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Simulation of Mileage and Fuel Economy of Plug-in Hybrid Electric Vehicle with Dual Clutch Transmission Considering Temperature Condition

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Summary

In Transmission Mounted Electric Device (TMED) Plug in Hybrid Electric Vehicle (PHEV), the transmission efficiency and power losses of the powertrain have a great influence on the fuel economy because all the power is always transmitted through the transmission. Dual Clutch Transmission (DCT), which consists of gears, bearings and shafts, etc., changes its transmission efficiency depending on input torque, rotational speed, gear status and oil temperature. We developed a performance simulator that can evaluate the driving distance and fuel consumption of PHEV with DCT by quantitatively calculating the power loss and transmission efficiency of DCT considering temperature conditions. The developed simulator will be useful for developing the fuel economy improvement strategy of TMED PHEV.

Keywords: PHEV (plug in hybrid electric vehicle), parallel HEV, efficiency, energy consumption

1 Introduction

DCT has synchromesh type gearbox and has advantage of high power transfer efficiency but also has advantage of automatic gear shifting by using Dual Clutch. For these reasons, DCT is emerging as a next-generation transmission and its market share is increasing. In TMED type powertrain, the power generated by the engine and the motor is transmitted through the transmission during acceleration, and also the kinetic energy of the vehicle is transmitted to the generator through transmission during regenerative braking. Therefore, it seems that transmission power losses have a significant effect on vehicle mileage and fuel economy [1]. Therefore, although studies have been made to quantitatively predict the power loss of the transmission [2], there is a lack of studies on the influence of transmission power loss on the fuel economy and mileage of the PHEV vehicle considering the temperature condition.

A study to derive quantitative power loss of a transmission was first started by predicting the power loss of single component. Petry-Johnson measured the power loss of the spur gear pair according to the rotational speed, transmitted power and lubricant type through the experiment and proposed a method to increase the transfer efficiency [3]. Seetharaman, S. experimented with the power loss of the emerged gear pair and presented the prediction equation [4]. However, complicated lubricant behaviour makes accurate power loss prediction difficult. Concli, F. conducted a study on the analysis of gear churning power loss phenomena using computational fluid dynamic [5]. Based on component level researches, MT (Manual Transmission) power loss prediction and transmission efficiency improvement [6] and improvement of the accuracy of power losses prediction research has been conducted [7].

2 DCT Power Losses Calculation

Power losses of DCT are caused by the interaction of elements such as gears, bearings, shafts, and lubricants. In order to obtain the total loss, it is necessary to sum up the losses per element, so it is necessary to arrange many losses by certain factors. Table 1 classifies the power loss occurring in DCT according to the factors and describes the cause of the loss.

Table 1: Dry type DCT power losses classification

Component	Type	Symbol	Causes of Losses
Gear	Mesh loss	P_{GM}	Frictional loss in contact point
	Churning loss	P_{GC}	Oil drag loss due to submerged gear rotation
Bearing	Rolling loss	P_{BR}	Bearing rolling loss due to roller deformation
	Sliding loss	P_{BS}	Frictional loss in contact point
	Drag loss	P_{BD}	Oil drag loss due to submerged bearing rotation
Etc.	Seal loss	P_{Seal}	Friction between oil seal and shaft
	Shaft loss	P_{Shaft}	Oil drag loss due to shearing stress from between two component that has speed difference
	Synchronizer loss	P_{Sync}	
	Fork loss	P_{Fork}	

Total power loss P_{V} can be divided into load dependent loss P_{L} and non-load loss P_{N}. The load dependent loss is mainly caused by the friction and dominantly influenced by the torque. Gear mesh loss occurring on the mating tooth surface, bearing rolling and sliding loss are the load dependent loss. On the other hand, the non-load loss is mainly caused by the viscosity of the lubricant and is dominantly influenced by the rotational speed. Gear churning loss, bearing drag loss and etc. are the non-load losses.

$$P_{\mathrm{V}} = P_{\mathrm{L}} + P_{\mathrm{N}} \tag{1}$$

$$P_{\mathrm{L}} = P_{\mathrm{GM}} + P_{\mathrm{BR}} + P_{\mathrm{BS}} \tag{2}$$

$$P_{\mathrm{N}} = P_{\mathrm{GC}} + P_{\mathrm{BD}} + P_{\mathrm{BS}} + P_{\mathrm{Seal}} + P_{\mathrm{Shaft}} + P_{\mathrm{Sync}} + P_{\mathrm{Fork}} \tag{3}$$

The non-load loss is caused by the viscosity resistance of the lubricant between the elements in which the relative motion occurs. In the case of a wet clutch, a large amount of resistance is generated between the clutch plates. However, this paper focuses on dry clutch, the non-load resistance generated in shafts, synchronizers, shift forks, etc. As shown in figure 1 below, the viscosity of the lubricating oil exponentially increases with decreasing temperature. Therefore, the fuel economy of the target PHEV is expected to decrease drastically when driving in EV mode at very low temperature condition.

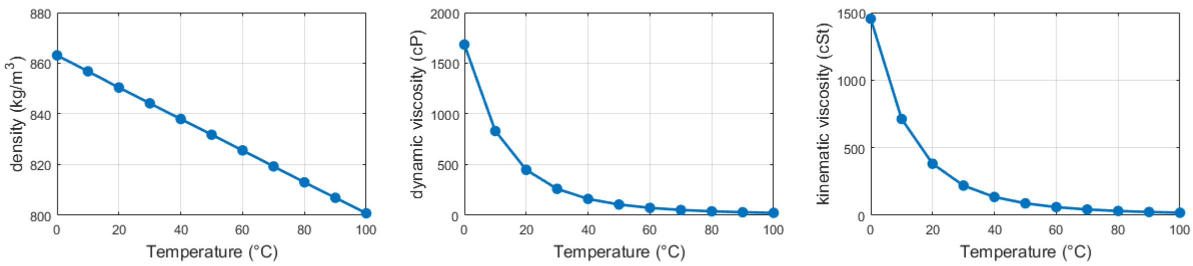

Figure 1 Target lubricant density, dynamic viscosity, kinematic viscosity according to temperature

2.1 Gear losses [8]

Mesh loss occurs in helical gear that transmits power, and churning loss occurs due to dipped rotating gear in lubricant. Gear mesh loss P_{GM} and churning loss P_{GC} are based on ISO / TR 14179-1. Gear mesh loss is calculated as follows: P_{a} is transmitted power and β is helix angle. Mesh mechanical advantage M is calculated using the sliding ratio calculated from the gear specification. The friction coefficient μ_{m} is obtained from the lubricant kinematic viscosity ν [cSt], load intensity K [N/mm] and tangential pitch line velocity V [m/s], α_{t} is transverse operating pressure angle, u is gear ratio, r_{o} is gear outside radius [mm], r_{p} is gear operating pitch radius [mm], z is tooth number, and b_{w} is face width in contact [mm]

$$P_{\mathrm{GM}} = P_{\mathrm{a}} \cdot \mu_{\mathrm{m}} \cdot \frac{\cos^2 \beta}{M} \tag{4}$$

$$M = \frac{2\,\cos \alpha_{\mathrm{t}}(H_{\mathrm{s}}+H_{\mathrm{t}})}{{H_{\mathrm{s}}}^2+{H_{\mathrm{t}}}^2} \tag{5}$$

$$H_{\mathrm{s}} = (u+1)\left(\sqrt{\frac{{r_{\mathrm{o2}}}^2}{{r_{\mathrm{p2}}}^2}} - \cos^2 \alpha_{\mathrm{t}} - \sin \alpha_{\mathrm{t}}\right) \tag{6}$$

$$H_{\mathrm{t}} = \left(\frac{u+1}{u}\right)\left(\sqrt{\frac{{r_{\mathrm{o1}}}^2}{{r_{\mathrm{p1}}}^2} - \cos^2 \alpha_{\mathrm{t}}} - sin\, \alpha_{\mathrm{t}}\right) \tag{7}$$

$$\mu_{\mathrm{m}} = \frac{\nu^{-0.223} K^{-0.4}}{3.239\, V^{0.7}} \tag{8}$$

$$K = \frac{1000\, T_1(z_1+z_2)}{2\, b_{\mathrm{w}}(r_{\mathrm{w1}})^2 z_2} \tag{9}$$

Gear churning loss calculation equation is as follows: h is immersion depth [mm], D is diameter of elements [mm], m_{t} is transverse tooth module, ω is rotational speed [rad/s], and b is tooth width [mm]. In the case of actual transmissions, the immersion depth will vary continuously with rotation speed, so an appropriate assumption must be used.

$$P_{\mathrm{GC}} = (1.474 \cdot D + 7.37 \cdot \frac{b}{\sqrt{\tan \beta}} \cdot (7.93 - \frac{4.648}{m_{\mathrm{t}}})) \cdot \frac{h\, \nu\, D^{3.7}}{A_{\mathrm{g}}} \cdot \omega^3 \times 10^{-26} \tag{10}$$

2.2 Bearing losses [9]

Helical gears are attached to shafts and generate axial and thrust forces when transmitting torque. Shafts and gears are supported by tapered roller bearings. Rolling and sliding losses are caused by the load applied to the bearings. Also, the bearings also create drag loss due to lubricant. Bearing losses are calculated as follows: R_1, R_2, Y, S_1, S_2, K_{L}, K_{Z}, V_{M} are standards related constant, n is rotational speed [rpm], d_{m} is bearing mean diameter [mm], and B is bearing width [mm].

$$P_{\mathrm{BR}} = R_1 \cdot {d_{\mathrm{m}}}^{2.38} \cdot (F_{\mathrm{r}} + R_2 \cdot Y \cdot F_{\mathrm{a}})^{0.31} \cdot \nu^{0.6} \cdot n^{1.6} \tag{11}$$

$$P_{\mathrm{BS}} = S_1 \cdot {d_m}^{0.82} \cdot (F_r + S_2 \cdot Y \cdot F_a) \cdot \mu_{sl} \cdot n \tag{12}$$

$$\mu_{\mathrm{sl}} = \Phi_{\mathrm{bl}} \cdot \mu_{\mathrm{bl}} + (1 - \Phi_{\mathrm{bl}}) \cdot \mu_{\mathrm{EHL}} \tag{13}$$

$$\Phi_{\mathrm{bl}} = \frac{1}{e^{2.6\times10^{-8}(\nu \cdot n)^{1.4} \cdot d_{\mathrm{m}}}} \tag{14}$$

$$P_{\mathrm{BD}} = V_{\mathrm{M}} \cdot \frac{K_{\mathrm{L}} \cdot K_{\mathrm{Z}}\,(d+D)}{D-d} \cdot B \cdot {d_{\mathrm{m}}}^4 \cdot n^3 \times 10^{-11} \tag{15}$$

2.3 Other non-load losses

The non-load loss of shaft, synchronizer, and shift fork is related to the viscosity of the lubricant due to the shear stress caused by the lubricant between the parts which has relative speed. The stress generated between the shaft and the gear occurs in the area of the disk shape, and the stress generated by the synchronizer is a coaxial cylindrical surface with a slight angle. The agitation resistance between the fork and the sleeve is both the disk shape and the cylindrical shape combined.

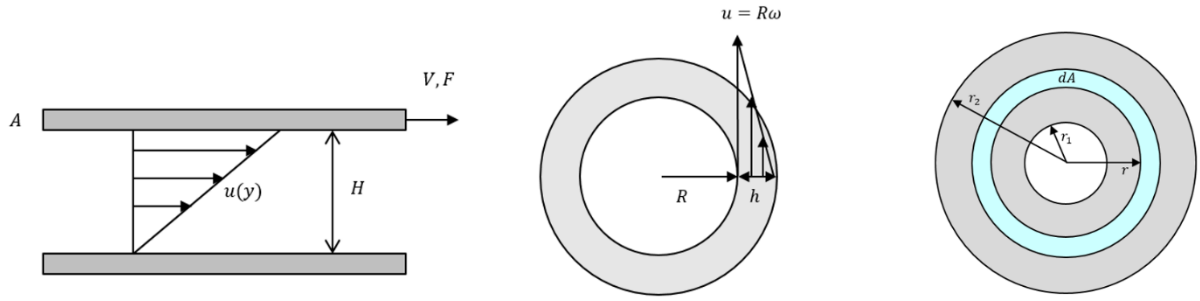

Figure 2 Newtonian flow in flat plate, coaxial cylinder and disk

In order to calculate the non-load power losses, the equation is derived assuming that the lubricant is Newtonian fluid and laminar motion [10]. The loss prediction equation for each shape type is derived as follows: F is friction force [N], T is torque [Nm], μ is absolute viscosity [cP], ν is kinematic viscosity [cSt], V is speed difference [m/s], A is plate area [m^2], H is film thickness [m]

$$F = \mu \frac{VA}{H} \tag{11}$$

$$P_{\mathrm{plate}} = F \cdot V = \mu \frac{VA}{H} \cdot V = \frac{\mu A}{H} \cdot V^2 \tag{12}$$

$$P_{\mathrm{cylinder}} = F \times R \cdot \omega = \left(\mu \frac{R\omega}{h}\right)(2\pi RL) \cdot R \cdot \omega = \frac{2\pi\mu R^3 L}{h}\omega^2 \tag{13}$$

$$P_{\mathrm{disk}} = \int dT \cdot \omega = \int_{r_1}^{r_2} \frac{2\pi\mu r^3 \omega}{H} dr \cdot \omega = \frac{\pi\mu}{2H}(r_2^4 - r_1^4) \cdot \omega^2 \tag{14}$$

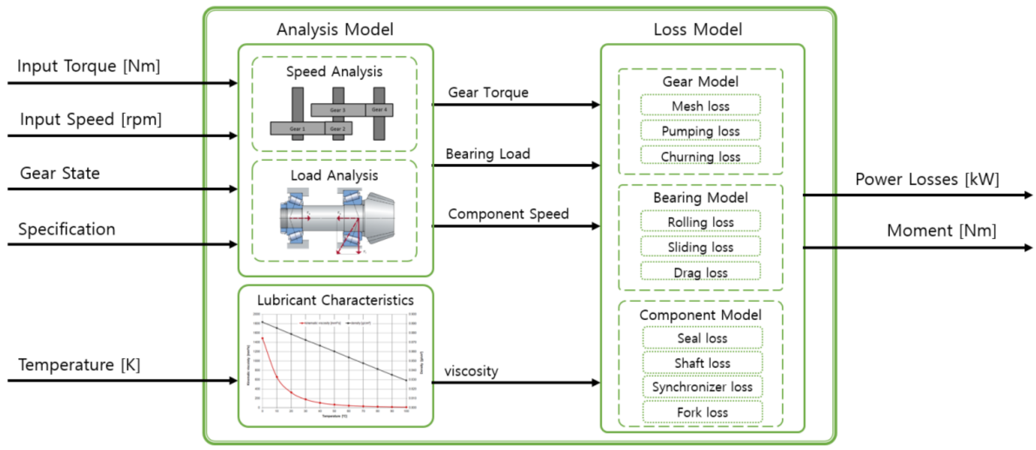

Figure 3 DCT power losses and moment calculation model

2.4 DCT power losses and moment calculation

In order to obtain the each component loss, we must calculate the input conditions: torque, load, and rotational speed. Therefore, component rotational speed and the load calculation model were developed. By combining component power loss prediction model and DCT analysis model, DCT losses calculation model was developed. Figure 3 shows the input and output variables and the calculation process. Figure 4 and 5 show the results of calculating the moment of DCT with torque and rpm at 20 ° C and 60 ° C, respectively. It can be seen that the influence of non-load loss moment is dominant at low temperature, high rotational speed.

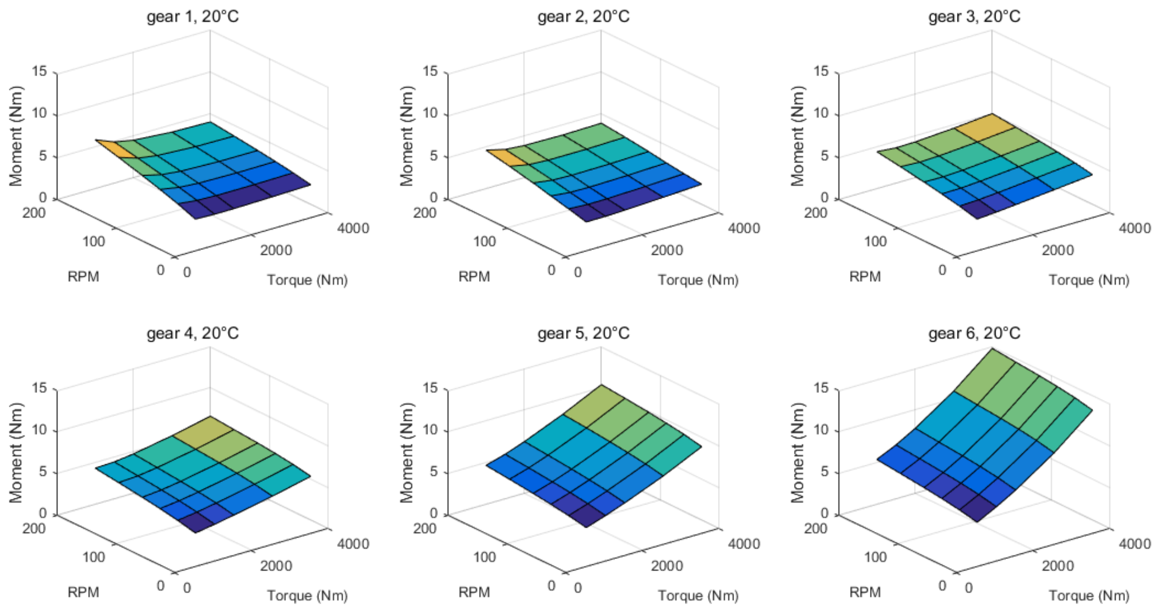

Figure 4 DCT resistance moment calculation result at 20°C

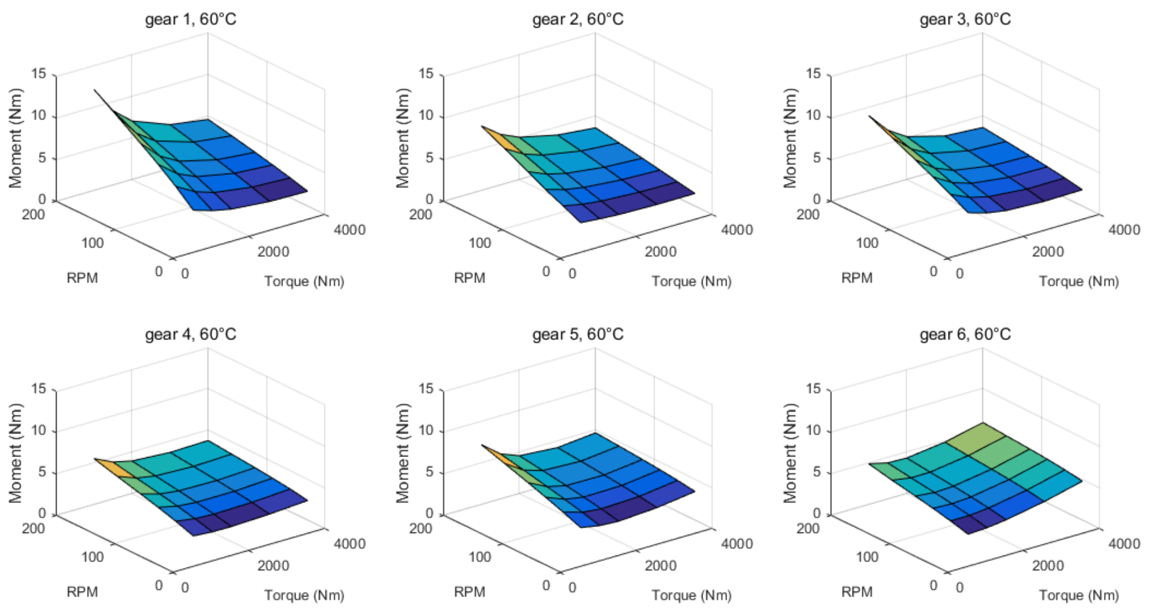

Figure 5 DCT resistance moment calculation result at 60°C

3 Plug-in Hybrid Electric Vehicle Modelling

The powertrain of the target PHEV consists of diesel ICE (Internal Combustion Engine), engine clutch, electric motor / generator, and DCT and its structure is shown in figure 4. PHEV fully charges the battery SOC (State of Charge) to 0.9 using external power when not driving. When the vehicle starts driving, the engine clutch is disengaged and motor is operated only in the EV mode before the battery SOC is reduced to 0.3. When the SOC falls below 0.3, vehicle operates in the HEV mode. The engine clutch is engaged to use engine power and motor power at the same time, or some of the engine power is converted into battery power through the generator to maintain SOC. When the vehicle decelerates, the engine clutch is disengaged and the kinetic energy of the vehicle is converted into electric energy through the generator and stored in the battery. Therefore, it can be expected that the DCT transfer efficiency will have a direct influence on the mileage and mileage since all the power is transmitted through the DCT during the driving and regenerative braking.

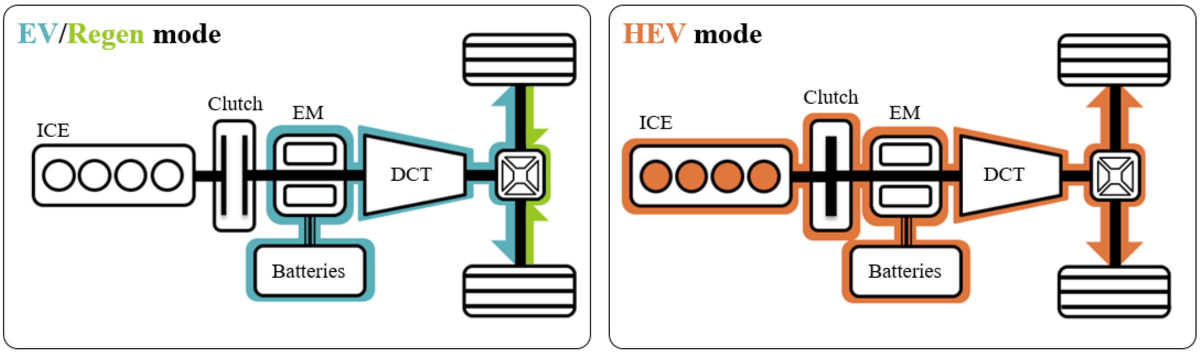

Figure 4 Series type PHEV structure and EV, HEV mode power flow

MATLAB Simulink software was used to construct the target PHEV powertrain dynamic model. The PHEV's powertrain consists of a 1L diesel engine, a 38kW motor / generator and DCT. The inertia and specification for each element are based on commercial vehicle values. The engine torque output curve and engine fuel consumption for rpm and torque were based on KATECH data. The engine operating range is 750 to 6000 rpm and the maximum torque occurs at 3200 rpm to 148 Nm. The mounted 6speed dry type DCT has a gear ratio of [3.53, 2.835, 1.81, 1.32, 1.02, 0.775] and the final differential gear ratio is 3.31. The shift control uses the shift map tuned according to the AP input and the vehicle speed. The motor / generator is modeled as 38kW class IPMSM (Interior Permanent Magnet Synchronous Motor) and efficiency map is applied. The has Li-ion battery capacity of 32.7Ah modeled using electromotive force, internal resistance and battery load current. The inverter efficiency is modeled as a constant. The driver model for the driving cycle test was composed of a PI controller. Table 2 lists the vehicle specifications.

. Table2: Target PHEV parameters

Parameters	Value	Unit	Parameters	Value	Unit
Vehicle Mass	1400	kg	Coefficient of Drag	3.3	
Max engine torque	148	Nm	Projection Area	2.6	m^2
Max motor power	38	kW	Tire Radius	0.33	m
Battery Capacity	32.7	Ah	Tire rolling coefficient	0.01	

4 Simulation

Table 3 shows the PHEV mileage and fuel economy simulation method and conditions considering the temperature condition. To measure the EV mode range, the initial battery SOC is set to 0.9, which is the fully charged state, and drives repeatedly FTP-72 driving cycle, until the SOC decreased to 0.3. To measure HEV mode fuel economy, the initial battery SOC was set to 0.3 and the FTP-72 cycle was run 10 times. The driving cycle is repeated 10 times in order to reduce the effect of the HEV mode algorithm because the algorithm is designed to keep the battery SOC at 0.25 ~ 0.3 repeatedly.

Table3: Simulation method and condition

Value	Driving cycle	Driving mode	Initial SOC	Final SOC
Mileage	FTP-72	EV mode	90	30
Fuel economy	FTP-72	HEV mode	30	30

As a result of simulations at 0 °C to 100 °C at 10 °C intervals, EV mode mileage and HEV mode fuel economy were calculated as shown in figure 5 and figure 6. In addition, the DCT power transfer efficiency for the FTP-72 driving cycle was calculated as shown in figure 7 and the load and non-load losses of DCT were calculated as shown in figure 8.

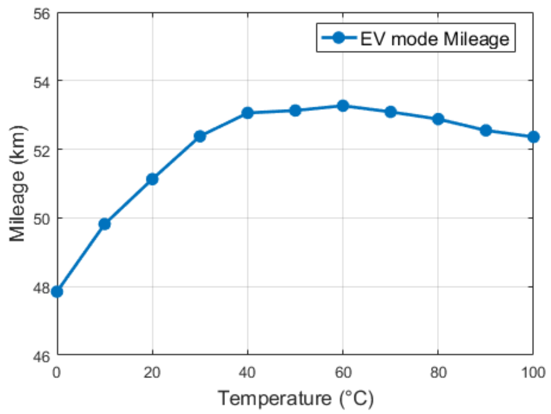

Figure 5 EV mode mileage simulation result

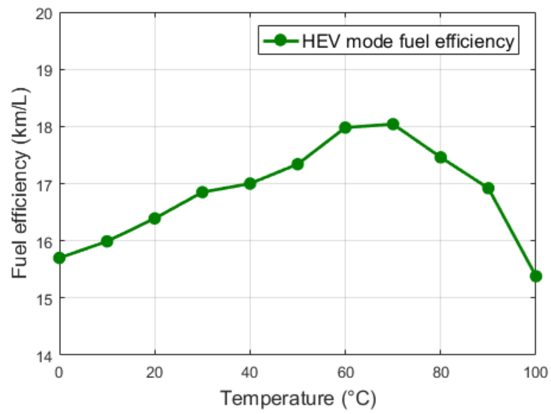

Figure 6 HEV mode fuel economy simulation result

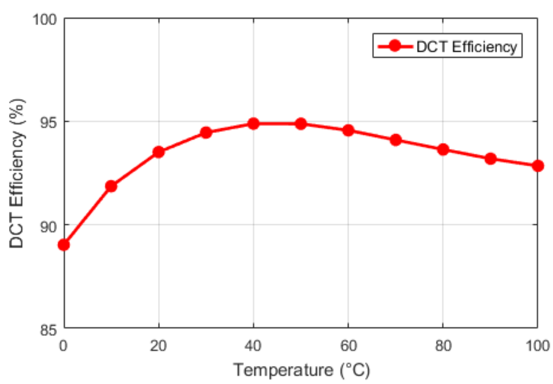

Figure 7 DCT power transfer efficiency result

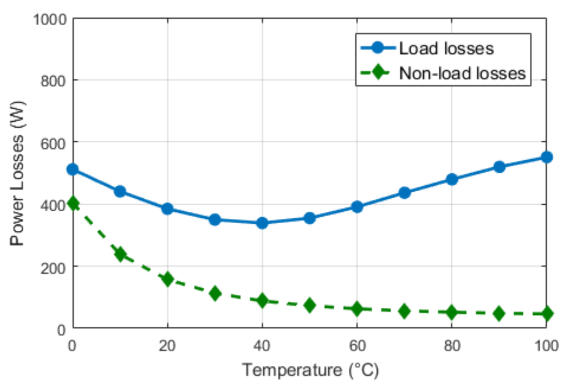

Figure 8 DCT power losses result

5 Conclusion

The simulations on the mileage and fuel economy for PHEV were performed considering the lubricant oil temperature based on theoretical and experimental formulas and the following conclusions were derived.

- The EV mode mileage has the highest value at 53.27 km at 60 ℃ and the lowest at 47.85 km at 0 ℃. The EV mode mileage decreased drastically when the temperature decreased at a temperature of 60 ℃. This means that when driving at low temperatures, the non-load loss will increase due to the increase in the transmission lubricant kinematic viscosity, which will reduce the EV mode mileage by about 10%. In a cold environment, mileage can be further reduced because the driver uses electric accessories such as a heater.
- The highest fuel economy was calculated as 18.04 km/L at 70°C, the lowest fuel economy was calculated as 15.38 km/L at 100°C. Unlike EV mode mileage tendency, fuel economy decreased drastically when temperature increased after the maximum fuel economy. As the temperature increases, the kinematic viscosity decreases cause the oil film breaks and the friction coefficient increases, which is predicted to be related to the increase of the load loss as shown in figure 8.
- Theoretically, when the lubricant temperature is 0 to 100 ° C, it is calculated that the EV mode mileage may vary by about 10% and fuel economy vary by about 15% based on the maximum value. The simulation results can be used to develop an optimal powertrain energy distribution control strategy that can improve the EV mode mileage at the low lubricant temperature condition and improve fuel economy at the high lubricant temperature condition

Acknowledgments

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References

[1] Baekyu Kim, et. Al., *A Theoretical Approach of Power Losses in Hyundai Hybrid Transmission*, KSAE Autumn Conference, 2010, 2991-3001

[2] Horim Yang, *Development of the Model for Analysing Automatic Transmission System Efficiency*, KSAE Spring Conference, 2015, 17-22

[3] Petry-Johnson, Travis T., et Al., *An experimental investigation of spur gear efficiency,* Journal of Mechanical Design, ISSN: 0738-0666, 130.6 (2008), 062601

[4] Seetharaman, S., et Al., *Oil churning power losses of a gear pair: experiments and model validation,* Journal of Tribology, ISSN: 0022-2305, 131.2 (2009), 022202

[5] Concli, F., et. Al., *Influence of lubricant temperature, lubricant level and rotational speed on the churning power loss in an industrial planetary speed reducer: Computational and experimental study,* International Journal of Computational Methods and Experimental Measurements, ISSN: 2046-0546, 1.4 (2013), 353-366

[6] Höhn, et. Al., *Optimization of gearbox efficiency*, Goriva imaziva, ISSN 0350-350X, 48.4 (2009), 462

[7] Changenet, et. Al., *Power loss predictions in geared transmissions using thermal networks-applications to a six-speed manual gearbox,* Journal of Mechanical Design, ISSN: 0738-0666, 128.3 (2006), 618-625

[8] ISO 14179-1, *Gears-Thermal Capacity Part 1: Rating gear drives with thermal equilibrium at 95℃ sump temperature,* 2001

[9] *SKF General Catalogue 6000*, SKF, 2005

[10] White, Frank M., *Fluid mechanics. 6th.*, ISBN: 978-89-6055-051-3, Boston, McGraw-Hill Book Company, 2005

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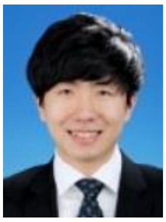

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