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Design Improvement of Mechanical Transmission for Tracked Small Agricultural Transporters through Gear Strength Analysis

Hong-Gon Kim[1], Yeon-Ju Jo[1], Chul-Soo Kim[1,2], Yong-Ho Han[3], Dae-Cheol Kim[1,2]*

[1]*Department of Bio-Industrial Machinery Engineering, Chonbuk National University, Jeonju, Korea*
[2]*Institute for Agricultural Machinery & ICT Convergence, Chonbuk National University, Jeonju, Korea*
[3]*Han's Mechatronics. Co., Ltd., Daejeon, Korea*

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Abstract

Purpose: The gear strength of a new mechanical transmission designed to increase the loading weight of small 4.8 kW tracked agricultural transporters was analyzed. Design improvements to increase the gear strength and reduce the gear weight were proposed after examining the parameters. **Methods:** Sixteen operators from three regions were surveyed to obtain the usage profile of small 4.8 kW transporters. Gear strength was evaluated by calculating contact stress and tooth root stress using commercial software following ISO 6336. **Results:** From the strength calculation for each gear pair, contact stress smaller than tooth root stresses were produced in all gear pairs. The safety factors in most cases exceeded 1.0, except in the case of gear pair II in group II. The design life of the transporter using gear pair II in group II was 42% under harsh conditions—thus, this design life needs improvement. A robust design was proposed by examining the relevant parameters (face width and profile shift coefficient) to increase the design life of the transporter. In addition, a lightweight design for gear pair I in group II that was considered overdesigned was proposed by examining the face width to reduce the weight of the drive gear by 42% and that of the driven gear by 30%. **Conclusions:** The Safety factor for the design life was examined through a gear strength analysis. After examining the relevant parameters, conditions for strength improvement were proposed to increase design life or adjust overdesigned gear. However, load conditions differ depending on the working conditions or user's preferences; therefore, it is necessary to conduct further studies in various regions.

Keywords: Design improvement, Gear strength analysis, Parameter design, Transmission, Transporter

Introduction

Mechanization for fruit farms in Korea is a challenge because most fruit farms are located in hilly areas (Jang, 2011). Multifunctional transporters for lifting and dumping are preferred over expensive agricultural machinery (Kwac and Kim, 2008). Further, in Korean fruit farms, tracked transporters are more commonly used than wheeled ones owing to the nature of the slopes.

It is important to understand the working conditions to improve powered vehicle transmissions because the working conditions of agricultural powered vehicles differ depending on soil conditions, types of works, and habits of the driver (Kim et al., 1998). Transmissions for such vehicles should allow change of speed and torque based on different working conditions, and they also should have appropriate life expectancy.

Gear design of such vehicles should satisfy the requirements for contact stress and tooth root stress and the load capacity for the tooth surface and tooth root so that damage during operation can be avoided (Kong et al., 2011).

Transmissions for small transporters are produced by empirical overdesigning or by design modification when a problem arises. However, damaged gear leads to breakage of power transmission apparatus and may result in human injury or loss of loads. Therefore, systemic gear design is necessary to improve the reliability and stability of the

***Corresponding author:** Dae-Cheol Kim
Tel: +82-63-270-2590; **Fax:** +82-63-270-2620
E-mail: dckim12@jbnu.ac.kr

transmission.

In fruit farms, small 4.8 kW tracked agricultural transporters are used widely, and mechanical transmissions are used to support the determined loading weight. In this study, the gear strength of a new mechanical transmission designed to increase the loading weight of these transporters was analyzed. Upon examination of the relevant design parameters, design improvements to increase the gear strength and reduce the gear weight were proposed. The objective of this study was to increase the strength of weak gears and to reduce the weight of overdesigned gears through strength analysis and parameter examination.

Materials and Methods

Small agricultural transporter

Figure 1 shows the small agricultural transporter used in this study, and Table 1 presents its specifications.

Figure 1. Small agricultural transporter used in this study.

Transmission

This study intended to develop a new transmission for small transporters with loading weights of 6,860 N because most of the transporters used had smaller loading capacity (4,900 N) than what is actually required. The specifications of the transmission are listed in Table 2.

Figure 2 shows the transmission used in this study: SI, SII, and SV represent the input shaft, reverse shaft, and output shaft, respectively. This study focused on analyzing the gear strength of the forward gear because the reverse gear was not used much. GP represents gear pair that engaged in both axes, and it includes the drive gear and driven gear. Each gear pair comprised spur gears, and the pressure angle was 20°. Table 3 lists the specifications of the transmission gears used in this study. Figure 3 shows gear structure and power flow of the transmissions.

Table 2. Specifications of the transmission used in this study

Item	Specification
Shifting type	Sliding gear
Rated input speed (rpm)	1,300
Gear material	SCM 415
ISO Accuracy grade	10
Lubricant type / lubricant	Oil bath lubrication / VG-220
Gear (forward/reverse)	3 / 3

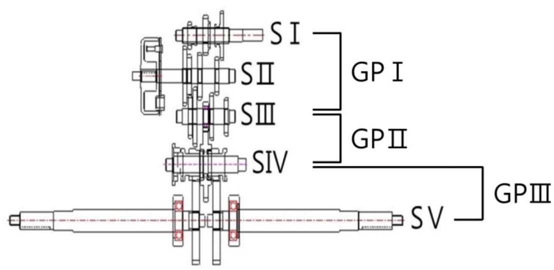
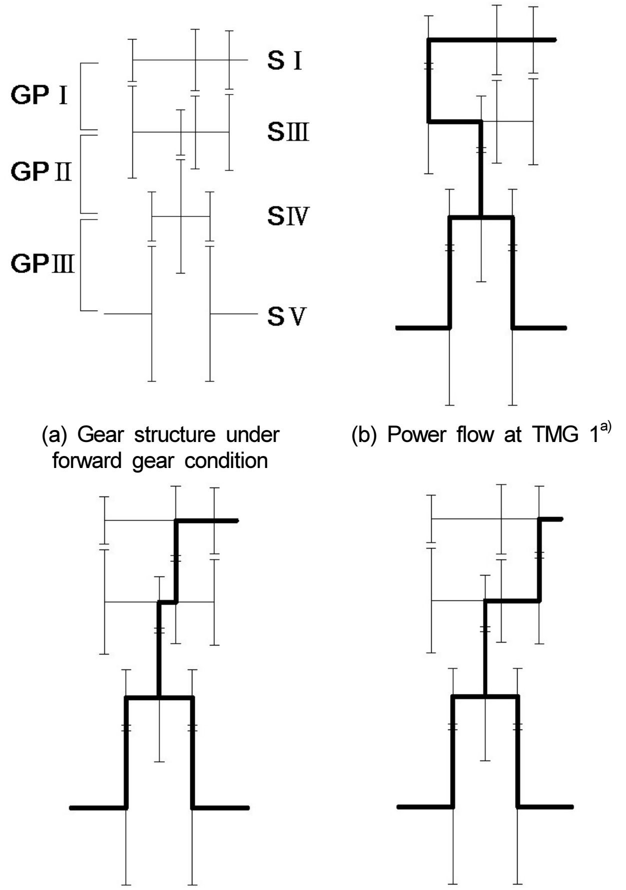


Figure 2. Gear train of the transmission used in this study.

Table 1. Specifications of the small agricultural transporter

Item		Specification
Drive system	Length × Width × Height (mm)	1,980 × 1,040 × 1,100
	Weight (N)	3,645
	Rated engine power (kW) / speed (rpm)	4.8 / 1,800
	Gear (forward / reverse)	3 / 2
	Operational speed (km/h)	1.4 / 2.8 / 4.5
Cargo system	Type	Hydraulic dump, lift
	Length × Width × Height (mm)	1,150 × 900 × 200
	Loading weight (N)	4,900



(a) Gear structure under forward gear condition

(b) Power flow at TMG 1[a]

(c) Power flow at TMG 2[a]

(d) Power flow at TMG 3[a]

[a]Transmission gears

Figure 3. Gear structure and power flow of the transmission.

Usage profile

A usage profile comprising the working speed, load, and time is necessary for the design and evaluation of mechanical components. The usage profile created by considering the actual life span and all operating conditions provides ideal and accurate data, but most usage profiles are created for selected typical operating conditions. In this study, a usage profile was created by considering annual working hours, driving ratio between flat areas and slopes, loading weight, and gear setting.

Driving axle torque calculation

Strength analysis was conducted using torque because the main load transmitted by the gear was torque. Driving axle torque was assumed as the torque required to overcome rolling resistance. The maximum driving axle torque was calculated by determining the driving axle torque corresponding to engine power and that to rolling resistance.

The driving axle torque derived from engine power was calculated by using the speed of the driving axles and engine power, and the speed of the driving axles was calculated using engine speed and the transmission gear ratio, as shown in equation (1). Driving axle power was calculated using engine output power (4.8 kW) and power efficiency. Therefore, the driving axle torque derived from engine power can be calculated by using equation (2). Power efficiency was determined as 0.94 because it

Table 3. Information of the transmission gears

		GP I			GP II	GP III
		TMG 1	TMG 2	TMG 3		
Number of teeth	Drive	15	23	29	13	12
	Driven	36	28	22	37	38
Normal module (mm)		2.5	2.5	2.5	3.0	3.5
Center distance (mm)		64.15	64.15	64.15	75.70	88.30
Face width (mm)	Drive	8.00	10.70	7.50	11.00	14.50
	Driven	8.00	8.00	8.00	11.00	14.00
Pitch circle diameter (mm)	Drive	37.50	57.50	72.50	35.00	42.00
	Driven	90.00	70.00	55.00	111.00	133.00
Profile shift coefficient	Drive	0.1508	0.1334	0.1508	0.2413	0.2600
	Driven	0.0129	0.0303	0.0129	0.0000	-0.0238
Root diameter (mm)	Drive	31.95	51.90	66.95	33.10	34.85
	Driven	83.75	63.85	48.75	103.40	123.95
Tip diameter (mm)	Drive	43.20	63.15	78.20	46.60	50.60
	Driven	95.00	75.10	60.00	116.90	139.70

[a]Transmission gears

was 98% for each axle of transmission and three axles were involved.

$$N_o = \frac{N_i}{i} \quad (1)$$

where, N_o : the speed of driving axles (rpm)
N_i : input speed of transmission (rpm)
i : gear ratio of transmission

$$T_E = \frac{60000 \times P}{2\pi \times N_o} \times \eta \quad (2)$$

where, T_E : driving axle torque from engine power (N·m)
P : engine power (kW)
N_o : speed of driving axles (rpm)
η : power efficiency

Rolling resistance between the track and ground on flat areas was calculated using equation (3), and the rolling resistance on slopes was calculated using equation (4). Finally, driving axle torque derived from rolling resistance was calculated using equation (5).

$$R_{r_1} = W_1 \times R_1 \quad (3)$$

where, R_{r_1} : rolling resistance on flat areas (N)
W_1 : gross weight on flat areas (N)
R_1 : rolling resistance coefficient on flat areas

$$R_{r_2} = (W_2 \times \cos\theta \times R_2) + (W_2 \times \sin\theta) \quad (4)$$

where, R_{r_2} : rolling resistance on slopes (N)
W_2 : gross weight on slopes (N)
R_2 : rolling resistance coefficient on slopes
θ : slopes angle (°)

$$T_R = R_r \times r \quad (5)$$

where, T_R : driving axle torque from rolling resistance (N·m)
R_r : rolling resistance (N)
r : rolling resistance coefficient on flat areas

Gear strength evaluation

ISO 6336 (ISO standards, 2006) was used for evaluating gear strength. Most gear failures were due to the breakage resulting from bending stress at the tooth root and contact stress at the tooth surface when two gear teeth were engaged.

Surface durability represents the safety of the pitting, and it was calculated using the contact stress acting on tooth surface and pitting stress limit (i.e., the allowable contact stress; Yang, 2010; Yim, 2010), which were calculated as shown in equations (6–8).

$$\sigma_{H0} = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}} \quad (6)$$

where, σ_{H0} : nominal contact stress at the pitch point (N/mm^2)
Z_H : zone factor
Z_E : elasticity factor
Z_ε : contact ratio factor
Z_β : helix angle factor
F_t : nominal tangential load (N)
b : face width (mm)
d_1 : reference diameter of pinion (mm)
u : gear ratio

$$\sigma_H = Z_B \sigma_{H0} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \quad (7)$$

where, σ_H : contact stress (N/mm^2)
Z_B : single pair tooth contact factor
σ_{H0} : nominal contact stress at the pitch point (N/mm^2)
K_A : application factor
K_V : dynamic factor
$K_{H\beta}$: face load factor for contact stress
$K_{H\alpha}$: transverse load factor for contact stress

$$\sigma_{HG} = \sigma_{Hlim} Z_{NT} Z_L Z_V Z_R Z_W Z_X \quad (8)$$

where, σ_{HG} : pitting stress limit (N/mm^2)
σ_{Hlim} : allowable contact stress number (N/mm^2)
Z_{NT} : life factor for test gears for contact stress
Z_L : lubricant factor
Z_V : velocity factor
Z_R : roughness factor
Z_W : work hardening factor
Z_X : size factor for contact stress

Safety factor for the surface durability was calculated using the contact stress and pitting stress limit as shown in equation (9).

$$S_H = \frac{\sigma_{HG}}{\sigma_H} \tag{9}$$

where, S_H : safety factor for surface durability
σ_{HG} : pitting stress limit (N/mm^2)
σ_H : contact stress (N/mm^2)

Tooth root bending strength indicated safety with regard to tooth breakage, and it was calculated using bending stress acting on a tooth root and the bending stress limit (i.e., allowable bending stress; Yang, 2010; Yim, 2010, which were calculated as shown in equations (10–12).

$$\sigma_{F0} = \frac{F_t}{b\, m_n} Y_F Y_S Y_\beta Y_B Y_{DT} \tag{10}$$

where, σ_{F0} : nominal tooth root stress (N/mm^2)
F_t : nominal tangential load (N)
b : face width (mm)
m_n : normal module (mm)
Y_F : form factor
Y_S : stress correction factor
Y_β : helix angle factor
Y_B : rim thickness factor
Y_{DT}: deep tooth factor

$$\sigma_F = \sigma_{F0} K_A K_V K_{F\beta} K_{F\alpha} \tag{11}$$

where, σ_F : tooth root stress (N/mm^2)
σ_{F0} : nominal tooth root stress (N/mm^2)
K_A : application factor
K_V : dynamic factor
$K_{F\beta}$: face load factor for tooth root stress
$K_{F\alpha}$: transverse load factor for tooth root stress

$$\sigma_{FG} = \sigma_{Flim} Y_{ST} Y_{NT} Y_{\delta relT} Y_{RrelT} Y_X \tag{12}$$

where, σ_{FG} : tooth root stress limit (N/mm^2)
σ_{Flim} : nominal bending stress number (N/mm^2)
Y_{ST} : stress correction factor
Y_{NT} : life factor for tooth root stress
$Y_{\delta relT}$: relative notch sensitivity factor
Y_{RrelT} : relative surface factor
Y_X : size factor related to tooth root strength

Safety factor for bending strength was calculated using the bending stress and bending stress limit as shown in equation (13).

$$S_F = \frac{\sigma_{FG}}{\sigma_F} \tag{13}$$

where, S_F : safety factor for bending strength
σ_{FG} : tooth root stress limit (N/mm^2)
σ_F : tooth root stress (N/mm^2)

Gear design and analysis software

Equations in the ISO 6336 are complicated because they include many coefficients. Therefore, commercial software (KISSsoft 03/2014D, KISSsoft AG, Switzerland) was used to calculate the gear strength based on the calculation formula. The face width in the drawing of gear was set to a small value, as shown in Table 4, and 18CrMo4 was considered as the material of the gear; this material is similar to the actual material SCM415. Table 5 lists the input data for KISSsoft. ISO 6336:2006 Method B was used for calculation because of its accuracy.

Results and Discussion

A survey for usage profile

Usage profile

The usage profile was created considering gears used on flat areas (Group I) and those used on both flat areas and slopes (Group II). Assuming a service life of 10 years, the design life based on the transmission gear set are listed Table 6.

Gross weights are listed in Table 7. The gross weights of the actual transporter, including the weight when empty, were 8,545 N on flat areas (both for group I and II) and 8,055 N on slopes.

Thus, the maximum loading weights were determined

Table 4. Information on the face width input to KISSsoft

		GP I			GP II	GP III
		TMG 1	TMG 2	TMG 3		
Face width (mm)	Drive	6.41	9.11	5.91	11.00	13.17
	Driven	7.21	7.21	7.21	11.00	14.00

Table 5. Data input to KISSsoft

	Data		Remarks
Basic data	Module, Center distance, Number of teeth, Face width, Profile shift coefficient	See Table 3	
	Pressure angle	20°	
	Quality	10	ISO 1328
	Material	18CrMo4	Original SCM415
	Lubrication	VG-220	Oil bath lubrication
Reference profile	Root diameter	See Table 3.	
	Tip diameter		
Tolerances	Tooth thickness tolerance	e27	DIN 3967
Rating	Calculation method	ISO 6336:2006 Method B	
	Application factor (ISO 6336)	1.25	Single load
		1.00	Load spectrum
	Torque	See Table 11	
	Speed		
	Required service life		
	Load spectrum		

Table 6. Design life based on transmission gear

Ground conditions		TMG 1	TMG 2	TMG 3	Sum
Group I	Flat areas (h)	680	420	200	1,300
Group II	Flat areas (h)	430	480	190	1,100
	Slopes (h)	840	190	70	1,100

Table 7. Gross weight information

Transporter	Group	Ground conditions	Empty weight (N)	Maximum loading weight (N)	Gross weight (N)
Survey	I, II	Flat areas	3,645	4,900	8,545
	II	Slopes	3,645	4,410	8,055
Research object	I, II	Flat areas	4,625	6,860	11,485
	I	Slopes	4,625	6,174	10,799

Table 8. Driving axle torque calculated from engine power

TMG	Gear ratio	Axle speed (rpm)	Driving axle torque from engine power (N·m)
1	21.63	60.1	717.8
2	10.97	118.5	364.1
3	6.84	190.1	226.9

Table 9. Driving axle torque calculated from rolling resistance

Ground conditions	Group	Rolling resistance (N)		Driving axle torque (N·m)	
		R = 0.12	R = 0.2	R = 0.12	R = 0.2
Flat areas	I, II	1,378	2,297	89.6	149.3
Slopes	II	3,152	4,002	204.9	260.2

as 6,860 N on flat areas and 6,174 N on slopes. The maximum loading weight ratio between the flat areas and slopes was 0.9, which was the same as that for the actual transporter. The gross weights, including the weight when empty, were 11,485 N on flat areas and 10,799 N on slopes in this study.

Driving axle torque calculation

Driving axle torque from the transmission gear to engine power was calculated using equations (1) and (2); the values are listed in Table 8. Engine output was assumed as the rated output power.

Driving axle torque by the rolling resistance was calculated using equation (5), and the values are listed in Table 9. The rolling resistance was calculated using equation (3) for flat areas and using equation (4) for slopes. The rolling resistance coefficient (Esaki, 1986) was 0.12 (normal conditions) in uplands and 0.2 (harsh conditions) in a paddy field. The gross weight values listed in table 7 and an inclination angle of 17.6% were used; the radius of the front wheel was 0.065 m.

Table 10 presents the driving axle torque from engine power and the driving axle torque from rolling resistance based on the transmission gear set. The driving axle torque curves for different transmission gear sets are shown in

Table 10. Driving axle torques calculated by different methods

TMG	Driving axle torque from engine (N·m)	Driving axle torque from rolling resistance (N·m)			
		Flat areas		Slopes	
		R = 0.12	R = 0.2	R = 0.12	R = 0.2
1	717.8				
2	364.1	89.6	149.3	204.9	260.2
3	226.9				

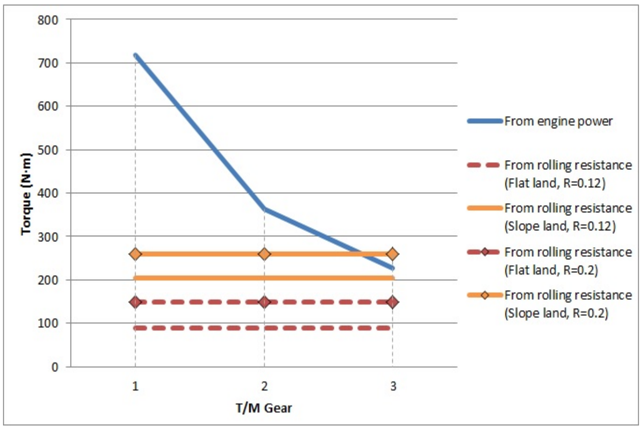


Figure 4. Driving axle torque curves.

Figure 4. Transmission gear set 3 under the harsh conditions (slope) produced the maximum driving axle torque from the engine power. In addition, the maximum driving axle torque was attributed to the rolling resistance.

The load spectrum of each transmission gear set, including design life and speed, was created as presented in Table 11.

Gear strength calculation

Torque, design life, and speed were obtained from Table 11. When the safety factor for tooth surface durability and tooth root bending strength exceeded 1.0, the gear strength was considered to satisfy the design life.

Table 12 lists the calculation results of gear strength, and all the safety factors, except the tooth surface durability of the drive gear of GP II in group II, exceeded 1.0.

Information on the design life that satisfied the safety factor of 1.0 for the tooth surface durability is presented in Table 13. The drive gear of GP II in group II needs improvement because it had a life of 913 h, which is 42% of the design life (2,200 h). Transmission gear set 2 of GP I was overdesigned because it had a life of 48,386 h compared to the design life of 670 h. The appropriate

Table 11. Load spectrum of the transmission

Gears			Items							
			Design life		Torque (N·m)			Speed (rpm)		
			Time (h)	Ratio (%)	S I	S III	S IV	S I	S III	S IV
Group I	Flat areas (R = 0.12)	1	680	52.5	4.1				542	190
		2	420	32.5	8.2	9.9	14.1	1,300	1,068	375
		3	200	15.0	13.1				1,714	602
	Flat areas (R = 0.2)	1	680	52.5	6.9				542	190
		2	420	32.5	13.6	16.6	23.6	1,300	1,068	375
		3	200	15.0	21.8				1,714	602
Group II	Flat areas (R = 0.12)	1	430	19.4	4.1				542	190
		2	480	21.9	8.2	9.9	14.1	1,300	1,068	375
		3	190	8.7	13.1				1,714	602
	Slopes (R = 0.12)	1	840	38.1	9.5				542	190
		2	190	8.7	18.7	22.7	32.4	1,300	1,068	375
		3	70	3.2	30.0				1,714	602
	Flat areas (R = 0.2)	1	430	19.4	6.9				542	190
		2	480	21.9	13.6	16.6	23.6	1,300	1,068	375
		3	190	8.7	21.8				1,714	602
	Slopes (R = 0.2)	1	840	38.1	12.0	28.9	41.1		542	190
		2	190	8.7	23.7			1,300	1,068	375
		3	70	3.2	33.2	25.2	35.8		1,714	602

Table 12. Gear strength calculation

Group	GP	TMG	Root safety (R = 0.12)		Flank safety (R = 0.12)		Root safety (R = 0.2)		Flank safety (R = 0.2)	
			Drive	Driven	Drive	Driven	Drive	Driven	Drive	Driven
I	I	1	5.48	6.52	1.30	1.53	3.88	4.62	1.12	1.32
		2	6.44	5.04	1.44	1.49	4.61	3.60	1.25	1.29
		3	3.67	4.17	1.43	1.34	2.64	3.00	1.24	1.16
	II	1,2,3	6.72	6.83	1.25	1.47	4.72	4.80	1.07	1.26
	III	1,2,3	7.32	8.16	1.34	1.69	5.05	5.63	1.15	1.44
II	I	1	3.62	4.30	1.09	1.26	3.06	3.64	1.01	1.18
		2	4.40	3.44	1.30	1.34	3.77	2.96	1.21	1.25
		3	2.40	2.73	1.28	1.20	2.36	2.69	1.22	1.14
	II	1,2,3	3.89	3.95	1.00	1.20	3.41	3.47	0.93	1.11
	III	1,2,3	4.11	4.59	1.09	1.36	3.54	3.95	1.01	1.25

Table 13. Life satisfying a safety factor of 1.0 for flank safety

Group	GP	TMG	Life satisfying 1.0 (h) (R = 0.12)	Remarks: design life (h) (R = 0.12)	Life satisfying 1.0 (h) (R = 0.2)	Remarks: design life (h) (R = 0.2)
I	I	1	∞	680	31,163	680
		2	∞	420	∞	420
		3	∞	200	5,579	200
	II	1,2,3	∞	1,300	12,877	1,300
	III	1,2,3	∞	1,300	28,716	1,300
II	I	1	18,379	1,270	1,799	1,270
		2	384,249	670	48,386	670
		3	3,947	260	2,929	260
	II	1,2,3	2,176	2,200	913	2,200
	III	1,2,3	5,608	2,200	2,480	2,200

design that satisfies the safety factor 1.0 should be realized through optimal designs.

Investigating parameters

The calculated gear strength values show that the drive gear of GP II in group II does not satisfy the safety factor of 1.0 for tooth surface durability under harsh conditions. The design life of transmission gear set 2 of GP I should be 670 h with a safety factor of 1.21; however, it was overdesigned to a design life of 48,386 h.

Tooth surface durability was influenced by the number of teeth, module, face width, profile shift coefficient, and tooth profile modification. Among these parameters, face width and profile shift coefficient were considered in this study. Tooth profile modification was not considered owing to the processing costs for considering this parameter.

Parameter design of gears

Face widths of gear pairs of GP II were 11 mm. Changes in gear strength were observed by increasing the face

Table 14. Parameter design for face width

Face width (mm) Drive	Face width (mm) Driven	Root safety Drive	Root safety Driven	Flank safety Drive	Flank safety Driven
9.0		3.23	3.28	0.87	1.04
9.5		3.28	3.34	0.89	1.07
10.0		3.34	3.39	0.91	1.09
10.5		3.36	3.42	0.91	1.09
11.0		3.41	3.47	0.93	1.11
11.5		3.46	3.52	0.94	1.13
12.0		3.50	3.56	0.96	1.15
12.5		3.57	3.63	0.97	1.16
13.0		3.66	3.72	0.99	1.18

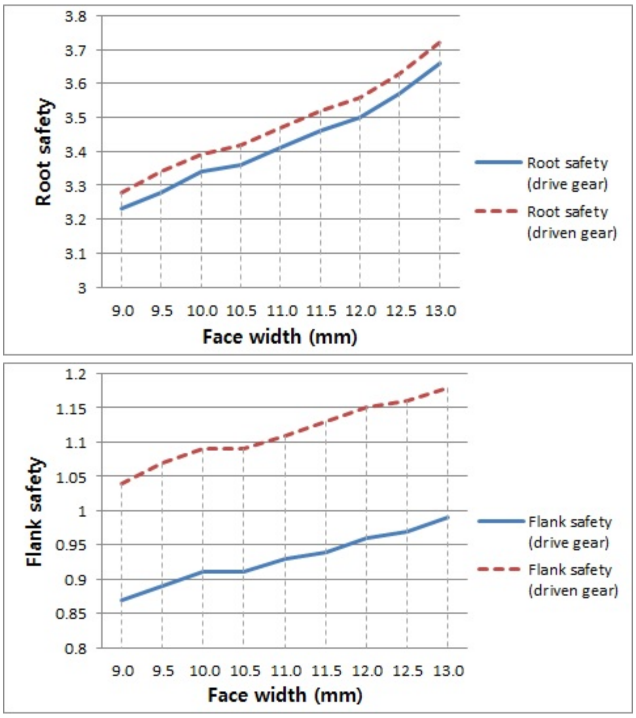


Figure 5. Curves of root safety and flank safety factors corresponding to tooth face width.

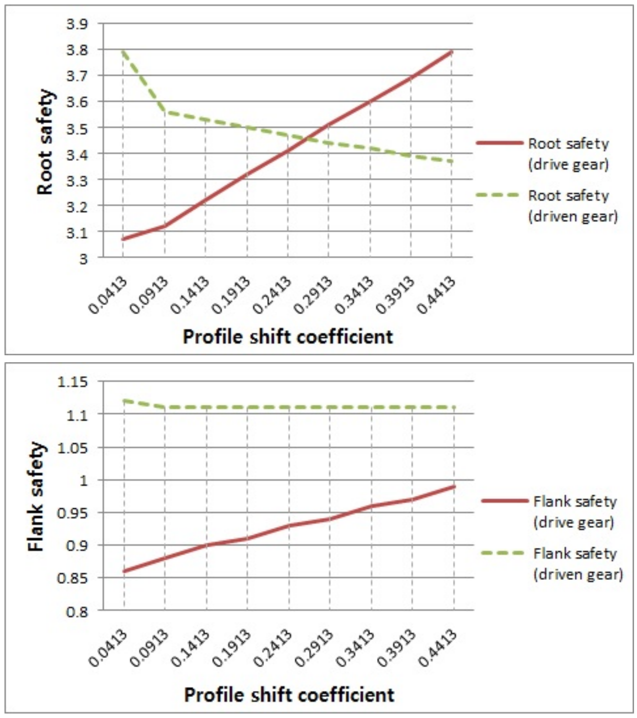


Figure 6. Curves of root safety and flank safety factors from the profile shift coefficient.

Table 15. Parameter design for the profile shift coefficient

Profile shift coefficient		Root safety		Flank safety	
Drive	Driven	Drive	Driven	Drive	Driven
0.0413	0.2000	3.07	3.79	0.86	1.12
0.0913	0.1500	3.12	3.56	0.88	1.11
0.1413	0.1000	3.22	3.53	0.90	1.11
0.1913	0.0500	3.32	3.50	0.91	1.11
0.2413	0.0000	3.41	3.47	0.93	1.11
0.2913	-0.0500	3.51	3.44	0.94	1.11
0.3413	-0.1000	3.60	3.42	0.96	1.11
0.3913	-0.1500	3.69	3.39	0.97	1.11
0.4413	-0.2000	3.79	3.37	0.99	1.11

Table 16. Parameter design results

	GP2 of Group II			
	Before		After	
	Drive	Driven	Drive	Driven
Face width (mm)	11	11	12	12
Profile shift coefficient	0.2410	0.0000	0.3913	-0.1500
Root safety	3.41	3.47	3.82	3.51
Flank safety	0.93	1.11	1.01	1.15
Life satisfying the safety factor of 1.0 (h)	913		2,319	
Remarks: design life (h)	2,200			

width at intervals of 0.5 mm from 9 to 13 mm. The results are presented in Table 14 and Figure 5.

The strength increased with the face width. However, the increase in face width was limited by the requirements for tooth surface durability; therefore, the profile shift coefficient was considered. The sum of the profile shift coefficient for GP II was 0.2413, and changes in gear strength were observed by increasing the profile shift coefficient at intervals of 0.05 from 0.0413 to 0.4413. The results are presented in Table 15 and Figure 6. The face width was fixed at 11 mm. The tooth root bending strength of drive gear increased with the profile shift coefficient, but the strength of the driven gear decreased.

Tooth surface durability of the drive gear increased with the profile shift coefficient, but that of the driven gear decreased slightly and showed no significant change. The increase in the profile shift coefficient was limited by the requirements for tooth surface durability; therefore, changes with increase in face width should also be considered.

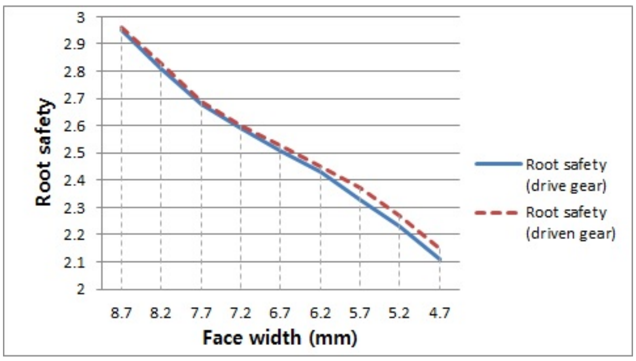
To improve the strength, the following parameters for the drive gear of GP II were proposed: a face width of 12 mm and a profile shift coefficient of 0.3913. Robust design is possible by modifying the face width and profile shift coefficient to satisfy the required life, as presented in Table 16.

Lightweight gear design

Face widths of transmission gear set 2 in GP I were 10.7 mm (drive gear) and 8 mm (driven gear). Changes in the strength were observed by decreasing the face width of the drive gear at intervals of 0.5 mm from 8.7 mm to 4.7 mm and that of the driven gear from 8 mm to 4 mm to find the conditions satisfying the safety factor. The results are presented in Table 17 and Figure 7.

Table 17. Lightweight design for the face width

Face width (mm)		Root safety		Flank safety	
Drive	Driven	Drive	Driven	Drive	Driven
8.7	8.0	2.95	2.96	1.21	1.25
8.2	7.5	2.81	2.83	1.18	1.22
7.7	7.0	2.68	2.69	1.15	1.18
7.2	6.5	2.59	2.60	1.11	1.15
6.7	6.0	2.51	2.53	1.07	1.11
6.2	5.5	2.43	2.45	1.03	1.07
5.7	5.0	2.33	2.37	0.99	1.03
5.2	4.5	2.23	2.27	0.96	0.99
4.7	4.0	2.11	2.15	0.91	0.93



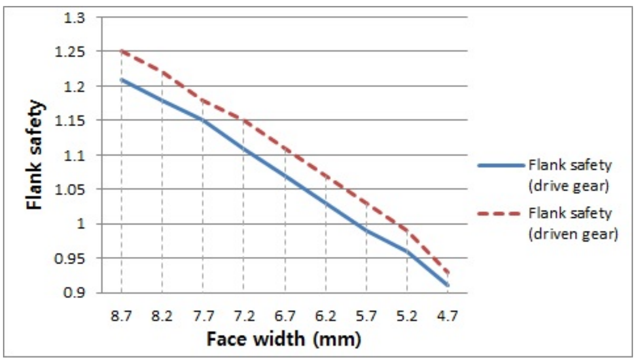


Figure 7. Curve of root safety and flank safety factors from tooth face width II.

Table 18. Lightweight design results

	TMG II of GP I of Group II			
	Before		After	
	Drive	Driven	Drive	Driven
Face width (mm)	10.7	8.0	6.2	5.5
Root safety	3.77	2.96	2.43	2.45
Flank safety	1.21	1.25	1.03	1.07
Life satisfying the safety factor of 1.0 (h)	48,386		989	
Remarks: design life (h)	670			

Under the harsh conditions of group II, the safety factor of GP I transmission gear set 2 for tooth surface durability became closer to 1.0 with decreasing face width. The face widths for transmission gear set 2 of GP I were proposed as 6.2 mm (drive gear) and 5.5 mm (driven gear) for lightweight design, and every condition for the safety factors, as in shown in Table 18, was satisfied. Therefore, the required life was realized by modifying the face width design, and the design reduced the weight of the drive gear by 42% (from 1.92 kg to 1.11 kg) and that of the driven gear by 30% (from 2.6 kg to 1.83 kg).

Conclusions

In this study, the working conditions of a small 4.8 kW agricultural transporter were analyzed, and a usage profile and load spectrum were created through a user survey. The safety factor and the design life of the contact strength and tooth root strength of the proposed transmission were calculated through gear strength analysis, and the parameters for strength improvement were examined. The major results are summarized as follows.

(1) The participants of the survey were divided into two groups: Group I worked on flat areas, and Group II worked on both flat areas and slopes. A usage profile with the gross weight and design life of each transmission gear set.

(2) The driving axle torque was calculated for a rolling resistance coefficient of 0.12 (normal conditions) and 0.2 (harsh conditions). The driving axle torque calculated using the engine power was the maximum torque.

(3) The load spectrum was constructed with the design life and shaft speed of each gear set. From the strength

calculation results for each gear pair, it was found that the contact stresses in all gear pairs were smaller than the tooth root stress. The design life of the transporter using gear pair II in Group II was 42% under harsh conditions; this design life needs to be improved.

(4) The design parameters were examined with gear pair II in Group II. The safety factor increased to 1.01 upon increasing the face width and profile shift coefficient. As a result, the design life increased from 913 h to 2,319 h.

(5) Parameters for lightweight design were examined with gear pair I in Group II. The safety factor decreased to 1.03 upon reducing the face width of the driving gear and the driven gear. As a result, the design life decreased from 48,386 h to 989 h, and became close to the design life of the product. The gear weight also decreased by 42% for the driving gear and 30% for the driven gear.

(6) The safety factor for the design life was examined through gear strength analysis. Conditions for strength improvement were determined by examining the parameters and were provided to increase the design life or to adjust the overdesigned gear. However, load conditions differ with the working conditions or the user's preferences; therefore, it is necessary to conduct further studies in various regions.

Conflict of Interest

The authors have no conflicting financial or other interests.

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