Study on performance and continuous durability test of single input multi output power take-off (PTO) gearbox

Studie zur Leistung und Dauerfestigkeit eines Nebenantriebsgetriebes mit Einzeleingang und Mehrfachausgang

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The power take-off (PTO) gearbox is widely used for special vehicles, large construction machinery, industrial machinery and ships, to distribute the power of a single input shaft from an engine or a motor to several shafts or to collect several shafts by single shaft power (Torque × rotation speed). The super large hydraulic excavator is equipped with a power take-off (PTO) gearbox between the main engine and a number of pumps for the purpose of improving the fuel efficiency and improving the system efficiency. The power take-off (PTO) gearbox of a large hydraulic excavator with a body weight of 100 ton (≈ 10⁵ kg) is connected with two large hydraulic pumps and one small hydraulic pump on three output shafts and different axial loads are generated depending on the working characteristics of the excavator. In this paper, we evaluated the performance and durability of a power take-off (PTO) gearbox of a large hydraulic excavator with a 100-ton body weight. The test evaluation criteria for non-fault life time calculation, performance and accelerated life test were established, and the types and characteristics of acceleration test equipment were analyzed. The torque distribution with the use of the gear phase difference to maintain the gearbox output condition was investigated, and energy was saved by circulating experimental power while controlling speed and load with two electric dynamometers. We analyzed the life of the gearbox by analyzing the metal particles generated in the lubricating oil of the gearbox.

Keywords: Power take-off (PTO) gearbox / hydraulic excavator / durability test / single-input multi-output / torque distribution

Das Zapfwellengetriebe wird häufig für Sonderfahrzeuge, große Baumaschinen, Industriemaschinen und Schiffe verwendet, um die Leistung einer einzelnen Eingangswelle von einem Antrieb oder einem Motor auf mehrere Wellen zu verteilen oder mehrere Wellen auf eine einzelne Welle zu bündeln (Drehmoment × Drehzahl). Der Hydraulikgroßbagger ist mit einem Zapfwellengetriebe zwischen dem Hauptmotor und einer Reihe von Pumpen ausgestattet, um die Kraftstoffeffizienz und die Systemeffizienz zu verbessern. Das Zapfwellengetriebe eines großen Hydraulikbaggers mit einem Eigengewicht von 100 Tonnen (≈ 10⁵ kg) ist mit zwei

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großen Hydraulikpumpen und einer kleinen Hydraulikpumpe auf drei Abtriebswellen verbunden, und abhängig von den Arbeitseigenschaften des Baggers werden unterschiedliche Axiallasten erzeugt. In diesem Beitrag wird die Leistung und Lebensdauer eines Zapfwellengetriebes eines großen Hydraulikbaggers mit einem Eigengewicht von 100 Tonnen bewertet. Die Testbewertungskriterien für die Rechnung der fehlerfreien Lebensdauer, die Leistung und den beschleunigten Lebensdauertest wurden festgelegt und die Typen und Eigenschaften der Beschleunigungstestgeräte analysiert. Die Drehmomentverteilung unter Verwendung der Getriebephasendifferenz zur Aufrechterhaltung des Getriebeausgangszustands wurde untersucht, und Energie wurde durch Zirkulieren von experimenteller Leistung bei gleichzeitiger Steuerung von Drehzahl und Last mit zwei elektrischen Dynamometern eingespart. Die Lebensdauer des Getriebes wurde analysiert, indem die im Schmieröl des Getriebes erzeugten Metallpartikel analysiert wurden.

Schlüsselwörter: Zapfwellengetriebe / Hydraulikbagger / Dauerfestigkeit / Mehrfachausgang mit Einzeleingang / Drehmomentverteilung

1 Introduction

In recent years, due to the increased concern regarding environmental and energy problems, wind energy as a renewable energy source has received increased attention. Since 2005, the average annual growth rate of the wind power capacity has reached 20%. It is expected that wind power will account for 12% of the world’s electricity supply by 2030 [1]. Wind power has the potential to become an important part of new power supplies and has achieved large-scale applications. The economics of wind power development and utilization significantly has been improved and the cost has dropped by approximately 30% over the past five years.

Hydraulic excavators are the most used construction equipment in the world. In recent years, the demand for large-sized hydraulic excavators (large-sized excavators) has been increasing in large plants and mines, which can increase the work efficiency and reduce the labor cost burden on x drivers. However, large-sized excavators are low in fuel efficiency, cause air pollution and are not able to cope with rising oil prices. Therefore, it is required to develop a super large excavator with improved fuel efficiency [2]. The super large excavator uses several hydraulic pumps for the purpose of increasing the fuel efficiency and supplies appropriate hydraulic pressure to a large number of hydraulic cylinders such as a boom, an arm, a bucket and so on, a traveling motor and a swing motor for excavation work. A single input is taken from the output shaft of a large engine, and two large hydraulic pumps equivalent to about 40% of engine power and one small hydraulic pump equivalent to about 20% are used, Figure 1. In this case, a single input multi-output power take-off (PTO) gearbox (hereinafter referred to as “gearbox”) is used to distribute power. As a result, the perform-
ance and life span of the gearbox are critical as a key component in the performance and life of the excavator.

In order to verify the performance and durability of the gearbox, it is necessary to determine the conditions of the load to be applied and the method of application. In order to reproduce the field operating conditions, a hydraulic pump should be installed in the gearbox to generate load, but the long-term economic efficiency should be considered. The gearbox is composed of a 3-axis output, and the shaft of the small hydraulic pump has a small cross section, so that when the same torque is applied, the shaft is damaged, Figure 1. Therefore, the gear phase difference technology of the gearbox should be applied in order to distribute 20% torque to one output shaft and 40% torque to each of two output shafts, respectively.

In this paper, we evaluated the performance and durability of power take-off (PTO) gearboxes of 100-ton class heavy-duty hydraulic excavator, calculate the required performance and durability life of power take-off (PTO) gearboxes, and establish test evaluation criteria for performance and accelerated life test. Also, the type and characteristics of the accelerometer were analyzed, and the gear phase difference torque distribution was used to maintain the gear box output condition. Two electric dynamometers were used to control speed and load while circulating experimental power, which resulted in significant energy savings and the analysis of the life of the gearbox was also conducted by analyzing the metal particles generated in the lubricating oil of the gearbox.

2 Accelerated life test and test equipment

2.1 Required performance and calculating no failure lifetime of power take-off (PTO) gearbox

The power take-off (PTO) gearbox transmits power from the engine to three hydraulic pumps, so that three gears of the same number of teeth engage in one pinion and rotate. Therefore, the losses due to the errors in three external gear power transmission, the frictions on six bearings and input/output shaft sealing and lubricating oil churning are generated, which are converted into heat, noise, and vibration.

Since the gearbox has the highest loss due to power transmission errors of gears, it focuses on the design of the optimal gear profile and the grinding and optimum lubricant supply to improve the tooth surface roughness and it requires more than 98% efficiency. In addition, the life expectancy of a super-large excavator is about 20,000 hours (h) when it is supposed to be used for 8 years, 250 days a year for 10 years, and so is that of the gear box because it is driven simultaneously when the engine is operated. Calculating the test time satisfying the no failure acceptance criterion in order to guarantee the life of the gearbox of 20,000 h (B5 life), we followed the Weibull distribution (β) of 1.7 as the shape parameter of the life distribution. When the guaranteed life is 20,000 h, the confidence level is 60%, and the sample is 2, the no failure test time is as shown in Equation 1 [3].

$$t_n = B_{100p} \cdot \left[ \frac{\ln(1 - CL)}{n \cdot \ln(1 - p)} \right]^{\frac{1}{\beta}}$$

$$t_n = B_{100p} \cdot \left[ \frac{\ln(1 - CL)}{n \cdot \ln(1 - p)} \right]^{\frac{1}{\beta}} = 20,000 \cdot \left[ \frac{\ln(1 - 0.6)}{2 \cdot \ln(1 - 0.05)} \right]^{\frac{1}{1.7}} = 72,511.70$$

Here in, $t_n$ is no failure test time, $B_{100p}$ is the assurance life, $CL$ is the confidence level, $n$ is the number of all items under test (number of samples), $p$ is the unreliability ($p = 0.1 \text{ for } B_{10}$ lifetime), $B$ is the shape parameter (1.7). The no failure life test time of the gearbox is 72,000 h, and the equivalent life test is required to ensure reliability. However, it is impossible to carry out a test with a large power consumption over 72,000 h on account of long test time and high cost, so an accelerated life test by overload should be performed. Therefore, the gear fitting was selected as the failure mode of the gear box, an acceleration model of inverse power law model was applied, and the life test design was carried out by using the torque generating main failure mode as an acceleration factor.

In Equation 2, life test time $N$ represents the number of contact of the gears, and is a constant $\Delta$ and a constant $\lambda$ for the strength of the contact times. Equation 3 is expressed as the product of the service life time $tf$ (hours), the number of rotations, and the number of times of contact $q$ per a rotation.
Equation 4 shows x surface stresses \( (1 \text{ kgf} \approx 9.81 \text{ N}) S_x \) (kgf cm\(^{-2}\)) with elastic modulus \( C_p \) (\( \sqrt{\text{kgf}/\text{mm}^2} \)), transfer tangent load \( W_t \) (kgf), overload index \( K_o \), dynamic factor \( K_s \), magnitude factor \( K_m \), surface condition for x fitting resistances \( C_n \), net surface width \( F \) (mm) for thin portion, and gear shape factor \( I \) for fitting resistance.

Equation 5 shows the tangential load \( W_t \) as the ratio of the transmission torque \( \tau \) to the revolution speed \( u \) per minute of the transmission torque gear and the working pitch diameter (mm) \( d \). Equations 6–8 can be obtained by putting Equations 3–5 into Equation 2.

Acceleration factor \( AF \) can be expressed as Equations 9–10, expressed by the ratio between the condition at the site and the test application condition. The \( \lambda_n \) value for the gear fitting can be found through the \( S-N \) curve, where the index value of \( S_x \) is 10 kgf cm\(^{-2}\) (98.1 N cm\(^{-2}\)), and the slope can be identified in ANSI / AGMA even though there is a difference, Figure 2. Torque was selected as the acceleration factor for the accelerated life test of the gear fitting of the gear box. Universe power law model was selected as the acceleration model and the equivalent torque of the normal condition was 1,372 N m, and the equivalent torque of the acceleration condition was 3,000 N m [4].

In addition, the acceleration index (m) 5 of the torque was applied, and the acceleration factor \( (AF) \) 45.03 was used according to the result of the Equation 11 [5].

**Figure 2. S-N Curve for the gear pitting.**

\[
N = \Delta S_x^{-\lambda} \tag{2}
\]
\[
N = 60n_{kgf} \tag{3}
\]
\[
S_x = C_p \sqrt{W_t K_o K_s K_m \frac{C_f}{d F T}} \tag{4}
\]
\[
W_t = \frac{2 \tau}{d u} \tag{5}
\]
\[
60n_{kgf} = \Delta \left( C_v \sqrt{W_t K_o K_s K_m \frac{C_f}{d F T}} \right)^{-\lambda_n} \tag{6}
\]
\[
t_f = \frac{1}{60 q} \cdot \left( C_v \sqrt{2 \tau K_o K_V K_s K_m \frac{C_f}{d^2 u F T}} \right)^{-\lambda_n} \tag{7}
\]
\[
t_f = \frac{1}{60 q} C_p^{1.5} \cdot \left( C_v \sqrt{\frac{F d^2 u}{2 K_o K_V K_s K_m C_f} I^{0.5 \lambda_n - 0.5 \lambda_n - 1}} \right)^{0.5 \lambda_n - 0.5 \lambda_n - 1} \tag{8}
\]
\[
AF = \frac{t_f}{t_f} = \frac{\tau_{test}}{\tau_{field}} \left( \frac{n_{test}}{n_{field}} \right)^{0.5 \lambda_n} \tag{9}
\]
\[
AF = \frac{\tau_{test}}{\tau_{field}} \left( \frac{n_{test}}{n_{field}} \right)^{\lambda_n} \tag{10}
\]
\[
AF = \left( \frac{T_{test}}{T_{field}} \right)^m \left( \frac{N_{test}}{N_{field}} \right)^n \tag{11}
\]

Here, \( AF \) is the acceleration coefficient, \( T_{field} \) and \( N_{field} \) are the use condition, and \( T_{test} \) and \( N_{test} \) are the torque and the rotation number of the acceleration test condition, and \( m \) and \( n \) are the acceleration index. Therefore, the minimum acceleration test time \( (t_m) \) of the pump drive gear box of the super large excavator is about 1,620 h as shown in Equation 12 [6].

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\[
\frac{t_{na}}{AF} = \frac{72,511.70}{45.03} = 1,610.461,620 \text{ h (12)}
\]

No failure time was calculated by regarding the gear fitting as the main failure mode as described above. The shape parameter is 1.7, acceleration index is 5, and continuous durability test time calculated 1,620 h.

When the bearings are regarded as the main failure mode part, the durability test time of about 8,200 h was calculated under the same acceleration condition as the shape parameter of 1.3 and the acceleration index of 10/3. Therefore, bearings can be installed with a large capacity, so that the design life is much longer than the gear design life. If the bearings are tested on a life-time basis, the test is required to take about three years, which is practically impossible.

In the test similar to the power take-off (PTO) gear box, the tooth fitting of gears which contact with high load of 30 times per second when rotating at 1800 min\(^{-1}\) had a higher failure rate rather than the bearing and the gear fitting was adopted as the main failure mode considering the test requirements of customers.

We also analyzed the chemical composition of the gearbox lubricant periodically for condition monitoring of tooth wear, speed, cracking, chipping damage and bearing wear and damage of gears inside the gearbox during the long accelerated life test with overload for 1,620 h. Accelerometer was attached to the main part of the test product to measure the frequency, and a fault was diagnosed by periodic frequency spectrum analysis.

### 2.2 Test equipment

The gearbox life test system used drive motors (electric motors, hydraulic motors, large engines, etc.) that supplied about 150% more power than the engine power for the accelerated life test, Figure 3. The input power was required to apply a load

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**Figure 3.** Original modelling of power take-off (PTO) gearbox test equipment.

**Bild 3.** Ursprüngliche Modellierung der Prüfausrüstung für das Zapfwellengetriebe.
using a hydraulic pump. However, since most of the input power was consumed by the frictional heat of the hydraulic oil resulting in the 100 % loss of the input power, an additional electric power was required to cool the heat oil.

Therefore, one large electric dynamometer (1,700 HP dynamometer) was used to drive one unit, and the other regenerated about 85 % of the test electric power by generating load power, Figure 4. The electric power required for the long time overload test was reduced to about 15 %. In addition, the reliability of test results was improved with the effect of testing two samples simultaneously by using two gearboxes.

2.3 Torque distribution technology

Since 40 % of power is used on each of two outputs and only 20 % on one shaft in the power take-off (PTO) gearbox, axial torque distribution technology was required in order to make a similar condition to the service condition. This study was designed to carry out the test with the initial axle torque distributed by varying the amount of contact of each axis with the use of the back lash and the phase difference of the torque distribution gear. The configuration of the test device in the power take-off (PTO) gearbox included two 1,700 HP dynamometer, five torques and rpm sensors, Figure 5. No. 1 shaft of three output shafts is connected with key to key combination and No. 2 and No. 3 axes are with key to power lock combination, Figure 6.

Initial torque of each axis was distributed by holding the output shaft and rotating the input shaft, and torque distribution was verified, Table 1, Figures 7–9.

2.4 Dynamic analysis and results of torque distribution

A model of the components of the equipment for dynamic analysis of test equipment was explained the main components being input/output gearbox, rotating shaft and coupling, Figure 10. The analysis

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**Figure 4.** Improved modelling of power take-off (PTO) gearbox test equipment.

**Bild 4.** Verbesserte Modellierung der Prüfausrüstung für das Zapfwellengetriebe.
elements to be considered were torque and rotational displacement applied to the gear of the gearbox on input side called \( T_{Gi}, \theta_{Gi} \) and the output side defined as \( T_{Go}, \theta_{Go} \). Torque and rotational displacement acting on the j-pinion were also defined as and tooth stiffness with input / output gear and j-pinion was defined as \( T_p^{(j)}, \theta_p^{(j)} \) and \( k_{in}^{(j)}, \theta_{os}^{(j)} \). The stiffness of the coupling connecting the Pinion rotating shaft was and the stiffness of the rotating shaft on both sides of the coupling was, while the factors determining the teeth rigidity and axial stiffness were one equivalent axial rigid \( k_c^{(j)}, k_n^{(j)} k_i \).

A series of calculations is omitted, and the analysis results by \( \zeta_{Tk} \) selected as the main factor determining the effectiveness of the torque distribution technique can be summarized as follows. \( \zeta_{Tk} \) is a coefficient representing the ratio of set torque \( (T'_p) \) to x axis stiffness \( (k_i) \) to be supplied to each axis, and \( \zeta_{Tk} = 1 \) if the set torque and the axis stiffness are same. The torque supplied to each

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**Figure 5.** Photo of power take-off (PTO) gearbox test equipment (a) components of test equipment, (b) output shafts and torque meters.

**Bild 5.** Foto des Prüfgeräts für das Zapfwellengetriebe (a) Komponenten des Testequipments, (b) Abtriebswelle und Drehmomentmesser.

**Table 1.** Torque distribution ratio [unit: %].

| Input Torque | Gear ratio | 1st Shaft | 2nd Shaft | 3rd Shaft |
|--------------|------------|-----------|-----------|-----------|
| 1,000 Nm     | 1 : 0.979  | 41.6      | 38.8      | 18.4      |
| 1,500 Nm     | 40.8       | 40.2      | 18.0      |
| 2,000 Nm     | 40.9       | 40.6      | 18.0      |
| 3,000 Nm     | 40.7       | 41.0      | 17.6      |
| Average      | 41.0       | 40.2      | 18.0      |
| Target distribution ratio | 40 | 40 | 20 |

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**Figure 6.** Connection type of output shaft. 1\(^{\text{st}}\), 2\(^{\text{nd}}\) and 3\(^{\text{rd}}\) Shaft.

**Bild 6.** Verbindungstyp der Abtriebswelle. Erste, zweite und dritte Welle.
axis in the case of $\xi_Tk$ 0.5, 1 and 1.5, Figure 11. The dynamic analysis has, where $\xi_Tk = 1$, the ratio of torque acting on each pinion axis does not change even if the applied torque is changed [5], Figure 12.

The efficiency before and after the acceleration test of the gearbox was compared. Efficiency tests were performed under various load conditions, Figure 13, Table 2. In addition, the efficiency before and after a life test was compared, and the average efficiency before and after the life test was 97.9 %, and the average efficiency after the life test was 98.2 %, which was somewhat higher than that before the life test.

3 Accelerated life test results of power take-off (PTO) gearbox

3.1 Comparison of efficiency before and after the life test

The efficiency before and after the acceleration test of the gearbox was compared. Efficiency tests were performed under various load conditions, Figure 13, Table 2. In addition, the efficiency before and after a life test was compared, and the average efficiency before and after the life test was 97.9 %, and the average efficiency after the life test was 98.2 %, which was somewhat higher than that before the life test.
3.2 Analysis of the contamination level of lubricants

The power take-off (PTO) gearbox uses a method of lubricating some gears that rotate inside the housing by raising the lubricant when they rotate with the face of some gears embedded in the lubricant. Therefore, during the acceleration test, an analysis of the increase in the level of contamination of lubricants in the gearbox was carried out by checking the degree of deterioration of the test target [7–8]. The criteria for analyzing the contamination level of lubricating oil divided into normal and careful states by analyzing iron (Fe) and chromium (Cr) in $10^{-6}$, the main components of the

![Figure 10. Dynamic modeling of two PTO gearboxes.](image1)

![Figure 11. Analysis result of distributed torque.](image2)

![Figure 12. Comparison between test result and dynamic analysis.](image3)
In the test procedure to check the contamination level of the gearbox, the contamination level of the lubricating oil was analyzed by dividing it into certain sections from the beginning of the test life test to completion. The analysis of wear components of gear lubrication has found that the components of iron and chromium are mainly wear elements, Figure 14.

In the beginning of the gearbox test, the life test was carried out without replacing the lubricant up to about 210 h after injecting new oil. Further analysis of lubrication oil was performed after 210 h to be marked separately on the graph. Lubricants were replaced after spending 498 h, 738 h, 936 h and 1019 h in testing. Results after 1,019 h indicated that $402 \times 10^{-6}$ and $165 \times 10^{-6}$ of iron (Fe) were de-

| Metal       | Normal | Marginal | Caution | Critical |
|-------------|--------|----------|---------|----------|
| Iron (Fe)   | 0–100  | 101–150  | 151–300 | > 300    |
| Chromium (Cr)| 0–4   | 5–10     | 11–15   | > 15     |
| Lead (Pb)   | 0–30   | 31–50    | 51–80   | > 80     |
| Copper (Cu) | 0–30   | 31–50    | 51–80   | > 80     |

Figure 13. Efficiency test result for power take-off (PTO) gearbox.

Figure 14. Analysis result of oil contamination.

For analysis of the contamination level, Table 3 [6].

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ected at both input and output respectively, significantly still higher than the reference values of $100 \times 10^{-6}$, Table 3. This has shown that contamination level tends to increase exponentially, and from 1,019 h later to 1,300 h, the results show a significant increase in iron (Fe) content. Therefore it is presumed from the results that the gear wear increased sharply and the gear reached almost the limited life.

4 Conclusion

In this study, a test device was built with torque distribution technology for durability test of single input multi-power power take-off (PTO) gearbox, accelerated life test was conducted, and efficiency and lubricating oil component analysis were performed in order to confirm durability during the test.

1. If three actual mounting pumps were used for load in the test of the power take-off (PTO) gearbox, the on-site operation conditions could be reproduced, but the overload test was not possible.

2. In conducting the test of the power take-off (PTO) gearbox for a long period of time, the control method of the 1700 HP AC-dynamometer was adopted, saving about 85 % of the test energy by circulating electric power.

3. As the technology for distributing torque was developed using the gear phase difference of the output power take-off (PTO) box, two of the three output shafts could be loaded 40 % and the remaining one 20 %.

4. Acceleration life test as endurance test has been conducted for 1,620 h under the test conditions of $1,800 \text{ min}^{-1}$ and 3,000 Nm, with rotational speed and torque as the accelerating factor.

5. The efficiency test was conducted before and after the endurance test to observe the change in efficiency and showed that the result of 98.2 % of efficiency could meet 98 % required by demander.

6. The lubrication oil contamination analysis was performed periodically to analyze the wear patterns of the gears by dint of predicting the life span of the gearbox, and the iron (Fe) content increased rapidly after 1019 h, which showed that there was a sharp increase in wear of the gears and the gear can be determined to have reached the limited life.

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