Failure Mechanism Analysis and Simulation Verification of An Unidirectional Transmission Mechanism

HUAN PANG, NING WANG, AND MENG LI
School of Automobile, Chang’an University, Xi’an 710064, China
Corresponding author: Huan Pang (panghuan@chd.edu.cn)

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Abstract

Unidirectional transmission mechanism (UTM) is widely used in mechanical field. The UTM studied in this paper is the core component of the aircraft hatch system, which is used to transfer the power of the motor to drive the hatch to be retracted and locked automatically after the motor closes. Aiming at the failure mode exposed in the experiment, the matching relations of both geometric parameters and mechanical parameters are studied, and the force analysis model of the UTM is established. The results show that the load of the UTM reaches its maximum at 65°, and if the friction coefficient between the friction plates is less than its critical value, braking failure will occur, which is in agreement with the test phenomena. In order to further verify the failure mechanism and obtain the effects of the gradual damage on the performance, a parameterized simulation model is established, through which failure phenomenon in the test is recovered, and the correctness of the failure mechanism analysis is verified. Finally, combined with its failure mechanism, the maintenance advice is proposed, and the validity is verified through the simulation model.

Index Terms

Unidirectional transmission mechanism, braking performance, failure mechanism, parameter matching, dynamic simulation.

I. INTRODUCTION

Unidirectional transmission mechanism (UTM), one of the important mechanical transmission devices, is to achieve unidirectional transmission of power but to prevent the reverse transmission of power, which is widely used in industry and daily life, including the hatch system of aircraft and ships, electric lifting system of doors and windows, electric sunshade system, et.al. The rear wheel transmission mechanism of bicycles and automatic brake mechanism of elevator are also belonged to UTM.

Almost every aircraft and ship has several hatches, traditionally, the opening and closing of these hatches are controlled by hydraulic system, which consists of hydraulic pump, pipe, valve, actuating cylinder, etc. The system is complicated and has a serious problem of reliability. What’s more, it is difficult to lock the hatch door in arbitrary position. For this reason, electric retractable system has been widely adopted, and the UTM is the core component of the electric retractable system which controls the retractable and locking of the hatch. During motion stage, the motor works, and UTM transfers power to the hatch, when the hatch needs to be stopped, the motor is closed, and the UTM automatically stops the hatch at the current position. Therefore, the performance of the UTM directly affects the movement of the hatch, and it is crucial to the electric retractable system and even the aircrafts or ships.

Since UTM is widely used, many scholars have been interested in its research [1]–[6]. The essence of the UTM analyzed in this paper is a friction clutch. Chen et al. [7] investigated the friction and load-bearing characteristics of the one-way clutch with multi-bodies contacting type, the result indicates that friction self-locking characteristic of this clutch is related with geometrical shape of contacting components, material characteristic, loading capacity and contacting state. Kudra et al. [8] compared three models of resultant contact force, which indicates that the approximate friction force models based on the Padé approximant and their extensions...
are presented more comprehensively. It can be seen that almost none of the papers considered manufacturing error or degradation factors. However, too many factors will lead to the change of the values. The failure of the mechanical system is mostly caused by manufacturing error or degradation factors [9]–[11].

NOTATION

- $T$: Load of the UTM acted on output shaft
- $\omega$: Rotation speed of the input shaft
- $H_0$: Penetrable depth of ideal case (no manufac- tural errors and clearance);
- $L$: Axial clearance of the output shaft bearing
- $W$: Total wear depth of friction discs
- $\beta$: Cone angle of output shaft
- $r$: Distance between cone axis to middle shaft axis
- $R_{in}$: Inner radius of friction disc
- $R_{out}$: Outer radius of friction disc
- $R$: Equivalent radius of friction disc
- $u$: Dynamic friction coefficient between the friction disc
- $F_s$: Force that corrugated spring acted on the middle shaft
- $F_T$: The component of steel ball contact force in shaft axis direction
- $F_r$: The component of steel ball contact force in radius direction
- $F \rho$: The component of steel ball contact force perpen-
dicular to the $F_T$ and $F_r$
- $n$: Number of contact surfaces of friction discs;
- $\mu_T$: Minimum friction coefficient corresponding to $T$
- $\theta_0$: The angle between door closed position (AE) and gravity direction
- $\theta_1$: The angle between the AD and the gravity direction
- $\gamma$: The angle of the door structure at the rotation shaft
- $\theta_A$: Angle of the Four-bar linkage at point A
- $\theta_B$: Angle of the Four-bar linkage at point B
- $\theta_C$: Angle of the Four-bar linkage at point C
- $\theta_D$: Angle of the Four-bar linkage at point D
- $\theta_{min}$: The minimum rotation angle of the door
- $\theta_{max}$: The maximum rotation angle of the door
- $l_1$: Length of BC rod
- $l_2$: Length of CD rod
- $l_3$: Length of AB rod
- $l_4$: Length of AE rod
- $l_5$: Length of AD rod
- $F_{ij}$: Reaction force that component $i$ acted on $j$
- $s_1$: The rotation arm that $F_{13}$ acted on the door
- $s_2$: The rotation arm that gravity acted on the door
- $G$: Gravity of the hatch
- $k$: Safety factor
- $K$: Elastic coefficient of corrugated spring
- $\Delta l$: Adjusting length of the corrugated spring

The UTM studied in this paper exposes some faults in the test. The aim of this study is to investigate failure reason of the UTM and obtain several useful suggestions for UTM design and maintenance. This paper is organized as follows: The work principle and failure phenomena of UTM are pointed out in Section II. Failure mechanism is deeply analyzed in Section III. Simulation model of both UTM and hatch system are established in Section IV. Failure mechanism obtained in Section III is verified and several maintenance suggestions are put forward in Section V. The last section is the conclusion of the paper.

II. FAILURE BACKGROUND OF UTM

A. THE COMPOSITION AND WORKING PRINCIPLE OF UTM

As shown in Fig. 1, the UTM mainly composed of input shaft, middle shaft, output shaft, dynamic friction disc, static friction disc, steel balls (each side has three steel balls) and corrugated spring.

![Figure 1. Composition and assembly drawing of UTM (1. output shell; 2. output bearing; 3. output shaft; 4. steel ball; 5. gasket; 6. corrugated spring; 7. input shell; 8. input bearing; 9. input shaft; 10. middle shaft; 11. dynamic friction disc; 12. static friction disc).](image-url)
the friction discs apart from each other, so that the output shaft rotates with the input shaft.

**B. TEST PRINCIPLE AND FAILURE PHENOMENA**

Two kinds of tests were carried out for UTM: 1) Individual test, in which only UTM is tested; 2) system test, in which the UTM is put into the hatch system.

1) INDIVIDUAL TEST AND FAILURE PHENOMENON

The principle of the individual test is shown in Fig.2: a constant forward load or reverse load $T$ is applied at the output shaft of the UTM, and a constant rotation speed is applied at the input shaft. After UTM is running smoothly, turn off the motor (that means the input shaft is released to free), then use angle sensor to measure the angle difference of the output shaft before and after the motor shuts off. Here the rotation speed of input shaft is set to 50 rpm and the torque of output shaft is set to 15.5 N·m.

- In the forward load condition (the direction of the torque is consistent with the speed direction), after the motor shuts off, the output shaft can brake immediately.
- In the reverse load condition (the direction of the torque is opposite to the speed direction), after the motor shuts off, the output shaft rotates a certain angle before brake.

**FIGURE 2.** Principle diagram of the individual test.

2) SYSTEM LEVEL TEST PRINCIPLE AND FAILURE PHENOMENON

The principle of system level test is shown in the Fig.3, it can be seen that the system consists of motor, UTM, reducer, four-bar linkage and hatch. The input shaft is driven by a motor that has a specific speed-torque characteristic. The output shaft, reducer, four bar mechanism and hatch are connected successively. In the test, turn off the motor during the opening or closing period of the hatch, and use angle sensor to measure the angle difference before and after the motor shuts off.

- Shuts off the motor when opening angle of the hatch is among 60° to 80°, the hatch will drop down a certain angle and then stop.
- When shut off the motor at other opening angle, the hatch can brake normally.

**III. FAILURE MECHANISM ANALYSIS**

**A. DIFFERENTIAL ANALYSIS OF LOAD AND REVERSE LOAD CONDITIONS**

In order to find out the reason of failure phenomenon in the individual test, difference of the forward condition and the reverse condition is analyzed, as shown in Fig.4.

1) FORWARD LOAD CONDITION

The motion state of the mechanism under the forward load condition is shown in the Fig.4 (a): Before motor shuts off, the direction of the torque generated by the hatch is consistent with the speed direction, actually, it is the driving force when the door closing, and the motor acts as a speed-limiting function. Therefore, the UTM always exhibits braking characteristics.

When shut off the motor, the input shaft become free, at this time, the direction of the torque generated by the hatch is still consistent with the original direction before, thus, the contact position of the steel ball in output remains unchanged, while the input shaft and the steel ball lose force. In this state, the middle shaft still keeps the movement tendency before the motor is closed. Therefore, the UTM still keeps the braking characteristics before the motor is closed, and the hatch can brake immediately.

2) REVERSE LOAD CONDITION

The movement state of the mechanism under the reverse load condition is shown in Fig.4 (b): Before the motor closes, the motor provides the power for the input shaft, and it drives the steel balls of the input side, middle shaft and the steel balls of the output side successively. In this state, the contact point between the steel balls and shaft as shown in Fig.4 (b).

When the motor is closed, the input shaft becomes a free state, due to gravity, the hatch will drop down, and the rotation direction is opposite to the direction before the motor closes. Under this condition, the contact point of the output shaft and steel balls will change, therefore, in the reverse load condition, the hatch will reverse rotate a certain angle before stopped.

**B. GEOMETRIC PARAMETER MATCHING RELATION ANALYSIS**

According to the braking principle, the input shaft is free and the output shaft has torque and initial speed. In order to realize braking function, the dynamic and static friction discs should compact each other to supply friction moment, which slows the output shaft and eventually make it stop rotation. Therefore, whether the friction disc can be compact is the necessary condition for braking function. So the matching
FIGURE 4. Movement analysis before and after motor closing.

FIGURE 5. Geometric matching relation of braking condition.

relation of the geometric parameters is analyzed, as shown in Fig.5.

It is obviously that when middle shaft and dynamic discs move from left to the right, if friction discs are compacted first, the basic condition of braking is achieved. Otherwise, if steel balls between the middle shaft and the input shaft are compacted first, the basic braking condition will lose.

In order to facilitate the analysis, it is assumed that the steel balls between the middle shaft and the input shaft are compacted completely. The penetration depth $H_0$ of the dynamic and static friction discs can reflect the brake condition. Positive value of $H_0$ represents that they can be compacted, while negative value of $H_0$ represents that they cannot compact each other.

According to the design parameters, the max penetration depth of friction discs is $H_0 = 0.189\text{mm}$. But many factors will lead to the change of contact depth, including: 1) axial clearance of the bearing, on which input shaft is assembled; 2) thickness of the friction disc, which is affected by machining error and wear effect, 3) diameter of the steel ball, which is also affected by machining error and wear effect. Further through the geometric relationship along the axis, it can be seen that wear of the friction discs can negatively influence the braking condition, while axial clearance of the bear can positively influence the braking condition.

From the above analysis, penetration depth $H$ can be expressed by Eq.1, and the braking failure criterion is $H < = 0$, which means dynamic disc and static disc cannot compact each other.

$$H = H_0 + C - W$$ (1)

where, $H_0$ is the penetrable depth of ideal case (no manu- factural errors and clearance); $C$ is the axial clearance of the input shaft bearing; $W$ is the total wear depth of friction discs.

C. MECHANICAL PARAMETERS MATCHING RELATION ANALYSIS

1) ANALYSIS OF MECHANICAL PARAMETER MATCHING RELATION

The force analysis of the braking condition is shown in Fig.6. After the motor is turned off, the input shaft is released and the load torque acted on output shaft is $T$, the force that corrugated spring acted on the middle shaft is $F_s$, which is directed to the right. In addition, there are four parameters affect performance of the UTM, including: the cone angle of output shaft ($\beta$), the distance between cone axis...
to intermediate shaft \((r)\), the equivalent friction radius \((R)\), dynamic friction coefficient between the friction disc \((\mu)\).

In practice, the speed of the mechanism is slow, the speed of the process is basically constant, and the influence of inertia force is small, so the influence of inertia force can be ignored in the analysis.

First, take the output shaft as the object of study—the forces shown in Fig. 7(a), they are the torque \(T\) and the contact forces of the three steel balls. In order to make the analysis convenient, the contact force is decomposed into three orthogonal components, including: (1) The axial force along the shaft axis, names \(F_{T}\); (2) The force along the perpendicular point to the vertical direction of the middle shaft axis, names \(F_{r}\); (3) The tangent direction of \(F_{\rho}\). Among them, three \(F_{T}\) overcome the centrifugal force of steel balls, the resultant force is 0; three \(F_{\rho}\) and load torque \(T\) are mutually balanced; the resultant force of the 3 \(F_{T}\) pushes the middle shaft to the right to provide positive pressure for the friction disc.

According to the equilibrium condition of the output shaft:

\[
3F_{\rho} \cdot r = T
\]

Thus, \(F_{\rho}\) can be obtained, namely:

\[
F_{\rho} = \frac{T}{3r}
\]

Then contact force on the cone surface is analyzed from the view of B-B. As shown in Fig. 7(b), it is obviously that:

\[
F_{T} = F_{\rho} \cot \beta = \frac{T}{3r} \cot \beta
\]

Finally, the middle shaft is taken as the object of study. Here, the dynamic friction disc and the intermediate shaft are considered as a whole, and the force analysis from the view of C-C is shown in Fig. 7(c): The middle shaft is subjected to contact force between three steel balls \((F_n)\), corrugated spring force \((F_{S})\), and friction force between the static and dynamic friction discs \((F_f)\). Among them, the contact force to the middle shaft and that to the output shaft are equal in size but opposite in direction. So they are represented by \(F_{\rho}', F_{T}'\) and \(F_r\) respectively.

In order to obtain balance condition of the middle shaft, equivalent radius of friction disc is analyzed, According to the Coulomb friction model, the friction torque of the friction disc can be expressed as

\[
M = \int_{R_{in}}^{R_{out}} \frac{F_n}{\pi} \frac{R^2 - R_{out}^2}{R_{out}^2 - R_{in}^2} 2\pi x^2 dx
\]

\[
= \frac{2}{3} \left( \frac{R_{out}^3 - R_{in}^3}{R_{out}^2 - R_{in}^2} \right) uF_n = uF_n R
\]

So the equivalent radius of friction disc can be expressed as

\[
R = \frac{2}{3} \left( \frac{R_{out}^3 - R_{in}^3}{R_{out}^2 - R_{in}^2} \right)
\]

According to the balance condition of the middle shaft names

\[
3F_{\rho}'r = u \cdot (F_{s} + 3F_{T}') R \cdot n
\]

Thus, the friction coefficient can be represented as follows:

\[
u = \frac{3F_{\rho}'r}{(F_{s} + 3F_{T}') R \cdot n} = \frac{3F_{\rho}'r}{(F_{s} + 3\frac{L}{3r} \cot \beta) R \cdot n}
\]

\[
= \frac{u}{(F_{s} + \frac{L}{3r} \cot \beta) R \cdot n}
\]

Thus,

\[
T = \frac{u \cdot R \cdot n \cdot F_{s}}{1 - uRn \cot \beta}
\]

Therefore, the braking failure condition is:

\[
3F_{\rho}'r > u \cdot (F_{s} + 3F_{T}') R \cdot n
\]

Names:

\[
u < \frac{T}{(F_{s} + \frac{L}{3r} \cot \beta) R \cdot n}
\]

where:

- \(T\): Load torque;
- \(u\): Friction coefficient between friction discs;
- \(n\): Number of contact surfaces of friction discs;
- \(R\): Equivalent radius of friction disc;
- \(r\): Distance between the contact points of the steel ball and the rotating shaft;
- \(\beta\): Cone angle at Output;

It can be seen that each load torque \(T\) corresponds to a minimum braking coefficient of friction \(u_T\), and when the actual friction coefficient is smaller than the minimum coefficient of friction to be braked, the UTM will lose the braking function.

2) RELATION BETWEEN THE LOAD OF UTM AND THE ANGLE OF HATCH

The configuration of the hatch mechanism is shown in Fig. 8(a), in which the solid line is in the closed position, the short dotted line is in the middle position, and the long dotted line is in the open position. It can be seen that the hatch...
mechanism is composed of the hatch, BC rod and CD rod. The essence of the mechanism is a four-linkage mechanism named ABCD. However, unlike the conventional four-bar mechanism, the original power of the mechanism comes from the UTM, which is installed at the movable point B, while the original power of conventional four bar linkage comes from fixed point A or D.

In order to obtain the relationship between the load of UTM and the hatch angle, the middle position is selected, and the geometry and force are analyzed. As shown in the Fig.8(b), the angle between door closed position AE and gravity is \( \theta_0 \), The angle between the AD and the gravity direction is \( \theta_1 \), The angle of the structure at the hatch shaft \( B'AE' = \gamma \), \( B'DAB' = \theta_A \), \( B'DC'B' = \theta_C \), \( \angle B'C'A = \theta_B \), where \( \theta_0, \theta_1, \gamma \) are consistent, and \( \theta_B, \theta_C \) and \( \theta_D \) are changed with \( \theta_A \).

Thus, the opening angle of the hatch door can be expressed as:

\[
\theta = \theta_A + \theta_1 - \gamma - \theta_0
\]  

According to range of the hatch angle \( \theta_{\min} < \theta < \theta_{\max} \), range of \( \theta_A \) can be obtained:

\[
\theta_{\min} + \gamma + \theta_0 - \theta_1 < \theta_A = \theta + \gamma + \theta_0 - \theta_1 < \theta_{\max} + \gamma + \theta_0 - \theta_1
\]  

Firstly, the C’D rod is chosen as the object of study. According to the mechanical principle, C ’D is a two force rod, and the load at both ends of the \( F_{12} \) and \( F_{42} \) are along the direction of the rod;

Then, take the B’C’ rod as the object of study, it receives the reaction force of C’D on C’ point, named \( F_{21} \), and receives the torque of T from UTM at B’ point. According to the equilibrium conditions of B’C’, The force \( F_{31} \) at B’ point should form a couple with \( F_{21} \), so the direction is contrary to \( F_{21} \), and the magnitude is

\[
F_{31} = \frac{T}{l_1 \times \sin(\theta_C)}
\]

Finally, the cabin is taken as the object of study. Its reaction force \( F_{13} \) and torque T at the B point by B ’C’ is subjected to
the gravity $G$ at the E point, and the counterforce is received at A.

The equilibrium condition of the hatch around the point A is:

$$T' + F_{13} \times s_1 - G \times s_2 = 0$$  \hspace{1cm} (12)

Thus, UTM load is obtained:

$$T = F_{13} \times s_1 - G \times s_2$$  \hspace{1cm} (13)

where,

$$s_1 = l_3 \times \sin(\theta_B - (180^\circ - \theta_C))$$  \hspace{1cm} (14)

$$s_2 = l_4 \times \sin(\theta_A + \theta_1 - \gamma)$$  \hspace{1cm} (15)

From Eqs. (9)∼(15), it can be seen that the key to solving the load is to calculate the $\theta_B$, $\theta_C$ and $\theta_D$ at different $\theta_A$. The method can be find in our previous study [12] and some other research of linkage mechanism [13]–[15].

The relationship between the load of the UTM and the angle of the hatch is shown in Fig.9, which shows that the load of UTM increases first and then decreases with the opening angle of the hatch, and when the hatch opens to 65 degrees, the load of UTM reaches its maximum value, and the magnitude is 12.69Nm.

![FIGURE 9. Relation between opening angle and the load.](image)

3) MINIMUM BRAKING FRICTION COEFFICIENT ANALYSIS

The design value of each parameter is brought into (6), and the minimum friction coefficient for brake is obtained:

$$u_T = \frac{T}{(F_s + \frac{T}{7} \cot \beta)R \cdot n} = \frac{12.96}{(240 + \frac{12.96}{0.02} \cot (55^\circ)) \times 0.025 \times 6} = 0.11$$

According to the results, the minimum friction coefficient for brake is 0.11, while the dry friction coefficient of the common metal materials is among 0.1 to 0.3, which is close to the friction coefficient of the common metal materials and the risk relatively high. Unfortunately, $F_s$ will decrease because of the stress relaxation effect, and it results in the increasing of $u_T$, which making the risk increasing. In order to enlarge margin of safety, the minimum friction coefficient for braking should be as small as possible. In design stage, it can be realized by adjusting $F_s$, $r$, $n$ or $\beta$. From (6), it can be seen that the critical friction coefficient can be reduced by increasing the spring preload, the equivalent radius of friction discs, the number of friction discs or reducing the distance between steel ball and rotating shaft, and the taper angle of the output end, which can fundamentally improve the reliability of the UTM. And in maintenance stage, it can be realized by adjusting $F_s$.

D. FAILURE MECHANISM ANALYSIS

According to the above analysis, the influence factors of the UTM can be obtained as shown in Table.1. Among which, three factors have obvious degradation characteristics: thickness of the friction disc and friction coefficient between discs degrades with the motion cycle because of wear, stiffness coefficient of corrugated spring degrades with the calendar time because of the stress relaxation effect.

![FIGURE 10. Braking performance analysis during opening process.](image)

As Fig.10 shows, maximum torque of the UTM is $T_{\text{max}}$, the corresponding minimum friction coefficient is $u_{T,\text{max}}$. Assuming that the initial friction coefficient is $u_0$, and $u_0 > u_{T,\text{max}}$, then the UTM can brake success at any opening angle.

According to the influence factors analysis, there are three degradation factors, when wear depth exceeds the maximum wear depth for brake, the UTM will lose the brake function. Wear also make the friction coefficient decreased from $u_0$ to $u_C$, while the decrease of the spring force $F_s$ will lead to an increase of minimum braking friction coefficient $u_{T,\text{max}}$. These two factors result in the degradation of the braking function. Even worse, when $u_C$ is smaller than $u_{T,\text{max}}$, the UTM braking failure occurs in the range of $\theta_L$ and $\theta_U$.

In general, when components do not fail, the performance degradation or function loss of mechanical products is usually caused by the mismatch of geometric parameters or mechanical parameters. Therefore, it is an effective method for failure analysis of mechanical products through the matching relationship of geometric parameters and mechanical parameters. The failure analysis method can be used not only in UTM, but also in other similar mechanisms. Furthermore, it has been applied to several mechanical products and has been proved to be effective.
TABLE 1. Influence factors of the UTM.

<table>
<thead>
<tr>
<th>No.</th>
<th>Main Influence factors</th>
<th>Degradation (Y/N)</th>
<th>Degradation reason</th>
<th>Degradation Influence factors</th>
<th>Degradation Trend</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Thickness of the friction disc/h</td>
<td>Y</td>
<td>wear</td>
<td>Motion cycle /N</td>
<td>↓</td>
</tr>
<tr>
<td>2</td>
<td>Friction coefficient between discs/f</td>
<td>Y</td>
<td>wear</td>
<td>Motion cycle /N</td>
<td>↓</td>
</tr>
<tr>
<td>3</td>
<td>Equivalent radius of the disc/R</td>
<td>N</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>stiffness coefficient of corrugated spring /F₂</td>
<td>Y</td>
<td>stress relaxation</td>
<td>Calendar time / t</td>
<td>↓</td>
</tr>
<tr>
<td>5</td>
<td>cone angular of output /β</td>
<td>N</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>clearance of the out bearing/C</td>
<td>N</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>radius of the steel ball/D</td>
<td>N</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Number of the friction disc/ n</td>
<td>N</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

IV. SIMULATION MODEL ESTABLISHMENT

A. SIMULATION MODEL OF THE UTM

For complex mechanisms, establishing and solving their dynamical equations are quite difficult or even impossible, especially when too many contacts exist. So, through applying multi-body dynamic simulation platform to simulate the dynamic response of mechanism becomes more and more popular. In this paper, LMS Virtual.Lab is adopted. The theoretical background of multi-body simulation can be found in [16]–[19].

The UTM model is established according to the failure mechanism, as shown in Fig.11. The color boxes represent components of the UTM, the lines describe constrains between components and the elliptical boxes represent influencing factors, which are parameterized in the model.

B. SIMULATION MODEL OF HATCH SYSTEM

The simulation model is established according to the test principle in Section 2. In the model, hatch, four-bar linkage and UTM are identical with the test. The hatch rotates around the fixed point A, CD rod rotates around the fixed point D. Here, length of BC rod is 310 mm and length of CD is 308 mm, the mass of the hatch is 142 kilograms and the center of gravity lies at point E.

The reducer is represented by a simplified model, which consists of two gear pairs and the transmission ratio is consistent with the actual reducer ($i = i_1 \times i_2 = 106$).

V. FAILURE MECHANISM VERIFICATION AND MAINTENANCE SUGGESTIONS

A. GEOMETRIC PARAMETER MATCHING RELATION VERIFICATION

Comparison of the braking function in both forward load condition and reverse load condition is shown in Fig.13(a). It can be seen that, in the forward load condition, the output shaft can brake immediately when motor shuts off, while in the reverse load condition, the output shaft rotates a certain angle before stopped. The simulation result is consistent with the individual test, which proved the correctness of the simulation model and the difference analysis of the two conditions.

Then, take four different values to wear depth, and compare the braking performance in reverse load condition. As shown in Fig.13(b), when the wear depth is less than 0.3 mm,
the UTM can brake at the moment motor closed. When the wear depth reaches 0.4 mm, the braking capacity will be lost, as the Fig.13(b) shows the output shaft has a serious reverse rotation. The result coincides with the phenomenon of the individual test, in which the brake capability lost when wear depth reaches 0.35 mm. Thus, correctness of the geometric parameter matching relation is verified.

B. MECHANICAL PARAMETERS MATCHING RELATIONSHIP VERIFICATION

According to the Load-Angle relation, the maximum torque of the UTM is at the angle of 65°, the corresponding open time is 38.5s. So, take three values to friction coefficient, and shut off the motor at 38.5s, then compare the brake capability.

As Fig.14 shows, the hatch can brake normally at the maximum torque position when friction coefficient is 0.2, but it fell from 65.4° to 29.3° when friction coefficient decreased to 0.1; even worse, the door fell from 65.4 ° to 23.7° when friction coefficient decreased to 0.09. The above results are in good agreement with the experimental results, and the validity of the mechanical matching relation is verified.

C. MAINTENANCE SUGGESTIONS

According to the failure mechanism, when performance of the UTM degrades and failure occurs, several actions can be used to recovery its performance. As shown in Figure.15:

1) PERFORMANCE RECOVERING BY ADJUSTING CLEARANCE

If the UTM failure occurs in a wide angle range, which means the performance has deteriorated seriously. In order to recover the performance, several components should be replaced and some clearance needs to be adjusted, including,

- Replacing friction discs
- Replacing gasket by a thicker one
- Increase the axial clearance of input shaft
2) PERFORMANCE RECOVERING BY ADJUSTING SPRING PRELOAD

If the UTM failure occurs in a small angle range, there is no need to replace components. The performance can be recovered by adjusting the preload of the spring. The detail steps are illustrated as follows:

Step 1: Calculate the maximum load that can be overcome at present according to the Fig.9.

Step 2: According to the (6) to calculate the current friction coefficient:

$$h_0 = \frac{T_\theta}{\left( F_s + \frac{T_\theta}{r} \cot \beta \right) \cdot R \cdot n} \quad (16)$$

Step 3: The preload required at present can be obtained according to Eq.16:

$$F_s' = k \cdot \left( \frac{T_{\text{max}}}{h_0 \cdot R \cdot n} - \frac{T_{\text{max}}}{r} \cot \beta \right) \quad (17)$$

where, $k$ is the safety factor.

Step 4: The adjusting length of spring compression calculation:

$$\Delta l = \frac{\Delta F_s}{K} = \frac{F_s' - F_s}{K} \quad (18)$$

where, $K$ is the elastic coefficient of corrugated spring.

As shown in Fig.14, when friction coefficient decreased to 0.09, the door drops from 65° to 29.3°. According to (16) and (17), in order to restore braking function, preload of spring should bigger than 507 N, so according to (18), increase the initial compression of spring by 1.9mm. Comparison of the brake performance before and after adjusting is shown in Fig.16. It can be seen that the maintenance method is effective.

VI. CONCLUSION

Failure mechanism of UTM is deeply analyzed in the study, which is verified by the simulation model. The following conclusions can be drawn from the present study.

1) It is an effective method for failure analysis of mechanical products through the matching relationship of geometric parameters and mechanical parameters. It can be used in UTM and other similar mechanisms.

2) The wear of friction discs and the stress relaxation of springs will lead to the performance degradation of UTM. The UTM failure in this paper is mainly caused by the wear of friction discs.

3) From the design point of view, by increasing $R$, $N$ and decreasing $\beta$ and $r$, the critical friction system can be reduced and the braking reliability of UTM can be improved.

REFERENCES


HUAN PANG received the master’s degree and the Ph.D. degree in aircraft design from Northwest Polytechnic University, in 2013 and 2016, respectively. He is currently a Lecturer with the School of Automobile, Chang’an University. His current research interests include dynamics simulation and reliability analysis of multi-body systems, reliability-based design of the multi-body systems, and automotive reliability analysis and maintenance management based on warranty data.

NING WANG received the M.S. and Ph.D. degrees in industrial engineering from Northwest Polytechnic University, in 2007 and 2012, respectively. He jointly trained at The University of Texas at San Antonio, in 2010, and was a Visiting Scholar with Texas State University. He is currently an Associate Professor of logistics engineering with the Automobile College, Chang’an University. His current research interests include equipment health management, maintenance decision-making methods and applications, and enterprise informatization.

MENG LI received the B.S. degree from the Department of Mechanical Engineering, Zhejiang University, in 2007, and the M.S. and Ph.D. degrees from the Department of Chemical Engineering, Xi’an Jiaotong University, in 2010 and 2016, respectively. He is currently a Lecturer with Chang’an University. His research interests include ultrasound nondestructive testing, lubrication state detection, and vibration monitoring and failure diagnosis of mechanical equipments.

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