A Study on the Efficiency of Tapered Roller Bearings

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Abstract: - An automatic transmission is one of the most popular systems to shift gears for passenger cars. But it has much more power losses than manual transmission. In this study, the power losses of bearings applied for automatic transmission are calculated. Internal geometry, lubrication and roughness of contact area tapered roller bearings are critical influential factors of the bearing frictional torque. Bearing frictional torque is theoretically investigated and verified by test according to the rotational speed of shaft.

Key-Words: - Sliding Friction, Rolling Resistance, Tapered roller bearing, Oil Film, Contact Area Roughness, Power loss

Symbols

T_0= load-independent frictional torque
T_1= load-dependent frictional torque
d_m= mean diameter of rolling bearing
f_1= coefficient for the load-dependent frictional torque for the reference condition
P_1= reference load
f_n= kinematic viscosity of the lubricant under the reference conditions
v_n= rotational speed of bearings

1 Introduction

As environmental regulations have been strengthened and vehicle performances are highly advanced recently, environment-friendly, high efficiency vehicles are in the spotlight of consumers. Accordingly, a number of auto manufacturers are willing to expend a lot of expenses and time for high efficiency, compactness, and lighter weight of car transmissions to live up to consumers' demands and stricter environmental regulations. In addition, most of the passenger cars being produced currently adopt automatic transmissions for convenience in driving although automatic transmissions are disadvantageous regarding the lower efficiency of power delivery than that of manual transmissions.

This study is to calculate loss of power in taper roller bearings applied to a 6-speed automatic transmission for passenger cars, grasp the friction torque characteristics of taper roller bearings according to the rotational speed based on the theoretical interpretation and experiment, and improve the efficiency of automatic transmission as to power delivery.[1]

2 Interpretation of Power Loss of Taper Roller Bearings

As for friction torques of bearings, the characteristics may vary depending on the operational conditions such as load and lubrication of the transmission, and ways of reducing friction torques can be sought by optimizing the internal design of bearings.

Fig. 1 presents the overview of friction torques of taper roller bearings. *T*, which indicates the friction torque of bearings, involves T_0, the rolling friction torque of the plane of the bearing orbit, and T_1, the sliding friction torque between the inner ring rib and roller, and between the roller and gauge as in equation (1) below:

$$T = T_0 + T_1 \quad (1)$$

T_0, the rolling friction torque, is presented in equation (2), which shows that this is affected to a large extent by the rotational speed of bearings and the viscosity of the lubricant.[2, 3]

$$T_0 = 10^{-7} \times f_n (v_n)^{2/3} d_m^{\,3} \quad (2)$$

T_1, the sliding friction, is presented in equation (3), which shows that this is in proportion to the size of the load and sliding friction.[4]

$$T_1 = f_1 \times P_1 \times d_m \quad (3)$$

As for sliding friction, the size of the friction torque between the cage and roller is quite small in general compared to the friction torque between surface of the roller and inner ring rip.

Fig. 2 shows the overview of the internal structure of 6-speed automatic transmission. The bearings to be interpreted are taper roller bearings that are embedded on the differential side in the transmission as in Fig. 2. Table 1 shows the size of the bearings.[5, 6]

In general, bearings applied to the differential side are located in the final reduction axis of the transmission so that they can operate in small size at a low rotational speed.

Table 2 shows the boundary condition of a transmission, Table 3 the boundary condition of a bearing, and Table 4 the condition of vehicle operation respectively. The maximum engine torque decides the size of load applied to the bearings, and the viscosity of lubricant may change depending on the operating temperature.

On the assumption that a lubricant is in the thermal equilibrium at 70 ℃, when the taper roller bearings are adopted to a transmission, a certain amount of preload is required, and a certain measure of preload is applied as load to the bearings.[7]

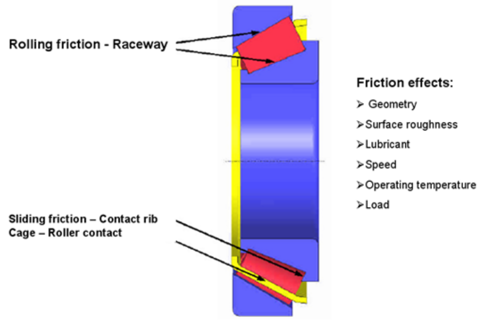

Fig. 1 Frictions of a tapered roller bearing

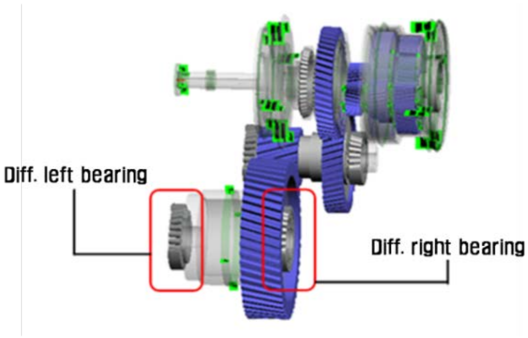

Fig. 2 Power loss analysis of transmission

Based on the boundary conditions presented in Tables 2 to 3, the amount of power loss during the time unit of operating existing bearings is calculated as in Table 4 and Table 5.

As indicated in the interpretation, the load of bearings in low gear is relatively heavy, which results in increasing the bearings friction torque accordingly. In high gear, the size of the bearings friction torque may be small while the power loss increases due to the fast rotation. In addition, it was confirmed that in high gear with fast revolution and the frequency of use, the power loss of bearings drastically increases.

Table 1 Bearing specification

Tapered roller bearing	32009
Inside diameter (mm)	45
Outside diameter (mm)	75
Width (mm)	20
Number of rollers	23
Dynamic load rating (N)	61000

Table 2 Boundary conditions of the transmission

Max. engine torque (N·m)	235
Operating temperature (℃)	70
Lubricant viscosity (mm^2/s at 40 ℃)	22
Lubricant density (kg/m^3)	849.0
Bearing preload (kN)	2

Table 3 Boundary conditions of the bearing

Axial load (kN)	5
Operating temperature (℃)	70
Lubricant viscosity (mm^2/s at 40 ℃)	22
Lubricant density (kg/m^3)	849.0

Table 4 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
1st coast	0.375	154.94	0.015
2nd drive	0.988	247.48	0.576
2nd coast	0.436	247.48	0.085
3rd drive	0.732	362.56	1.668
3rd coast	0.484	362.56	0.368
4th drive	0.641	470.86	4.741
4th coast	0.517	470.86	1.275
5th drive	0.640	652.61	9.841
5th coast	0.563	652.61	2.886
6th drive	0.665	845.35	16.645
6th coast	0.605	845.35	5.048
Rev.drive	0.363	192.79	0.016
Rev.coast	0.838	192.79	0.013
Sum.			43.363

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Table 5 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
1st coast	1.046	154.94	0.042
2nd drive	0.330	247.48	0.192
2nd coast	0.850	247.48	0.165
3rd drive	0.261	362.56	0.595
3rd coast	0.766	362.56	0.582
4th drive	0.273	470.86	2.019
4th coast	0.736	470.86	1.815
5th drive	0.402	652.61	6.181
5th coast	0.721	652.61	3.696
6th drive	0.481	845.35	12.040
6th coast	0.727	845.35	6.066
Rev. drive	1.189	192.79	0.054
Rev. coast	0.282	192.79	0.004
Sum.			33.512

3 Interpretation of the new designed taper roller bearings

This study involves the following two basic design variables for the optimal designing of bearings: first, roughness of the sliding friction side; and second, the number of bearing rollers in the taper roller bearings.

Table 6 shows the three new bearings reflecting the design variables in this study.

Fig. 3 shows the result of interpreting the friction torque in application of the design variables above by means of the bearing program developed and used by Scheffler Korea.

In application of the interpretation results based on the roughness of the sliding part and the number of rolling bodies, the new three bearings with higher efficiency than existing ones are selected.[8]

Based on the interpretation results above, the newly designed three bearing were applied to the transmission model, and the power loss due to bearings friction was interpreted as in Fig. 4 and Tables 7~8. All of the boundary conditions are the same with those in section 2.

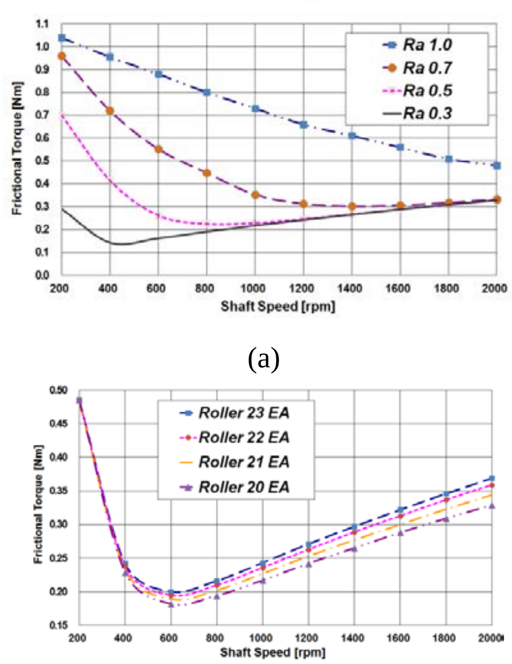
In examination of the interpretation results of the new bearings (New bearing #3), it turned out that the value of the friction torque was less than in the case of existing bearings in every load condition.

Especially in the 5th and 6th speed conditions, where the driving load is high, the reduction effect was even doubled.

Based on the findings above, it turned out that the application of the new bearings to the transmission resulted in friction torque reduction effects of 65.2 % for the left-side bearing and 61.8 % for the right-side bearing respectively.

Table 6 Modification of design parameters

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



(a)

(b)

Fig. 3 Analysis results of frictional torque
(a) Effect by roughness of sliding contact area
(b) Effect by number of rollers

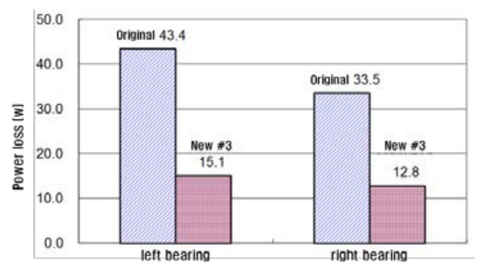


Fig. 4 Results of power loss reduction

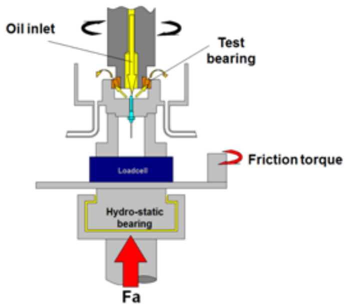


Fig. 5 Test rig of frictional torque

Table 7 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
1st coast	0.357	154.94	0.014
2nd drive	0.660	247.48	0.385
2nd coast	0.318	247.48	0.062
3rd drive	0.304	362.56	0.693
3rd coast	0.219	362.56	0.166
4th drive	0.216	470.86	1.598
4th coast	0.188	470.86	0.463
5th drive	0.203	652.61	3.121
5th coast	0.192	652.61	0.984
6th drive	0.224	845.35	5.607
6th coast	0.218	845.35	1.819
Rev. drive	0.337	192.79	0.015
Rev. coast	0.784	192.79	0.012
Sum.			15.129

Table 8 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
1st coast	1.040	154.94	0.042
2nd drive	0.219	247.48	0.128
2nd coast	0.568	247.48	0.110
3rd drive	0.115	362.56	0.262
3rd coast	0.312	362.56	0.237
4th drive	0.109	470.86	0.806
4th coast	0.233	470.86	0.574
5th drive	0.166	652.61	2.553
5th coast	0.211	652.61	1.081
6th drive	0.200	845.35	0.061
6th coast	0.230	845.35	0.042
Rev. drive	1.009	192.79	0.128
Rev. coast	0.261	192.79	0.110
Sum.			12.829

Table 9 Test conditions of bearings

Axial load (kN)	5±0.1
Inlet oil temperature (℃)	70±2
Oil inlet flow (L/min)	0.5±0.01
Speed of inner ring (rpm)	200 ~ 2000

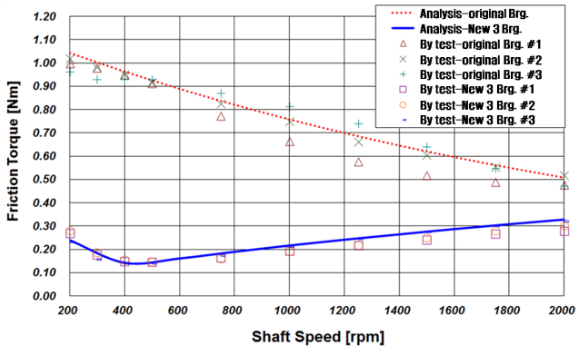


Fig. 6 Comparison of analytic results and test results

4. Verification Test

The verification test of the new bearings was implemented by means of the bearing friction torque measuring unit as shown in Fig. 5.

The test conditions are presented in Table 9. In referent to the driving conditions of the bearings applied to the transmission, the temperature of the lubricant, quantity, load in direction of the axis, etc are decided.

The bearing friction torque values depending on the rotational speed of the main axis were measured and analyzed.

The rest results show the changes in torque values depending on roughness of the sliding friction side, which indicates that improvement of the surface roughness can drastically decrease the loss due to bearing friction.

In addition, as the number of roller decreases, the rolling resistance decreases accordingly. However, the effects turned out to be insignificant due to the low viscosity of the lubricant.

Based on the interpretation and test results above, the final results are presented as a graph in Fig. 6. As indicated in the graph, the interpretation values are similar to those of the test.

Besides, it turned out that the new bearings showed an excellent efficiency especially in an applicable area of 2,000 rpm or less compared to existing bearings.

5. Conclusion

This study is to grasp the characteristics of friction

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torque generation based on the friction loss interpretation of taper roller bearings applied to 6-speed automatic transmissions for passenger cars. The way of increasing the efficiency of transmissions and its possibility are presented based on the theoretical, experimental verification process on changes in the friction torque depending on the design factors of bearings.

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References:

[1] Yoo In Shin, Jin-Young Lee, Chul Ki Song, Jung-Wan Park, Jung Kyu Jeong, "Lightning Design for Transmission Gears", *Proceeding of the KSME Fall Annual Meeting*, pp. 292-294, 2010

[2] Harris, T. A., "Rolling Bearing Analysis", 4th *Edition, Wiley,* pp. 483 - 495, 2001

[3] ISO 15312 “Rolling bearing – Thermal speed rating – Calculation and Coefficients ”, 2003

[4] Brandlein Johannes, Eschmann Paul, Hasbargen Ludwig, Weigand Karl, "Ball and Roller Bearings 3th Edition", *Wiley*, pp. 213 - 224, 1999

[5] ISO 281 “Rolling bearings – Dynamic load ratings and rating life”. 2007

[6] Tae Jo Park, “Elastohydrodynamic Film Thickness in Elliptical Contacts with Rolling and Spinning”, *KSTLE* Vol. 24, No. 6, December 2008, pp. 355~361

[7] ISO 15, “Rolling bearings – Radial bearings – Boundary dimensions, general plan”, 1998

[8] Berthold Martin, Harold E. Hill, “Design and Selection Factor for Automatic Transaxle Tapered Roller Bearings” *SAE Technical Paper Series* 920609