Fatigue Life Simulation of Tractor Spiral Bevel Gear According to Major Agricultural Operations

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Abstract: The spiral bevel gear in a tractor, unlike the other gears in the transmission, is one of the most vulnerable gears in terms of fatigue life, as it is consistently driven throughout the operations of the tractor. Conventional fatigue life tests of transmission gears require expensive equipment and repeated tests, and do not reflect dynamic field loads. The aim of this study is to develop a simulation model which can replace conventional fatigue life tests for actual gears, in order to evaluate the fatigue life of a tractor using dynamic field load data. A transmission simulation model including a spiral bevel gear was developed using commercial software. In order to measure the dynamic load of the tractor according to various field operations, an axle torque measurement system was developed, and field experiments were performed for major agricultural operations occurring in the field. Fatigue life was calculated using Rainflow cycle counting (RFC), the Smith–Watson–Topper (SWT) model, and S–N curves based on torque data measured in the field. The fatigue life under moldboard plow tillage, subsoiler tillage, rotary tillage, and baler operation were 13,599, 285, 278,884, and 525,977 h, respectively. The fatigue life of the tractor, according to subsoiler tillage and baler operation, was 0.104 and 192 times the service life, respectively, where the difference between these two operations was about 1846 times. The fatigue life of the tractor, according to the attached implement type, was significantly different. Therefore, it can be seen that the fatigue life of a tractor can be significantly different, depending on agricultural operation type which the farmer uses most often; this can be used as basic data for tractor design and evaluation. In addition, it is considered that the developed simulation model can be applied to fatigue life evaluation using dynamic field load data.

Keywords: agricultural tractor; spiral bevel gear; fatigue life; simulation model; dynamic field load
1. Introduction

The global tractor market is expected to increase gradually from USD 45,300 million in 2017 to USD 59,100 million in 2022, with a compound annual growth rate (CAGR) of 5.5% [1]. The production of tractors in Korea was 49,056 units in 2016, accounting for about 70% of major agricultural machinery (e.g., cultivators, combines, rice transplanter, and so on) production, and their utilization rate is 98.7%, which is the highest among agricultural machinery [2]. In particular, the import of agricultural machinery in Korea has increased by about 156%, from USD 85 million in 2015 to USD 133 million in 2018, most of which was high horsepower tractors (75 kW or higher), which are difficult to produce in Korea [2]. In view of this trend, the demand for high horsepower tractors has steadily increased; however, there is still a lack of original technology for developing high horsepower tractors in Korea. Therefore, it is necessary to secure original technology to design and evaluate high horsepower tractors in Korea.

The transmission is a key component of the tractor, which accounts for the highest proportion of the tractor’s total price when compared to other components [3]. In particular, among the various gear pairs of the transmission, the spiral bevel gear receives power from the range shift shaft and distributes the power to the left and right rear wheels at a right angle; thus, a high level of design technology is required [4]. In addition, it is driven throughout the entire cycle of tractor operations (unlike other gears, whose frequency of use is determined by the gear stages) and has a high reduction gear ratio (i.e., a higher-load act) than other gears. Therefore, the spiral bevel gear is the most basic core part, which is considered the most vulnerable in terms of fatigue life.

In order to secure design reliability and to optimally design a tractor transmission, it is essential to evaluate its fatigue life. A conventional method for fatigue life evaluation of a tractor is an accelerated life test (ALT) [5], which is widely used as it can reduce the time required for fatigue life evaluation of a transmission. It is performed under rotational speed or torque conditions which are harsher than field conditions, in order to reduce test time [6]. However, the ALT requires a high cost for the construction of the test equipment, and its repeatability and reproducibility are low, as it takes a long time to repeat experiments under various conditions. In the development and performance evaluation stages of a tractor transmission, as verification through frequent design changes are performed, an alternative method is required. For this purpose, computer-aided engineering simulation technology provide useful alternatives [7]. In the era of digital farming, computer simulation technology can replace the actual testing of complex mechanical systems and is mainly used for the design, analysis, and evaluation of mechanical systems. This makes it possible to test various conditions in a short time without additional test equipment, making design optimization and performance analysis easy for each condition [8,9]. Litvin et al. [10] conducted contact and stress analysis of spiral bevel gears based on the finite element method using the commercial software ABAQUS. Kong et al. [11] conducted strength analysis by speed for multistage gear units using ROMAX Designer, a commercial gear analysis software. They verified the gear modeling by comparison with the American Gear Manufacturers Association (AGMA) 2001 standard. Wu and Tran [12] proposed a novel honing method for longitudinal tooth flank crowning of a helical gear using the KISSsoft software. Study of the performance of simulation-based gear systems, as described above, has several strengths in terms of saving time and costs but, if the actual environmental conditions are not reflected, it is difficult to guarantee the reliability of the results. Most of the previous studies did not reflect the actual dynamic field load conditions and, thus, simulation-based performance evaluations reflecting actual conditions are required.

Gear breakage frequently occurs in the form of fatigue breakage, which occurs due to the repeated action of a dynamic load lower than the allowable load [13]. As the tractor is operated under soil conditions, the load generated on the tractor during field operations has the characteristics of irregular fluctuations in size and direction [14,15], and certain soil types (e.g., reclaimed land) cause high loads. In addition, the load on the tractor varies depending on the various working conditions, such as attached implement type [16,17] and gear stages [3,18]. This load directly affects the fatigue life and, so, the fatigue life of the tractor may vary depending on the user’s work characteristics.
Therefore, fatigue life evaluation using dynamic field load data of major agricultural operations is necessary. However, to date, most fatigue life tests are conducted with the engine-rated load as the input condition [8]. Therefore, fatigue life evaluation using dynamic field load data of major agricultural operations, as measured through actual field experiments, is necessary. Research on the load measured through field experiments has mainly consisted of basic statistical analyses [19], creating the load spectrum [18,20–23], relative severity [3,18,24,25], and damage evaluation [26,27]. Kim et al. [18] measured the load data of a 78 kW tractor, according to the disk plow and rotary tillage, and evaluated the relative severity for evaluating the transmission fatigue life by analyzing the load spectrum. Bai et al. [27] analyzed the influence of key factors on the load dynamic characteristics and fatigue damage of the powertrain. They reported that the harmonic engine torque and vibration characteristics of the powertrain system had a great influence on the fatigue damage of the shaft parts, and that the average engine torque had an effect on the fatigue damage of the gear parts. However, the above studies only allowed for the relative comparison of torque, load spectrum, and severity according to conditions such as major agricultural work, number of stages, soil, and so on, and were not applied to actual durability tests. In conclusion, studies evaluating the fatigue life of a simulation-based tractor considering actual field dynamic loads are insufficient. Therefore, it is necessary to evaluate the fatigue life based on simulations utilizing dynamic field load condition data measured for each agricultural operation through field experiments.

The purpose of this study is to develop a simulation model of a tractor transmission, in order to evaluate the fatigue life of an agricultural tractor transmission spiral bevel gear using dynamic field load data measured in the field. The novelty of this study is that the fatigue life evaluation can be reproduced and repeated using the simulation model and dynamic field load data which fluctuates according to various conditions (e.g., soil conditions, gear stage, attached implements, and so on) without additional experimental equipment. This can contribute to securing original technology for the optimal design and durability evaluation of tractors. The major objectives of this study are as follows: (1) analysis of dynamic field load data for each agricultural operation measured through field experiments; (2) developing a simulation model of a transmission including the spiral bevel gear based on commercial software; (3) evaluation of the fatigue life of the spiral bevel gear for each agricultural operation.

2. Simulation Model Development

2.1. Software

The simulation program used for the life evaluation of spiral bevel gears in this study is KISSsoft (Ver. 03/2017, KISSsoft AG, Bubikon, Switzerland), which has been widely used for the design and analysis of mechanical systems. KISSsoft is a design program which specializes in the design and analysis of complex mechanical systems. It performs analyses of mechanical elements based on international gear strength standards such as international organization for standardization (ISO). In this study, bending stress was calculated using the “Bevel and Hypoid gears” module of KISSsoft.

2.2. Tractor Power Transmission System

In this study, we used a mechanical front wheel drive (MFWD) tractor (S07, TYM Co., Ltd., Gongju, Korea) which is widely used in Korea; its major specifications are shown in Table 1. The dimensions (Length × Width × Height) of the tractor are 4225 × 2140 × 2830 mm. The tractor’s empty vehicle weight is 3985 kg, and the ratios of the front and rear axles are 42.4% and 57.6%, respectively. The tractor is equipped with a 78 kW engine (D34P, Doosan Infracore Co., Ltd., Incheon, Korea). The rated torque of the tractor is 324 Nm at the rated rotational speed of 2300 rpm and its maximum torque is 430 Nm at 1400 rpm, as shown in Figure 1. The type of tractor transmission is a power shuttle, and the number of gear stages are Forward 32 and Reverse 32. The transmission consists of two power shifts (High and
Low), four driving shifts (1, 2, 3, and 4) of synchromesh type, and four range shifts (C, L, M, and H) of constant mesh type.

**Table 1. Specifications of agricultural tractor used in this study.**

| Items                        | Specifications                                      |
|------------------------------|-----------------------------------------------------|
| Dimension                    | 4225 × 2140 × 2830 mm                               |
| Weight (ratio of front and rear axle) | 3985 (42.4% and 57.6%) |
| Model                        | D34P                                                |
| Engine                        |                                                     |
| Model                         |                                                     |
| Rated power                   | 78 kW at 2300 rpm                                  |
| Rated torque                  | 324 Nm at 2300 rpm                                 |
| Maximum torque                | 430 Nm at 1400 rpm                                 |
| Type                          | Power Shuttle                                       |
| Number of gear stages         | 32 Forward/32 Reverse                              |
| Powertrain                    |                                                     |
| Power shift                   | 2 (High and Low)                                   |
| Driving shift                 | 4 (1, 2, 3, and 4)                                  |
| Range shift                   | 4 (C, L, M, and H)                                 |
| Tire (Front and rear axle)    |                                                     |
| Type                          | Bias ply                                            |
| Size                          | 13.6–24 8PR and 18.4–34 10PR                       |
| Pressure                      | 216 and 157 kPa                                     |
| Static loaded radius          | 0.545 and 0.740 m                                   |

**Figure 1.** Engine performance map of the tractor used in this study.

The tractor power transmission system was constructed as shown in Figure 2. The output shaft of the engine is engaged/disengaged with the drive shaft through a clutch. The drive shaft is first connected to the forward/reverse shift gear and, then, sequentially connected to the high/low shift gear, driving shift gear, range shift gear, and spiral bevel gear. In the spiral bevel gear, the power input from the range shift shaft is diverted to the left and right rear wheels.
2.3. Spiral Bevel Gear

Spiral bevel gears are a kind of bevel gear which has helical teeth. The spiral bevel gear used in the differential gear system of a tractor transmission receives the power of the driving shaft of the transmission from the drive pinion gear and transmits power to the left and right wheels in a right angle direction. The spiral bevel gear consists of a pinion gear and a ring gear, in a structure that intersects at right angles. Table 2 shows the main specifications of the spiral bevel gear used in this study. The torque and rotation speed of the spiral bevel gear were calculated using the gear ratio and efficiency, based on the torque and rotation speed data of the wheel measured in the field, as shown in Equations (1) and (2). The efficiencies of the spiral bevel gear and the final reduction gear are 0.98 and 0.94, respectively, and the gear ratios of the spiral bevel gear and the final reduction gear are 3.82 and 6.00, respectively.

Table 2. Specifications of spiral bevel gear used in this study.

| Parameters                  | Pinion Gear | Ring Gear |
|-----------------------------|-------------|-----------|
| Tooth profile               | Gleason     |           |
| Module (mm)                 | 6.5         |           |
| Pressure angle (°)          | 20          |           |
| Helix angle (°)             | 35          |           |
| Center distance (mm)        | 89          |           |
| Material                    | SCM822      |           |
| Number of teeth             | 11          | 42        |
| Face width (mm)             | 42.5        | 42.5      |
| Pitch circle diameter (mm)  | 71.5        | 273       |
| Cone distance (mm)          | 141.104     | 141.104   |

\[
T_s = \frac{1}{\eta_s} \frac{1}{\eta_f} \left( G_s G_f (T_{rl} + T_{rr}) \right) \tag{1}
\]

where \( T_s \) is the spiral bevel gear input torque (Nm), \( \eta_s \) is the efficiency of the spiral bevel gear, \( \eta_f \) is the efficiency of the final reduction gear, \( G_s \) is the gear ratio of the spiral bevel gear, \( G_f \) is the gear ratio of the final reduction gear, \( T_{rl} \) is the torque of the rear left wheel (Nm), and \( T_{rr} \) is the torque of the rear right wheel (Nm).

\[
N_s = G_s G_f \left( \frac{N_{rl} + N_{rr}}{2} \right) \tag{2}
\]
where $N_s$ is the rotational speed of the spiral bevel gear (rpm), $N_{rl}$ is the rotational speed of the rear left wheel (rpm), and $N_{rr}$ is the rotational speed of the rear right wheel (rpm).

2.4. Simulation Model

The transmission simulation model, including the spiral bevel gear of the tractor used in this study, was developed (as shown in Figure 3a) based on the tractor power transmission structure. It implemented the entire transmission gear system, including the F/R shift gear, High/Low shift gear, driving shift gear, range shift gear, spiral bevel gear, and rear axle. In particular, in the developed simulation model, the spiral bevel gear was modeled as shown in Figure 3b. It consists of a pinion gear that receives the power from the range shift shaft and a ring gear that transmits power to the rear axle. The developed model can be used for gear strength analysis and for evaluating the fatigue life of the transmission system.

![Simulation model of the tractor used in this study: (a) transmission and (b) spiral bevel gear.](image)

2.5. Fatigue Life Evaluation Method

2.5.1. Procedure

Fatigue life testing is often performed under the relatively simple conditions of constant amplitude and frequency [28]. However, the load conditions of a tractor are not simple, as the stress level fluctuates according to various agricultural operations under irregular soil surface environments [3]. Therefore, the fatigue of a tractor can be defined as the sum of damages to its mechanical components due to repetitive loads in the field. Figure 4 shows the procedure for evaluating the fatigue life using torque data measured in the field. In general, agricultural operations of the tractor such as tillage are performed only during the forward movement, and since the reverse movement is only used for simple steering, not specific agriculture operation, almost no load is generated. Such low load is a level that can be neglected in terms of the overall fatigue life. Thus, in this study, only data during forward movement was considered. Fatigue life was evaluated using the torque data computed from data measured in the field using Equation (1). Torque data computed is converted into frequency using Rainflow cycle counting (RFC), which counts fatigue cycles throughout the transmission’s history. The load spectrum (Number of cycles, Torque) was created using the data obtained through RFC, while the load spectrum (Number of cycles, Stress) was modified through torque–stress conversion. The damage of each load cycle was assessed using the S–N curve and the Miner cumulative damage rule, and the fatigue life was evaluated based on the service life.
2.5.2. RFC and Load Spectrum

As the torque data measured in the field fluctuated irregularly, it was converted into a 3D map of mean, range, and number of cycles using the RFC [29–32]. To quantify the spectrum load magnitude, the equivalent torque was calculated, based on the Smith–Watson–Topper (SWT) model, using the mean and range in the 3D map obtained from RFC, as shown in Equation (3) [33,34]. In this study, the load data was measured for a relatively short time (55–90 s) during field operations, and the load data cannot be measured during the entire life cycle of the tractor. Nevertheless, in order to evaluate the fatigue life of a tractor spiral bevel gear, it is necessary the load history for the entire life cycle. In this case, generally, the load data measured in a short time can be extended by considering the entire life of the tractor, as shown in Equation (4) [3,18]. Data of a previous study, obtained through the survey of the actual conditions of users in Korean agricultural operational times, were used to extend the number of cycles [35]. The tractor’s service life was applied as 2736 h [9,35].

\[ T_e = \sqrt{(t_a + t_m)t_a} \]  

(3)

where \( T_e \) is the equivalent torque (Nm), \( t_a \) is the range of torque, and \( t_m \) is the mean torque (Nm).

\[ N_T = 3600NLh \]  

(4)

where \( N_T \) is the extended number of cycles of the load (cycles), \( N \) is the number of cycles of measured load data (cycles/s), \( L \) is the service life of the used tractor (years), and \( h \) is the annual usage time of the tractor (h/year).

The modes of gear fatigue failure can be divided into tooth bending fatigue and contact fatigue. In general, failure of the tooth root is caused by bending stress in the gear, while wearing of the tooth surface is caused by contact stress [36]. In particular, although fitting is known as the most common failure mode, it is nevertheless difficult to determine the exact failure time, and power transmission is possible even if fitting occurs. Therefore, Korean agricultural machinery manufacturers regard bending stress as the more important major mode of fatigue failure [37]. The bending fatigue failure causes gradual damage to the gear teeth, which may result in the failure of the entire transmission. Therefore, in this study, the bending fatigue of the gear was regarded as a factor influencing the fatigue life and the fatigue life was evaluated based on the bending stress. For fatigue analysis of the spiral bevel gear, the measured torque was converted into bending stress, based on the specifications of the gears and major parameters presented Table 3 using ISO 10300-1:3, as shown in Equations (5)–(7) [38–40]:

\[ F_{mt} = \frac{2000T}{d_m} \]  

(5)
where $F_{mt}$ is the nominal tangential force at mid-face width of the reference cone (N), $T$ is the torque at the driving and driven gear (Nm), and $d_{mn}$ is the mean pitch diameter of the driving and driven gear (mm);

$$\sigma_{F0} = \frac{F_{mt}}{b m_{mn}} Y_{Fa} Y_{Sa} Y_\varepsilon Y_K Y_{LS}$$

(6)

where $\sigma_{F0}$ is the nominal tooth-root stress (N/mm²), $b$ is the gear face width (mm), $m_{mn}$ is the mean normal module (mm), $Y_{Fa}$ is the tooth form factor, $Y_{Sa}$ is the stress correction factor, $Y_\varepsilon$ is the contact ratio factor, $Y_K$ is the bevel gear factor, and $Y_{LS}$ is the load distribution coefficient;

$$\sigma_F = \sigma_{F0} K_A K_V K_{FB} K_{Fa}$$

(7)

where $\sigma_F$ is the tooth-root stress (N/mm²), $K_A$ is the application factor, $K_V$ is the dynamic factor, $K_{FB}$ is the face load factor, and $K_{Fa}$ is the transverse load factor.

Table 3. Major parameters for calculating the tooth root bending stress.

| Parameters | Values |
|------------|--------|
| $Y_{Fa}$  | 2.89   |
| $Y_{Sa}$  | 1.58   |
| $Y_\varepsilon$ | 0.63 |
| $Y_K$     | 1.027  |
| $Y_{LS}$  | 0.963  |
| $K_A$     | 1      |
| $K_V$     | 1      |
| $K_{FB}$  | 3.37   |
| $K_{Fa}$  | 1      |

2.5.3. Damage Sum

In order to evaluate the fatigue life, an S–N curve is required. To apply a more accurate S–N curve, an S–N curve was created using the results of previous studies which performed fatigue life tests using the same model as the spiral bevel gear used in this study. In the previous study, three samples of spiral bevel gears were used to evaluate the fatigue life based on the ALT. In this study, the S–N curve was obtained by calculating the bending stress using Equations (5)–(7), based on the test results of the previous study, as shown in Figure 5. The number of cycles of the S–N curve was calculated considering the rotational speed and the service life of the tractor. The coefficient of determination ($R^2$) of the obtained S–N curve was 0.99.

**Figure 5.** S–N curve of spiral bevel gears obtained based on fatigue life test results presented in previous studies [41].
The damage sum can be calculated based on the Palmgren–Miner cumulative damage rule, using the S–N diagram and the load spectrum of the measured data, as shown in Equation (8) [34,42]. This method assumes that the total life of the tractor can be estimated by adding the percentage of life consumed at each stress level.

\[ D = \sum_{i=1}^{k} \frac{n_i}{N_i} \]  

(8)

where \( D \) is the damage sum, \( n_i \) is the number of cycles of measured data at the \( i \)th stress level, and \( N_i \) is the number of cycles of the S–N curve at the \( i \)th stress level.

2.5.4. Fatigue Life

Fatigue life evaluation was performed to analyze the field load for each agricultural operation on the guaranteed life, based on the service life (2736 h). The predicted fatigue life was calculated by dividing the sum of the service life and damage sum, as shown in Equation (9). Here, when the damage sum in the field is 1, the fatigue life can be guaranteed to be the same as the service life; meanwhile, when the damage is less than 1, the fatigue life is guaranteed to be higher than the service life [28,41,43].

\[ L_f = \frac{L_s}{D} \]  

(9)

where \( L_f \) is the predicted fatigue life (h) and \( L_s \) is the service life of the used tractor (h).

2.6. Simulation Conditions

In this study, fatigue analysis of the spiral bevel gear based on simulation was conducted using ISO 10300-1:3 [38–40]. The strength analysis method is divided into Methods A, B, and C, where the most widely used Method B was used in this study [8]. The heat treatment of the spiral bevel gear was set to carburizing. ISO-VG 220, which is widely used in general, was selected as the lubricant [12]. In addition, the precision of the gear was selected as Class 6 of KS B ISO 1328-1, which is applied when manufacturing gears for agricultural machinery [44]. Load conditions input into the simulation for strength and life analysis were values obtained from the analyzed load spectrum using dynamic load data of each agricultural operation. The required time for the simulation was calculated using the value collected in the previous study. In the previous study, it was reported that the annual usage time of the tractor in Korea was 342 h, while the durability life of the tractor suggested by the agricultural mechanization promotion law was 8 years. Thus, in this study, the entire service life of the tractor was applied as 2736 h [9,35].

3. Field Data Acquisition

3.1. Data Measurement System

The data measurement system was built to measure the torque and rotational speed of the rear wheel axle generated by the tractor during field operations. The torque acting on the left and right rear wheel axles was measured using two wheel torquemeters (MW_30 kNm, MANNER Sensortelemetrie GmbH, Spaichingen, Germany). The wheel torquemeter can measure up to 30 kNm. The wheel rotation speed was calculated by detecting 100 gear teeth attached to the axle using a two-proximity sensors (MP-981, ONO SOKKI, Yokohama, Japan), both for left and right wheels. Field data were measured with a sampling rate of 100 Hz during data acquisition (CRONOS compact CRC-400-11, IMC, Berlin, Germany).

3.2. Field Sites

A total of four field sites were selected and used for each agricultural operation. Sites 1, 2, 3, and 4 were located in Seosan (36°46'43.8″ N 126°33'38.3″ E), Dangjin (36°55'50.6″ N 126°38'00.3″ E), Buan (35°46'43.9″ N 126°45'00.0″ E), and Jeonju (35°53'04.5″ N 126°59'53.9″ E) in Korea, respectively.
The size of sites 1, 2, and 4 were all 4000 m² (40 × 100 m), while the size of site 3 was 6000 m² (60 m × 100 m). The physical properties of soil for each field site, such as soil texture, moisture content, and cone index, are provided in Table 4. The soil texture of sites 1, 2, and 3 was Loam, while the soil texture of site 4 was clay loam. These results are consistent with the results of previous studies, that most Korean paddy soils are loam [45]. The soil moisture content of sites 1, 2, 3, and 4 were 27%, 32%, 42%, and 37%, respectively. The cone index can be expressed as an indicator of soil strength, which is different for each depth.

### Table 4. Physical properties of soil for each field site.

| Sites  | Soil Texture | Soil Moisture Content (%) | Cone Index (kPa) by Depth |
|--------|--------------|--------------------------|--------------------------|
|        |              |                          | 0 cm | 5 cm | 10 cm | 15 cm | 20 cm | 25 cm |
| Site 1 | Loam         | 27                       | 199  | 474  | 828   | 1484  | 2467  | -     |
| Site 2 | Loam         | 32                       | 270  | 875  | 1146  | 1670  | 3549  | 3743  |
| Site 3 | Loam         | 42                       | 114  | 389  | 720   | 1552  | 2794  | 2750  |
| Site 4 | Clay loam    | 37                       | 191  | 423  | 476   | 742   | 2241  | 3052  |

#### 3.3. Major Agricultural Field Operations

Experiments were conducted across four agricultural operations—moldboard plow tillage, subsoiling, rotary tillage, and baler operation—at the four locations in Korea, as shown in Figure 6. Moldboard plow tillage, subsoiler tillage, rotary tillage, and baler operations were performed at sites 1, 2, 3, and 4. The major specifications of each attached implement are shown in Table 5. The moldboard plow tilling operation was performed at a tillage depth of about 15–20 cm under the gear stage M3 Low (7.09 km/h) [4]. Subsoiler and rotary tillage were performed at a tillage depth of about 35–40 and 15–20 cm, respectively, under the gear stage L3 Low (2.38 km/h). The baler operation was performed under the gear stage M1 Low (3.78 km/h).

![Figure 6. Field experiments of major agricultural tractors: (a) moldboard plow tillage, (b) subsoiler tillage, (c) rotary tillage, and (d) baler operation.](image-url)
Table 5. Detailed specifications of major attached implements used in this study.

| Implements       | Model                  | Dimensions (Length × Width × Height; mm) | Weight (kg) | Number of Ridges (Blades) | Working Width (mm) | Required Power (kW) |
|------------------|------------------------|------------------------------------------|-------------|---------------------------|--------------------|---------------------|
| Moldboard plow   | WISP-8 (WOONGJIN) WHDP500 (WECAN GLOBAL) SW 230 GL (SUNCGWOO) L325 (LIVEMAC) | 2180 × 2800 × 1285 1800 × 2095 × 1380 930 × 2640 × 1280 3490 × 2430 × 2400 | 495 870 735 3180 | 8 5 54 - | 2800 2095 2300 2000 | 78 † 75 † 60 † 67 † |

During operations, the engine speed was set as full throttle (2510 rpm). The moldboard plow, subsoiler, and rotary tillage operations were performed by a skilled expert of TYM, while the baler operation was performed by a skilled expert of the LIVEMAC company.

4. Results

4.1. Dynamic Load Analysis

Figure 7 shows the dynamic load profiles corresponding to the four field operations. Each task was divided into a working and steering period, where the dynamic load profile for each period was different. The torque of the spiral bevel gear in the working period during moldboard plow tillage was approximately 500–700 Nm, while the torque in the steering period showed large variation (Figure 7a). The maximum torque was temporarily displayed. The torque during subsoiler tillage gradually increased with an increase of the tillage depth in the working period, showing a torque of about 900–1300 Nm in the subsoiler period (Figure 7b). In the steering period during subsoiler tillage, the torque range was lower than that of the working section; however, the torque temporarily increased. This temporary torque increase was determined to be due to the tractor stopping for steering, according to the working pattern of the tractor: Driving (Tillage)—Stop—Three-point hitch ascending—Driving (Steering). During rotary tillage, the torque of the spiral bevel gear showed a maximum value when starting to drive the tractor from a standstill, with a torque of less than 100 Nm in the tillage section (Figure 7c). Unlike moldboard and subsoiler tillage, rotary tillage drives a power take-off (PTO) blade and, so, it is considered that low torque is required, as the soil is pushed through the PTO blade and driven by the reaction force in the forward direction of the tractor. The torque in the steering period during rotary tillage was higher than that of the working period, due to the influence of the steering force required for steering. The baler operation, similar to rotary tillage, shows the maximum torque value when driving a tractor stopped at the starting point of the tractor (Figure 7d). In the working period, the tractor performs operations such as baling for collecting Italian ryegrass, and binding for bale compression, for 10–50 s, showing a torque range of 0–150 Nm. From 50 to 75 s, it showed almost constant torque as it was the process of discharging the bale after stopping. In the turning section, the torque range was about 0–300 Nm. Table 6 provides a statistical description of the spiral bevel gear torque for each operation, during both the working period and steering period. The average torque data during the working period of moldboard plow and subsoiler tillage was higher than that in the steering period, while the average torque data during the steering period in rotary tillage and baler operation was higher than the working period. In the working period, the average torque was highest in the order of subsoiler tillage, moldboard plow tillage, baler operation, and rotary tillage, while the average torque in the steering period was highest in the order of subsoiler tillage, rotary tillage, moldboard plow tillage, and baler operation. As the required time for the working period accounts for more than about 80% of the total torque profile, the torque of the spiral bevel gear in the working section is considered to have a greater influence on the fatigue life.
Figure 7. Dynamic load profile of spiral bevel gear by field operations: (a) moldboard plow tillage, (b) subsoiler tillage, (c) rotary tillage, and (d) baler operation.

Table 6. Statistical description of the spiral bevel gear torque by field operations during working and steering periods.

| Field Operations         | Spiral Bevel Gear Torque (Nm) | Working Period (A) | Steering Period (B) |
|--------------------------|-------------------------------|--------------------|---------------------|
|                          | Max.  | Avg. ± Std. | Max.  | Avg. ± Std. |
| Moldboard plow tillage  | 750.7 | 614.7 ± 45.1  | 784.8 | 214.0 ± 202.1 |
| Subsoiler tillage       | 1324  | 876.5 ± 293.9 | 1205  | 400.3 ± 238  |
| Rotary tillage          | 617.8 | 47.2 ± 46.3   | 481.4 | 305.1 ± 74.1 |
| Baler operation         | 584.0 | 55.9 ± 49.5   | 358.9 | 212.8 ± 87.0 |

4.2. Load Spectra

Figure 8 shows the load spectra of the spiral bevel gear during the four tractor operations. The load spectrum is expressed as 3D Map of a range, mean, and number of cycles. Figure 8a shows the load spectrum during moldboard plow tillage, showing the highest number of cycles in the mean of 500–700 Nm and the range of 0–200 Nm. Figure 8b shows the load spectrum during subsoiler tillage, showing the highest number of cycles in the mean of 750–1250 Nm and the range of 0–100 Nm. Figure 8c,d show the load spectra during rotary tillage and baler operation, showing the highest number of cycles in the mean of 0–100 Nm and the range of 0–50 Nm. The 3D Load spectra were converted into stress and used to predict the fatigue life, as detailed below.
4.3. Fatigue Life Evaluation Based on Simulation Model

The fatigue life of the spiral bevel gear was evaluated for each agricultural operation, using the developed simulation model. Figure 9a shows the accumulated damage sum for each operation acting on the spiral bevel gear, respectively. It can be seen that subsoiler tillage exceeded a damage sum of 1 (which is the criterion for fatigue failure based on the Miner cumulative damage rule) and, thus, leads to a shorter life than the service life. In addition, the damage sum in the other three operations are less than 1, indicating that the fatigue life is longer than the service life. Figure 9b shows the fatigue life for each agricultural operation. This is the value of the service life divided by the damage sum, which shows the opposite tendency to the sum of the damage. Subsoiler tillage, with the harshest load conditions, had a fatigue life of less than 1000 h. Table 7 shows the results of damage sum and fatigue life under the four operations. As a result, the damage sums for moldboard plow tillage, subsoiler tillage, rotary tillage, and baler operation were 0.2012, 9.6000, 0.0098, and 0.0052, respectively. The fatigue life for moldboard plow tillage, subsoiler tillage, rotary tillage, and baler operation were 13,599, 285, 278,884, and 525,977 h, respectively.
Figure 9. Results of fatigue analysis of spiral bevel gears based on simulation model: (a) damage sum and (b) fatigue life.

Table 7. Damage sum and fatigue life of the spiral bevel gear by each agricultural operation based on our simulation model.

| Field Operations          | Damage Sum | Fatigue Life (h) |
|---------------------------|------------|------------------|
| Moldboard plow tillage   | 0.2012     | 13,599           |
| Subsoiler tillage        | 9.6000     | 285              |
| Rotary tillage           | 0.0098     | 278,884          |
| Baler operation          | 0.0052     | 525,977          |

Figure 10a shows the ratio of fatigue life (obtained through simulation) to service life (2736 h). The ratio of fatigue life for each agricultural operation ranged from 0.104 to 192. In subsoiler tillage, with the most severe load conditions, the fatigue life ratio was 0.104, indicating it was lower than the service life. This means that, when working with the subsoiler during the entire life cycle of the tractor, the life of the tractor can be guaranteed at about 10.4% of its service life. The fatigue life ratio of plow tillage, rotary tillage, and baler operation were 4.97, 101, and 192, respectively, indicating that tractor may be used longer than the service life. In particular, the fatigue life during rotary tillage and baler operation was shown to last 100 times longer than the service life. This was considered to be due to the fact that, as these operations are performed by driving the PTO, a rather low load is applied to the spiral bevel gear to drive the tractor. Figure 10b shows the relative fatigue life obtained by dividing the fatigue life value of each agricultural operation based on the subsoiler tillage, which had the lowest fatigue life. The results of the Relative Fatigue Life Analysis indicate that the tractor during moldboard plow, rotary tillage, and baler operations can be used 48, 979, and 1846 times longer than subsoiler tillage, respectively. Consequently, as the life of the tractor is differently affected by the frequency of use of various implements, the designer needs to design and develop tractors taking into consideration the frequency and characteristics of use of the attached implements.
In this study, a simulation model was developed for the fatigue life evaluation of the spiral bevel gear of tractors. Although this simulation model has strengths, in terms of repeatability and reproducibility, it is essential to confirm whether the simulation model shows a level similar to that of the actual fatigue life evaluation results. Thus, the simulated fatigue life prediction results of spiral bevel gears were compared with previous studies based on ALT. Previous studies have evaluated the fatigue life of spiral bevel gears through ALT based on constant load conditions, not dynamic load conditions. In this study, the simulation model was compared under the two load condition cases (Case 1: Torque = 1128 Nm, Speed = 559 rpm; Case 2: Torque = 621 Nm, Speed = 482 rpm) performed in previous studies [41]. Figure 11a,b show the fatigue life obtained by our developed simulation model and that obtained from the previous study (i.e., based on ALT) under cases 1 and 2, respectively. Table 8 shows the fatigue life results of simulations and the previous study under the two load cases. In load case 1, the fatigue life of the simulation was 197 h, while the fatigue life of samples 1, 2, and 3, and their average obtained in the previous study were 273, 157, 184, and 184 h, respectively. In load case 2, the fatigue life of the simulation was 43,315 h, while the fatigue life of samples 1, 2, and 3, and their average obtained in the previous study were 26,330, 58,453, 33,702, and 39,495 h, respectively. There was a slight difference between the fatigue life obtained through simulation and the fatigue life obtained in the previous study. These differences are related to the differences between samples in the previous study. It has been reported that the fatigue life obtained in the previous study differed by up to 222% between samples, which was judged to be due to environmental conditions and test environments during gear processing. Nevertheless, the fatigue life obtained by the developed simulation model under the two load cases were distributed within the minimum and maximum fatigue life ranges of the previous ALT-based study. In particular, the differences between the average fatigue life of the three samples and that of the developed simulation model was not significant. However, verification of the simulation model considering dynamic load conditions is still insufficient. In order to verify whether the simulation model reflects dynamic field load conditions well, an actual life test based on dynamic field load conditions is required. Studies improving the accuracy of and verifying the developed simulation model will be addressed in future work.
was found to be excessively low during subsoiler tillage, which may mean that the combination of the used subsoiler and the tractor is not appropriate. In this case, the manufacturer of the subsoiler should recommend the user to use a tractor with a higher horsepower. The developed simulation model is based on Miner cumulative damage rule. To evaluate the fatigue life of the spiral bevel gear, the damage sum was calculated using RFC, SWT, and S–N curves, based on torque data measured in the field. The fatigue life for moldboard plow tillage, subsoiler tillage, rotary tillage, and baler operation was 192 times that of the service life. The difference of fatigue life under these two operations was 0.104 times that of the service life, while the fatigue life under baler operation was 0.104 times that of the service life. Therefore, tractor designers must design the transmission in consideration of the working conditions (e.g., implement type) under which the tractor will be mainly used. In addition, the fatigue life of the spiral bevel gear was found to be excessively low during subsoiler tillage, which may mean that the combination of the used subsoiler and the tractor is not appropriate. In this case, the manufacturer of the subsoiler should recommend the user to use a tractor with a higher horsepower. The developed simulation model can be used for fatigue life evaluation according to design conditions when designing a transmission.

Table 8. Fatigue life of spiral bevel gear, according to the simulation results and a previous study based on ALT.

| Cases | Simulation | Previous Study Based on ALT [41] |
|-------|------------|---------------------------------|
|       | Sample 1   | Sample 2 | Sample 3 | Average |
| 1     | 197        | 123      | 273      | 157     | 184     |
| 2     | 43,315     | 26,330   | 58,453   | 33,702  | 39,495  |

6. Conclusions

We aimed to develop a simulation model which can replace the fatigue life testing of conventional actual gears for the evaluation of the fatigue life of a tractor by using dynamic field load data. To this end, a simulation model of a spiral bevel gear was developed using commercial gear analysis software. To apply dynamic load conditions to the simulation model, an axle torque measurement system was developed and the dynamic field load data of the tractor under major agricultural operations performing in the field were measured. The fatigue life evaluation using the developed simulation model is based on Miner cumulative damage rule. To evaluate the fatigue life of the spiral bevel gear, the damage sum was calculated using RFC, SWT, and S–N curves, based on torque data measured in the field. The fatigue life for moldboard plow tillage, subsoiler tillage, rotary tillage, and baler operation were 13,599, 285, 278,884, and 525,977 h, respectively. In addition, the fatigue life of the tractor under subsoiler tillage was 0.104 times that of the service life, while the fatigue life under baler operation was 192 times that of the service life. The difference of fatigue life under these two operations was approximately 1846 times.

The results of this study show that the life of a spiral bevel gear may differ, depending on which agricultural operation the user mainly uses, and that the maximum life of the spiral bevel gear may also differ by 1846 times or more. The fatigue life of a spiral bevel gear according to its agricultural operation type was significantly different, indicating that the life of the spiral bevel gear might be significantly different depending on which operation type the user mainly performs. Therefore, tractor designers must design the transmission in consideration of the working conditions (e.g., implement type) under which the tractor will be mainly used. In addition, the fatigue life of the spiral bevel gear was found to be excessively low during subsoiler tillage, which may mean that the combination of the used subsoiler and the tractor is not appropriate. In this case, the manufacturer of the subsoiler should recommend the user to use a tractor with a higher horsepower. The developed simulation model can be used for fatigue life evaluation according to design conditions when designing a transmission.

Figure 11. Comparison between fatigue life analysis results by simulation and obtained fatigue life from a previous study with three samples for each load condition: (a) case 1 and (b) case 2.
by reflecting various working environment conditions. In particular, if a database constructed using field dynamic load data according to various tractor working conditions was combined with the developed simulation model in this study, it can be used for designing optimal tractor transmissions, thereby securing original technology, improving design reliability, and reducing costs.

Despite these contributions, this study calculated the torque of the spiral bevel gear by using an equation applying only the gear ratio and power transmission efficiency based on the data measured on the wheel. This may be difficult to sufficiently reflect the dynamic effect of the final reduction gear. According to the dynamic characteristics of the gear, when the torsional stiffness of the drive shaft of the gear increases, the vibration average of the gear and the stress amplitude of the gear root increase, and the reliability of the gear decreases [31]. Therefore, there is a need for a calculation model that can sufficiently consider the dynamic effect of the final reduction gear. This work will be conducted in our future study. In addition, as an expanded study based on the results of this study, a real-time fatigue life prediction system will be developed using the real-time dynamic field loads generated during tractor operations.

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