart from the losses in power train. This experimentation utilized the advantages of combining multiple gearbox architecture to develop a new gearbox to achieve maximum fuel efficiency and thereby reducing the carbon footprint. A systematic analysis of multiple gearbox architecture enabled the development of topological layout to achieve desired transmission architecture.

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20.1 Introduction

Transmission system in vehicle transmits power from the propulsion source to power axle and ensure the availability of adequate torque for different driving conditions. For a conventional combustion engine driven vehicle, the fuel efficiency and emission is a major issue for its contribution to environmental pollution caused by vehicle emission via burning of fossil fuel. The transmission efficiency depends on the engine and power train for such vehicles. The thermal efficiency of IC engines, normally, lies in the range 35–40% [1], whereas gearbox efficiency lies in the range of 85–90% [2]. Thus, to improve the vehicle efficiency both of these factors has to be given adequate weightage. A suitable method to improve the overall efficiency is the electrification of the vehicle, but due to costly battery technology, it is difficult to produce cost-efficient electric vehicles. Another solution to this problem is hybrid vehicles. The hybrid vehicles possess higher source efficiency and there is scope for improving in overall efficiency of the vehicle by optimizing its transmission performance [2]. There are four major types of transmission variants are available, namely, manual, automated manual, automatic, and continuous variable transmission. Among these transmission variants, manual have highest fuel efficiency, but it is near obsolete from the car market. In automated manual transmission, dual clutch transmission is most popular as it provides higher performance in overdrive driving situations in other hand automatic transmission or planetary transmission provides better performance in underdrive condition and continuous variable transmission provide the option of infinite speed ratio in a certain gear ratio range [3]. The development of electric continuously variable transmission (eCVT) and power split device proves to be a significant application of planetary gear transmission. The different transmission systems have their own advantages and disadvantages and hence, a combination of two different types of transmission can produce a better design/architecture, also, reported in different scientific articles. Vacca et al. combined a planetary gear-train with conventional DCT to achieve a Hybridized AMT (HAMT) and the results show that fuel efficiency is increased by more than 9% [2].

AMT provides better shift quality by reducing shifting time, 0.1 s for AMT, as compared to manual transmission (0.3 s) and also it possesses lower weight and higher efficiency as compared to the automatic transmission [4]. It is worth to point out that one of the limitations of the AMT could be reduced driving comfort due to the lack of traction during gearshift actuation. However, this drawback is not the intrinsic limitation of the AMT and proper gearshift management can solve it [4]. The planetary gear arrangement has compact gear structure and higher torque bearing capability and can produce multiple gear ratios with single planetary gear set [5, 6]. So it is easy to implement and control single planetary gear set a transmission in hybrid vehicle [7]. Combining two types of transmission, namely dual clutch transmission (Automated manual transmission) and planetary gear set with an electric motor can improve the torque continuity during gear shift [2]. Also, the application of the planetary gear set with motor can provide infinite gear ratios in a particular range of gear ratio values [8].

In this article, different transmission variants combined and tested to analyze the performance. Using a vehicle simulation model, the different variants are tested for HWFET driving cycles. The fuel consumption and the emission from the vehicle are calculated. Based on the experimentation, the best transmission architecture has been selected. In the first section, a summarized description has been provided for the previous works on combining different transmission architectures. Then, the description of tools and methodology is followed by the simulation result discussion and at the last the conclusion is provided.

20.2 Tools and Methodology

MATLAB Simulink, a simulation tool, is considered for this study. The MATLAB, Simulink environment has been utilized for designing the transmission models and model analysis. Majorly, three transmission models CVT, DCT, and PGT have been combined and different architectures are obtained. Highest and lowest gear ratios are required to be defined beforehand, based on vehicle requirement, in order to design the transmission system and it is discussed in Sect. 20.2.2. The methodology of this article is presented in the following flowchart (Fig. 20.1).

The hybrid vehicle model can be segregated as engine model, motor model, generator model, battery model, vehicle body model, and transmission model. Hybrid vehicle is also represented in this section.

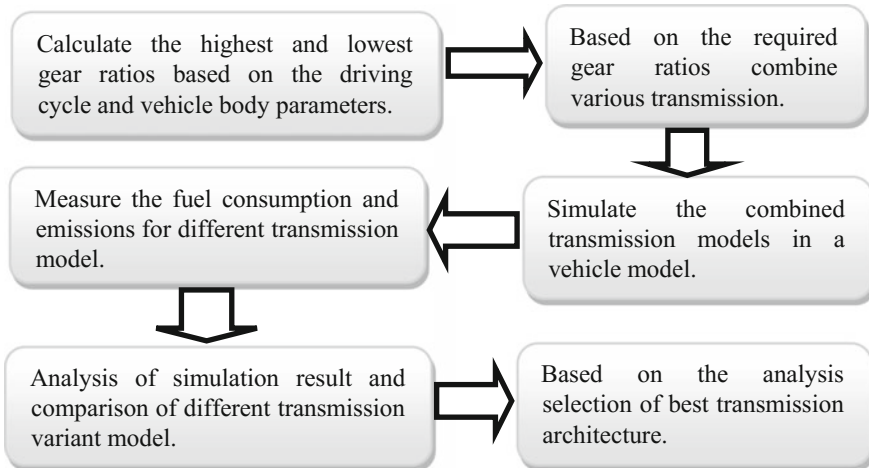


Fig. 20.1 Methodology

20.2.1 Powertrain Model

Engine Model:

A simple engine model has been considered in this article. The engine model can be represented by Eqs. (1) and (2). The engine torque speed curve is provided in Fig. 20.2.

$$P(\Omega, T) = T \cdot g(\Omega) \tag{1}$$

$$\tau = P/\Omega \tag{2}$$

where, P : Actual power delivered by Engine, T : throttle input signal, Ω : Engine speed, $g(\Omega)$: Engine power demand—a polynomial function.

Motor—Generator Model:

Motor and generator model are quite similar to each other; as permanent magnet motor model is considered. The motor torque speed diagram has been provided in Fig. 20.3.

Battery Model:

The basic battery model is considered for this study. The battery equivalent circuit has been provided in Fig. 20.4. The battery model can be represented by Eqs. (3) and (4) [9].

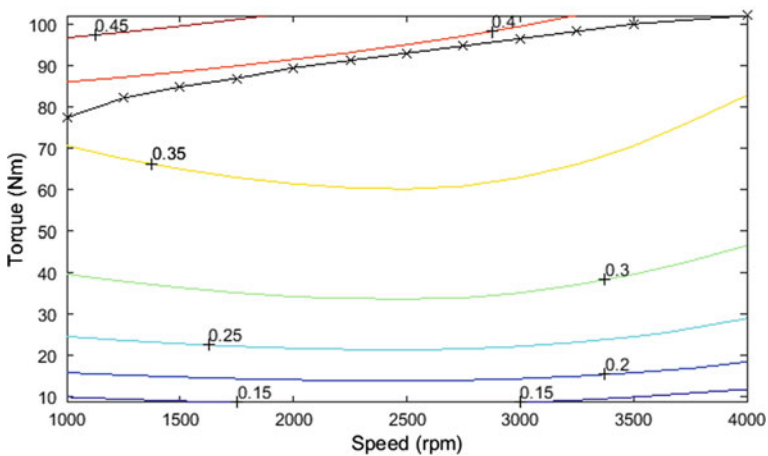


Fig. 20.2 Engine speed torque model

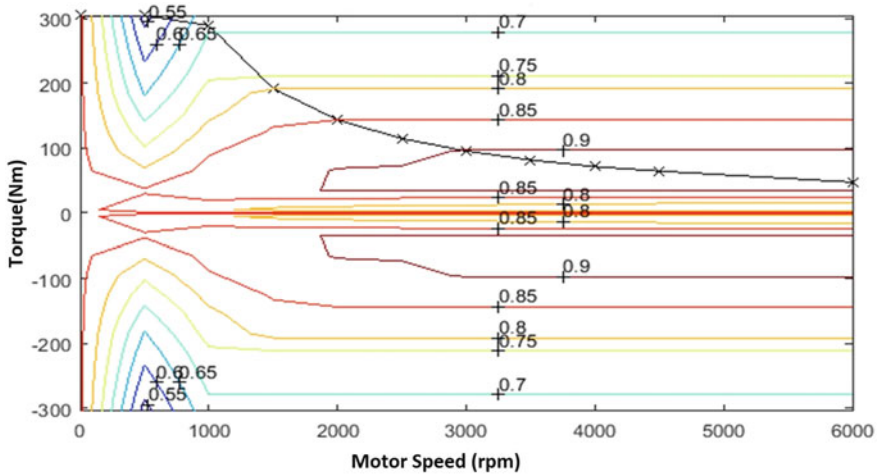
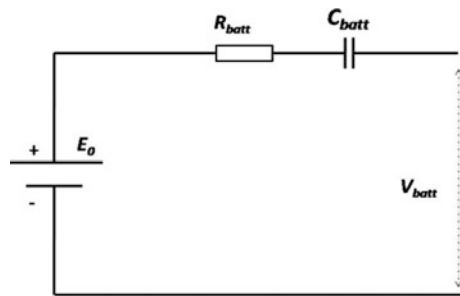


Fig. 20.3 Motor torque speed characteristics

Fig. 20.4 Battery equivalent circuit



$$I_{bat} = \frac{E_0 - \sqrt{E_0^2 - 4R_{batt}P_{batt}}}{2R_{bat}} \tag{3}$$

$$SOC_t = SOC_{t-1} + \int_0^{t_s} \frac{I_{batt}}{C_{batt}} dt \tag{4}$$

where I_{bat} is battery current, R_{batt} : battery internal resistance, C_{batt} : battery capacitance in Ah, SOC_t : battery state of charge at time t , t_s : simulation time, V_{batt} : battery terminal voltage, E_0 : open circuit voltage.

Vehicle body model:

The vehicle model is selected based on the force acting on the vehicle. A vehicle needs to overcome three types of resistive forces, namely aerodynamic drag force

(F_a), rolling resistance (F_r), and grade force (F_g), to move. Thus, at equilibrium, the traction force (F_t) generated by the engine should be equal to or greater than that of the three combined forces. It can be represented by Eqs. (5–10) [10].

$$F_t = F_a + F_r + F_g \tag{5}$$

$$F_a = m_v \times a \tag{6}$$

$$F_a = \frac{1}{2} c_d \rho A_f V^2 \tag{7}$$

$$F_r = c_{roll} m_v g \cos \theta \tag{8}$$

$$F_g = m_v g \sin \theta \tag{9}$$

$$F_t = \frac{\tau_{eng} n \eta}{r_t} \tag{10}$$

Transmission Model:

Four combinations of transmission have been considered for this study (i) CVT-DCT, (ii) CVT-PGS (iii) DCT-PGS, and (iv) CVT-DCT-PGS. These four combination further utilized to simulate the vehicle model as shown in Figs. 20.5, 20.6, 20.7 and 20.8. As shown in the following figures different, combinations of available transmission system are used to test the fuel efficiency. In CVT-DCT model, CVT transmission is clubbed with the shaft carrying even number of gears where higher torque is required at second gear and moderate torque is required at 4th gear. It may be noted that other combinations of gearbox are also possible and that needs to be examined. For time being, this particular arrangement of transmission

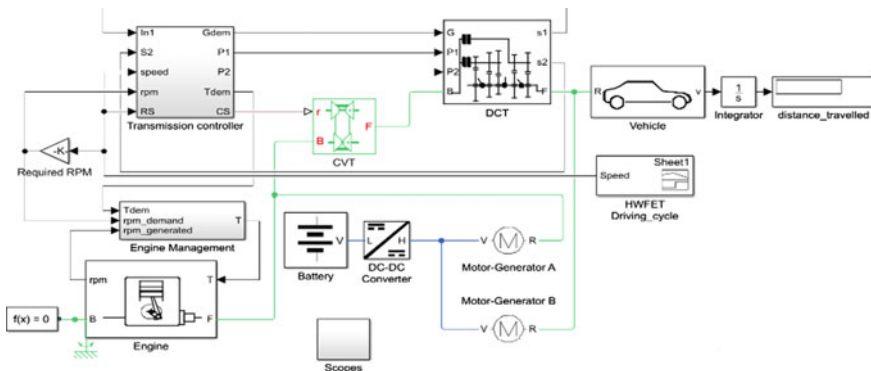


Fig. 20.5 CVT-DCT combination

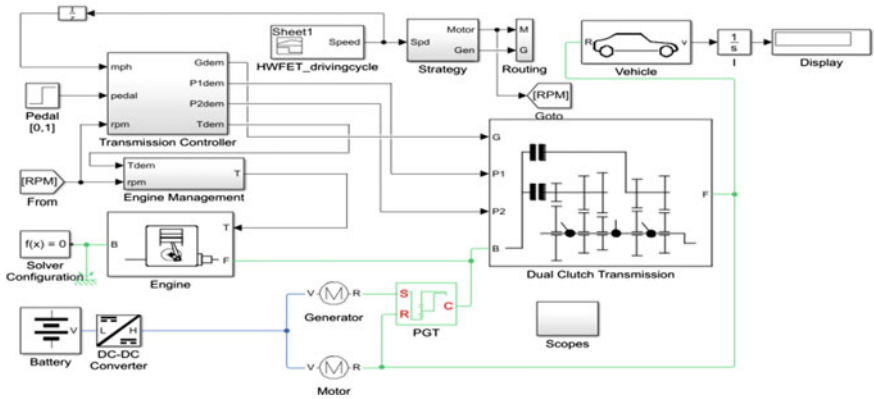


Fig. 20.6 PGT-DCT combination

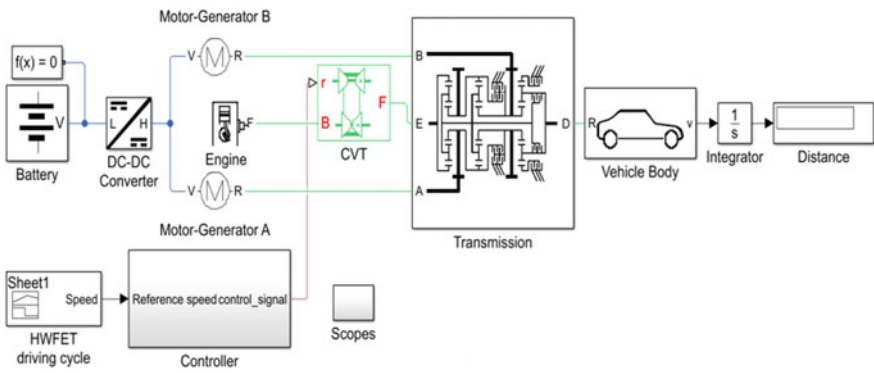


Fig. 20.7 CVT-PGT combination

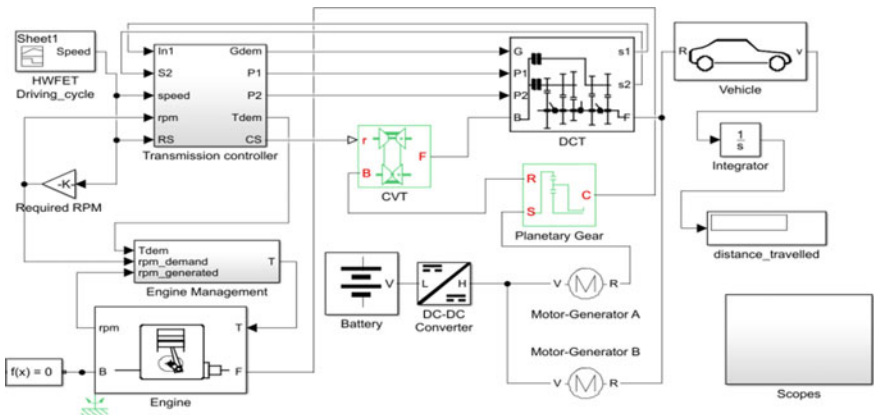


Fig. 20.8 CVT-PGT-DCT combination

system is tested for each combination. But in the case of CVT-PGT-DCT combination, PGS is used as power split facilitator in vehicle transmission.

20.2.2 Calculation of Highest and Lowest Gear Ratio

Based on the vehicle parameter and traction force demand, the highest and lowest gear ratio has been calculated. Highest and lowest gear ratios are calculated based on Eqs. (11) and (12) [11].

$$n_L = \omega_{s \max} \times \frac{2\pi r_t}{60V_{\max}n_f} \tag{11}$$

$$n_H = \frac{F_{T\max}r_t}{\mu_d n_f \tau_{s\max}} \tag{12}$$

where n_H : highest gear ratio, n_L : lowest gear ratio, $\omega_{s \max}$: maximum rotational velocity form the source, $\tau_{s \max}$: maximum engine torque, V_{\max} : maximum vehicle speed, n_f : final drive ratio, μ_d : final drive ratio.

An initial simulation has been run with the vehicle information [12], Table 20.1 and maximum traction force required is calculated accordingly. The calculated

Table 20.1 Simulation parameters

Parameters	Value	Parameters	Value
Vehicle weight	1335 kg	Engine peak efficiency	0.40
Vehicle Cd	0.35	Battery module number	40
Vehicle front area	2 m ²	Battery nominal voltage	308
Engine maximum torque	100 Nm	Motor maximum power	31 kW
Engine speed at max. power	3000 RPM	Motor peak efficiency	0.90
Motor maximum torque	300 Nm	Generator maximum power	15 kW
Motor RPM at max. power	4000 RPM	Generator peak efficiency	0.84
Vehicle maximum speed	27.78 m/s	SOC higher threshold	0.75
IC engine maximum power	43 kW	SOC lower threshold	0.45

Table 20.2 Gear ratios for transmission design

	Gear box ratio	Overall ratio
First gear	2.15	4.52
Second gear	1.23	2.58
Third gear	0.64	1.34
Forth gear	0.33	0.70
Final drive	2.1	–

highest gear ratio is 2.15 and the lowest gear ratio is 0.33, Table 20.2. The intermediate gear ratios have been calculated based on the geometric progression rule [13] of gear selection.

20.3 Simulation and Result Discussion

The simulation has been performed on a workstation (memory size 16 GB, i7 processors). All transmission models are tested for same HWFET driving cycle, which is of approximately 766 s, and 16.5 km driving cycle. The driving cycle speed pattern is provided in Fig. 20.9. The result shows that the vehicle model with DCT-PGS combination has the lowest fuel consumption. It may be noted that consumption of gasoline is only measured to compare the performance indices of these different combinations of transmission systems. Wind velocity and road grade are assumed as 5 km/h and 5%, respectively. The result has been provided in Table 20.3.

Fuel consumption with respect to time for different combinations of gearbox is shown in Fig. 20.10. Results suggest that DCT-PGS combination consumed fairly

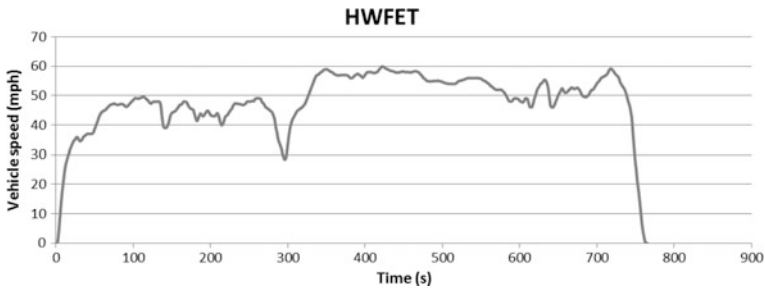


Fig. 20.9 HWFET driving cycle

Table 20.3 Simulation result: fuel consumption

	Distance traveled	CVT-DCT	CVT-PGS	DCT-PGS	CVT-DCT-PGS
Fuel consumption	16.5 km	513.16 g	956.9 g	475.14 g	861.13 g

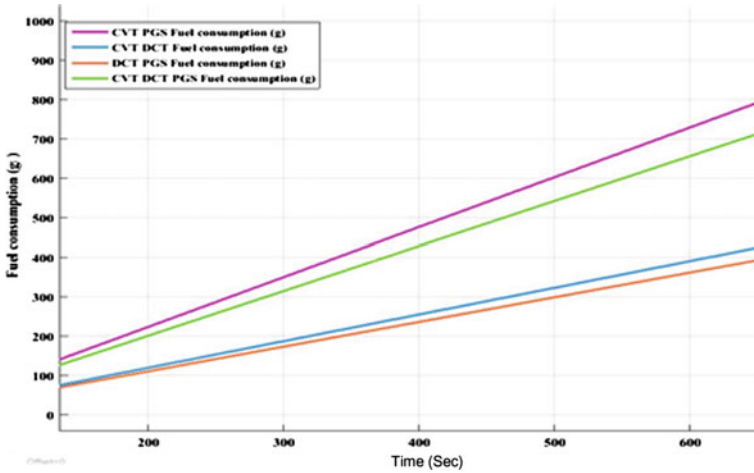


Fig. 20.10 Fossil fuel consumption versus time for HWFET driving cycle

less fuel as compared to CVT-PGS and CVT-DCT-PGS combinations whereas there is no much difference in fuel consumption as compared to CVT-DCT combination. This may be happening due to dynamic behavior of system and needs to be investigated.

20.4 Conclusion

Different combinations of CVT, DCT, and PGS were analyzed which focus on the fuel consumption for each possible pair in the power train. Experiment shows that the fuel consumption is minimum in case of DCT-PGS combination. Emission and fuel consumption are closely related with each other and hence, the emission will have observed to be least in DCT-PGT model. As fuel consumption is minimal, thus it would be more economical to use this combination with hybrid vehicles at low operating cost. As shown in Table 20.3, fuel consumption in DCT-PGT is 8% less than CVT-DCT combination and approximately 40% less as compared to the other two combinations. The difference in fuel consumption may occur due to other operating parameters and can be investigated in the future.

Acknowledgements The authors would like to acknowledge the authority of PAM Lab, Rajendra Mishra School of Engineering Entrepreneurship for helping in conducting the experiment using the available infrastructure.

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