International Journal of Science and Engineering Applications
Volume 14-Issue 12, 29 – 34, 2025, ISSN:- 2319 - 7560
DOI: 10.7753/IJSEA1412.1006

# Simulation Analysis of Random Vibration Fatigue of Urea Tank Bracket

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**Abstract**: This paper in order to investigate the crack and fracture problems of a truck frame fixed structure. Taking a urea tank bracket as the research object, the fatigue reliability of the bracket is evaluated by the fatigue damage results obtained by the random vibration fatigue simulation method. Firstly, the finite element model of the urea tank bracket was built, and the modal analysis and frequency response analysis were carried out. Taking the power spectral density (PSD) converted from the acceleration signal of the bracket collected in the experiment as the load excitation, the fatigue simulation analysis of the structure was carried out. The fatigue damage results of the urea tank bracket were obtained. The risk position of the simulation results was basically consistent with the bracket fracture position feed backed by the market, which verifies the accuracy of the simulation method. Structural improvement to the risk location, the fatigue damage results of the improved structure was significantly reduced compared with damage results of the original structure, which verified the rationality and effectiveness of the simulation method, and also provided a reference for the subsequent development and design of similar structures.

**Keywords**: random vibration; frequency response analysis; power spectral density; fatigue damage

## 1. Introduction

With the continuous tightening of emission regulations in the truck industry, the rear exhaust system has become increasingly critical in the overall vehicle structure. In particular, the layout design and structural reliability of components such as the aftertreatment system and urea tank in recent years have emerged as key factors influencing normal vehicle operation and driving safety. Among these, the urea tank assembly is mounted on the frame as a suspended structure. The urea tank bracket must support the weight of the urea solution and the tank itself while also being subjected to road-induced excitations transmitted through the frame—both of which are major contributors to fatigue failure.

In engineering, the primary form of fatigue failure is structural damage caused by alternating loads or repeated stresses. This is especially pronounced when the frequency of the alternating load coincides with or approaches one or more of the structure's natural resonant frequencies. Under such conditions, structural resonance can amplify the response to a given excitation, making failure more likely a phenomenon referred to as vibration fatigue or dynamic fatigue failure[1]. Although literature on structural reliability analysis of urea tanks is relatively limited, the research methodologies employed for frame-mounted structures such as fuel tanks and battery packs are highly relevant and worth referencing. For example, in addressing fuel tank bracket fracture issues, Zhao Weiyan[2] conducted fatigue life analysis by integrating static strength results from HyperMesh finite element simulations with measured acceleration signals from the bracket. A modified design incorporating reinforced weld designs at stiffener joints was developed and validated through testing, demonstrating significantly enhanced structural strength and fatigue performance. Liu Longtao[3]employed ANSYS finite element software to perform modal analysis and random vibration analysis on the structure. The finite element model was calibrated using experimental modal test data, while stress response power spectral density was derived through random vibration analysis. Ultimately, structural fatigue damage was quantified using a frequency-domain methodology. Suo Minghe[4] et al. employed measured acceleration signals to conduct frequency response analysis through a structural finite element model when validating whether a battery bracket met strength requirements. They implemented a neighborhood cultivation multi-objective genetic algorithm to optimize the inner and outer plates, achieving both mass reduction and structural stress below the yield limit. The optimized design successfully passed full-vehicle road testing. Cheng Chuansheng[5] et al. integrated finite element analysis with experimental testing to perform vibration analysis on a urea tank bracket. Structural enhancements were implemented at stress concentration areas, and the optimized design successfully passed both bench testing and durability validation.

This research adopts an integrated methodology combining finite element fatigue simulation and experimental validation to investigate fracture failures in the urea tank bracket of a domestic truck model. By utilizing measured excitation data as loading boundary conditions, a random vibration-based fatigue simulation approach was implemented to pinpoint fatigue-prone areas in the bracket structure. The results establish a theoretical foundation for subsequent structural enhancements and optimization efforts.

## 2. Vibration Fatigue Analysis Methods

Power Spectral Density (PSD), also known as power spectrum density, is a measure of the mean-square value of a random variable. It serves as a frequency-domain description of stationary random processes and represents a probabilistic statistical method. The power spectral density function $S_x(\omega)$ is obtained through the Fourier transform of the autocorrelation function R(τ) of a stationary random process.

$$S_x(\omega) = \int_{-\infty}^{+\infty} R_x(\tau)\, e^{-j\omega t} d\tau \tag{1}$$

In the formula:

$\omega$ represents the angular frequency of vibration;

denotes the autocorrelation function

$$R_x(\tau) = \lim_{s\to\infty} \frac{1}{s} \int_{-s/2}^{s/2} x(t)\, x(t+\tau) dt;\ j = \sqrt{-1}$$

The Power Spectral Density (PSD) contains substantial statistical characteristics of random processes. These statistical properties can be derived through the spectral moments of the PSD. The definition of the i-th order spectral moment is:

$$m_i = \int_0^\infty \omega^i G_v(\omega)\, d\omega \tag{2}$$

In the formula:

i=0,1,2,……;

$G_v(\omega)$ is the one-sided power spectral density function of stress at angular frequency $\omega$。

The number of stress cycles $n_i$ within the stress range （$S_i$，$S_i+\Delta S_i$） over time duration T is given by:

$$n_i = E(P) T P(S_i) \Delta S_i \tag{3}$$

In the formula:

*E(P)* represents the expected value of the peak frequency in the random response signal $E(P) = \sqrt{m_4/m_2}$ ;T represents the duration of the random response excitation; $P(S_i)$ denotes the probability density function of the stress amplitude $S_i$。

According to the material's stress-life curve (*S-N* curve), the total number of cycles to failure *N* is known:

$$N = S^{-m} C \tag{4}$$

*C* is the material constant; m is the Basquin exponent.

The empirical Dirlik formula[6,7] for rainflow cycle counting is employed to obtain the peak probability density function of the stress *P(S)*:

$$\mathrm{P(S)} = \frac{\frac{D_1}{Q} e^{\frac{-Z}{Q}} + \frac{Z D_2}{R^2} e^{\frac{-Z^2}{2R^2}} + D_3 Z e^{-\frac{Z^2}{2}}}{2\sqrt{m_o}} \tag{5}$$

In the formula, $D_1 = \frac{2(x_m - \gamma^2)}{1+\gamma^2}$; $D_2 = \frac{1-\gamma-D_1+D_1^2}{1-R}$; $D_3=1-D_1-D_2$; $Z = \frac{S}{2\sqrt{m_0}}$; $Q = \frac{1.25(\gamma - D_3 - D_2 R)}{D_1}$;

$\gamma = \frac{m_2}{\sqrt{m_0 m_4}}$; $R = \frac{\gamma - x_m - D_1^2}{1-\gamma-D_1-D_1^2}$; $x_m = \frac{m_1}{m_0}\sqrt{\frac{m_2}{m_4}}$;

According to Miner's linear cumulative damage rule, the fatigue damage of a structure is expressed as:

$$\mathrm{D} = \sum D_i = \sum \frac{n_i}{N_i} \tag{6}$$

By combining equations (2) through (6), we obtain:

$$D = \sum \frac{n_i}{N_i} = \frac{E(P)}{C} T \int_{-\infty}^{+\infty} P(S) S^m\, dS \tag{7}$$

## 3. Data Signal Acquisition and Processing

The acquisition of acceleration load spectra provides essential input for subsequent fatigue analysis of the urea tank bracket, serving as critical data for fatigue strength assessment and life cycle calculations. In this study, acceleration signal collection was conducted at the proving ground of a commercial vehicle manufacturer, where experimental investigations were performed on the loading conditions of the urea tank bracket system of a specific truck model under various road conditions.

Based on the mounting configuration of the urea tank bracket structure on the vehicle frame, six triaxial accelerometers were deployed throughout the system. Specifically, one accelerometer each was installed at positions 1 and 4 on the frame, two were mounted at transition bracket points 2 (left-right symmetric positions a and b), and two were positioned on the urea tank bracket points 3 (left-right symmetric positions a and b). The sensor signal acquisition directions were aligned with the vehicle coordinate system, with Z representing the vertical direction, X denoting the longitudinal direction, and Y indicating the lateral direction. Key testing conditions included: idle operation, acceleration operation, cobblestone road operation, and gravel road operation. Three cyclic tests were performed for each condition. The experimental signals were sampled at a frequency of 4096 Hz. The urea tank system model and accelerometer installation locations are illustrated in Fig. 1.

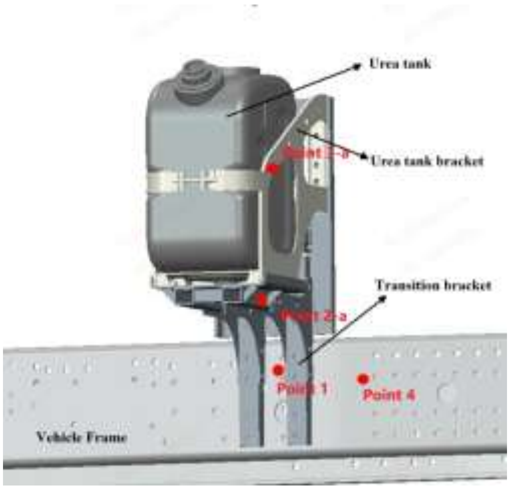



Figure. 1 Urea tank system model and accelerometer layout

One set of the most stable results from the three test groups was selected for response signal analysis and processing. To eliminate interference from external factors, signal data processing was necessary. The collected acceleration signals underwent resampling, band-pass filtering, zero-drift correction, and removal of outliers and spike values. After these processing and correction steps, the acceleration signals at each measurement point of the urea tank bracket system were obtained. The processed acceleration signal at measurement point 1 is shown in Fig. 2 below.

Using time-frequency conversion data processing methods, the measured time-domain acceleration signals were transformed into acceleration power spectral density (PSD) spectra. PSD better displays the distribution of vibration energy in terms of power density and also serves as the load input for random vibration fatigue simulation analysis. The resulting PSD is shown in Fig. 3 below.

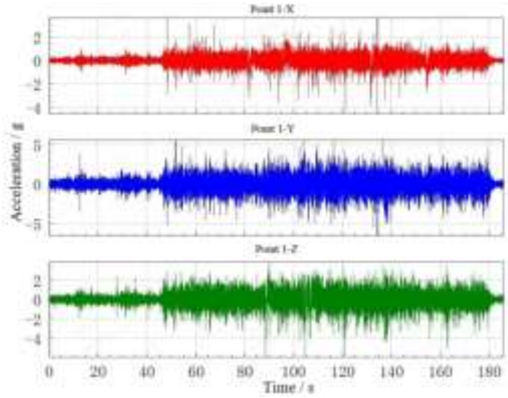



Figure. 2 Acceleration signal at Measurement Point 1 of the urea tank bracket system

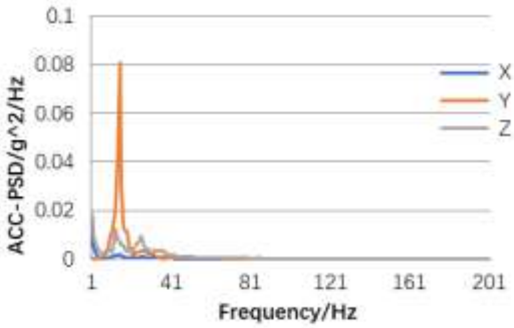



Figure. 3 Acceleration power spectral density at Measurement Point 1 of the urea tank bracket system

## 4. Vibration Fatigue Simulation Analysis

### 4.1 Finite Element Model

The urea tank bracket system primarily consists of the vehicle frame, transition bracket, urea tank bracket, urea pump, and urea tank. The modeling process also accounts for the effect of the urea solution, with its mass equivalent to the fluid mass during actual vehicle testing. The 3D geometric model of the system is shown in Fig. 1.

Model simplification aims to balance accurate representation of actual conditions with computational efficiency for subsequent solving. The simplified model was imported into Hypermesh finite element analysis software for preprocessing operations, including geometry cleanup and mesh generation. Bolt connections were simulated using RBE2 elements, clamp bands were defined via coupling constraints, and welded joints were modeled with rigid connections. Contact relationships between components were properly defined, and boundary conditions were applied according to the actual mounting configuration. The simplified finite element mesh model of the urea tank system is shown in Fig. 4 below.

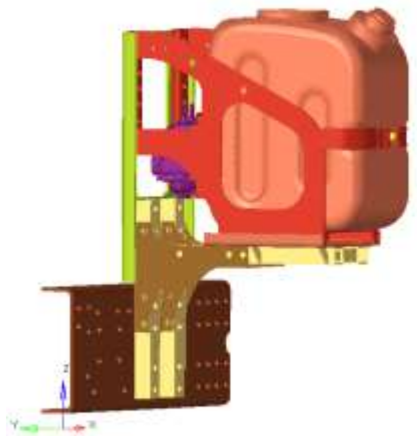


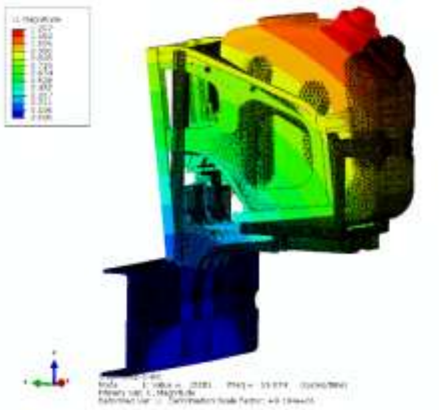
Figure. 4 Finite element mesh model of the urea tank system

### 4.2 Modal analysis

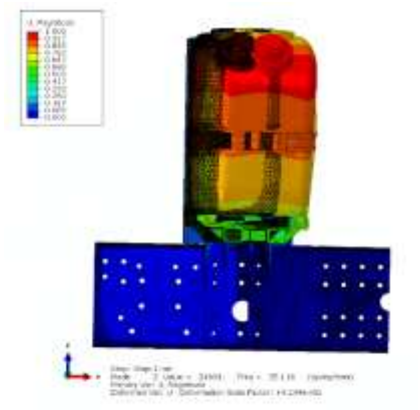
Modal analysis reveals the natural frequencies and mode shapes of the system, providing critical frequency range inputs for subsequent frequency response analysis and accurate acquisition of frequency response functions. The Lanczos method[8], known for its computational efficiency and accuracy, was employed for modal analysis due to its suitability for large-scale models. The simulation results of the urea tank system modal analysis are summarized in Table 1 below, while the first three mode shapes are illustrated in Fig. 5.

**Table 1. Natural Frequencies of the Urea Tank System**

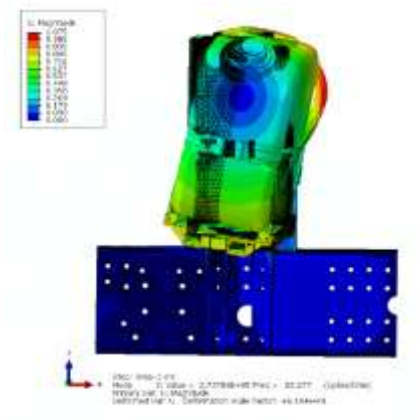
| Mode Order | Natural Frequency | Mode Shape Description |
|---|---|---|
| 1 | 19 | Vertical oscillation of the system |
| 2 | 25 | Fore-aft oscillation of the system |
| 3 | 83 | Fore-aft torsional motion of the system |
| 4 | 96 | Lateral sway of the system |
| 5 | 144 | Local vibration mode of the urea tank bracket |
| 6 | 153 | Local vibration mode of the urea tank bracket |



Mode 1



Mode 2



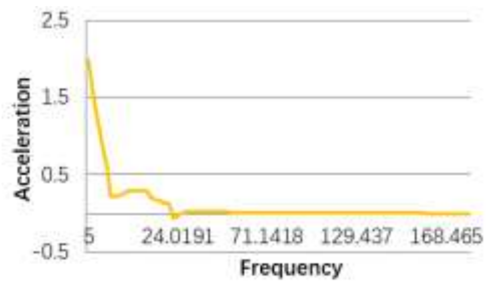
Mode 3

Figure. 5 First three mode shapes of the urea tank bracket system

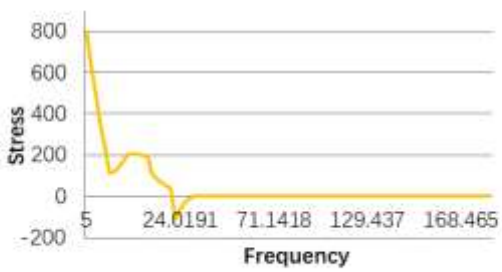
### 4.3 Frequency Response Analysis

The frequency response calculation module in Abaqus software includes three methods: Direct, Modal, and Subspace. Based on the characteristics of each computational approach, the Direct method exhibits significantly slower solution speed for frequency response functions compared to the Modal method[9]. Therefore, this study employs the Modal method for frequency response analysis.

In the finite element model, a base excitation point was established at the same location as the actual measurement point 1. Unit load excitations were applied separately along the X, Y, and Z directions at this base excitation point. Building upon previous modal calculation results, the excitation frequency range was set from 5 to 512 Hz. For structural damping of steel materials, typically ranging from 0.02 to 0.06, this study adopted a damping ratio coefficient of 0.03 for the urea tank bracket. Through the Modal-based frequency response analysis method, the relationship between frequency and displacement/stress distribution was obtained. By appropriately setting frequency points and bias parameters, the frequency response functions of the urea tank bracket system were derived. Figure 6 shows the stress and acceleration frequency response curves at critical node 219,398 under unit Y-direction load excitation.




(a) Computation Results of Acceleration Frequency Response at a Specific Node




(b) Calculation Results of Stress Frequency Response at a Specific Node

Figure. 6 Stress and acceleration versus frequency curves at Node 219398

## 5. Fatigue Simulation Calculation

During actual vehicle operation, excitation from engine ignition and road surface irregularities simultaneously act on the frame, which subsequently transmits these loads to the urea tank bracket system. Consequently, the component is subjected to alternating cyclic loads, with fatigue failure being its primary failure mode. Therefore, it is essential to perform fatigue simulation analysis on this structure.

### 5.1 Fatigue Damage Calculation

The frequency response functions obtained from previous frequency response analysis and the processed acceleration power spectral density (PSD) data from experimental tests were imported into finite element fatigue analysis software. Combined with the material S-N curve, random vibration fatigue simulation analysis was conducted. Based on the material parameters of the urea tank bracket, the S-N curve of the bracket material was fitted, as shown in Fig. 7. The fatigue software applied the Miner Modified method to correct the S-N curve based on Miner's linear cumulative damage theory.

The duration of PSD loading directly influences the magnitude of the calculated fatigue damage. In this study, a loading time of 960,000 seconds was adopted, consistent with the durability test standards for bad road conditions (cobblestone and gravel roads) at the proving ground.

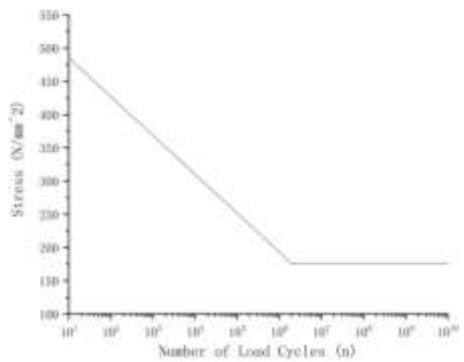


Figure. 7 S-N curve of the urea tank bracket material

Using fatigue analysis software, the fatigue performance parameters of the urea tank bracket material were modified to account for factors such as surface roughness, loading mode, stress gradient, and mean stress effects. These parameters were adjusted within the spectral analysis module. The Dirlik method was selected as the solution algorithm, and the fatigue damage calculation was performed with a survival rate set at 99.99%. The resulting fatigue damage contour plot of the urea tank bracket is shown in Fig. 8 below.

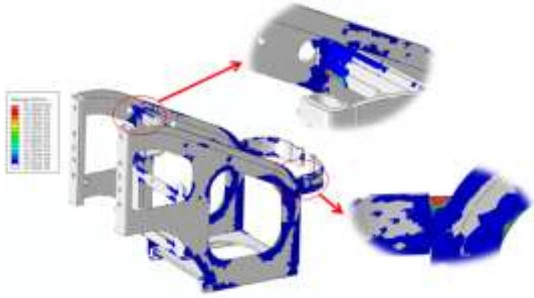


Figure. 8 Fatigue damage contour of the urea tank bracket

From the fatigue damage results of the random vibration analysis on the urea tank bracket, it can be observed that the location of maximum fatigue damage is at the root of the clamp band (as shown in Fig. 8 above), with a cumulative fatigue damage value of D1 = 0.0704. The second highest damage location is at the root of the back beam on the bracket plate (as indicated in Fig. 8), with a cumulative fatigue damage value of D2 = 0.0304. According to Miner's rule, although these values are below the limit of 1 and will not cause failure, these positions still represent the highest-risk areas in the urea tank bracket structure.

## 5.2 Comparison of Simulation and Experimental Results

Based on the field-reported failure locations of the urea tank bracket fractures, a comparison with the identified highest-risk locations from the simulation results shows that the position of maximum simulated damage closely aligns with the actual fracture locations on the vehicles. The comparison between the maximum damage location from simulation and the actual bracket fracture location is illustrated in Fig. 9 below.

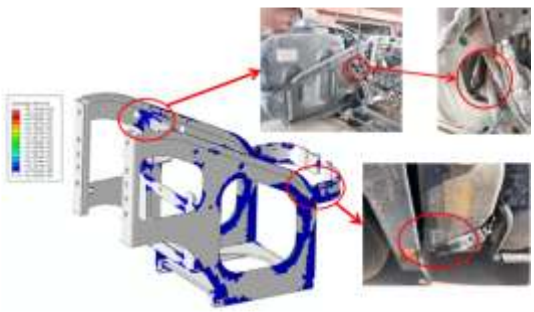


Figure. 9 Comparison between the maximum simulated damage location and the actual vehicle fracture location

## 5.3 Fatigue Analysis of the Optimized Bracket

Based on the critical locations identified through the aforementioned method, the existing structure of the bracket was improved. This included optimizing the welding process near the crack-prone areas, increasing the thickness of the clamp band by 1 mm, and adding reinforcing ribs between the back beam and the side plate of the bracket. A new model of the optimized urea tank bracket was developed, followed by modal analysis and frequency response analysis to obtain updated frequency response functions. Using the same PSD inputs as in the previous analysis, a random vibration fatigue simulation was performed. The resulting fatigue damage contour of the improved urea tank bracket is shown in Fig. 10 below.

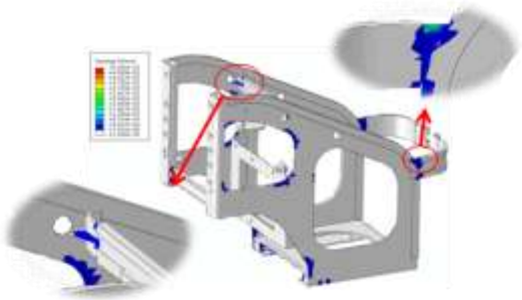


Figure. 10 Fatigue damage contour of the optimized urea tank bracket

As shown in Fig. 10, the fatigue damage results of the optimized bracket structure indicate that the location of maximum fatigue damage remains consistent with the original design, primarily at the root of the clamp band and the root of the back beam on the bracket plate. However, the maximum cumulative fatigue damage value is reduced to D3=0.00285, demonstrating a significant improvement compared to the fatigue damage results at the same locations in the original structure.

# 6. Conclusion

This study employs an integrated experimental and simulation approach to investigate the urea tank bracket system of a specific truck model. Based on measured load spectra, fatigue damage calculations were performed, and design improvements were proposed. The following conclusions are drawn:

(1) The finite element method was utilized to establish a model, with measured load spectra converted into PSD as load boundaries for random vibration fatigue analysis. This approach closely approximates real-world conditions while offering advantages of low cost, short cycles, and ease of implementation, making it suitable for engineering applications. Furthermore, this methodology provides valuable reference experience for addressing similar issues in comparable structures.

(2) Comparison between field failure locations and simulated maximum fatigue damage positions shows strong consistency, effectively reproducing the actual working condition failures of the urea tank bracket. This validates the feasibility and accuracy of the frequency-domain-based random vibration fatigue simulation method, while also providing a reference basis for subsequent structural improvements.

(3) Structural enhancements were implemented targeting the weak areas of the original design, resulting in significantly reduced fatigue damage compared to the initial structure. This outcome not only demonstrates the rationality of the improved structural solution but also further validates the effectiveness of the random vibration fatigue simulation methodology.

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