Fatigue Life Analysis for Electronic Automatic Gearshift of Vehicles

XIANG ZHANG, PENG MENG, WEI YANG and JIAQIN ZHANG

ABSTRACT

This paper is studied a type of electronic automatic gearshifts made of glass-fiber reinforced polypropylene. Finite element model was established in order to implement the nonlinear static strength analysis, which can determine the von Mises stress and the weak region of the gearshift. The strain test result confirmed the validity of the finite element analysis model. The relevant factors for mechanical performances of nonlinear material were determined through the static tensile test and fatigue loading test. Fatigue life analysis was conducted using the nominal stress method. It has been found that gearshift life satisfied design requirement. In addition, the methodology applied plays a significant role to the design theory of fatigue analysis for nonlinear material and gearshift life

INTRODUCTION

Electronic automatic gearshift has widely been used in automobile areas as it is convenient, safe, and comfortable, and has excellent anti-theft feature. However, fatigue is appeared to be the major failure form for the automatic gearshift [1]. Scientists have recently performed numerous studies on the fatigue life of automotive transmission system. Considering the stress concentration coefficient and load magnification factor, Ding [2] employed Goodman method to modify the stress in predicting the fatigue life of transmission gear pair. On the basis of revolution and working time for each gear, the equivalent load of transmission bearings was calculated and the fatigue life was predicted [3]. Repetto [4] generated the stress-time history using rain-flow counting method. Then, the cycle counting

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procedure was applied to estimate the stress in the fatigue analysis of the front axles in vehicles.

There are few studies on electronic gearshift for tiptronic compared to conventional mechanical automatic gearshift [5]. In addition, most studies focused on the linear material including the optimization for linear structure, the shifting curve and the dynamic simulation. However, the influence of nonlinear mechanical material properties on the fatigue life of electronic automatic gearshift has been rarely considered. Therefore, the aim of this study is to analyze nonlinear static strength for a certain type of electronic automatic gearshift. In addition, constitutive relationship and fatigue life of nonlinear material are analyzed based on the experimental results of nonlinear material properties. The methodologies developed in this work have both important theoretical significance and practical values which guarantee the driving safety, guide the designing and developing of automatic gearshift, and improve the domestic design condition of automatic gearshift.

1 NONLINEAR STATIC STRENGTH ANALYSIS OF AUTOMATIC GEARSHIFT

1.1 Theories

The section headings are in boldface capital and lowercase letters. Second level headings are typed as part of the succeeding paragraph (like the subsection heading of this paragraph). During loading process of nonlinear material, both different loads and variation of circumstances in structure cause nonlinear stress-strain relationship. In addition to constitutive relation, elastic rate constitutive equation for nonlinear material on the basis of linear elastic constitutive equation is shown in equation (1).

\[
\sigma = D_{\text{ep}} \varepsilon
\]  

Where \(\varepsilon\) is strain; \(\sigma\) is stress; \(D_{\text{ep}}\) is elastic-plastic matrix which can be expressed in equation (2)

\[
D_{\text{ep}} = D_e - D_e \frac{\partial g}{\partial \sigma} \left( \frac{\partial f}{\partial \sigma} \right)^T D_e / \left[ A + \left( \frac{\partial f}{\partial \sigma} \right)^T D_e \frac{\partial g}{\partial \sigma} \right] \]  

(2)
Where $D_e$ is the elastic matrix; $g, f$ are the plastic and yield function, respectively. $A$ is hardening function.

Finite element equation for each unit, was connected to obtain finite element equation matrix for structure, are shown in equation (3).

$$
\begin{bmatrix}
\sigma_1, \sigma_2, \sigma_3 \ldots
\end{bmatrix}^T = \begin{bmatrix}
D_{ep1}, D_{ep2}, D_{ep3} \ldots
\end{bmatrix}^T \begin{bmatrix}
\varepsilon_1, \varepsilon_2, \varepsilon_3 \ldots
\end{bmatrix} \tag{3}
$$

Newton-Raphson method is applied to solve equation (3) and the procedure is shown in figure 1 [6]. Equation (3) is updated to equation (4) as $[\sigma_1, \sigma_2, \sigma_3 \ldots]^T$ and $[D_{ep1}, D_{ep2}, D_{ep3} \ldots]^T$ are substituted by $[\sigma]$ and $[D_{ep}]$ respectively.

$$\varphi(\varepsilon) = [D_{ep}]\varepsilon - [\sigma] \tag{4}$$

According to Equation 1, $[\sigma]$ is a differentiable continuous function of $\varepsilon$, so $\varphi(\varepsilon)$ has its first derivative. $\varepsilon^{(0)}$ is initial guess, $\varepsilon^{(n)}$ is the iterative value at the end of nth iteration. Equation (5) shows the Taylor series expansion of $\varphi(\varepsilon)$ in $\varepsilon^{(n)}$ by truncating the series at the second derivative.

$$\varphi(\varepsilon^{(n)}) + \left[D_{ep}^{(n)}\right] \varepsilon^{(n+1)} - \varepsilon^{(n)} = 0 \tag{5}$$

The iterative value, $\varepsilon^{(n+1)}$, at the end of (n + 1)th iteration is given in equation (6).

$$\varepsilon^{(n+1)} = \varepsilon^{(n)} - \left([D_{ep}^{(n)}]\right)^{-1} \varphi(\varepsilon^{(n)}) \tag{6}$$

Where $\left[D_{ep}^{(n)}\right] = \frac{\partial \varphi}{\partial \varepsilon} |_{\varepsilon^{(n)}}$ is the tangent stiffness matrix for structure comprising of the corresponding unit tangent stiffness matrix. The convergence procedure of N-R was regulated by comparing the results for ith and (i+1)th iteration.

N-R procedures are:
(1) Assuming n=0, so the initial $\varepsilon^{(0)}$ was set.
(2) The tangent stiffness matrix is calculated.
Figure 1. Geometrical illusion of the Newton-Raphson method.

\[
[D_{ep}]_{T}^{(n)} = \frac{\partial \varphi}{\partial \varepsilon} \bigg|_{\varepsilon = \varepsilon^{(n)}}
\]

(3) The unbalanced value is calculated.

\[\varphi^{(n)} = \varphi(\varepsilon^{(n)}) = [D_{ep}]^{(n)} \varepsilon^{(n)} - [\sigma]\]

(4) The equations are solved.

\[[D_{ep}]_{T}^{(n)} \Delta \varepsilon^{(n)} = -\varphi^{(n)}\]

(5) The approximation value at the end of \((n + 1)\)th iteration is calculated.

\[\varepsilon^{(n+1)} = \varepsilon^{(n)} + \Delta \varepsilon^{(n)}\]

(6) The errors and convergence are achieved if it is under tolerance, otherwise continue the iteration with \(n = n + 1\).

1.2 Static Strength Analysis

Electronic control board unit and circuit for transferring signal were simplified in the finite element model to reduce the high complexity of the gearshift assembly unit. Further, gearshift arm and rest of the structures were separately meshed. Gearshift arm was meshed into 8 nodes solid element with 31188 meshes and 35293 nodes in total. On the other hand, gearshift box and base were meshed in 4 nodes with 129449 meshes and 133876 nodes in total and 2mm element thickness. The maximal gear shifting force, \(F_{\text{max}}\), was about 15N measured on a test platform when the gear shifted from Park (P) to Reverse (R) and R to Neutral (N). The maximum compressive force, \(F_{\text{max}}\), for inner springs was about 20N. In the FE
model, \( F_{\text{ymax}} \) and \( F_{\text{ymin}} \) were loaded to the gearshift arm and inner springs, respectively. Figure 2 shows the von Mises stress distribution from the nonlinear static strength analysis of the gearshift using FEM model. The maximal von Mises stress for gearshift arm and gear shift shell box was 11.2MPa and 1.79 MPa, respectively. Weak regions in gear shift arm were observed around the middle part of arm and the connecting parts between arm and shell. However, weak regions in gear shift shell were observed near the connections between shift arm and box between shell box and ground, and between shell components.

2 STATIC STRENGTH ANALYSIS

For confirming the accuracy of results obtained through nonlinear static strength analysis, the strain test was conducted on the platform for testing automatic gearshift, as shown in figure 3. The regions and parts sensitive to the signals were chosen as measure points. Figure 3(a) shows the locations of measure points. Straingauges were mounted to the measure points located on the surface of gear shift (Figure. 3(b)). The test equipment consists of C307 test platform for automatic gearshift, KYOWA resistance strain gauge and DRA-30A digital strain instrument.

During the strain test, automatic gearshift was mounted on the platform and the control system designed for simulating the automatic gearshift. Under the real gear shifting conditions, gear shifting process was finished. The strain of each measuring point over time was recorded using digital strain instrument and saved in the computer. Figure 4 shows the real measurement data for three measure points. A comparison between theoretical and experimental result is shown in table 1.

(a) Von Mises stress distribution for shift arm  (b)Von Mises stress distribution for transmission box

Figure 2. Stress distribution for automatic gearshift for vehicles.
(a) Locations of measure points  
(b) Layout of Strain gauges  
(c) Strain test platform for the gearshift

Figure 3. Experiment setup details.

Figure 4. Strain data for 3 measure points.

<table>
<thead>
<tr>
<th>Measure point 1 (MPa)</th>
<th>Measure point 2 (MPa)</th>
<th>Measure point 3 (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experimental results</td>
<td>11.01</td>
<td>1.68</td>
</tr>
<tr>
<td>Theoretical results</td>
<td>11.21</td>
<td>1.61</td>
</tr>
<tr>
<td>Error (%)</td>
<td>1.8</td>
<td>4.3</td>
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</tbody>
</table>

TABLE 1. EXPERIMENT DATA AND THEORETICAL.

3 GEARSHIFT FATIGUE LIFE PREDICTION

3.1 Material Constitutive Relation

Static tensile test was conducted to obtain mechanical material properties, such as Young’s modulus, Poisson ratio and ultimate yield strength (UTS) of glass fiber-reinforced PP40. Experimental procedure followed GB/T 1447-2005
standard and the samples were flat tensile bars. Figure 5(a) shows the experiment setup for tensile test where external force was used on samples using a hydraulic unit of 10KN universal tensile test machine. Variation in stress and displacement on the two sides of samples were measured. Figure 5(b) shows fracture tensile samples after the test.

True strain, $\varepsilon$, is defined as the logarithm of the ratio of transient displacement, $l$, and original displacement, $l_o$. The volume changes caused by elasticity can be neglected when the plastic deformation is significantly large.

$$l = l_o + \varepsilon l_o$$  \hspace{1cm} (7)

$$Al = A_o l_o$$  \hspace{1cm} (8)

Where $\varepsilon$ is the engineering strain, $A$ and $A_o$ are the original and the cross section area after tensile test, respectively.

The true strain, $\varepsilon$ and true stress, $\sigma_2$, can be described in equation (9) and (10), respectively.

$$\varepsilon = \int_{l_o}^{l} \frac{1}{l} dl = \ln \frac{l}{l_o} = \ln(1 + \varepsilon)$$  \hspace{1cm} (9)

$$\sigma_2 = \frac{F}{A_o} \cdot \frac{A}{A_o} = \sigma_1 \left( \frac{l}{l_o} \right) = \sigma_1 (1 + \varepsilon)$$ \hspace{1cm} (10)

Where, $\sigma_1$ is the engineering stress.

True stress-strain curves calculated using the above equations are shown in Figure 6. The peak value of the curves is considered to be the ultimate tensile strength of the material. Mechanical properties calculated from the true strain-stress curves are shown in Table 2.
Figure 5. Machine setup of tensile testing and fracture samples.

Figure 6. Strain-stress curves for PP40.

Table 2. Mechanical Properties for PP40.

<table>
<thead>
<tr>
<th>Material</th>
<th>Young’s Modulus (MPa)</th>
<th>Poisson Ratio</th>
<th>Yield Stress (MPa)</th>
<th>Ultimate Tensile Strength (MPa)</th>
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<tr>
<td></td>
<td>2000</td>
<td>0.36</td>
<td>33</td>
<td>35</td>
</tr>
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</table>
3.2 Fatigue Loading Test

An S-N curve for PP40 was obtained from the fatigue test performed by a 10KN universal testing machine. Pull-pull fatigue test method was used and R was chosen to be 0.1 as glass fiber-reinforced PP40 material has high strength and excellent impact-resistance. Figure 7(a) is the “dumb bell” design for fatigue test sample and (b) shows the PP40 samples after fatigue failure.

Figure 8 shows the S-N curve of PP40 generated using least square method based on the combination of average stress-cycle numbers from fatigue test and empirical data. The PP40 logarithm S-N curve is plotted in figure 8 as it is expressed in equation (11).

\[ S = 60 - 5.66 \log N \]  

(11)

![Figure 8. S-N curve of PP40.](image)

![Figure 9. Fatigue life analyses.](image)

(a) Life value distribution  
(b) Logarithm distribution of life value
3.3 Fatigue Damage Analysis

On the basis of nonlinear static mechanical analysis, the nominal stress method is used to predict the fatigue life of gearshift employing the linear fatigue damage theory [7].

The results are shown in figure 9 indicate that fatigue damage tends to occur near the connections between gearshift arm and shell, and all the bolt-holes. The fatigue damage reaches its threshold value after 1.48E+6 cycles under the maximum load. From the statistic archives, it is estimated that several hundreds of thousands of gear shifting [8] occur during the recommended number of years for a vehicle driving on the city road. Our simulation and experiment results illustrates, the number of statistical actual shifting times is less than the calculated fatigue life limit indicating that the gearshift satisfies the design requirements.

4 CONCLUSIONS

On the basis of nonlinear static strength analysis for automatic gearshift made of PP40 material, the fatigue life study was conducted and conclusions can be drawn as follows:

1) Nonlinear static strength analysis was simulated successfully after constructing the finite element model for the gearshift. Results indicate that the weak regions positioned at the connection between the middle part of gearshift arm and the shell.

2) Young’s modulus, yield strength, ultimate tensile strength, Poisson ratio and S-N curve for PP40 were determined from the static tensile and fatigue loading test.

3) Fatigue life was determined using the nominal stress method for gearshift and results indicated that fatigue damage reached its limitation after 1.48E+6 cycles under the maximum load. It satisfies the design requirement.

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