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Fatigue life evaluation of mechanical components using vibration fatigue analysis technique †

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Abstract

Unit brackets attached on a cross member and subjected to random loads often fail due to self-vibration. To prevent such failures, it is necessary to understand the fatigue failure mode and to evaluate the fatigue life using test or analysis techniques. The objective of this study is to develop test specifications for components, which are applicable to predict fatigue life at the stage of initial product design, for the unit brackets by using a vibration fatigue technique. For this objective, the necessity of a fatigue analysis considering resonant effect was reviewed. Also, a series of vibration fatigue analyses were carried out by changing the acceleration's direction and magnitude. Then, a methodology was proposed to determine the optimum vibration fatigue test specification of the component, which gives an equivalent failure mode with the vehicle test condition.

Keywords: CAE; Damage; Test Specification; Vibration Fatigue

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1. Introduction

Fatigue is the localized structural damage that occurs when a material is subjected to cyclic loading, and fatigue fracture is one of the most frequent failure mechanisms in daily life or industry. In the automotive industry, it is necessary to evaluate fatigue life and to design components with guaranteed durability at the stage of initial product design. Automotive structural engineers often validate the durability of components with component fatigue tests or driving fatigue tests, but the use of these evaluation methods may make a timely review of the durability difficult due to constraints of time and money. By performing virtual fatigue analysis using finite element analysis, the time and money consumed at the stage of development of new vehicles can be decreased. Because of this benefit, both fatigue tests and virtual fatigue analyses are widely used in industry [1].

Fatigue analysis can be generally categorized into quasi-static fatigue analysis and resonant fatigue analysis. Quasi-static fatigue analysis is appropriate to evaluate the durability of relatively stiff systems such as suspension arms, and resonant fatigue analysis is suitable to consider the resonant effect in cases in which the frequency of excitation loads includes the natural frequency of the system. Load time history data are generally used for fatigue analysis, but they are not available to consider the massive amount of loads or loading cases. Also, it is impossible to perform fatigue analysis when loading conditions are presented in the form of PSD (Power Spectral Density) defined in a frequency domain. In this case, structural response of systems can be computed by using transfer function of target systems and PSD of excitation loads [2-6].

Meanwhile, unit brackets used to suspend some sensors in an arbitrary position are attached on a vehicle frame in a form of the cantilever. The acceleration load input at the supported position makes the brackets vibrate, and then the brackets often fail during driving tests in cases in which the excitation frequency includes the natural frequency of the brackets. To prevent these failures, various types of vibration fatigue test specifications for components specified in ASTM, ISO, KS, JIS and so forth are utilized, but those don' t reflect the characteristics of driving test roads or field. Also, previous researches [7-9] have already attempted to establish vibration fatigue specifications, but these studies were confined to only uniaxial test condition because testing structures at three axes simultaneously is not feasible with testing hardware used [10].

The objective of this study is to develop the vibration fatigue test specification for the component, which is equivalent to the vehicle test condition and applicable to predict fatigue life at the stage of initial product design, for unit brackets by using a vibration fatigue analysis technique. For this objective,

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the necessity of a fatigue analysis considering resonant effect was reviewed. Also, a series of vibration fatigue analyses were carried out by changing acceleration, direction and magnitude. Then, a methodology to determine the optimum vibration fatigue test specification of the component, which gives an equivalent failure mode with the driving test condition, was proposed.

2. Quasi-static durability analysis

In order to evaluate the fatigue life of structures, it is necessary to calculate the accurate stress or strain state. Usually, stress analysis can be categorized into static response analysis and transient response analysis according to types of loads.

The purpose of a transient response analysis is to compute the behavior of a structure subjected to time-varying excitation, and structural response is computed by solving a set of coupled equation written as follows:

$$[M]\ddot{x}(t)+[B]\dot{x}(t)+[K]x(t)=f(t) \tag{1}$$

where $[M]$, $[B]$ and $[K]$ are mass, damping and stiffness matrices, respectively, and $x(t)$ and $f(t)$ are displacement and load vectors, respectively. When static load is applied, the stiffness matrix is only considered without term of velocity and acceleration. Quasi-static fatigue analysis is a fatigue life estimation method using superposition of static stress obtained by finite element analysis and load time history data, and it can be used in cases in which the dynamic effect is negligible.

Fig. 1 [11] is a schematic diagram of a cantilever beam structure composed of beam elements with a thickness of 5mm and two concentrated masses, and the 1st - 3rd natural frequencies of the structure are 5.4, 25.1 and 81.7 Hz, respectively. Half-sine waves of 1.0 and 25.1 Hz were input at the right end of the structure to calculate the structural behavior so that the dynamic effect could be observed. Fig. 2 represents the first principle stress in the element, 400 mm from the supported position. Fig. 2(a) shows that quasi-static analysis and transient response analysis results showed a good agreement in stress level under the 1.0 Hz half-sine wave. Fig. 2(b) is the analysis result calculated under the half-sine wave excitation of 2nd natural frequency (25.1 Hz) of the structure, and the results showed a big discrepancy with the quasi-static analysis result. Therefore, in cases in which the frequency component of excitation loads includes the natural frequency of the systems, the dynamic effect should be considered to evaluate accurate structural behavior. The fatigue analysis method for considering dynamic effect is briefly described in the following chapter.

In order to carry out quasi-static fatigue analysis, the stress or strain is calculated under the unit load, and this is superposed with load time history data to obtain stress time history data. The stress time history data can be converted to a stress range-mean histogram through rainflow cycle counting. Then, the damage fraction for each cycle is determined by using a S-

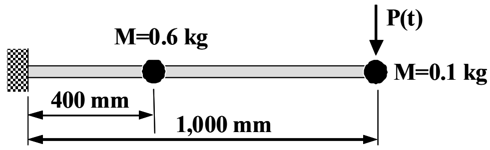

Fig. 1. Schematic diagram of cantilever beam structure with excitation load.

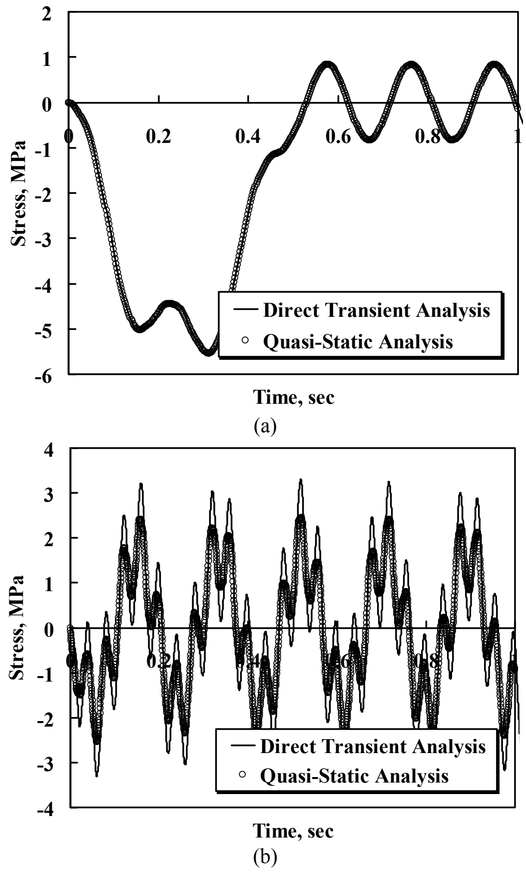

Fig. 2. Stress analysis results for excitation of (a) 1.0 Hz half-sine wave and (b) 25.1 Hz half-sine wave.

N curve, and fatigue life is finally obtained by calculating the cumulative damage fraction.

It is generally known that fatigue failure happens when the summation of the fatigue damage fraction for a structure is 1.0, and this is expressed as follows:

$$\sum D_i = \sum\left(\frac{n_i}{N_i}\right) \geq 1.0 \tag{2}$$

where D_i, n_i and N_i are the damage fraction at the stress level S_i, cycle and fatigue life, respectively.

3. Vibration fatigue analysis

In cases in which the frequency component of excitation loads includes a natural frequency of the structure, the structure shows dynamic characteristics. The dynamic behavior can be expressed in a time domain or a frequency domain, but computing dynamic behavior in a time domain is time-consuming and produces massive result data. On the other hand, dynamic behavior in a frequency domain can be calculated more easily and rapidly by using a transfer function for the structure.

In the motion equation of Eq. (1), if the load vector can be expressed in the form of a harmonic function, the displacement vector can be also expressed in the form of a harmonic function. These are given by the following equations:

$$f(t) = F \cdot e^{i \cdot \omega \cdot t},\ x(t) = X \cdot e^{i \cdot \omega \cdot t} \tag{3}$$

where F and X are load and displacement amplitude, respectively.

By taking the first and second derivatives of Eq. (3) and substituting into Eq. (1), the following is obtained:

$$X(\omega) = H(\omega) \cdot F(\omega) \tag{4}$$

where $X(\omega)$ and $F(\omega)$ are displacement and load function, and $H(\omega)$ is linear transfer function as expressed by Eq. (5).

$$H(\omega) = \left[-[M] \cdot \omega^2 + [C] \cdot i \cdot \omega + [K]\right]^{-1} \tag{5}$$

For vibration fatigue analysis considering resonance, stress PSD is used, which is calculated from the superposition of the transfer function obtained from the frequency response analysis and the PSD of the excitation load. Many researchers proposed the cycle counting methods from stress PSD [3-5]. After the stress cycle counting, the procedure of fatigue life evaluation is identical with that of quasi-static fatigue analysis. Fig. 3 shows the procedure of vibration fatigue analysis.

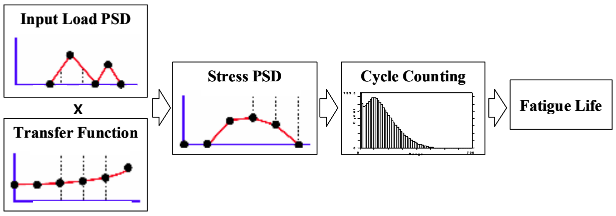

Fig. 3. Procedure of vibration fatigue analysis.

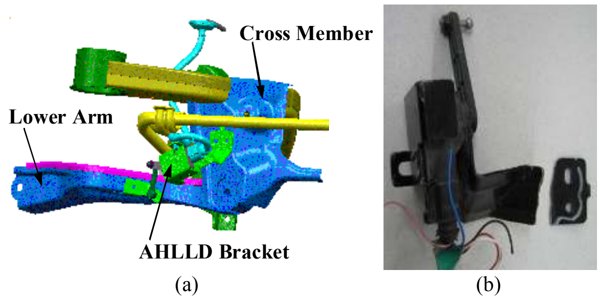

Fig. 4. Rear suspension assembly and failed bracket.

4. Determination of optimum vibration fatigue test specification for component

In this chapter, a vibration fatigue analysis technique is applied to predict the fatigue life of an AHLLD (Auto Head Lamp Leveling Device) bracket. The AHLLD bracket is attached on a rear cross member and suspends a sensor, measuring the relative motion between the cross member and the lower arm and controlling the aimed direction of the head lamp to enhance driving safety. This bracket has failed at 7,000 km in an X/C (cross country) driving test. Fig. 4(a) shows the rear suspension assembly including the AHLLD bracket, and Fig. 4(b) shows the failed bracket.

The component durability of these kinds of unit brackets is usually evaluated by using KS/JIS specifications, but those don't reflect the characteristics of driving test roads or field. Table 1 shows a fatigue test specification defined with acceleration magnitude, excitation frequency and excitation time for each x, y and z direction. Although the AHLLD bracket satisfied the component fatigue test specification, the bracket failed in a driving test as presented in Fig. 4(b), and a number of failure cases were reported. Thus, it is necessary to present fatigue test specifications for components, taking field conditions into account.

In this study, a vibration fatigue test specification for the AHLLD bracket will be proposed through a series of vibration fatigue analyses.

4.1 Determination of analysis method

In order to determine the fatigue analysis method, a mode analysis for the AHLLD bracket was carried out and acceleration data were analyzed. NASTRAN Version 2006 package was used for the mode analysis. Fig. 5 shows a finite element mesh for the AHLLD bracket, and the finite element mesh was constructed by using 3-node element (CTRIA3) and 4-node element (CQUAD4) with the number of 14,854. The auto head lamp leveling sensor was modeled by a concentrated mass with mass moment of inertia. The translational and rotational D.O.F. (Degree Of Freedom) of the mounting point of the bracket was constrained for the mode analysis, and the 1st - 5th natural frequencies were presented in Table 2.

Fig. 6 shows the acceleration time history data measured while driving the X/C road, and Fig. 7 is the PSD for the acceleration data, which was obtained from FFT (Fast Fourier Transform) for the time history data. It can be known from Fig. 7 that most of the frequency components were distributed in the region below 120 Hz. Thus, the fatigue analysis, which can consider the resonant effect, should be performed because the excitation frequency includes the first natural frequency of the AHLLD bracket.

Table 1. Component fatigue test SPEC for AHLLD bracket.

Acceleration	4.4g	6g
Excitation Frequency	33/63Hz	33/63Hz
Excitation Time	1Hr (X, Y Axis) 4Hr (Z Axis)	300,000 Cycle (X, Y, Z Axis)

Table 2. Natural frequency of AHLLD bracket.

Mode Number	Natural Frequency (Hz)
1	52.2
2	137.1
3	283.1
4	279.3
5	383.9

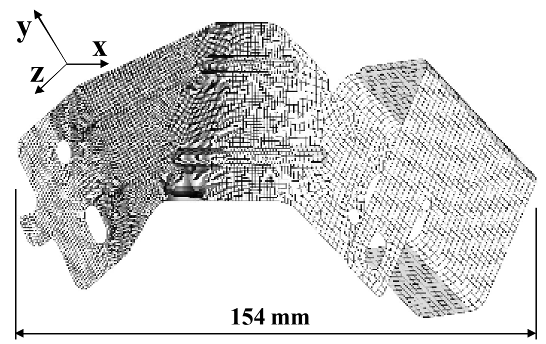

Fig. 5. Finite element model for AHLLD bracket (thickness = 2.3 mm).

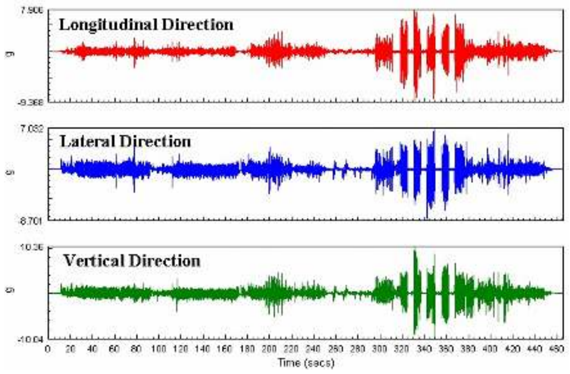

Fig. 6. Acceleration time history data.

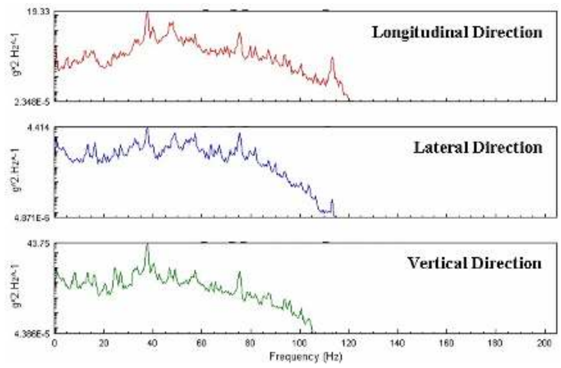

Fig. 7. PSD of acceleration.

4.2 Vibration fatigue analysis

PSD's loads are required to perform a vibration fatigue analysis. To perform FFT for the time history data makes the load amplitude at arbitrary frequency averaging so that the loads expressed in the form of PSD showed less risk than those in the original data. The equivalent amplitude, of which damage is identical with that of original amplitudes of load at arbitrary frequency, can be determined by using S-N curve [12], and then equivalent PSD, representing equal damage, can be also determined. In case the equivalent PSD is used for fatigue analysis, more useful and accurate results can be obtained. The general PSD obtained by FFT and equivalent PSD for vertical direction load in Fig. 6 were plotted in Fig. 8. It can be known that the equivalent PSD includes more energy than general PSD.

The transfer function was calculated by the unit load, and the transfer function for the hot spot was represented in Fig. 9. The vibration fatigue module of the DesignLife Version 5.1 package was used to predict the fatigue life for the AHLLD bracket. The equivalent PSD and transfer function presented in Figs. 8 and 9 were used for the fatigue analysis. Fig. 10 shows contour plots of fatigue damage and the fatigue life was presented in Table 3 with that calculated by MSM (Mode Superposition Method). The MSM technique estimated the tested fatigue life well within the difference of 13%. The vibration fatigue analysis result was compared with the MSM fatigue analysis result under the z-direction acceleration load, and the estimated result showed a good agreement within 15%. Therefore, the validation of vibration fatigue analysis method was reviewed.

4.3 Methodology for determination of optimum vibration fatigue test specification

Vibration fatigue tests are generally carried out independently according to the direction of loads, x, y and z as presented in Table 1. This condition is different from the real loading condition at the driving test so that the failure mode of a component test may differ from that of the driving test state. Therefore, in this study, a component test mode identical to the damage distribution with driving test was determined. Because the fatigue life is determined by loads and shape of components, it was assumed that the test mode of components is the function of direction and magnitude of acceleration.

In order to determine the optimum acceleration direction, a case study was performed by considering the acceleration direction in an arbitrary axis. The acceleration of the arbitrary axis, was determined by the sum of components of the acceleration presented in Fig. 6 in the arbitrary direction, and this is represented as follows:

Table 3. Fatigue life evaluated by using MSM and vibration fatigue technique.

Position	Fatigue Life (km)		
	MSM Fatigue*	MSM Fatigue**	Vibration Fatigue**
1	6,100	10,620	12,180
2	31,360	41,340	42,310
3	40,810	63,100	60,690

* 3 axes (x, y and z) load

** z axis load

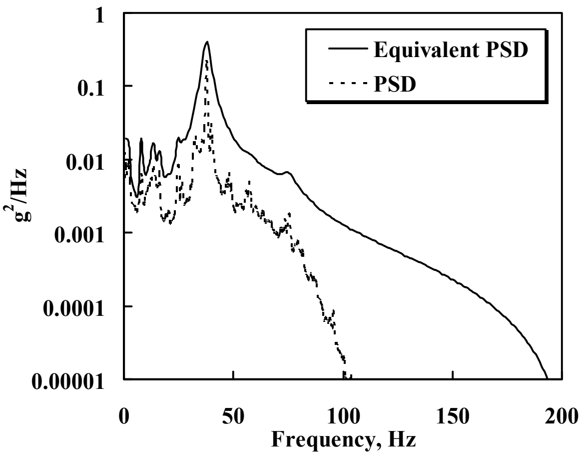

Fig. 8. Equivalent PSD for vertical direction load.

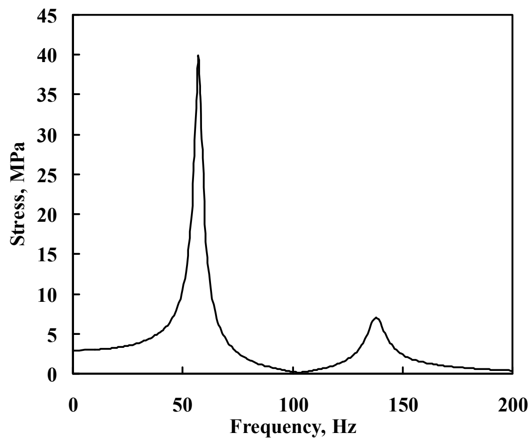

Fig. 9. Transfer function.

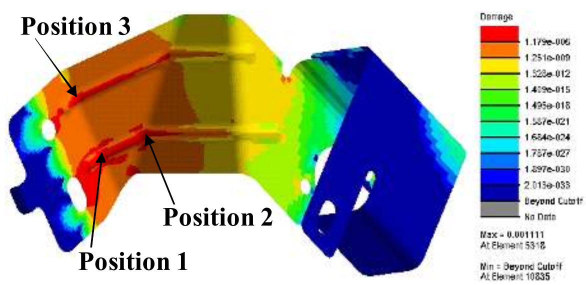

Fig. 10. Fatigue damage distribution and main position.

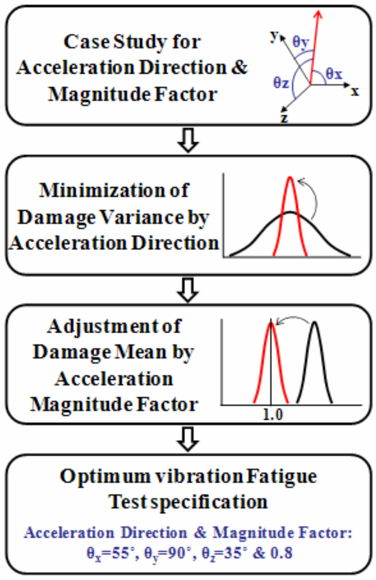

Fig. 11. Schematic procedure for determination of optimum test condition for component.

$$a = \sum a_i \cos\theta_i,\ i = x, y, z \quad (6)$$

where a_i and $\cos\theta_i$ are the acceleration component and directional cosine for i axis, respectively.

Through case study with the variation of acceleration directions, the optimum acceleration direction, which minimizes the difference of damage fraction with driving test condition at all positions of the bracket, was determined. Table 4 is the case study results, which represent the fatigue life at the main damaged position shown in Fig. 10. The life ratio indicates the ratio of vibration fatigue analysis result to MSM fatigue analysis result. In MSM fatigue analysis, the 3-axis acceleration data were used. The SN (Signal-to-Noise) ratio [13], the ratio of mean to deviation, was used to find out the optimum acceleration direction. The large SN ratio means that the data distribution from the average is relatively small. Therefore, the direction of θ_x=55°, θ_y=90° and θ_z=35° was selected as the optimum acceleration direction.

Acceleration magnitude factor was also used as another parameter for the case study. When the factor of 0.8 and the optimum acceleration direction was used to estimate the fatigue life, the life ratio was predicted as a value of 0.98 close to 1.0. As a result, the acceleration magnitude factor of 0.8 was selected as the optimum acceleration magnitude factor. In this study, the acceleration direction was used to minimize the variance of damage fraction at all positions of the bracket, and the acceleration magnitude was utilized to adjust the mean of damage fraction to 1.0. Fig. 11 shows the schematic procedure for determination of the optimum acceleration direction and magnitude factor for AHLLD bracket.

Thus, the optimum acceleration direction and magnitude were selected, and it is thought that these can be used for a

Table 4. Effect of load input direction on fatigue life.

NO.	θ_x (deg)	θ_y (deg)	θ_z (deg)	Life Ratio at Position 1	Life Ratio at Position 2	Life Ratio at Position 3	Mean	Deviation	S/N
1	50	80	40	0.38	0.25	0.27	0.29	4.57×10^{-3}	12.5
2	50	90	40	0.29	0.22	0.20	0.24	1.56×10^{-3}	15.6
3	50	100	40	0.31	0.25	0.23	0.26	1.06×10^{-3}	18.0
4	50	110	40	0.49	0.38	0.37	0.39	5.97×10^{-3}	14.0
5	55	80	35	0.32	0.24	0.22	0.26	2.20×10^{-3}	14.8
6	55	90	35	0.25	0.21	0.20	0.22	4.31×10^{-3}	20.5
7	55	100	35	0.30	0.24	0.23	0.25	1.09×10^{-3}	17.4
8	55	110	35	0.48	0.37	0.37	0.37	7.00×10^{-3}	13.0
9	60	80	30	0.36	0.24	0.22	0.27	3.56×10^{-3}	13.1
10	60	90	30	0.29	0.22	0.21	0.24	1.32×10^{-3}	16.2
11	60	100	30	0.31	0.25	0.25	0.25	2.26×10^{-3}	14.4
12	60	110	30	0.50	0.40	0.40	0.39	9.41×10^{-3}	12.1

vibration fatigue test specification for the AHLLD bracket. By using this methodology to determine the optimum vibration fatigue test specification, the component failure mode that closely resemble the real failure mode, can be predicted, and then the durability of the components will be assured at the stage of initial product design.

5. Conclusions

In this paper, the necessity of the fatigue analysis, considering resonant effect, was reviewed. Also, the methodology to determine the optimum vibration fatigue test specification of a component, which gives an equivalent failure mode during driving test conditions, was proposed. The key findings and results are as follows:

(1) The component vibration fatigue test specification was determined for a auto head lamp leveling device (AHLLD) bracket by using vibration fatigue analysis technique. The acceleration direction and magnitude factor of the specification are θ_x=55 °, θ_y=90 ° and θ_z=35 ° and 0.8, respectively. The failure mode of the specification is closely equivalent to that of vehicle test condition from the view of the failure location and fatigue life.

(2) The methodology to determine the optimum vibration fatigue test specification for components, which gives equivalent failure mode with driving test condition, was proposed by considering the direction and magnitude of acceleration.

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