The strength analysis of a complex planetary gear system for tunnel boring machine reducer

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Abstract

The power train on the reducer of driving cutter head device for a tunnel boring machine makes use of a 3-stages complex planetary gear system. The 3-stages planetary gears are very important parts of the reducer for tunnel boring machine because of the strength problem. The purpose of this study is to calculate the specifications of the 3-stages planetary gears and to analyze the gear bending and compressive stresses of the planetary gears. It is necessary to analyze gear bending and compressive stresses confidently for optimal design of the complex planetary gear system in respect of cost and reliability. In the paper, researchers analyze actual gear bending and compressive stresses of the 3-stages planetary gears using Lewes and Hertz equation and verify the calculated specifications of the complex planetary gear system by evaluating the results with the data of allowable bending and compressive stresses from the Stress - No. of cycles curves of gears. Researchers also analyze *PVT* factor of gear scoring and evaluate the possibility of scoring failure on the 3-stages planetary gears for the tunnel boring machine reducer.

Keywords: gear bending stress, gear compressive stress, gear scoring, planetary gear, reducer, tunnel boring machine

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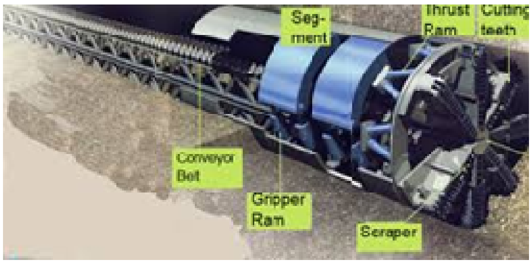
Introduction

Tunnel boring machine (TBM) is a piece of large-scale and highly effective tunnel construction equipment with multi-functions such as excavation, advance, muck discharge, supporting, ventilation and dust prevention. It drives cutter head and disc cutter to rotate, crushes face rock through disc cutter rolling and accomplishes tunnel excavation.

Currently, 90% of domestic roads, railways, subways, power stations, communication districts, and water supply and sewage tunnel works are using NATM (New Austrian Tunneling Method) construction method. The NATM integrates the principles of the behavior of rock masses under load and monitoring the performance of underground construction during construction because the NATM method involves problems such as noise and vibration caused by blasting.

The TBM method is rapidly expanding in the overseas construction market including the high-speed railway projects of China and Brazil. Modern TBMs typically consist of the rotating cutting wheel, called a cutter head, followed by a main bearing, a thrust system and trailing support mechanisms. The type of machine used depends on the particular geology of the project, the amount of ground water present and other factors. Since most domestic TBMs depend on imports, the development of the localization is an urgent field. The cutter head is one of the key parts of TBM with complex structure. TBM cutter heads can come in different type, either for soft or mixed ground geologies or for very compact massive rock grounds. The reducer is driven by an electric motor or a hydraulic motor for TBM.

In this paper, the simulation was adopted to analyze the strength and effects of planetary gear train have on static and dynamic performance of cutter head. The shield type TBM is shown in (Figure 1), TBM typically consist of the rotating cutter head, a thrust system and trailing support mechanisms.

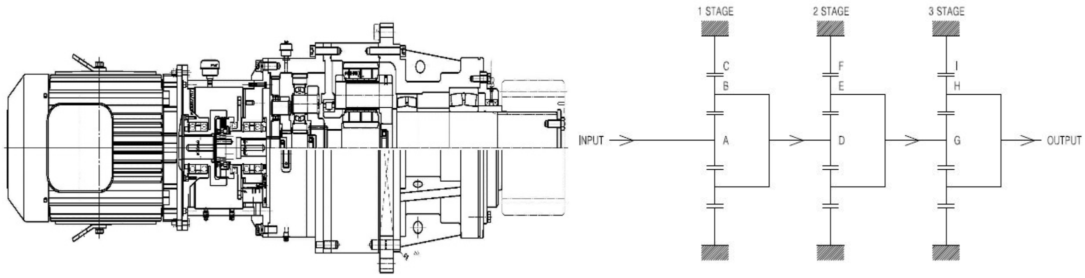


(a) Structure of shield type TBM

(b) Exterior photo of TBM

Figure 1 Photograph of the shield type TBM.

(Figure 2) shows a schematic diagram for the analytical model of TBM, (a) shows a combined body of the TBM reducer objective to be analyzed and (b) shows a gear train of the 3-stage planetary gear system. (Table 1) shows the specifications of the TBM reducer.



(a) Design of TBM (b) Structure of gear train

Figure 2 Schematic diagram of analytical model.

Table 1 Specifications of the TBM reducer.

gear ratio	life span (h)	input torque/speed (Nm/rpm)			output torque/speed (Nm/rpm)
		1 stage (8.333:1)	2 stage (5.777:1)	3 stage (4.750:1)	
228.7 : 1	10,000	492 / 1,460	4,100 / 175.2	23,684 / 30.32	112,500 / 6.38

Gear teeth are damaged due to the lack of fatigue strength, compound planetary gears and severe operating conditions of a TBM reducer that become a problem.

Several investigations have been reported, as cited by Imwalle (1972), load equalization in planetary gear systems. Seager (1972) established load distribution calculation of the planetary gears. Cunliffe *et al.* (1974), dynamic tooth loads inepicyclic gears for planetary gears. Castellani and Castelli (1980) also cited the gear strength analysis method. Coy *et al.* (1975) further emphasized the dynamic capacity and surface pressure durability life of spur and helical gears. Oda and Tsubokura (1981) similarly stressed the effect of bending endurance strength for addendum modification of spur gears and was likewise investigated. There is also an inclusion of typical bending strength calculation of planetary gears AGMA 218.01 (1982) and Gear Handbook by Dudley (1984) that shows the bending strength calculation method of planetary gears. This study, developed the gear specifications calculation program and produced detailed specifications of the differential planetary gear system for TBM reducer based on Gear Handbook by Dudley.

Moreover, it also developed the stress analysis program of differential planetary gear system by Lewis (1982) and Hertz equation and analyzed the safety factor of gear bending and compressive stresses consider required life time of TBM reducer and the S/N curve presented in the Gear Handbook by Dudley. It also verified the predictive validity with respect to the developed programs. (Figure 3) shows the equation system solving with gear specifications calculation and strength analysis of the differential planetary gear system for TBM reducer. It also verified the predictive validity with respect to the development and estimate of the scoring failure of differential planetary gears by scoring factor analysis.

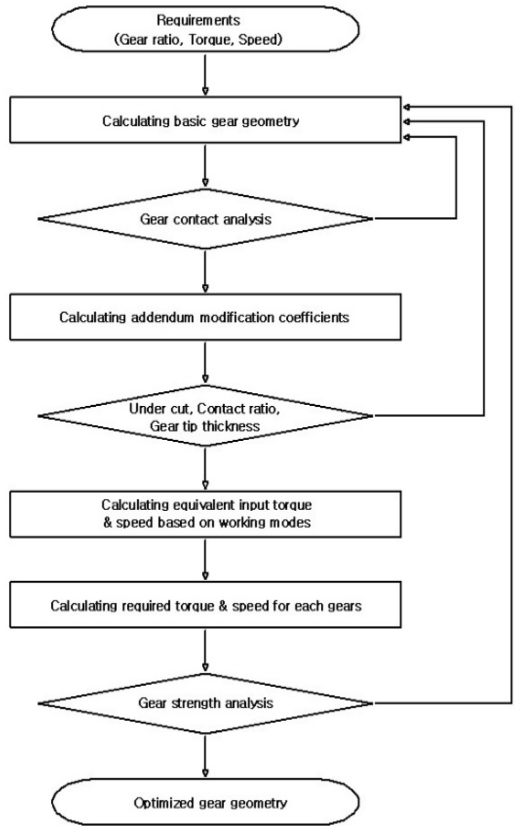


Figure 3 Equation systems solving with gear specifications calculation and stress analysis.

Methodology

1. Calculation of gear specifications

(Table 2) shows the calculated specifications of the planetary gears for TBM on No. 3^{rd} stage and (Figure 4) indicates the results of calculated Lewes bending factors on No. 3^{rd} stage gears.

Table 2 Specifications of planetary gears on no. 3rd stage gear.

items	sun gear (G)	pinion gear (H)	ring gear (I)
module	9	<---	<---
press angle	25^0	<---	<---
No. of gear teeth	16	22	60
tooth modification factor	0.4670	0	0.4670
outside dia.	170.4	216	530.4
over pin measurement	$184.383^{-0.163}_{-0.626}$($\phi$20)	$222.541^{-0.202}_{-0.774}$($\phi$16.5)	$524.859^{+0.293}_{+1.171}$($\phi$16)
face width	135	140	140
backlash	0.22-0.84		0.25-0.97
center distance	175		175
contact ratio	1.2723		1.4768

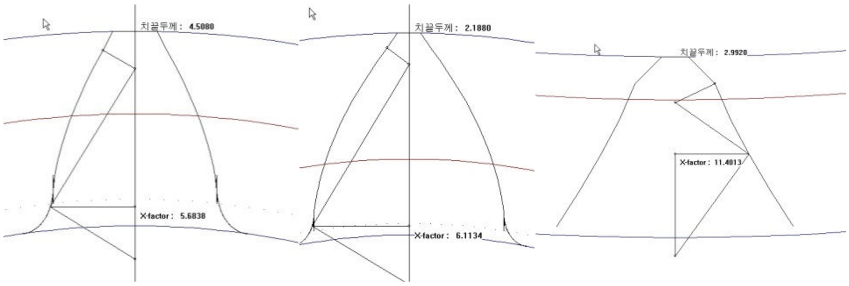


Figure 4 The results of the gear specifications calculation program.

2. Torque and number of rotation analysis

From the schematic diagram in (Figure 2), the gear ratio of TBM reducer calculated by a relative speed diagram method is as follows:

$$\mathrm{Y} = \left\{\frac{Z_{S1}+Z_{R1}}{Z_{S1}}\right\} \times \left\{\frac{Z_{S2}+Z_{R2}}{Z_{S2}}\right\} \times \left\{\frac{Z_{S3}+Z_{R3}}{Z_{S3}}\right\} \quad (1)$$

The number of rotation for each planetary gear calculated by relative speed diagram method is as follows:

$$N_{P_{1,2,3}} = N_{S_{1,2,3}} \times \left\{\frac{Z_{S_{1,2,3}} Z_{R_{1,2,3}}}{Z_{P_{1,2,3}}\left(Z_{S_{1,2,3}} - Z_{R_{1,2,3}}\right)}\right\} \quad (2)$$

$$N_{C_{1,2,3}} = N_{S_{1,2,3}} \times \left\{\frac{Z_{S_{1,2,3}}}{\left(Z_{S_{1,2,3}} + Z_{R_{1,2,3}}\right)}\right\} \quad (3)$$

3. Gear bending stress analysis

The actual gear bending stress equation by Lewes formula is as follows:

$$s = \frac{29{,}400\pi T}{N_a F X Z} \quad (4)$$

Whereas S is actual gear bending stress (N/mm^2), T is torque on gears(Nm), N_a is contact length of action (mm), F is face width of gear (mm), X is Lewes bending factor (mm), Z is number of teeth in gear.

Allowable gear bending stress equation by Gear Handbook of Dudley and AGMA Standard 218.01 including gear bending S/N curve is as follows:

$$Sab = \frac{C_1}{N_F^{\frac{1}{20.8}}} \quad (5)$$

Whereas *Sab*is allowable gear bending stress (N/mm^2), N_F is No. of cycles, C_1 is coefficient.

4. Gear compressive stress analysis

The actual gear compressive stress, P (N/mm^2) applied at the tip of the planetary gears based on contact formula of Hertz is as follows:

In the case of external gear contact, the actual gear compressive stresses of sun gear and pinion gear are,

$$P_S = 19.43\sqrt{\frac{2\pi T_S \times CDSIN\alpha}{A_S(CDsin\phi - A_S)\times F_c \times N_a \times Z_S}} \quad (6)$$

$$P_P = 19.43\sqrt{\frac{2\pi T_S \times CDSIN\alpha}{A_P(CDsin\phi - A_P)\times F_c \times N_a \times Z_S}} \quad (7)$$

In the case of internal gear contact, the actual gear compressive stresses of pinion gear and ring gear are,

$$P_P = 19.43\sqrt{\frac{2\pi T_P \times CDSIN\alpha}{A_P(CDsin\phi + A_P)\times F_c \times N_a \times Z_P}} \quad (8)$$

$$P_R = 19.43\sqrt{\frac{2\pi T_P \times CDSIN\alpha}{A_R(A_R - CDsin\phi)\times F_c \times N_a \times Z_P}} \quad (9)$$

Whereas α is normal pressure angle, ϕ is transverse pressure angle, $T$is torque on driving gear (Nm), F_c is active face width in contact (mm), Z is No. of gear teeth, CD is operating center distance, N_a is contact length of action (mm), $A = \sqrt{OR^2 - BR^2}$, OR is outside radius of gear, BR is base radius of gear.

Allowable gear compressive stress equation by Gear Handbook of Dudley and AGMA Standard 218.01 including gear compressive S/N curve is as follows:

$$Sac = \frac{C_2}{N_F^{\frac{1}{6.8433}}} \quad (10)$$

Whereas *Sac* is allowable gear compressive stress (N/mm^2), N_F is No. of cycles, C_2 is coefficient.

5. Gear Scoring factor analysis

Gear scoring factor is based on Gear Handbook of Dudley. To predict scoring failure of differential planetary gears, *PVT* (Nm/sec· mm) is as follows:

In the case of external gear contact, the actual gear scoring factor of sun gear and pinion gear are,

$$PVT_S = \frac{\pi N_S}{0.08525}\left(\frac{Z_P+Z_S}{Z_P}\right)(A_S - PR_S \times SIN\phi)^2 \quad (11)$$

$$PVT_P = \frac{\pi N_P}{0.08525}\left(\frac{Z_P+Z_S}{Z_S}\right)(A_P - PR_P \times SIN\phi)^2 \quad (12)$$

In the case of internal gear contact, the actual gear compressive stresses of pinion gear and ring gear are,

Whereas P is actual compressive stress (N/ mm^2), V is sliding velocity (m/sec), T is contact length from pitch point to contact point (mm), N is speed of gears (rpm), Z is number of gear teeth, ϕ is operating pressure angle.

Gear scoring factor, *PVT* must not exceed 38, 860 (Nm/sec· mm) to prevent scoring failure under HRC 60, carburized gears in mineral oil condition.

Results and discussion

1. The results of gear bending and compressive stress analysis

Calculating actual gear bending and compressive stresses of planetary gear system and considering allowable gear bending and compressive stresses, produce safety factors and verify the problems of gear strength for the calculated specifications of the planetary gear system in TBM reducer.

(Figure 5) shows the results of gear bending stress analysis and (Figure 6) the results of gear compressive stress analysis of planetary gear system which consists of nine gears (A to I) in TBM reducer.

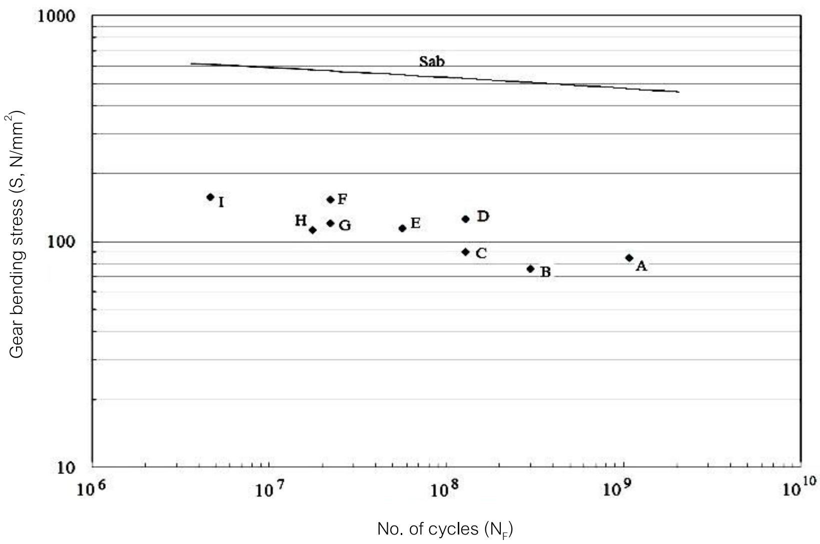


Figure 5 The results of gear bending stress analysis.

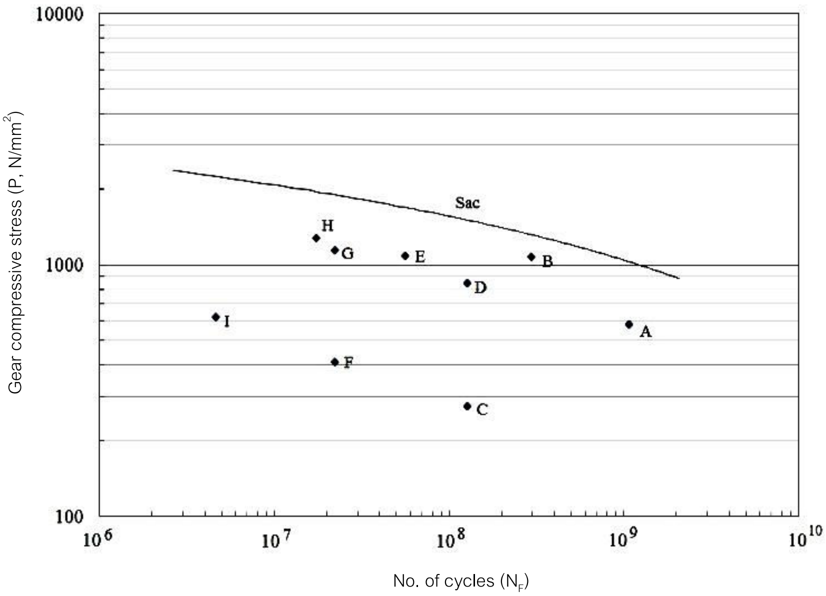


Figure 6 The results of gear compressive stress analysis.

Table 3 Calculated actual gear scoring factor.

items	sun gear (A)	pinion gear (B)		ring gear (C)
actual scoring factor (Nm/s· mm)	694	267	1344	153

(a) No.1 stage, sun gear (A) + pinion gear (B) + ring gear (C)

items	sun gear (D)	pinion gear (E)		ring gear (F)
actual scoring factor (Nm/s· mm)	253	98	527	53

(b) No.2 stage, sun gear (D) + pinion gear (E) + ring gear (F)

Items	sun gear (G)	pinion gear (H)		ring gear (I)
actual scoring factor (Nm/s· mm)	336	159	709	163

(c) No.3 stage, sun gear (G) + pinion gear (H) + ring gear (I)

It can be shown that actual gear bending and compressive stresses of the differential planetary gears are under the allowable gear bending and compressive stresses in these S/N curves. The compressive stress safety factor satisfies in all respects. Thus, calculation results are set safely and have been verified as valid.

2. The results of Gear Scoring factor analysis

This calculates the actual scoring factor, *PVT* and judges the safety of gear scoring failure considering the limit value, 38,860 (Nm/sec· mm). (Table 3) shows the results of calculated actual gear scoring factor.

Conclusion

This study, analyzed actual gear bending and compressive stresses of the differential planetary gears using Lewes and Hertz equation and the calculated specifications of differential planetary gears on the 75 ㎾ grade TBM cutter head and the results with the data of allowable bending and compressive stress from the Stress-No. of cycles curves of gears, based on Gear Handbook of Dudley and AGMA Standard 218.01 have been evaluated. A single pinion type planetary gear is used for simulation. The outcomes of this study can be summarized as:

(1) The strength of the differential planetary gears and developed programs have been verified, in respect of the result of gear bending and compressive stress analysis of calculated specifications of differential planetary gears of TBM cutter head.

(2) In respect of the result of actual scoring factor analysis of calculated specifications on the differential planetary gears, below the limit value 38,860 (Nm/sec· mm), scoring failure is not predicted.

(3) The developed programs calculating the specifications and analyzing the gear bending and compressive stresses and gear scoring factor of the planetary gear system for TBM reducer are expected to be effectively utilized. Also future researches on more excellent planetary gear system of the various reducers for construction machines are expected to be still performed.

Acknowledgement

This work was supported by the Ministry of Trade, Industry and Energy (project number: 10053018-Korea, The 75 ㎾ grade planetary gearbox for cutter head device of tunnel boring machine) and the authors are gratefully appreciative of the support.

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