Failure Life Prediction of Hub Bearing in Composite Tooling

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Abstract: Composite work tools have many components, and complex shapes when compared to standard tools such as rotaries and plows. In addition, the component durability for the tools is critical because the load varies severely according to the soil conditions. This study predicts the fatigue life of the hub bearing in composite work tooling. The loads acting on the tools were measured based on field tests, and the loads acting on the hub bearing were derived using the load reconstruction method. The static safety factor and fatigue life of the hub bearing, loads, and contact stress acting on the inner and outer raceways and presence of truncation were analytically predicted based on the derived loads. The fatigue life of the bearing changed depending on the preload amount of the hub bearing. The bearing life was more than 3000 h for preloads of less than 40 µm, which satisfied the target life of 1200 h. The load acting on the inner and outer raceways of the bearing decreased and then increased as the preload amount of the bearing rose. The bearing contact area, maximum contact stress, and number of balls increased as the load applied to the hub bearing rose. The fatigue life, load, contact stress, and static safety factor of the hub bearing met all requirements, and no truncation occurred on the inner and outer raceways of the bearing. The test verified the achievement of the target life of 1200 h and confirmed that there was no breakage, cracking, or deformation of the bearing.

Keywords: composite tooling; disk harrow; failure life; hub bearing; prediction

1. Introduction

A farm tractor is a machine for various farm jobs by installing attachments such as a rotary and a plow in the vehicle’s power take-off (PTO) or a three-point hitch system. Attachments must be replaced according to the different jobs, which lowers efficiency. Composite work tools have been found to perform two or more concurrent operations [1,2].

A composite tool can concurrently perform tillage operation and hilling work. It consists of the mainframe, disk harrow assemblies, chisel plows, and a hiller. A disk harrow assembly includes a disk harrow arm, a hub bearing, and a disk harrow.

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The part durability of composite tooling is much more critical compared to general working tools such as rotaries and plows because composite working implements have many components and diverse shapes of working implements and are subject to relatively large load fluctuations according to soil environments. Field loads should be measured considering field environments and working conditions and reflected based on the design [3] to ensure the durability of composite work tools.

In the case of the hub bearing used for smooth rotation of the disk harrow in composite working implements, severe misalignment occurs because of combined loads acting on the disk harrow, and there are difficulties in lubrication because of poor working environments. This misalignment dramatically affects the accuracy of the fatigue life of the bearing predicted through the ISO standard. Moreover, the most crucial factor in predicting the fatigue life through the ISO standard is the load acting on the bearing, and it is not easy to derive the reliable load level considering working environments.

Collins and Cupera installed torque meters on the axle to measure the load acting on the tractor transmission [4,5]. This method is very easy to measure when the torque meters are installed on the tractor’s wheels, but it is necessary to reduce the measurement error of wheel speed in order to measure the transmitted output power exactly. It also may differ slightly from the actual load since it is assumed that the measured load is the same as the tractor’s tractive force and this load is applied equally to all components of the working implement.

Park and Kim installed a six-component load cell on the three-point hitch system of the tractor to measure the tractive force required for the working implements [6,7]. A six-component load cell is installed between the three-point hitch system of the tractor and the composite working implement, and the tractive force was calculated by converting the force and moment components measured by 3-axis load cells. The load acting on a disk harrow was modeled as the resistance of soil crushing force, shear force and soil weight [8], and calculated using the equation of motion. However, there is an advantage in measuring the load acting on all of the working implements, but it is impossible to measure the load of the component itself since the six-component load cell is installed as an integrated measuring device.

Kim measured and analyzed the loads measured by a torque meter installed on the PTO shaft of a tractor during the rotary work and a six-component load cell installed on the three-point hitch system during the plow work [9]. However, the loads acting on the transmission input shaft, the steering and the three-point hitch system could not be measured, so the load on the component itself could not be measured accurately.

Oh and Bae analyzed the effects of rotational speed, radial load and misalignment on the bearing fatigue life [10,11]. Bearing fatigue life decreased as the radial load increased in the low speed region. Whereas since the contact force caused by the centrifugal force acting on rolling elements became relatively larger than it caused by the radial load in the high speed region, the effect of the radial load decreased. It was also shown that the bearing life decreased drastically regardless of the operating conditions when the bearing misalignment was more than 1 mrad. However, this study confirmed the effects of the bearing’s operating conditions on fatigue life analytically, and these effects were not validated through bearing tests or any other method.

Park analyzed the influence of the type change of the output shaft bearings of the pitch reducer for wind turbine system on the load distribution over the gear tooth flank and load sharing among the planets [12]. This study confirmed that the gear life in the pitch reducer increased when the combination of the spherical roller bearing (SRB) and cylindrical roller bearing (CRB) was changed to two taper roller bearings (TRBs).

Zhang has developed a methodology to predict bearing life under a corrosive environment based on accelerated life testing data and the inverse power law. Bearing life tests under various corrosion stress levels were performed and the test result shows that the accelerated life test effectively assess the bearing life based on accelerated environmental testing [13]. Hongwu presented the durability test methods for wheel bearings. Through studying on the offset between the wheel center line and the position where a force is applied to the tire, the simulating load position on the wheel bearing was...
calculated. Depending on the influence of inner circle or outer circle rotation on the wheel bearing life, the test method on the bench was determined [14]. From the above two papers, information on the accelerated life test method and test devices for the hub bearing in the working implement could be obtained.

Few studies have derived the loads to evaluate the life of the components of working implements, such as bearings, and few studies have conducted experimental verification to check the accuracy of the predicted life.

This study was conducted to predict the fatigue life of the hub bearing used in composite working implements. The load acting on the composite working implement was measured through field tests, and the combined loads acting on the hub bearing were derived using the load reconstruction method. Based on the derived loads, the static safety factor, and fatigue life of the bearing, the load acting on the inner and outer raceways of the bearing, contact stress, and truncation were analytically predicted. Besides, the predicted fatigue life of the bearing was verified through a life test.

2. Derivation of Loads on the Hub Bearing

The entire content of this chapter was written based on the results of load measurement, analysis, and reconstruction in a previous study conducted by Han, et al. [3].

2.1. Structure of Composite Working Implements

A composite working implement is a working implementation that performs two or more tasks simultaneously with the tractive force of a tractor. The composite working implement in this study is one that simultaneously performs tillage operation and hillig work. As shown in Figure 1, the composite working implement consists of a three-point hitch connector for mounting working tools on the three-point hitch system of the tractor, disk harrow assemblies for tillage operation for paddy fields or fields, chisel plows for crushing subsoil, a hiller for forming ridges, and the mainframe for mounting components.

![Figure 1. Structure of composite working implements.](image)

There are 18 disk harrow assemblies in total. Nine versions of assembly are installed at equal intervals in each of the rows in the front and rear of the mainframe, and four chisel plows are mounted in a row in the middle between the front and rear disk harrow assemblies. As shown in Figure 2, a disk harrow assembly consists of a disk harrow that performs tillage operation, a hub bearing that enables the rotational movement of the disk harrow and supports loads and a disk harrow arm for connecting the hub bearing and the frame.

2.2. Field Test Workload Measurement of the Composite Working Implement

The specifications of the tractor selected to derive the workload of the composite working implement are as shown in Table 1. A GPS speedometer was used to measure the actual driving speed,
considering the slip of the tractor wheel. A six-component load cell was installed at the three-point hitch connection between the tractor and the composite working implement. Strain gauges were attached to the disk harrow assemblies of the composite working implement to measure the loads acting on the composite working implement.

Finite element analysis determined select positions at the disk harrow arm for attachment of strain gauges. Four locations (S1, S2, S3, S4) where large strains were shown were selected and attached with strain gauges, as shown in Figure 3. An accurate measurement system was configured with the data acquisition system (DAQ) model Quantum X (MX840B, MX1615B) of HBM Co. We also used an industrial PC model UNO-2483G of ADVANTECH Co. that is robust against tractor vibrations. An inverter converted the tractor 12 V direct current into the alternating current required by DAQ and PC, and an uninterrupted power supply for stable data acquisition, since the tractor is turned off during the measurement. The measurement equipment was installed inside the tractor cabin.

The experimental field for the field tests is a rectangular pavement located in Baeksan-myeon, Gimje-si, Jeonbuk, Korea. The soil strength and moisture content were measured using the soil cone penetrometer model DIK-5530 ((DAIKI Co.), which measures strength according to soil penetration depths, and the soil moisture sensor model WT1000B (Mirae Sensor Co.), which measures moisture contents in soil. The measured soil strength and penetration depths were 1939 kPa and 18 cm, respectively, and the determined moisture content was 8.6%.

The field test condition for the composite working implement was set to 15 km/h, which is the maximum speed. A travel distance was selected to allow an acceleration of 15 s after the tractor starts, a deceleration of 15 s before the tractor stops, and the maximum speed lasted for 24 s.
A total of five field tests were conducted. The representative workload was determined to calculate the equivalent load for fatigue life prediction of the hub bearing, as shown in Figure 4. Stain data were taken only when the traveling speed was in a range of 14–16 km/h, and the plowing depth varied from 20 to 30 cm, which helped to minimize changes in the workload according to travelling speed and plowing depth of the tractor. The traveling rate was measured using a tractor-attached GPS speedometer, and the plowing depth of the disk harrow was identified by analyzing the images of the tractor-attached CCD camera. Although the strain data obtained in these methods are judged to be severer than actual working conditions, they are worthy of bringing about somewhat conservative results in the prediction of the fatigue life of the hub bearing. Figure 4 shows the results of field tests (1st–5th test) of the disk harrow assembly at the traveling speeds of 14–16 km/h. The data are measured during 224.3 s in total (results of five trials).

2.3. Derivation of Loads on the Hub Bearing

Static load tests on the disk harrow assembly were performed to derive the equivalent load on the hub bearing. The static load test equipment was configured, as shown in Figure 5. The bolt fastening region of the disk harrow arm connected to the frame was fixed to the jig. An actuator was connected to the hub bearing mounting area. A 3000 N load was applied for each direction (X, Y, Z direction).
The strain transformation matrix was calculated using the strain data obtained through the static load tests and cyclic loadings. Here, the strain transformation matrix refers to the sensitivity of the strain to cyclic load and is a matrix for transforming the strain data of field tests into the load history. Using the strain data obtained through field tests, as shown in Figure 4 and the strain transformation matrix obtained through the static load test, the load histories (Fx, Fy, Fz) of the disk harrow assembly were reconstructed as shown in Figure 6. The reconstructed load histories in the x, y, and z directions were calculated in the form of the load sizes (Figure 7) and the principal directions (Table 2) over time through the synthesis of load vectors.

Table 2. Principal direction of disk harrow assembly.

| Direction | θx | θy | θz |
|-----------|----|----|----|
| Angle, °  | 82.4 | 41.0 | 50.0 |

The load sizes and principal directions were converted into equivalent loads using Equation (1) presented in ISO 6336-6 [15] to predict the fatigue life of the hub bearing.

\[
F_{eq} = \left( \sum_{i=1}^{k} \frac{n_i F_i^x}{n_i} \right)^{\frac{1}{x}}
\]

(1)

where,

- \( F_{eq} \): Equivalent load acting on the bearing
- \( n_i \): Number of cycles for divided range i
- \( F_i \): Load for divided range i
- \( x \): Slope of the Wohler curve (acceleration index)

If it is assumed that the wheel and disk harrow of the tractor does not slip on the ground, when the tractor’s working speed is 15 km/h, and the diameter of the disk harrow is 600 mm, the rotation speed of the disk harrow and hub bearing will be 132.6 rpm. The number of cycles, \( n_i \), was calculated using the rotation speed of the hub bearing and the total time of load history 224.3 s. Although the
slope of the Wohler curve varies with the parts and the processing method, three is generally applied in the case of ball bearings [11]. The load data in Figure 7 and derived by the equation (1) leads to an equivalent load at 1.69 kN. The equivalent load was split into loads in the x, y, and z directions using the angle of the principal direction of the disk harrow assembly like Table 2. The separated loads for x, y, and z directions were calculated as 0.22 kN, 1.27 kN, and 1.08 kN respectively as shown in Figure 8. The fatigue life of the hub bearing was analyzed using the results of the calculation of the loads.

Figure 6. Load reconstruction for disk harrow assembly in all directions.
Figure 7. Load amplitude history for disk harrow assembly.

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|-----------|----|----|----|
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\[
F_{eq} = \sum_{i} F_i \left[ x_k^i \right]_n
\]

where,

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- \( n_i \): Number of cycles for divided range \( i \)
- \( F_i \): Load for divided range \( i \)
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3. Prediction of the Life of the Hub Bearing

3.1. System Model and Analysis

The hub bearing that connects the disk harrow arm with the disk harrow is a special bearing used exclusively for the disk harrow. The overall shape of the disk harrow assembly was modeled to analyze the hub bearing. A finite element model [12] was applied to the disk harrow arm and the disk harrow. The shape of the element is a second order tetra mesh. The numbers of nodes and elements of the disk harrow are 4639 and 2148, respectively, and the numbers of nodes and elements of the disk harrow arm are 20,405 and 10,994, respectively, as shown in Figure 9. Six degrees of freedom (Fx, Fy, Fz, Mx, My, Mz) were the boundary conditions for the disk harrow arm connection to the frame and the shaft of the hub bearing [10,11,14]. Five degrees of freedom except for torque were set for disk harrow and housing of the outer raceway of the hub bearing.
Figure 9. FE model of disk harrow assembly. (a) Disk harrow arm (b) Disk harrow.

The equivalent load of the hub bearing according to the ISO 281 standard was calculated with Equation (2) and input loads in the x, y, and z directions derived in Section 2.3; the value was 1.98 kN [16].

\[ P = XF_r + YF_a \]  

where,

- \( P \): Equivalent load
- \( X \): Radial load factor
- \( Y \): Axial load factor
- \( F_r \): Radial load
- \( F_a \): Axial load

### 3.2. Analysis of the Life and Characteristics of the Hub Bearing according to Changes in the Preload

The hub bearing is located between the disk harrow and the disk harrow arm. The disk harrow is bolted to the outer raceway cover of the hub bearing, and the threaded part of the shaft protruding from the inner raceway of the hub bearing is inserted into a hole located in the disk harrow arm and fastened with a nut. The fastening torque determines the preload of the hub bearing used when the disk harrow arm is attached to the inner raceway of the hub bearing, and the fatigue life of the hub bearing varies according to changes in the preload. Therefore, changes in the fatigue life according to the preload of the hub bearing were analyzed.

Utilizing the finite element model of the disk harrow assembly, the fatigue life of the hub bearing was predicted using the ISO 281 standard while increasing the preload of the hub bearing by 5 \( \mu m \) from 0 to 50 \( \mu m \). [12,16]. The bearing life was the longest as \( B_{10} \) 3600 h when the preload of the hub bearing was in a range of 20–30 \( \mu m \). The bearing life sharply decreased when the preload exceeded 40 \( \mu m \). The bearing life was \( B_{10} \) 3245 h when the preload amount was 0 \( \mu m \) and \( B_{10} \) 3154 h when the preload was 40 \( \mu m \). Therefore, when assembling the disk harrow, the fastening torque should be determined so that the preload of the hub bearing is within a range of 0 to 40 \( \mu m \), as shown in Figure 10.

Changes in the loads applied to the inner/outer raceways of the two rows of bearings according to the preload of the hub bearing, the maximum contact stress, and truncation, and the static safety factor of the bearings according to ISO 76 were identified [17]. The maximum contact stress of ball bearings should not exceed 4200 MPa, and the safety factor should not be smaller than 3 [17]. The loads applied to the inner/outer raceways of the two rows of bearings decreased and increased with the preload of the hub bearing, as seen in Figure 11 [12]. When the preload was 0 \( \mu m \), the maximum load was shown to be 0.98 kN in row 1 and 1.41 kN in row 2. Figure 12 showed contact stress levels of the inner/outer raceways of the bearing when the preload was 0 \( \mu m \) and when the preload was 25 \( \mu m \), at which the maximum life was shown. The maximum contact stress levels were shown to be 2230 MPa and 2171 MPa at the inner raceway of the bearing, and it was identified that the number of balls that meet raceways increased. The contact stress of the inner raceway of the bearing tended to be higher than that of the outer raceway, stemming from the difference between the radius of curvature of the inner raceway and that of the outer raceway. The position where the maximum contact stress acts
in row 1 of the bearing is different by 180° from that in row 2 of the bearing. This difference is because the contact occurs at the 90° position in row 1 while occurring at the 270° position in row 2 because of the deformation of the shaft located in the inner raceway of the bearing and bearing misalignment. The static safety factor of the bearing was shown to be 20.57. Figure 13 is the graph that shows the contact pattern to check the presence of truncation in the inner/outer raceways of the bearing. The thick solid lines placed vertically on the x-axis at equidistant intervals show the positions where actual contact occurred when a load was applied, and the balls met the inner/outer raceways. When a load acts in the y-direction from the outer raceway to the balls, 0° is the center of the cross-section of the width of the inner/outer raceways. The positions along the y-axis mean that contacts occur on the left or right of the center. The reason that the contacts occur on the left or the right is the occurrence of axial loads. Since no contact occurred at 65° and −65°, which are the ends on two sides based on the 0° of the inner/outer raceways, no truncation occurred in the inner or outer raceway.

![Graph 1](image1.png)

**Figure 10.** Bearing life over preload of hub bearing.

![Graph 2](image2.png)

**Figure 11.** Raceway load over preload of hub bearing.
Figure 12. Cont.
Figure 12. Bearing contact stress over preload of hub bearing. (a) Preload 0 μm (b) Preload 25 μm.

All requirements for the fatigue life and static safety factor of the bearing, loads on the inner/outer raceways, contact stress, and truncation according to ISO 76 and 281 were satisfied [16,17].
Figure 13. Cont.
4. Experimental Evaluation of the Life of the Hub Bearing

4.1. Determination of Accelerated Life Test Conditions

Although the target life of the composite working implement is $B_{10} = 800$ h, the target life of the hub bearing was determined as $B_{10} = 1200$ h because the hub bearing should have higher reliability than the composite working implement. The zero-failure test time to ensure the target life can be calculated with equation (3). The primary failure mechanism of the bearing is fatigue failure, and the shape parameter was 2.0 when the life distribution was assumed to follow the Weibull distribution [18].
When calculated with the confidence level of 0.9 (90%), the number of test samples of 1, and the reliability of 0.9 (90%), the zero-failure test time was shown to be 5610 h.

\[
T_{ZFT} = B_p \left[ \frac{\ln(1 - CL)}{n \ln(R)} \right]^\frac{1}{\beta}
\]

(3)

where,

- \(T_{ZFT}\): Zero-failure test time
- \(B_p\): Target life of the hub bearing
- \(CL\): Confidence level
- \(n\): Number of test samples
- \(R\): Reliability
- \(\beta\): Shape parameter

As for the zero-failure test time, one sample of the hub bearing should be tested under the equivalent load condition for 5610 h, and no failure should occur during the test. However, since it is challenging to conduct the test for 5610 h, conditions for the accelerated life test were selected. The load and rotation speeds for the accelerated life test of the hub bearing were determined using the inverse-power model, which is mainly used as an accelerated life test model of bearings [13,18]. The equivalent speed of the hub bearing was 132.6 rpm, and 200 rpm was selected as the rotation speed for the acceleration test considering the temperature of the hub bearing, etc. The equivalent load imposed on the actual hub bearing is 1.98 kN, but a static load of 6 kN was applied as the accelerated test load in consideration of the measured load during the field test of the composite working implement. The acceleration index was set to 3 because the hub bearing is a ball bearing. The acceleration factor (AF) and the zero-failure accelerated life test time (T_{ZFALT}) for the calculation of the accelerated life test time of the hub bearing were calculated using Equations (4) and (5).

\[
AF = \left( \frac{P_{test}}{P_{eq}} \right)^x \left( \frac{N_{test}}{N_{eq}} \right)
\]

(4)

\[
T_{ZFALT} = \frac{T_{ZFT}}{AF}
\]

(5)

where,

- \(AF\): Acceleration factor
- \(P_{test}\): Accelerated test load
- \(P_{eq}\): Equivalent load
- \(x\): Slope of the Wohler curve

(Acceleration index)

- \(N_{test}\): Accelerated test rotation speed
- \(N_{eq}\): Equivalent rotation speed
- \(T_{ZFALT}\): Accelerated life test time

The resultant total \(AF\) is 42.23, and the accelerated life test time is 133 h.

The static safety factor and fatigue life of the hub bearing, the loads acting on the inner and outer raceways of the hub bearing, the contact stress, and whether there were any contact stress and truncation were analytically checked under the conditions of the rotation speed of 200 rpm and the static load of 6 kN. The operating condition was changed so that the outer raceway of the hub bearing would be fixed, and the inner raceway of the hub bearing would rotate to better consider the boundary conditions of the actual accelerated life test.
The static safety factor of the hub bearing was shown to be 6.03, which was greater than 3 recommended in ISO 76, and the fatigue life was shown to be 149.8 h with a 95.4% decrease compared to the life when the preload was 0 µm as shown in Section 3.2 [17]. It was longer than the accelerated life test time 133 h. The loads acting on the inner/outer raceways of the hub bearing were shown to be 1.83 kN in row 1 and 4.42 kN in row 2, and the maximum contact stress levels of the inner/outer raceways of the hub bearing were shown to be 2413 MPa and 3282 MPa, respectively, as shown in Figure 14. Whereas row 1 in Figure 12a showed a tendency for the maximum stress to occur at 90°, row 1 of in Figure 14 showed a tendency for the maximum stress to occur at 270°. The maximum stress of row 2 showed a phase difference of 180° from row 1, attributable to the inner raceway of the bearing, fixed as described in Section 3.2, the outer raceway of the bearing, as described in Section 4.1 in consideration of the accelerated life test conditions. However, when the analysis was conducted considering the boundary conditions in Section 3.2 and the load conditions in Section 4.1, the results of life analysis tend to be larger than 149.8 h, indicating that these conditions are more conservative than the actual conditions. When Figure 12a was compared with Figure 14, the contact area and the maximum contact stress of the bearing showed a tendency to increase as the load applied to the hub bearing increased.

Figure 15 is the graph that shows the contact pattern for checking the presence or absence of truncation in the inner/outer raceways of the bearing, in which contact area showed a tendency to increase similarly to the results of the analysis of contact stress shown in Figure 14. No truncation occurred even when the load applied to the hub bearing increased.

The accelerated life test conditions were calculated based on the target life of hub bearing B₁₀ 1200 h. All the static safety factor and fatigue life of the bearing, the loads on the inner/outer raceways, contact stress, and truncation satisfied the requirements according to ISO 76 and 281 even under loads applied to the hub bearing, that is, test load conditions [16,17].

4.2. Configuration of the Test Rig

For fatigue life tests of the hub bearing, a test rig was designed, as shown in Figure 16 [13,14]. The inner raceway is fixed to the disk harrow arm, and the outer raceway rotates together with the disk in actual hub bearing operating conditions. Despite this fixture, the boundary condition was changed in the life test so that the outer raceway would be fixed, and the inner raceway would rotate. The test rig was configured so that the rotation speed of the hub bearing would be controlled using an electric motor, and the equivalent loads in the x, y, and z-axis directions would act on the hub bearing by using an actuator. Figure 17 shows the configuration of the test rig and sensor locations for fatigue life tests of the hub bearing [13,14]. The main specifications of the test rig and sensors are shown in Table 3.
Figure 14. Bearing contact stress of hub bearing under test load condition.
Figure 15. Cont.
Figure 15. Contact footprint between balls and raceways under test load condition. (a) Inner/Outer raceway of row 1 (b) Inner/Outer raceway of row 2.
Figure 16. Schematic diagram of test rig.

(a).

(b).

Figure 17. Test rig for hub bearing. (a) Test rig (b) Locations of measuring sensors.

Table 3. Specifications of components for test rig.
was no abnormality, additional tests were carried out to meet the life test time in the analysis. The atmospheric temperatures were in a range of 9~12°C. Figure 18 is a graph that shows changes in the actuator load during the accelerated life test of the hub bearing. The command load applied to the actuator during the test was 6 kN of static load, but it is assumed that the variable load was caused by the rolling body passing through the load zone of the bearing as the inner raceway of the bearing rotating.

4.3. Results of Fatigue Life Tests

Fatigue life tests were conducted at 6 kN and 200 rpm, which are analytically validated accelerated life test conditions. The torque measured by the torque meter during the test ranged from 1.96 to 5.89 Nm, and the range of maximum measured temperatures of the temperature sensor attached to the surface of the housing of the hub bearing was 28~30°C indicating that almost a constant temperature was maintained. In this case, the atmospheric temperatures were in a range of 9~12°C. Figure 18 is a graph that shows changes in the actuator load during the accelerated life test of the hub bearing. The command load applied to the actuator during the test was 6 kN of static load, but it is assumed that the variable load was caused by the rolling body passing through the load zone of the bearing as the inner raceway of the bearing rotating.

Any abnormal noise and vibration of the bearing was checked for after 133 h through the hearing and touch of the test engineer, which is the accelerated life test time. After identifying that there was no abnormality, additional tests were carried out to meet the life test time in the analysis. The accelerated life test was conducted for a total of 151 h, and it was confirmed that the bearing was operating normally without any abnormal noise during the test. After completion of the test, the visual inspection of the hub bearing was performed and, as shown in Figure 19, no damage to the hub bearing such as breakage, cracking and deformation of the hub bearing was found in the visual inspection. Based on the test results, it was identified that the hub bearing satisfied the life of B10 1200 h under...
reliability of 0.9 (90 %), confidence interval of 0.9 (90%) and the equivalent load conditions of 1.98 kN and 132.6 rpm.

![Appearance of hub bearing after test.](image)

**Figure 19.** Appearance of hub bearing after test.

5. Conclusions

This study was conducted to predict the fatigue life of the hub bearing used in composite working implements. The loads acting on the composite working implement were measured through field tests, and the loads acting on the hub bearing were derived using the load reconstruction method. Based on the derived loads, static safety factor, and fatigue life of the hub bearing, the loads acting on the inner and outer raceways, contact stress, and truncation were analytically predicted. An accelerated life test was performed to validate the fatigue life of the hub bearing. The results obtained through the series of studies are as follows.

1. In order to confirm the relationship between the strain and the load acting on the hub bearing, a static load test was performed to obtain a strain transformation matrix. Using the strain transformation matrix, the strain history was reconstructed as a load history, and the equivalent load condition for the direction with the greatest fatigue damage was obtained.

2. In order to predict the hub bearing life, the change of bearing life according to the difference in the preload amount was confirmed. In addition, a preload value that maximizes bearing life could be found.

3. The characteristics of the bearing were analyzed under the equivalent load condition of the hub bearing derived from the field tests and the accelerated life test conditions determined using the foregoing, and according to the results, all the bearing life, loads, contact stress, and static safety factor satisfied the requirements, and no truncation occurred in the inner or outer raceway.

4. According to the results of the accelerated life test of hub bearing, there was no abnormal noise, breakage, cracking, or deformation during operation. Therefore, it was verified that the life of $B_{10,1200}$ was satisfied under reliability of 0.9 (90 %), confidence interval of 0.9 (90 %).

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