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A Study on Efficiency of Tapered Roller Bearing for an Automatic Transmission

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승용차 자동변속기용 테이퍼 롤러 베어링의 효율개선 연구

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ABSTRACT

Automotive fuel efficiency regulations and air pollution control are hot issues of recent years in the automotive industry. To solve these regulation problems, many studies are continuing to improve the transmission efficiency of transmissions. Tapered roller bearings are useful to improve the transmission efficiency in the recent automobile parts. The frictional losses in the tapered roller bearings are mainly composed of the rolling friction and the sliding friction, and are dependent upon the load, the lubrication, the rotation speed of bearings, and etc.

In this paper, the operating conditions of the transmission are defined and then the power losses of each bearing are calculated. In addition, improvement options are suggested after identifying the design factors influenced much by the improvement effect of power loss under the operating conditions of each bearing.

We compare the power losses of the entire transmission system due to bearing improvements by comparing the friction losses between the original design and the improved design. Lastly, it is shown that the calculated power losses are valid by comparing the test values and the theoretical values for the frictional torque characteristics of the original and improved bearings.

Key Words : Sliding Friction(미끄럼마찰), Rolling Resistance(구름저항), Transmission Efficiency(전달효율), Contact Area Roughness(접촉부 거칠기), Tapered Roller Bearing(테이퍼 롤러 베어링)

1. Introduction

Recently developed high-performance vehicles must be environmentally friendly and exhibit high efficiency in accordance with reinforced environmental regulations such as Euro 6.

Most vehicles provide an automatic transmission as a basic option in order to fulfill the convenience of the driver, but the power transmission efficiency of the automatic transmission is lower than that of the manual transmission.

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Efforts to improve the efficiency of the automatic transmission have focused on increased compactness, increased number of gears, and weight reduction[1-6].

In this study, it was calculated the power loss of a tapered roller bearing where it was installed on a 6-speed automatic transmission. Then, the friction torque characteristics of the tapered roller bearing according to the rotational speed of the bearing are compared with theories and experiments. The efficiency of power delivery in the 6-speed automatic transmission is discussed.

2. The Analysis of Power Loss of Tapered Roller Bearings

The friction torque of the tapered roller bearing has different characteristics depending on the load conditions of the transmission and the operating conditions such as lubrication. This torque may be reduced by optimizing the internal design of the bearings. A general description of the friction torque acting on the tapered roller bearing is shown in Figure 1[7-10]. The friction torque of the bearing can be divided into the following components: rolling friction of the bearing raceway surface, sliding friction between the inner ring rib and the roller, and the friction between the roller and the cage. It is expressed as follows (1)[11-12].

$$M = M_0 + M_1 \tag{1}$$

The rolling friction is influenced by the rotational speed of the bearings and the viscosity of the lubricant, and may be expressed as follows:

$$M_0 = f_0 (\nu n)^{2/3} d_m^3 \cdot 10^{-7} \tag{2}$$

Where, f_0 is the bearing type coefficient associated with the rotational speed, ν is the kinematic viscosity of the lubricant, n is the rotational speed, and d_m is the average diameter of the bearing.

The sliding friction is proportional to the bearing load and the bearing diameter, and can be expressed as follows:

$$M_1 = f_1 P_1 d_M \tag{3}$$

Where, f_1 is the bearing type coefficient associated with the bearing load, and P_1 is the magnitude of the equivalent load of the bearing action.

In general, the friction torque between the cage and the roller in sliding friction is considerably smaller than the friction torque between the roller end face and the inner ring rib[13].

The internal structure of the 6-speed automatic transmission is shown in Fig. 2. BearinX, a dedicated analysis software for Schaeffler Group, was employed in this study.

The analyzed tapered roller bearing is installed on the differential side of the transmission. Generally, the bearings applied to the differential side are located on the final deceleration shaft of the transmission and,

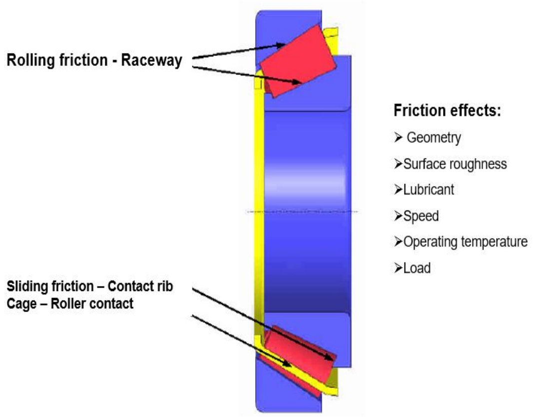


Fig. 1 Frictions of tapered roller bearing

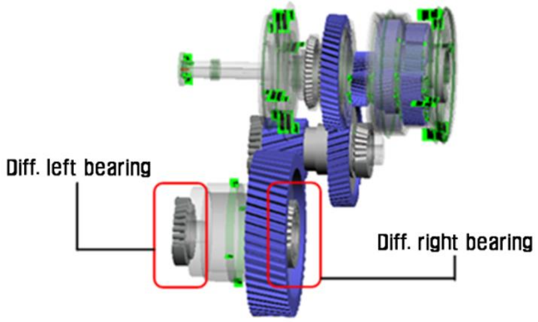


Fig. 2 Power loss analysis of transmission

Table 1 Bearing specification

Bearing Designation	Tapered roller bearing 32009 series
Inside diameter	45 mm
Outside diameter	75 mm
Width	20 mm
Number of rollers	23 EA
Dynamic load rating	61,000 N

Table 2 Boundary conditions of the transmission

Max. engine torque	235 N·m
Operating temperature	70℃
Viscosity of oil	22 mm^2/s at 40 ℃
Density of oil	849.0 kg/m^3
Bearing preload	2,000 N

Table 3 Boundary conditions of the bearing

Axial load	235 N·m
Operating temperature	70 ℃
Lubricant viscosity	22 mm^2/s at 40℃
Lubricant density	849.0 kg/m^3

hence, the bearings are large and are driven in a relatively low-speed range. The bearing specifications are shown in Table 1.

Table 2, Table 3, and Table 4 show the boundary conditions of the transmission, boundary conditions of the bearings, and driving conditions of the vehicle, respectively. The magnitude of the load acting on the bearing is determined by the maximum torque of the engine, and the viscosity of the lubricant is determined by the operating temperature.

Table 5 and Table 6 show the power loss values of the original bearings. These values are determined based on the boundary conditions shown in Table 2, Table 3, and Table 4.

According to Table 5 and Table 6, the large bearing friction torque associated with the low speed gear results from the relatively large bearing load value of this gear.

In the case of a high-speed gear, the bearing friction torque is small, but the power loss increases due to the relatively high rotation speed. Furthermore, in the case of friction energy, the bearing energy loss of high-frequency high-speed gears with numerous revolutions is considerably higher than that of the other gears.

Table 4 Load cases for power loss calculation

Load case	Duty(%)	Transmission input	
		Torque (N·m)	Engine speed (rpm)
1st drive	0.750	138.321	2,000
2nd drive	2.250	138.321	2,000
3rd drive	6.000	138.321	2,000
4th drive	15.000	138.321	2,000
5th drive	22.500	138.321	2,000
6th drive	28.275	138.321	2,000
Rev. drive	0.225	138.321	2,000

Table 5 Analysis results of the original left bearing

Load case	Frictional torque (N·m)	Speed (rpm)	Power loss (W)
1st drive	1.537	154.94	0.187
2nd drive	0.988	247.48	0.576
3rd drive	0.732	362.56	1.668
4th drive	0.641	470.86	4.741
5th drive	0.640	652.61	9.841
6th drive	0.665	845.35	16.645
Rev. drive	0.363	192.79	0.016
Sum.			33.674

Table 6 Analysis results of the original right bearing

Load case	Frictional torque (N·m)	Speed (rpm)	Power loss (W)
1st drive	0.501	154.94	0.061
2nd drive	0.330	247.48	0.192
3rd drive	0.261	362.56	0.595
4th drive	0.273	470.86	2.019
5th drive	0.402	652.61	6.181
6th drive	0.481	845.35	12.040
Rev. drive	1.189	192.79	0.054
Sum.			21.142

Table 7 Modification of design parameters

No.	Sliding contact roughness	Number of rollers
Original bearing	Ra 1.0	23 EA
Model 1	Ra 0.7	22 EA
Model 2	Ra 0.5	21 EA
Model 3	Ra 0.3	20 EA

3. Design Analysis of New Tapered Roller Bearings

In this study, two design variables are selected for design of the optimized bearing. The first variable is the roughness generated during sliding friction between the inner ring rib and the sliding friction side. The second variable is the number of taper roller bearings (this variable has significant influence on the rolling friction). Friction torque graphs for both variables are respectively shown in Fig. 3.

The design variables considered in this study are applied to three different bearings (see Table 7). Depending on the analytical results corresponding to the variables, model 3 (with higher efficiency than the other bearings) is selected. We analyze model 3 under the same conditions considered in the previous analysis. The results obtained for the power loss associated with the bearing friction are shown in Tables 8 and 9 as well as Fig. 4.

The results of the new bearing (model 3) show that, for each load condition, the friction torque is lower than that of the other bearings. As shown in Table 4, at high driving frequency, the driving loss is reduced 2.9 times and 2.7 times at the 5th drive and 6th drive, respectively.

When a new bearing is applied to the transmission based on the considered result, the friction torque decreases by 65.2%.

4. Verification Test

Based on the analysis results of the new bearing (see Fig. 5), a bearing friction torque measuring device is used to perform the test.

The test conditions are listed in Table 10. The temperature, flow rate, and axial load of the lubricating oil are determined by the driving environment of the bearing installed on the transmission. Furthermore, the frictional torque of the bearing is determined from the rotational speed of the main shaft.

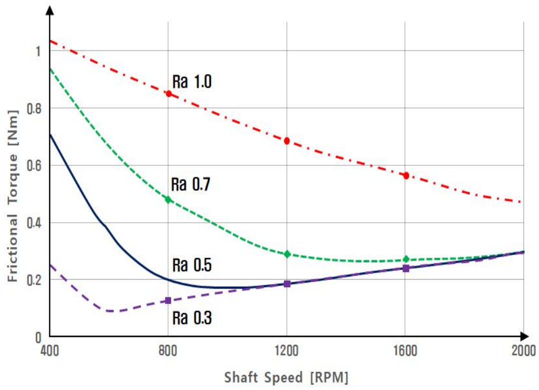
These results revealed that the friction torque value of the bearing varies with the roughness of the sliding friction surface, i.e., the bearing friction loss can be significantly reduced by improving the friction-surface roughness. In addition, the rolling resistance is if the number of rollers is small. However, this effect is slight, owing to the low kinematic viscosity value of the lubricating oil and the relatively low rotational speed.

Table 8 Analysis results of the modified left bearing

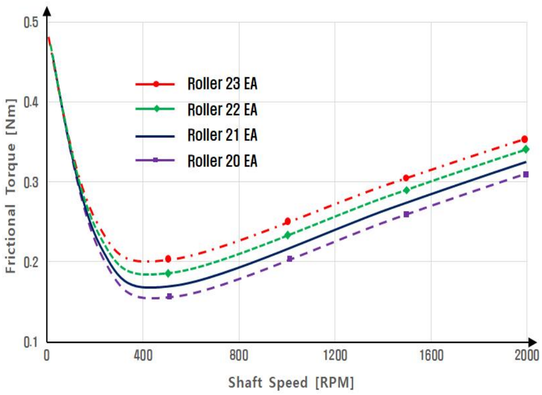
Load case	Frictional torque (N·m)	Speed (rpm)	Power loss (W)
1st drive	1.559	154.94	0.190
2nd drive	0.660	247.48	0.385
3rd drive	0.304	362.56	0.693
4th drive	0.216	470.86	1.598
5th drive	0.203	652.61	3.121
6th drive	0.224	845.35	5.607
Rev. drive	0.337	192.79	0.015
Sum.			11.609

Table 9 Analysis results of the modified right bearing

Load case	Frictional torque (N·m)	Speed (rpm)	Power loss (W)
1st drive	0.499	154.94	0.061
2nd drive	0.219	247.48	0.128
3rd drive	0.115	362.56	0.262
4th drive	0.109	470.86	0.806
5th drive	0.166	652.61	2.553
6th drive	0.200	845.35	5.006
Rev. drive	1.009	192.79	0.046
Sum.			8.862



(a) Effect by roughness of sliding contact area



(b) Effect by number of rollers

Fig. 3 Analysis results of frictional torque

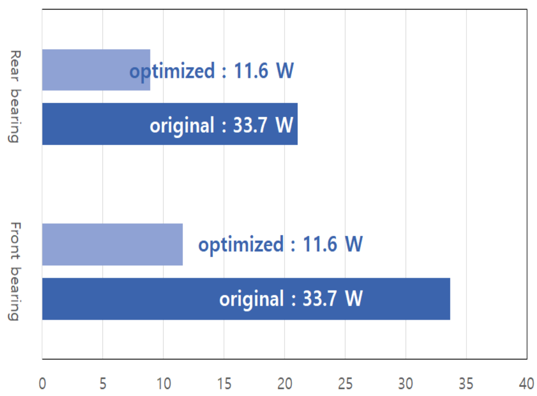


Fig. 4 Results of power loss reduction

Table 10 Test condition of bearing

Thrust Load	5,000 ± 0.2 N
Temp. of lubricant	70 ± 5 ℃
Lubricant	SK AFT 6S SP-4
Flow of oil	0.5 ± 0.01 L/min
Rotation speed	200 ~ 2,000 rpm

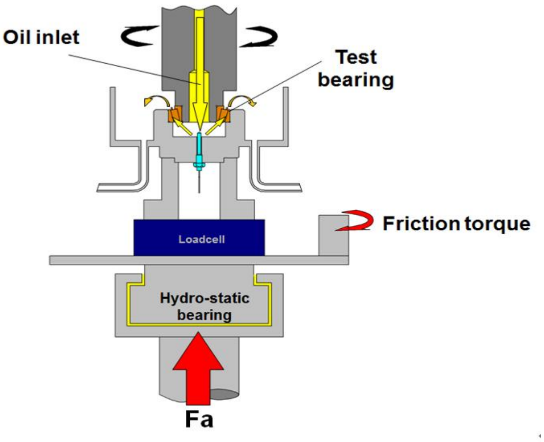


Fig. 5 Test rig of frictional torque

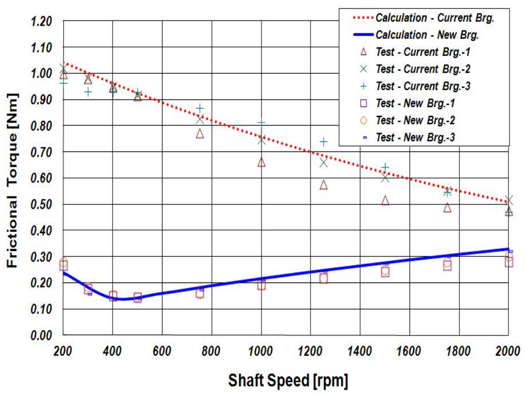


Fig. 6 Comparison of calculation results and test results

5. Analysis and Test Results

The final analysis and test results are shown in Fig. 6. As shown in the figure, the theoretical analysis value is similar to the test value. This indicates the validity of the power loss value obtained from simulation of the actual driving environment inside the transmission (see Fig. 4). Moreover, for values lower than 2,000 rpm, the efficiency of the new bearing is higher than that of the original bearing.

6. Conclusion

In this work, the friction characteristics of tapered roller bearings are evaluated through the friction loss analysis of a 6-speed automatic transmission.

We have demonstrated, through theoretical and experimental verification, that the transmission efficiency can be improved by the friction torque bearing design variables.

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