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An Experimental Investigation into the Durability of the Main Clutch Friction Disc in Tracked Vehicles

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Abstract:

This study investigates the wear durability of the main clutch friction disc in tracked vehicles through the application of an accelerated testing methodology. The wear resistance of the friction disc is a key indicator of the main clutch's operational reliability. Conventional durability assessments under standard working conditions require prolonged testing durations and multiple test samples, resulting in substantial time and financial costs. To overcome these limitations, an accelerated testing approach is proposed, wherein the friction disc is subjected to continuous high-stress operation using a dedicated test rig, thereby inducing rapid wear. Experimental data obtained from this process is subsequently analyzed to predict the friction disc's lifespan under normal service conditions, providing a cost-effective and efficient alternative for durability evaluation.

Keywords:

accelerated testing, friction disc, main clutch, tracked vehicle, Weibull distribution

1 Introduction

The main clutch is a critical assembly within the power transmission system of tracked vehicles, located between the engine and the gearbox (Fig. 1). Its primary functions include transmitting torque from the engine to the subsequent components of the drivetrain, disengaging the power flow from the engine to the gearbox when required, and serving as a safety mechanism to prevent sudden overloads within the powertrain system. Additionally, it facilitates smooth vehicle startup [1]. This clutch is a multi-disc type consisting of 19 plates, operating under dry friction conditions with steel-on-

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steel contact surfaces (Fig. 2). The compressive force required for torque transmission is generated by coil springs [2].

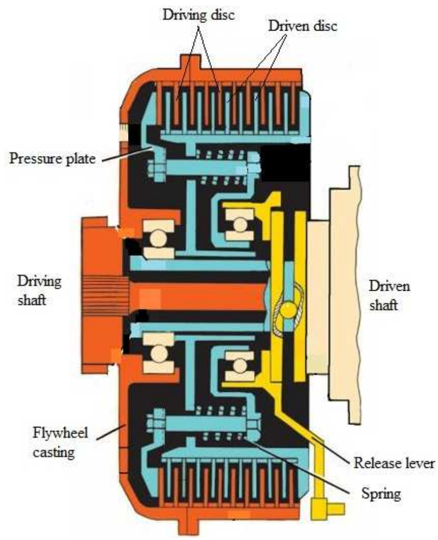

Fig. 1 Structure of the main clutch in a tracked vehicle

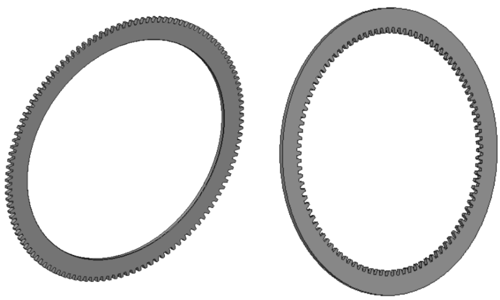

Fig. 2 Structure of the driven and driving friction disc assembly

The friction discs of the main clutch operate under harsh dry-friction conditions characterized by high sliding speeds, elevated temperatures, and significant contact pressures. As a result, they tend to wear rapidly and are typically the components with the shortest service life within the power transmission system of tracked vehicles. To address this issue, the materials used for manufacturing friction discs must possess a high and thermally stable coefficient of friction, excellent wear resistance, as well as high thermal conductivity and thermal stability [3].

In the development of friction discs, it is essential to evaluate their operational durability. However, applying conventional (statistical) methods to assess the reliability of these components can be extremely time-consuming, labor-intensive, and costly. To overcome these limitations, accelerated testing methods are employed to rapidly predict the reliability of the main clutch friction discs. In this approach, the components are tested under conditions that are significantly more severe than their actual operating environment. The data obtained under high load conditions is then used to estimate the service life of the product under normal working conditions [4–6].

2 Experimental Setup

2.1 Determining Working Load of Friction Disc

The specific pressure on the friction surface of the main clutch is a key parameter that significantly influences the operational performance of the clutch [7]. To ensure that the main clutch can transmit high torque while maintaining reasonable wear durability and allowing for smooth clutch engagement, the value of specific pressure must remain within the allowable limits of the friction material. The schematic diagram for calculating the clutch is shown in Fig. 3. Fig. 4 shows the schematic diagram used to determine the specific pressure on the friction surface.

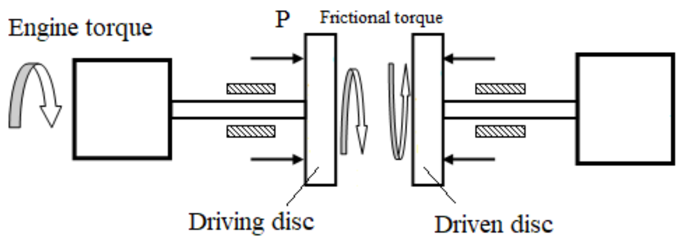

Fig. 3 Schematic diagram for calculating the clutch assembly

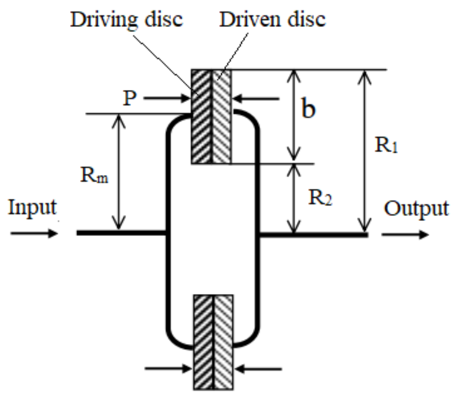

Fig. 4 Schematic diagram for determining the specific pressure

The specific pressure on the friction surface (q) is determined as follows [2]:

$$q = \frac{P}{2\pi R_m b \lambda} \tag{1}$$

where P is the total clamping force exerted by the springs; R_m is the mean radius of the friction disc; b is the width of the friction disc; λ is the coefficient that accounts for the reduction of the friction surface area due to grooving (for a dry steel friction disc without groove, $\lambda = 1$).

In this study, the operating conditions are determined under the startup mode, and the clutch load for testing is defined as the total clamping force exerted by the springs on the working surface of the main clutch friction discs.

The total clamping force exerted by the springs is calculated using the following equation:

$$P = n \times C \times f \tag{2}$$

where n is the number of clutch springs (n = 18); f is the initial deformation of the clutch spring when the clutch is in the engaged state (f = 21.5 mm); C is the spring stiffness. The stiffness of the main clutch spring is determined using the following equation [8]:

$$C = \frac{d^4 E}{8D^3 n_{sp}} \tag{3}$$

where E is the modulus of elasticity of the material used for the clutch spring, made of steel ($E = 8.5 \times 10^4$ MPa); d is the wire diameter of the spring (d = 4 mm); D is the mean diameter of the spring coil (D = 27 mm); n_{sp} is the number of coils in the spring (n_{sp} = 9).

Substituting into Eq. (3), the spring stiffness is obtained as: C = 15 350 N/m. Substituting into Eq. (2), the result is P = 4 860 N. This value is determined as the load condition (level) under normal operating conditions of the vehicle. The specific pressure on the friction surface is $q = 0.2 \times 10^6$ N/m^2 = 0.2 MPa.

2.2 Determination of Enhanced Load

Based on the operating conditions of the main clutch, the accelerated load selected for the experiments is the total clamping force of the springs, converted to the specific pressure on the friction surface. Specifically, the following accelerated load levels were chosen for testing: P_1 = 5 830 N (120 %), P_2 = 6 800 N (140 %), P_3 = 7 290 N (150 %).

The load levels and number of test samples are presented in Tab. 1.

Tab. 1 Load levels and number of test samples

Load Level	Clamping Force [N]	Specific Pressure [MPa]	Number of Samples
Low (P_1)	5 830	0.24	15
Medium (P_2)	6 800	0.28	12
High (P_3)	7 290	0.30	10

In this study, the experiment concludes when all the samples have failed, thus the failure time (the number of sliding cycles of the friction disc) is a random variable. At each load level, the test samples are sequentially placed into the test apparatus and subjected to the test until failure occurs. After all samples at the low load level have been tested, the samples at the higher load levels are tested.

2.3 Development of Testing System

For conducting experiments, a specialized testing device was designed and fabricated. This device includes an electric motor, a main clutch assembly, a flywheel, a loading device, a hydraulic system to control the clutch engagement and disengagement process, a wear measurement device for the friction disc, and a control panel (Figs 5 and 6). The specialized device operates based on simulating the real working conditions of the main clutch in a tracked vehicle as closely as possible.

The friction disc is placed within the clutch, and the movement of the pressure disc is controlled by a hydraulic system. The testing device automatically stops when the friction disc fails (when the wear of the disc reaches 0.175 mm [9]). The failure time (measured in terms of the number of clutch cycles) is recorded by the measuring device.

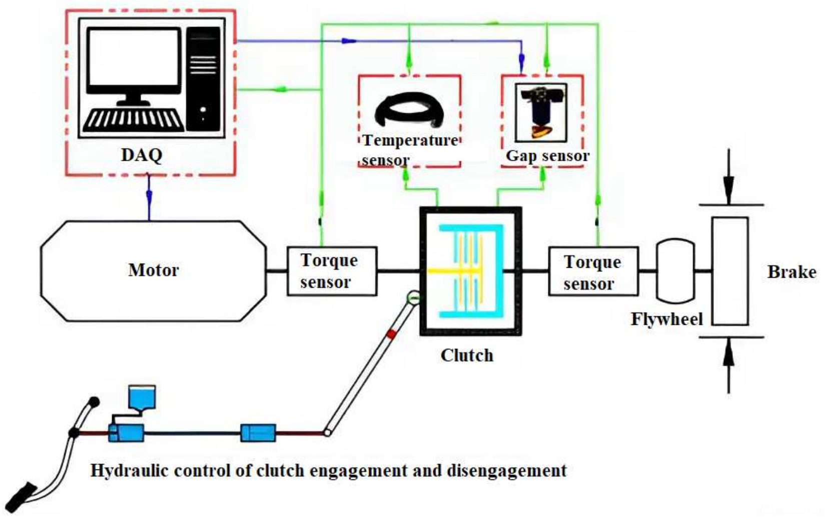

Fig. 5 Schematic diagram of testing device

Fig. 6 Experimental setup for accelerated testing of friction disc

3 Evaluation of Test Results

3.1 Test Results

After conducting the experiments, the results showing the number of clutch cycles until the friction disc fails are presented in Tab. 2 (sorted in ascending order).

Tab. 2 Test results

Low (P_1)	**Medium** (P_2)	**High** (P_3)
20 368	15 054	13 752
20 990	16 197	14 178
21 334	16 518	14 304
21 474	16 976	14 422
21 623	17 301	14 564
21 850	17 385	14 740
22 184	17 479	14 961
22 346	17 597	15 151
22 395	18 111	15 247
22 715	18 265	15 587
23 015	18 341	
23 119	18 612	
23 525		
23 614		
24 028		

3.2 Evaluation of Results

Based on the test results obtained, the first step is to determine the probability distribution that best fits the collected data. The Maximum Likelihood Estimation (MLE) method is used to estimate the parameters of the distributions. Subsequently, the Anderson-Darling (A–D) method was used to test the goodness-of-fit for the three probability distributions identified above [10, 11]. The results showed that the Weibull distribution best fits the data obtained from the experiments. Histogram of the clutch cycles corresponding to the load levels are shown in Figs 7, 8, and 9. Fig. 10 shows probability density function (PDF) of the Weibull distribution. The parameters of the Weibull distribution are shown in Tab. 3.

Tab. 3 Estimated parameters of the Weibull distribution

Load Level (P)	**Weibull shape parameter** (β)	**Weibull scale parameter** (η)
Low (P_1 = 5 830 N)	25	22 785
Medium (P_2 = 6 800 N)	23	17 750
High (P_3 = 7 290 N)	31	14 945

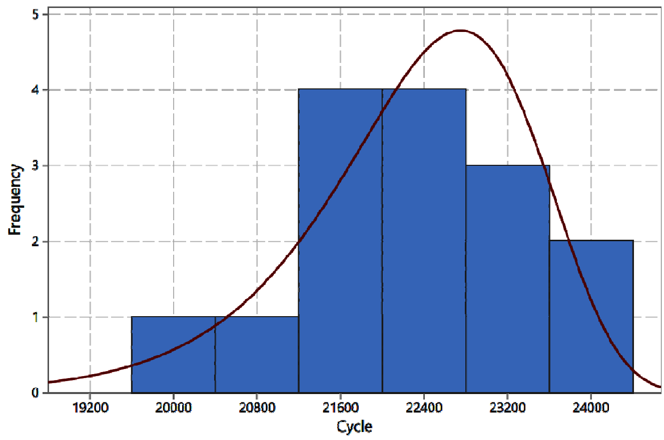

Fig. 7 Histogram of clutch cycles at low load level

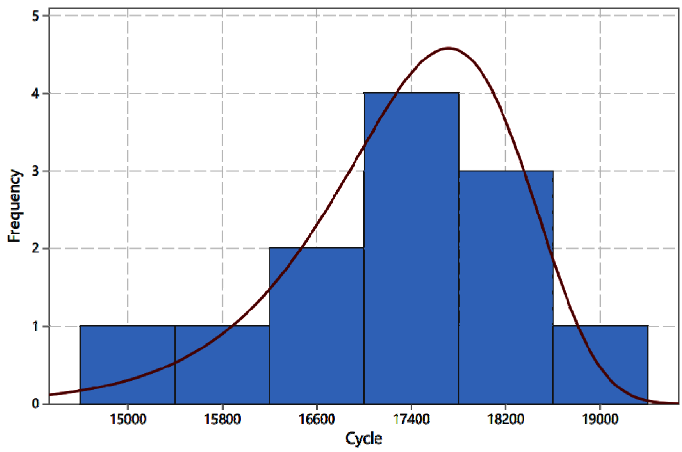

Fig. 8 Histogram of clutch cycles at medium load level

The next step is to determine the parameters of the lifetime-load relationship. In this study, the Inverse Power Law (IPL) model is used to extrapolate the reliability parameters under normal operating conditions [12]:

$$L(P) = \frac{C}{P^a} \tag{4}$$

where $L(P)$ represents a quantifiable life measure, such as mean time to failure (MTTF); P is the stress (total clamping force); C is the model parameter; a is the model parameter dependent on the behavior of the stress.

The IPL-Weibull model for the lifetime-load relationship is [13]:

$$\eta = L(P) = \frac{C}{P^a} \tag{5}$$

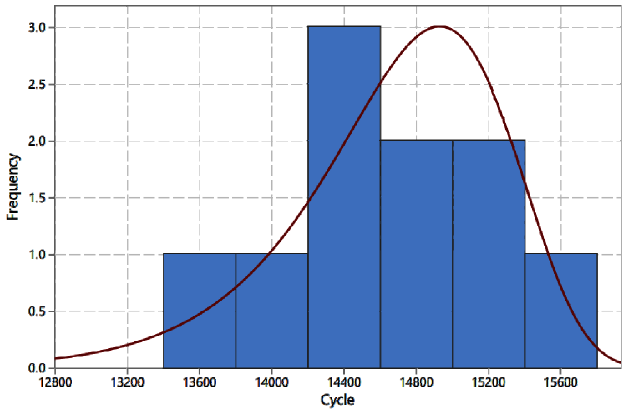

Fig. 9 Histogram of clutch cycles at high load level

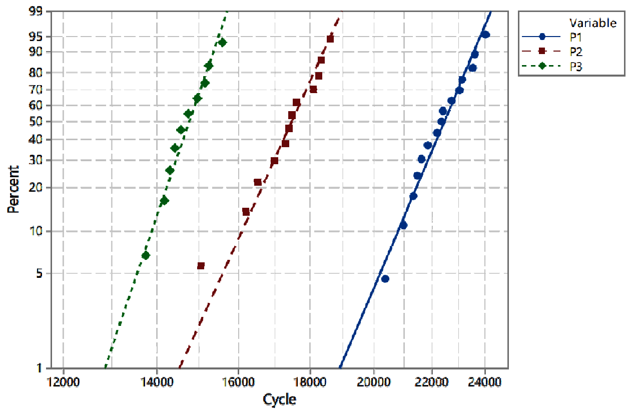

Fig. 10 PDF of Weibull distribution

The relationship is linearized by taking the natural logarithm of both sides of the IPL-Weibull model:

$$\ln\eta = -a \ln P + \ln C \tag{6}$$

where a is the slope of the line, and ln C is the y-intercept of the line.

Tab. 4 is constructed based on the testing conditions and the estimated parameters of the Weibull distribution.

Tab. 4 Conversion Data of IPL-Weibull Model

Load Level (P)	**Scale Parameter** (η)	ln P	ln η
$P_1 = 5\,830$ N	22 785	8.67	10.03
$P_2 = 6\,800$ N	17 750	8.82	9.78
$P_3 = 7\,290$ N	14 945	8.89	9.61

Using the least squares method, the IPL-Weibull relationship is shown in Fig. 11, with the model parameters obtained as a = 1.843; ln C = 26.025. The estimated scale parameter of the Weibull distribution under normal operating conditions for the friction disc (with the calculated load level of P_0 = 4 860 N) is:

$$\eta_0 = \frac{e^{26.025}}{4860^{1.843}} = 32\,214$$

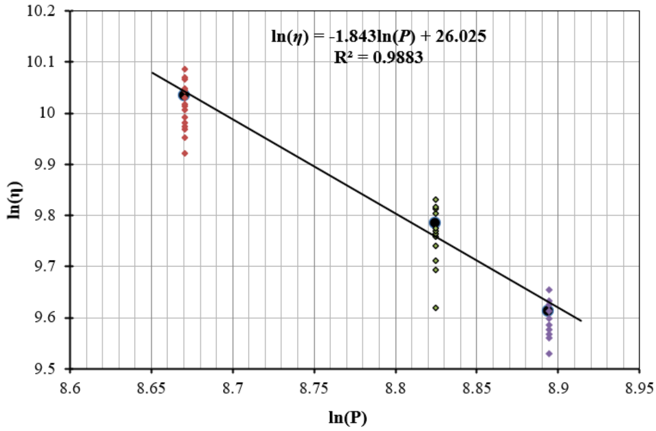

Fig. 11 Lifetime-Load Relationship

The estimated shape parameter (β_0) of the Weibull distribution under normal operating conditions for the friction disc is calculated using the following equation [11]:

$$\beta_0 = \frac{\sum N_i \beta_i}{\sum N_i} \tag{7}$$

where N_i is the number of failures of the test sample at the i-th stress level. Substituting the known values into Eq. (7), the shape parameter of the Weibull distribution in this case (N_1 = 15, N_2 = 12, N_3 = 10) is determined to be:

$$\beta_0 = \frac{15 \cdot 25 + 12 \cdot 23 + 10 \cdot 31}{15 + 12 + 10} = 26$$

The *MTTF* of the friction disc according to the Weibull distribution is determined as follows [11]:

$$MTTF = \eta_0 \, \Gamma\left(\frac{1}{\beta_0} + 1\right) \tag{8}$$

where $\Gamma\left(\frac{1}{\beta_0} + 1\right)$ is the gamma function at the value $\left(\frac{1}{\beta_0} + 1\right)$.

Substituting the available parameters into Eq. (8) gives:

$$MTTF = 32\,214 \cdot \Gamma\left(\frac{1}{26} + 1\right) = 32\,214 \cdot \Gamma(1.04) = 32\,214 \cdot 0.979 = 31\,538$$

Thus, the lifetime of the friction disc obtained from accelerated testing is 31 538 engagement cycles. The main clutch friction disc meets the required operational durability.

4 Conclusion

The study defined the test conditions and established the experimental setup for the friction disc in the main clutch of a tracked vehicle and conducted accelerated testing at three different load levels. The test results show that the friction disc satisfies the required service life. Furthermore, the accelerated testing method proves to be a highly effective tool for predicting the reliability and service life of component assemblies, contributing to significant savings in both time and cost.

From the results obtained, the paper proposes the process and technical parameters for the accelerated testing of the main clutch of the tracked vehicle, aiming to quickly and accurately assess the quality of the product during the production and inspection process. This process can be widely applied to different types of clutches, contributing to improving the reliability and service life of the tracked vehicle in particular, and motor vehicles in general. This study opens a new direction in the application of accelerated testing methods in the field of quality control and service life prediction of machine parts, especially in harsh working environments. Further research can focus on optimizing enhanced load levels, as well as building more accurate service life prediction models, based on rich test data and other influencing factors such as temperature and environment.

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