System Dynamics Analysis Based on Steel Tube Frame

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Abstract. The paper focuses on the chassis of the Dongfeng HUAT Team's 11th generation car and conducts a comprehensive study on its vibration characteristics using the finite element method. Firstly, a finite element model of the chassis structure is constructed using CATIA. Subsequently, the modal analysis is performed by applying ANSYS technology to calculate the finite element analysis results. Furthermore, the dynamic characteristics of the chassis are explored in greater depth by investigating the impact of engine vibrations on the chassis structure and examining the interaction between the chassis design features and the race car's drive system. Finally, random vibration analysis is conducted using ANSYS to further validate the theoretical and simulation errors. This research provides valuable insights and a solid foundation for effectively addressing mechanical vibration issues in tubular chassis systems. The findings have significant theoretical significance and practical implications.

Keywords: Steel pipe frame · finite element · dynamic characteristics · modal analysis · random vibration analysis

1 Introduction

The FSCC racing frame system is a complex vibration system with multiple degrees of freedom. When subjected to external excitation forces, it experiences vibrations. Furthermore, when the frequency of external vibrations matches the natural frequency of the system, resonance occurs, which can have detrimental effects on the structural integrity of the frame and the overall performance of the car, including handling, driving, comfort, and safety.

To enhance the system's vibration resistance and reduce the occurrence of resonance, comprehensive research and analysis are necessary. It is crucial to implement measures to optimize the design and strengthen the structure of the frame. This endeavor holds significant importance in improving racing performance and driving experience.

2 Basic Theory of Modal Analysis

2.1 Introduction

FSCC racing frame is a complex vibration system with multiple degrees of freedom, which will vibrate when stimulated by the outside world, and resonance will occur when the frequency of the external vibration is close to the frequency of the system

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itself. Resonance not only destroys the structure of the frame itself and the integrity of the body, but also affects the handling, driving comfort and safety of the car. As a continuous elastic structure system, the natural vibration frequency of the frame also has an infinite number of corresponding natural modes. The low order modes of the frame are generally the global mode, such as bending mode and global torsion. The high-order modes are generally local modes, such as front ring mode and main ring mode. At the same time, due to the low local stiffness of the frame, there will be some local modes in the low frequency range, or at the same time with the vehicle mode. Reasonable frame mode distribution is very important to improve the safety and NVH performance of FSCC racing vehicles.

2.1.1 Modal Assumption

The modal analysis can determine the fixed frequency and vibration mode of the frame structure, which are also the key data in the design of dynamic load architecture. Before applying modal analysis, a series of assumptions are made about the system:

(1) The system linearity hypothesis assumes that the entire system is linear, in other words, the response of the system to any set of incentives acting simultaneously is a linear superposition of their responses.
(2) The system time invariant hypothesis assumes that the whole system is steady, then all the characteristic parameters of the whole system are also constant, and this system is called a steady system. If it is assumed that the system does not conform to the time-invariant system, the data obtained at different times will be inconsistent, that is, the fixed system characteristic parameters cannot be obtained.
(3) Observability of the system For the standard model that accurately describes the characteristics of the system, the input and output measurement signals of the control system must include sufficient information, at which time the control system becomes a fully observable system. The modal superposition theory and coordinate system transformation are the cornerstone of the modal analysis method of multi-degree-of-freedom system analysis, and the key of the modal mode analysis method. The modal coordinate system is constructed by using the modal mode matrix, and the corresponding coordinate system is obtained.

2.1.2 Mathematical Assumption Prerequisite

Modal analysis is the analysis of the dynamic characteristics of the structure of the part, and the dynamic characteristics of the structure can be expressed by modal parameters. Generally, the physical coordinate system on the differential equations (principal modes of each order) of the vibration of the linear fixed constant system is used as the modal coordinate system, and is decoupage from the standard parameters. The frame can be regarded as a rigid body model placed on the assembly shock absorber spring, and connected to the tire through the shock absorber, the wheel is supported on the road through the elastic and damped tire, so the frame can be simplified in the mode of multi-degree of freedom system in the modal analysis.

2.1.3 Analysis Process

The classical modal analysis process consists of four stages: model construction, modal calculation, expansion mode and analysis results.

(1) The modeling process of modal analysis is very similar to that of the statics method in ANSYS, and can be roughly divided into the definition of the element model, real constant, structural characteristics, the construction of geometric models, and the analysis process of finite element mesh.
(2) Modal solution The modal calculation process usually includes setting analysis mode, analytic options, design constraints, setting load options, and completing the calculation of fixed frequencies.
(3) Extended modes If you want to see the calculation results from the post-processing process, you must first write the mode to the data document by adding the mode. The whole process includes reloading the solver, launching the extension selection, determining the load step options, and extension processing.
(4) The results of modal analysis are usually divided into the fixed frequency, mode, corresponding stress and force of the structural form.

3 Finite Element Analysis

3.1 Finite Element Modal Analysis of the Frame

3.1.1 Modal Objective

To analyze and study the inherent characteristics of the frame through modal analysis, while the natural frequency and vibration mode of the frame can be used to directly evaluate and optimize the design of the frame. The vibration form when the component resonates is called natural vibration type, and the main parameter of evaluating the dynamic performance of the system structure is natural frequency. When the working frequency and natural frequency are similar, resonance phenomenon will occur, and the system will produce large amplitude and deformation, which greatly reduces the reliability and service life of the frame. Therefore, the modal analysis of the existing frame structure is a good way to determine whether the structure has resonance phenomenon.

3.1.2 Free Mode

Free boundary and bound boundary (free mode and bound mode) The difference between the two boundary conditions for modal analysis is whether there is a rigid body mode. The so-called rigid body mode means that the structure itself has no elastic deformation, and the structure as a whole (rigid body) moves in the direction of six degrees of freedom. In finite element analysis, for free boundary conditions, the frequency of the first 6 modes is 0 or the value is very small, starting from the 7th mode is the elastic mode of the structure (there is elastic deformation inside the structure). The free boundary has not only rigid body mode, but also elastic mode; The constrained boundary also has elastic modes. The following is a simulation evaluation of the free mode and the constrained mode respectively (Figs. 1, 2, 3, 4, 5, 6 and Table 1).

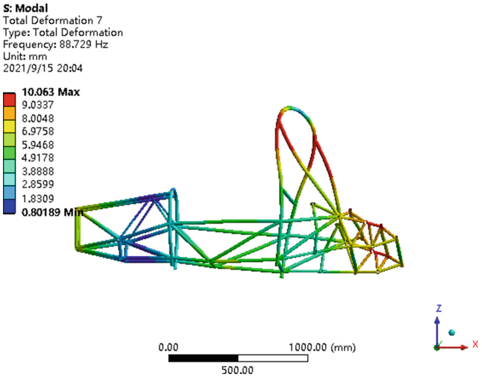

Fig. 1. The 7th mode

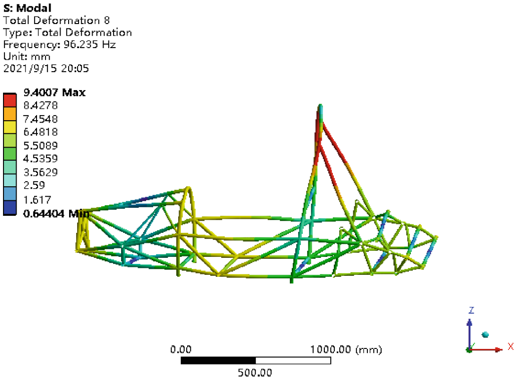

Fig. 2. The 8th mode

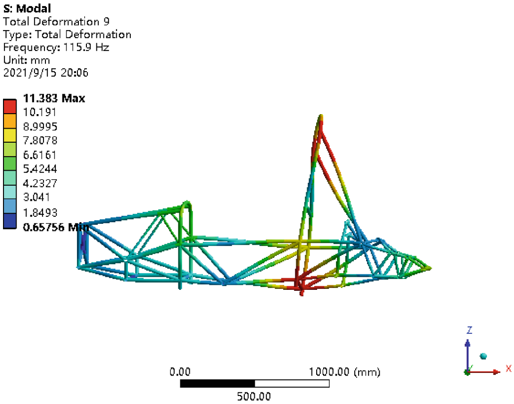

Fig. 3. The 9th mode

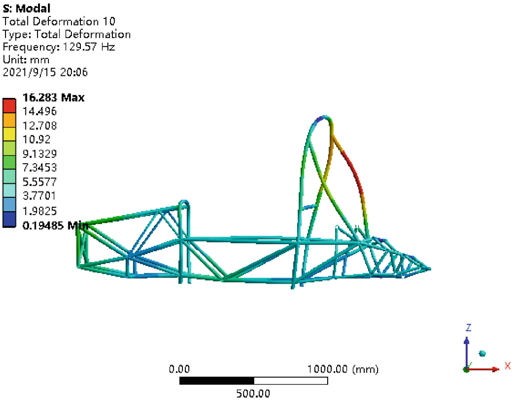

Fig. 4. The 10th mode

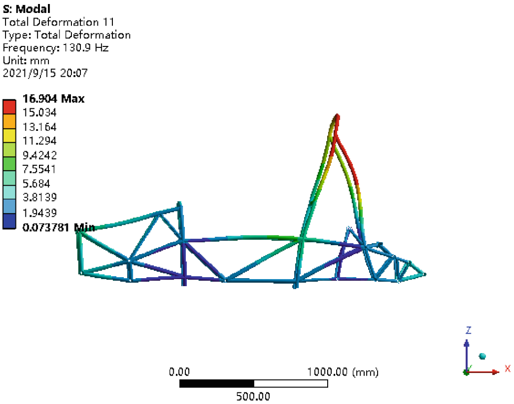

Fig. 5. The 11th mode

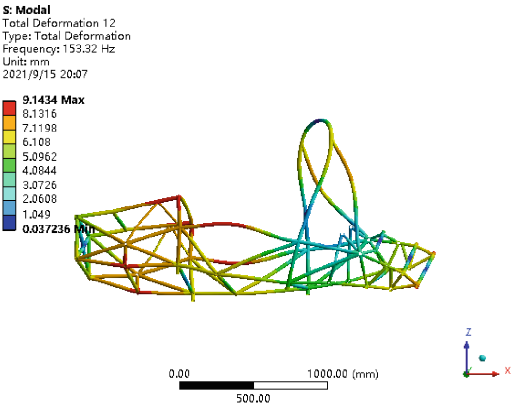

Fig. 6. The 12th mode

Table1. Free mode frequency and mode of frame

order	frequency	mode of vibration
7	88.729	The front and rear compartments are rotated in reverse to the left and right based on the support transverse pipe under the main ring
8	96.235	The rear compartment is rotated around the X-axis
9	115.9	The front and rear cabins are rotated in reverse to the left and right based on the cockpit seat belt mounting tube
10	129.57	The center of the support tube under the main ring is twisted in the same direction
11	130.9	The front and rear cabins twist up and down with the cockpit pipe as the benchmark
12	153.32	The center of the support tube under the main ring is twisted in the same direction

3.1.3 Constraint Modal Analysis

Constrains the hard points connected to the frame and suspension, which constrains the hard points behind the left and right upper A-arm of the front frame and the hard points UZ at the left and right upper A-arm of the rear frame, the hard points UY at the left and right upper A-arm of the front frame, and the hard points UX at the left upper A-arm of the rear frame. Ensure the vehicle in UX, UY, UZ, RotX, Rot Y, RotZ six degrees of freedom constraints, while no constraints. The first 16 order natural frequencies of the frame under constrained vibration were extracted by Block Lanczos method. The natural frequencies of the first 16 orders are shown in the table, and the natural frequencies of the frame from the seventh to the twelfth orders and the corresponding vibration modes are shown in the figure (Figs. 7, 8, 9, 10, 11 and 12).

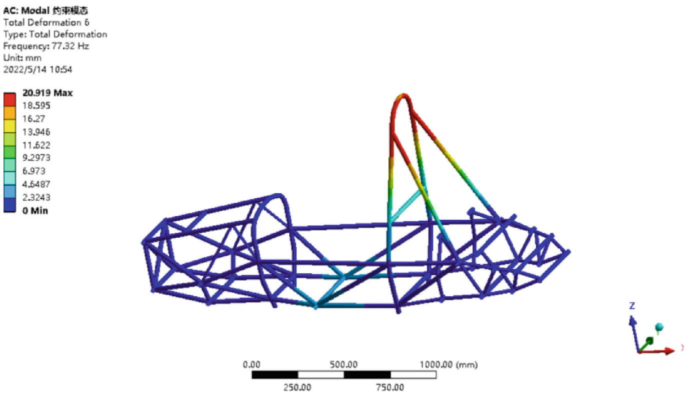

Fig. 7. The 7th constraint mode

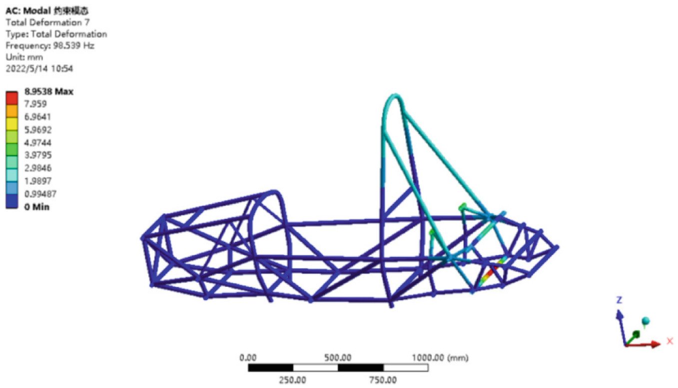

Fig. 8. The 8th constraint mode

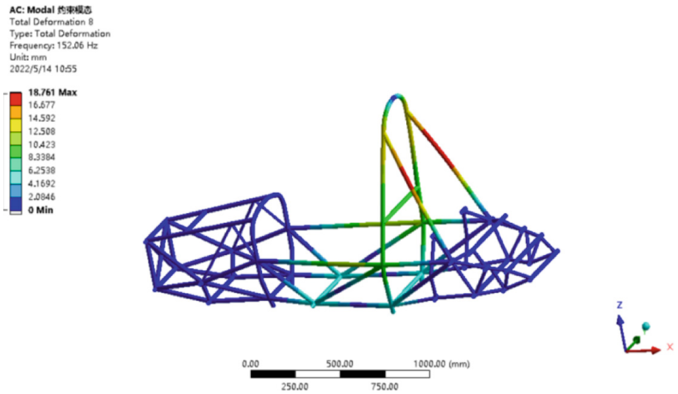

Fig. 9. The 9th constraint mode

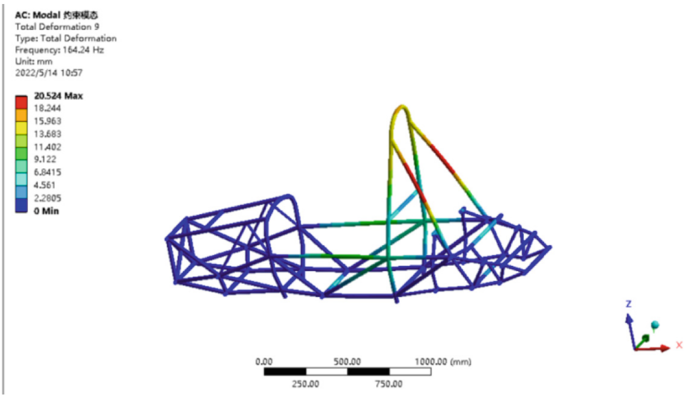

Fig. 10. The 10th constraint mode

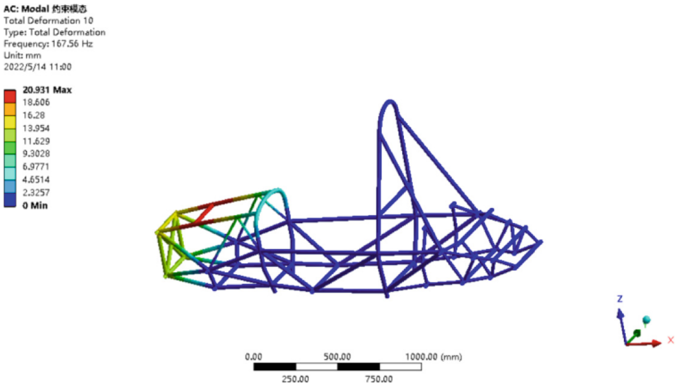

Fig. 11. The 11th constraint mode

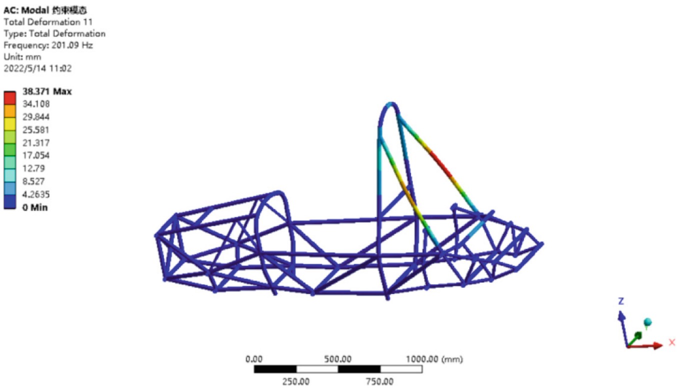

Fig. 12. The 12th constraint mode

3.2 Analysis of Modal Results

During the driving of the car, the external excitation mainly comes from the road surface, wheels, engine, etc. In particular, the frequency of the engine will cause vibration to the whole frame, which will have an impact on the frame. Through the analysis of each mode of the frame, the frequency of the frame when it reaches each mode is calculated to avoid the coincidence with the external excitation frequency as far as possible. The road excitation is determined by the road conditions, the racing track is a good road surface, and the excitation is mostly below 3 Hz, and the excitation frequency caused by the wheel imbalance is generally lower than 11 Hz. The engine excitation frequency is calculated as follows:

$$f = \frac{2nz}{60k} \tag{1}$$

n -- is the engine speed, r/min;

z - number of engine cylinders

k - number of engine strokes

The same engine used this season is the Triumph 675, three cylinders, four strokes, the engine idle speed of 2500r/min. The vibration frequency range of the engine is 62.5 Hz. Working speed range: 4000-10000r/min. The range of engine vibration frequency is obtained: 100–250 Hz. Therefore, the natural frequency of the frame should avoid the two frequency ranges above 3 Hz–11 Hz and 200 Hz. From the modal analysis results, it can be concluded that the low order frequency of the frame free mode natural frequency is above 11 Hz, and the high order frequency is less than 200 Hz. Although the 13th, 14th, 15th and 16th natural frequencies of the constrained modes of the frame are higher than 200 Hz, these fourth-order modes are not in the continuous working state of the engine and have little impact on the frame. Therefore, the frame can prevent resonance problems in a large range and has excellent dynamic stability.

3.3 Random Vibration Analysis

3.3.1 Basic Theory

Random vibration analysis is a spectral analysis method based on the idea of probability statistics, also known as power spectral density research. Its purpose is to study the response of structures under random vibration loads by converting the time history of changes in vibration load samples into power spectral density index (PSD). From the point of view of probability theory, the conclusion of random vibration analysis is random change, and is related to the standard deviation of stress, displacement and other results.

Therefore, it belongs to the frequency-domain analysis method and requires modal analysis. Through modal analysis, the spectrum signal range and mode information of different structural modes are obtained. In order to ensure that the full spectrum signal range is obtained to cover the modes that have an important impact on the structure, the spectrum signal coverage of the PSD curve cannot be too small, and needs to be continuously extended to the range of smaller spectral values.

This can ensure the accuracy and comprehensiveness of the calculation results. Random vibration analysis can provide the response characteristics of structures under random vibration loads. By evaluating the vibration characteristics and dynamic response of the structure, the possible resonance problems, the vibration resistance of the structure and the measures to reduce the impact of vibration are identified. This is of great significance for the design and optimization of the structure, and helps to improve the reliability and durability of the structure.

3.3.2 Road Surface Theory

The excitation of road surface to vehicle is a random process. Since it cannot be expressed by defined parameters, this process is often represented by the power spectral density of road roughness in a statistical way. The roughness of the road surface refers to the vertical deviation of the road surface from the ideal reference plane. The reference standard is ISO/DIS8608 and the Chinese national standard GB/T7031, which is the road roughness obtained by the automobile vibration input, generally through the road power spectral density to express its statistical characteristics.

Road roughness excitation will be transmitted to the tire and frame and affect the dynamic characteristics of the vehicle. Vibration and deformation can cause discomfort to the driver and fatigue damage to the frame. Uneven road surface will affect the stability and handling performance of the car, increasing the risk of driving. The vibration of the frame caused by the excitation force generated by the unevenness of the road surface is closely related to the speed.

Its vehicle speed at resonance is

$$v = 3.6\mathrm{L}\omega\mathrm{f}(\mathrm{km/h}) \tag{2}$$

The experimental results of the unevenness wavelength of different road spectra in our country are shown in Table 2.

Table 2. Wavelength of different road roughness

road surface	Unpaved surface	gravel road	Washboards		Flat highway
Road roughness wavelength(m)	0.77–2.5	0.32–6.3	0.74–5.6	0.75	1.0–6.3
Resonance speed(km/h)	226 ~ 734	94 ~ 1850	217 ~ 1644	220	293 ~ 1850

Therefore, as long as the critical speed of the frame resonance is increased to the maximum speed of the car (set speed of 110km/h), it can effectively prevent the frame resonance caused by the uneven road surface. Assuming that the wavelength of road roughness is the minimum value of 0.32m, the minimum natural frequency of the frame structure is taken as:

$$\mathrm{f}_{\mathrm{min}} = \frac{\mathrm{v}_{\mathrm{max}}}{3.6\mathrm{L}_{\mathrm{min}}} = \frac{110}{3.6\times0.32} = 95.486(\mathrm{HZ})$$

From the results of frame analysis, it can be known that the seventh mode frequency of the frame (the first six are rigid body modes) is 98.539 Hz, and according to the evaluation principle, it can be known that the resonance of the frame caused by road excitation is not very likely (Figs. 13, 14, 15 and 16).

After the modal solution is completed, the random vibration analysis is carried out. The boundary conditions are fixed constraint, that is, the hard point of the suspension is constrained. The direction is set to Z (the vertical vibration direction of the frame), the density spectrum is set to 10, and the maximum value of the range should be less than half of the maximum mode, which is set to 160. Due to default to excitation spectrum distribution is gaussian distribution, take its incredible parameters for 1 sigma.

$$\mathcal{U} = \left[\frac{\pi\gamma^2\varphi^2\mathrm{PSD}}{4\varepsilon\mathrm{M}^2(2\pi)^4\mathrm{f}_0^3}\right]^{0.5} \tag{3}$$

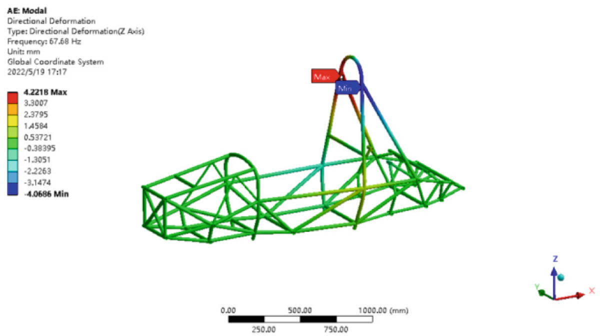

Fig. 13. Maximum deformation in Z direction of random vibration

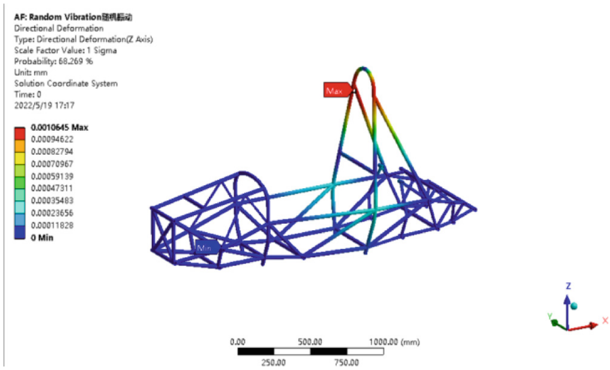

Fig. 14. Maximum deformation of random vibration with 1Sigma confidence

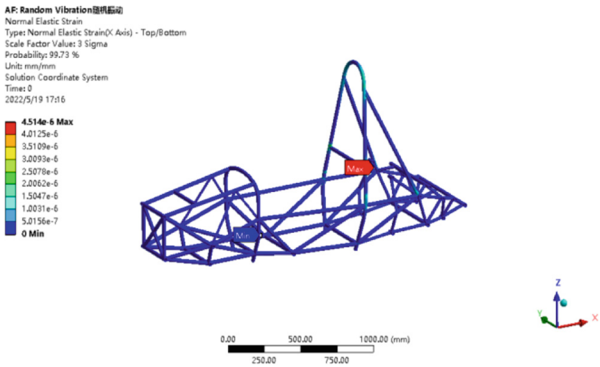

Fig. 15. Maximum deformation of random vibration under 3Sigma

1 response under the position of sigma U is the displacement response of a given location, gamma is given under the order number of modal participation FACTOR (the results of modal PARTIC. FACTOR), phi is a given modal vibration mode (maximum

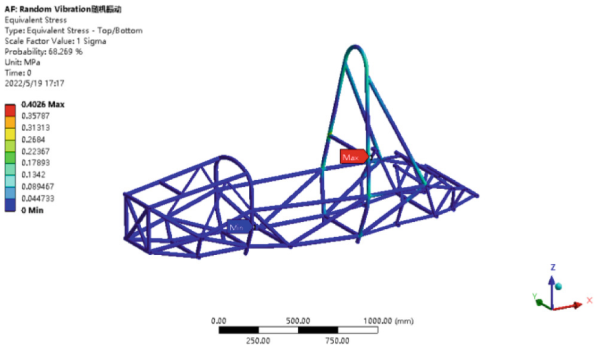

Fig. 16. Effective stress of random vibration at 1Sigma

amplitude, The amplitude of the mode is a relative value, not an absolute value, because the mode mode is a relative quantity), εis damping, ansys defaults to 0.02, M is the generalized mass at a given order mode (for the first order mode, M = 1), f0 is the frequency of the given order mode, and PSD is the power spectral density (where PSD G Acceleration, The calculation method is PSD = G acceleration*9800*9800, in mm^2/(s*Hz)). For the above formula, the values of my simulation are:$\gamma = 64.570$; $\varphi = 4.2218$ mm; $\varepsilon = 0.02$; f0 = 67.68 Hz; M = 1; PSD = 10*9800*9800 Finally calculated u = 0.001876 mm.

The simulation result is 0.0010645 mm, the two are different, but the difference is not big, and on the same order of magnitude, so it is considered that the simulation is more reasonable. Based on the above analysis, after considering the exclusion of objective factors such as suspension dynamic expansion, tire and tire pressure required for dynamic race, the maximum time and maximum load are obtained. The track surface provided by the arena will not cause fatigue damage to the frame, which further verifies the excellent reliability and durability of the frame.

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