Design of Transmission For Electric Vehicle

Chetan Karan[1], Nidhip Naik[2]
[1, 2] Dept of Mechanical Engineering
[1, 2] K. J. Somaiya College of Engineering, Mumbai-77

Abstract- *As environmental and economic interests increase, the need for eco-friendly vehicle such as an electric vehicle (EV) has increased rapidly. Various research of enhancing EV powertrain efficiency and relibility have been studied. In this study, 2-speed shift gears mechanism is designed by using simpson type planetary gear train. This transmission has two planetary gear unit. Gear position is determinded by which ring gear is fixed. Internal components of the transmission are designed for satisfying the required specification of EV. We analyze gear strength, gear mesh efficiency, and transmission efficiency. By manufacturing the transmission prototype and performing some experiments, we verify the application suitability of this transmission.*

Keywords- Electric vehicle, Transmission, Powertrain, Planetary gear

NOMENCLATURE

D_m	: mean diameter of friction plate, (mm)
Q	: total torque capacity of all friction plates
Q_1	: torque capacity each friction plate
R	: reduction gear ratio
T	: torque with axial direction, (N.m)
Z	: number of friction plates
Z_S	: number of sun gear teeth
Z_{P1}	: number of 1st planet gear teeth
Z_{P2}	: number of 2nd planet gear teeth
Z_{R1}	: number of 1st ring gear teeth
Z_{R2}	: number of 2nd ring gear
μ	: friction coefficient of friction plate
ω_S	: angular velocity of sun gear
ω_R	: angular velocity of ring gear
ω_C	: a angular velocity of carrier

I. INTRODUCTION

Due to automobile exhaust causing air pollution and global warming, the interest of an electric vehicle(EV) continues to increase. Improving the efficiency and the reliability of the vehicle is a key issue for developing the EV powertrain. Using the transmission on the EV has some advantages. For maximization of the powertrain efficiency, the output shaft needs high torque at the low vehicle speed. On the other hand, it needs high rotation speed at the high vehicle speed. Hence, it is important to control torque and speed by using the transmission.

The purpose of this research is to design the transmission of the EV, especially commercial vehicle. In this study, the 2-speed transmission system is designed for optimizing electric commercial vehicle (ECV) powertrain. The suitability of the transmission is verified by using gear design program and testing the prototype. This prototype has two efficiency graphs, so this transmission has more wide efficiency area comparing to the single step gear box.

II. DESIGN OF THE TRANSMISSION INTERNAL STRUCTURE

2.1 Calculation of the Gear Specification

As shown in Figure 1, 2-speed Simpson type planetary gear train is established. This kind of transmission has the interesting feature of changing gear ratios without having to engage or disengage individual gears. This allows for smooth gear changing even under load. This transmission has simple shift gears mechanism which is provided in chapter 2.2. Thus, it is easy to change gear position among 1st gear, 2nd gear, parking gear, reverse gear, and neutral. In comparison with conventional system, especially the hydraulic automatic transmission, has low efficiency. Because, it experiences viscous and pumping losses in the torque converter. The torque converter has an efficiency curve that resembles ∩ as shown in Figure 2. Zero efficiency at stall, generally increasing efficiency during the acceleration phase and low efficiency in the coupling phase. The torque converter maximum efficiency is rotate in the opposite direction. By rotating all component including sun gear, ring gear, and carrier at angular velocity- ωR, ring gear stops spinning, and, ωS (angular velocity of sun gear), ωC (angular velocity of carrier) is defined as

$$\left(1 + \frac{Z_S}{\ }\right)\omega_1$$

$$\omega_S = \left(1 + \frac{Z_S}{Z_R}\right)\omega_1 \qquad (2)$$

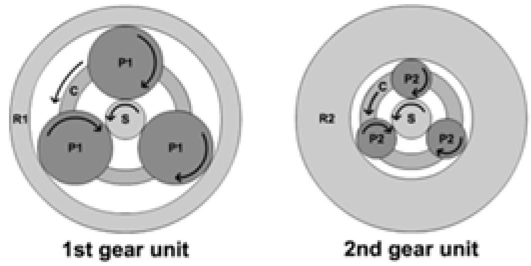

Figure 1. 1st and 2nd planetary gear unit.

$$\omega_c = \frac{Z_S}{Z_{R1}}\omega_1 \qquad (3)$$

From the angular velocity in Table 1, reduction gear ratio is defined as

$$i = \frac{Z_{R1}}{\frac{Z_S}{Z_{R1}}\omega_1} = \frac{Z_S + Z_{R1}}{Z_S} = 1 + \frac{Z_{R1}}{Z_S} \qquad (4)$$

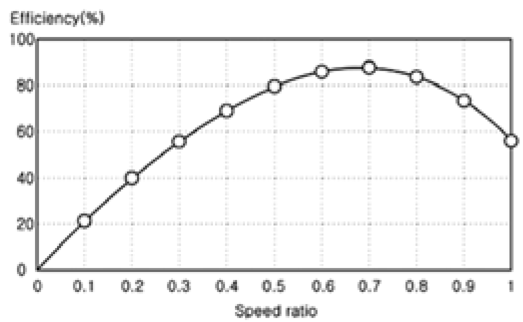

Figure 2. Torque converter efficiency.

typically 95%, which means that the loss of energy is more than 5% (Strachan, 1992).

The planetary gear train designed in this study consists of a sun gear, two ring gears, and three planet gears of each ring gear. Ring gears are selectively fixed by shifting gears. Every planet gears is mounted on a single carrier which itself rotates to the sun gear and acts as a transmission output shaft(June, 1995). Calculation process of reduction gear ratio is shown in Table 1. ZS is the number of sun gear teeth, ZP1 , ZP2 are the number of 1st planet gear and 2nd planet gear, ZR1 , ZR2 are the number of 1st ring gear and 2nd ring gear. Assume that carrier is fixed and sun gear rotates at angular velocity ω1, then angular velocity of ring gear ωR is defined as

$$\omega_R = -\frac{Z_S}{Z_{R1}}\omega_1 \qquad (1)$$

In this equation (4), gear teeth of all gears in this planetary gear train which satisfy required specification of electric commercial vehicle powertrain can be calculated, and its specification is shown in Table 2 (Dudley, 1984; Litvin, 1994).

2.2 Transmission Mechanism

The transmission mechanism of 2-speed Simpson type planetary gear train is shown in Figure 3.

For shifting gears on this transmission, each ring gear has clutch B1, B2 which is operated by hydraulic to stop each ring gear spinning. By shifting from neutral to 1st gear, 1st clutch B1 is operated, 1st ring gear R1 is fixed, input provided to the sun gear which rotates planet gears P1, and output rotation is produced from the carrier. There is no difference shifting mechanism between 1st gear and 2nd gear, except which clutch is operated. Operating condition of gears, carrier and clutches, which is controlled by shifting gears, is shown in Table 3. "0" means stop, "1"

Table 2. Data of reduction gears.

	Number of teeth	
Gear position	1st	2nd
Sun gear	12	12
Planetary gear	32	11
Ring gear	78	36
Reduction gear ratio	7.5:1	4:1
Module	2	2

The negative sign indicates that the ring and sun gears

Table 1. Calculation of reduction gear ratio.

		S (Sun gear)	$R_1(R_2)$ (Ring gear)	C (Carrier)
1	C is fixed, S rotate at ω_1	ω_1	$-\frac{Z_S}{Z_{R1}}\omega_1$	0
1	Rotate all +	$\frac{Z_S}{Z_{R1}}\omega_1$	$\frac{Z_S}{Z_{R1}}\omega_1$	$\frac{Z_S}{Z_{R1}}\omega_1$
1+2	R (ω_R) is fixed	$\left(1 + \frac{Z_S}{Z_{R1}}\right)\omega_1$	0	$\frac{Z_S}{Z_{R1}}\omega_1$

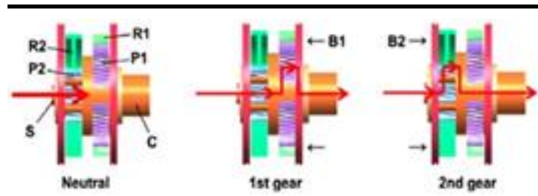

Figure 3. Shift gears mechanism.

Table 3. Shift gears operating condition.

Gear position	S	P_1	R_1	P_2	R_2	C	B_1	B_2
Parking	0	0	0	0	0	0	0	0
Reverse	1,ccw	1	0	-1	-1	1,ccw	1	0
Neutral	0	-1	-1	-1	-1	-1	0	0
1st	1,cw	1	0	-1	-1	1,cw	1	0
2nd	1,cw	-1	-1	1	0	1,cw	0	1

means rotating, and “-1” means idling condition (Yang *et al.*, 2009).

2.3 Design of the Transmission Clutch

Based on existing technologies and patent of clutch, multiple plate clutch system is selected as a suitable clutch for the transmission. This type of clutch has several driving members interleaved or stacked with several driven members. It is used in many different automobile transmissions, and also used in some electronically controlled all-wheel drive systems. The equation to calculate braking power of multiple plate clutch is defined as

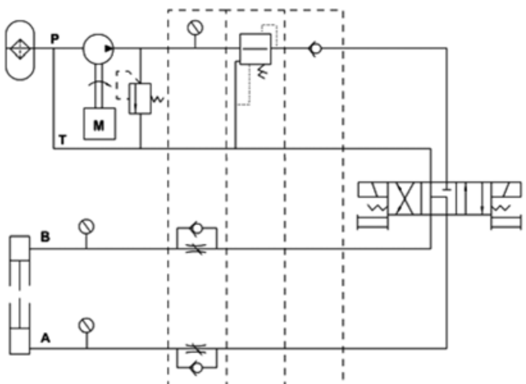

Figure 4. Hydraulic circuit diagram.

than 270 mm. Number of friction plates, friction coefficient, and friction plate pressure are calculated and shown in Table 4.

For applying transmission clutch system, hydraulic circuit diagram is designed and it is shown in Figure 4.

III. DESIGNING AND MANUFACTURING THE TRANSMISSION

3.1. 3D Design of the Transmission

$$T = Z(\mu Q_1)F\frac{1}{2} = \mu QF\frac{1}{2} \qquad (5)$$

where Dm is mean diameter of friction plate, μ is the friction coefficient of friction plate. Total torque capacity Q is increased by increasing number of friction plates Z. It means, multiple friction plates increase the torque capacity without the size of the unit.

In consideration of the motor size, the size of the transmission is limited to less than 300 mm × 300 mm × 200 mm, and diameter of friction plates is limited to less

Table 4. Specification of components.

			1st(7.5:1)	2nd(4:1)
Motor	Power	kW	150	
	Rotation speed	Rpm	3,500	
	Torque	Nm	409.4	
Friction plate	Plate number	Ea	4	3
	Frictional face number	Ea	8	6
	Outer diameter	mm	269	
	Inner diameter	mm	236	
	Coefficient of friction		0.1	
Piston	Outer diameter	mm	270	
	Inner diameter	mm	150	180
Axial direction braking force		kN	21.0831	97.3064
Friction plate pressure		MPa	16.2004	7.43187
Piston pressure		MPa	5.32432	3.05811
Required pressure		MPa	5.32432	3.05811

By using design of the transmission internal structure and multiple clutch system, 2-Dimension diagram of the transmission is designed by using CAD program and is shown in Figure 5. based on 2-Dimension diagram, 3- Dimension model of the transmission is designed by using 3D CAD program (Pro-Engineer 4.0) and is shown in Figure 6.

3.2 Analysis of Gear Strength and Gear Mesh Efficiency Suitability of gears we designed is verified by using gear design and analysis program, and the result of analysis including Root Safety Factor, Flank Safety Factor, Scuffing Table 5 (Kissling and Beerman, 2007).

Gear mesh efficiency is calculated when rotation speed of motor is increased from 1,000 RPM to 10,000 RPM. Plate

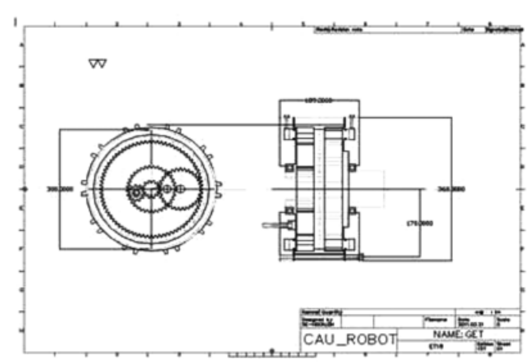

Figure 5. 2-D diagram of the 2-speed transmission

Table 5. Gear analysis result.

			1st(7.5:1)			2nd(4:1)	
Gear	Z_s		Z_{P1}	Z_{R1}	ZR_s	Z_{P2}	Z_{R2}
Root safety	2.70		1.92	1.56	3.51	1.21	2.65
Flank safety	1.11		1.24	1.63	1.09	1.11	1.02
Scuffing safety		2.88		4.78		3.89	8.90
Micro-pitting safety			1.42			1.11	

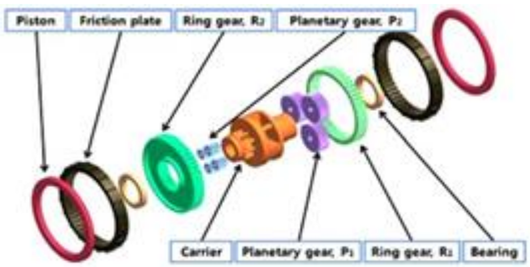

Figure 6. 3-D structure of the 2-speed transmission.

Table 6. Transmission efficiency analysis result.

Rotation speed(rpm)	1st(7.5:1)	2nd(4:1)
1,000	97.6%	95.9%
2,000	98.2%	96.9%
3,000	98.5%	97.4%
4,000	98.5%	97.5%
5,000	98.6%	97.8%
6,000	98.7%	97.8%
7,000	98.7%	97.9%
8,000	98.7%	98.0%
9,000	98.8%	98.0%
10,000	98.8%	98.1%

Maximum gear mesh efficiency is 98.8% which is occured in 1st gear when rotation speed of motor is 10,000 RPM which is maximum rotation speed. And also, maximum gear mesh efficiency in 2nd gear is occured in maximum rotation speed (Parker *et al*., 2000; Del Castillo, 2002; Bajer and Demkowicz, 2002).

Mean value of gear mesh efficiency is 98.51% in 1st gear and 97.53% in 2nd gear. and it come up to high standard. Gear mesh efficiency by sun gear angular velocity is shown in Table 6.

3.3 Manufacturing the Transmission Prototype

3D design of the transmission, verified by Analysis process and modified partially, is manufactured as shown in Figure 7. Compared with the first 3D design model, oil-cooled path flow and hydraulic power cylinder are added, and assembly convenience is improved.

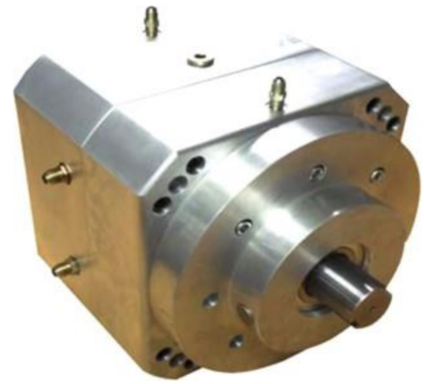

Figure 7. Prototype of the 2-speed transmission.

IV. TESTING THE TRANSMISSION PROTOTYPE

In order to verify suitability of the transmission prototype, measure of its vibration, noise, and power transmission efficiency is needed. The transmission test bed for these measurements is shown in Figure 8. The torque sensor in the nearby motor measures input shaft torque, and the other torque sensor measures output shaft torque.

For precise measurements, high-precision torque sensor and speed sensor are used. All measured data, including transmission power, efficiency, temperature and noise, is shown and saved in real-time by using CAN (Controller Area Network).

4.1 Noise Test

Sound-level meter is located 1m off the transmission and measures its noise in minimizing ambient noise environment. In common with vibration test, maximum noise occurs in 9,000 RPM, and its value is 88.99 dB. Noise test result is shown in Table 7.

4.2 Vibration Test

A pair of acceleration sensors measure vertical vibrationTable 7. Transmission noise test result.

Table 7. Transmission noise test result.

1st gear		2nd gear	
Rotation speed (rpm)	Noise level (dB)	Rotation speed (rpm)	Noise level (dB)
1,000	76.86	4000	83.28
2,000	78.98	5000	83.04
3,000	82.41	6000	86.52
4,000	84.38	7000	85.82
5,000	83.48	8000	87.73
6,000	84.31	9000	88.99
7,000	85.53		

Table 8. Transmission vibration test result.

Rotation speed (rpm)	Input housing vibration (mm/sec)		Output housing vibration (mm/sec)	
Direction	Vertical	Horizontal	Vertical	Horizontal
1,000	0.1663	0.2050	0.1464	0.1794
2,000	0.4649	0.5613	0.4142	0.4029
3,000	0.7492	1.0382	0.7022	0.7229
4,000	0.7236	0.9540	0.8474	0.8468
5,000	0.9223	0.9721	0.8209	0.8810
6,000	0.9551	1.1709	1.0401	1.1585
7,000	1.1714	1.4069	1.1345	1.3745
8,000	1.2939	1.7362	1.3381	1.5842
9,000	1.4456	1.8424	1.6069	1.6329

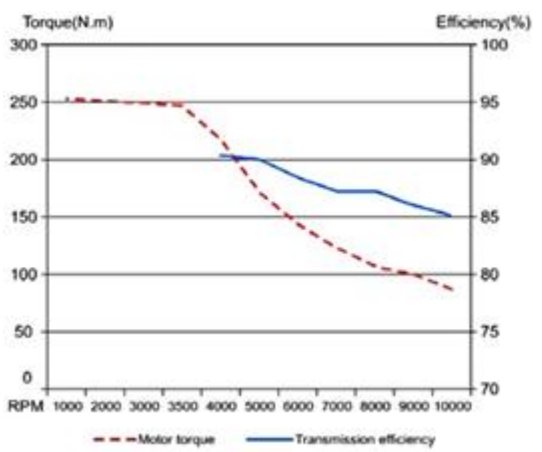

Figure 9. Power Transmission efficiency.

and horizontal vibration and each pair of sensors is mounted on input shaft housing and output shaft housing. The transmission vibration of each input shaft rotation speed is calculated by an integral of measured acceleration. The maximum vibration is 1.6369 mm/s, and it occurs in 9,000 RPM which is maximum speed of loaded motor. Vibration test result is shown in Table 8.

4.3. Power Transmission Efficiency Test

In order to know the power transmission efficiency, input shaft speed, input torque, output shaft speed, and output torque are measured by two torque sensors and two RPM sensors as shown in Figure 8. By using these mesured values, the input shaft power and the output shaft power can be calculated. The power transmission efficiency can be evaluated by dividing the output shaft power by the input shaft power. The efficiency of power transmission is over 85% as shown in Figure 9 (Irimescu *et al*., 2011).

Every experimental value of vibration test, noise test, and power transmission efficiency test meets the demand of required specification.

V. CONCLUSION

In this study, 2-speed shift gears mechanism is designed using simpson type planetary gear train for the electric commercial vehicle.

The transmission internal structure is designed by calculation of gear specification which has reduction ratio 7.5:1 and 4:1. And it is verified by analysis of gear strength and gear mesh efficiency. Mean value of gear mesh efficiency is above 97.53%. As a suitable clutch for the transmission, multiple plate clutch system is selected and its specification is calculated. Based on design of the transmission internal structure and multiple clutch system, 3D model of transmission is designed and its prototype is9,000 1.4456 1.8424 1.6069 1.6329 manufactured with some correction including oil-cooled path flow, hydraulic power cylinder and assembly convenience.

Through the vibration test, the noise test, and the power transmission efficiency test, the results as follow were obtained. The maximum vibration is 1.6369 mm/s and the maximum noise is 88.99 dB at 9,000 RPM which is maximum input shaft speed of loaded motor. Power transmission efficiency, calculated based on input shaft speed, input torque, output shaft speed, and output torque, is over 85%. Every experimental value meet the demand of required specification.

VI. ACKNOWLEDGEMENT

This paper was supported by the MKE(Ministry of Knowledge Economy), Korea, under the Advanced Technology Center support program.

REFERENCES

[1] Bajer, A. and Demkowicz, L. (2002). Dynamic contact/ impact problems, energy conservation, and planetary gear trains, *Computer methods in Applied Mechanics and Engineering*, 191, 4159−4191.

[2] Del Castillo, J. M. (2002). The analytical expression of theefficiency of planetary gear trains. *Mechanism and Machine Theory*, 37, 197−214.

[3] Dudley, D. W. (1984). *Handbook of Practical Gear Design*. McGraw-Hill. New York. 1.27−3.153, 7.1−7.51. Irimescu, A., Mihon, L. and Padure, G. (2011). Automotive transmission efficiency measurement using a chassis dynamometer. *Int. J. Automotive Technology* 12, 4, 555−
559.

[4] June, A. K. (1995). Planetary gear train dynamics. *J. Mechanical Design, Trans. ASME*, 116, 241−247.

[5] Kissling, U. and Beermann, S. (2007). Face gears: Geometry and strength. *Gear Technology*, Jan/Feb, 54−61.

[6] Litvin, F. L. (1994). *Gear Geometry and Applied Theory*.

[7] Prentice-Hall. New Jersey. 1−84, 331−345.

[8] Parker, R. G., Agashe, V. and Vijayakar, S. M. (2000). Dynamic response of a planetary gear system using a finite element/contact mechanics model. *J. Mechanical Design*, 122, 304−310.

[9] Strachan, P. J., Reynaud, F. P. and von Backstrom, T. W. (1992). The hydrodynamic modeling of torque converters. *N&O Joernaal*, Apr, 21−28

[10] Yang, H., Kim, B., Park, Y., Lim, W. and Cha, S. (2009). Analysis of planetary gear hybrid powertrain system part 2: Output split system. *Int. J. Automotive Technology* 10, 3, 381−390.