# Chapter 4
# Fatigue Durability Analysis of a Frame Based on Multi Body Dynamics

**Qin Zhang, Fengchong Lan, Jiqing Chen and Qinsheng Huang**

**Abstract** A fatigue analysis procedure for a frame based on virtual prototype and multi-body dynamics is established. The first 5 modes frequencies of the finite element model are calculated and verified by experimental test. The frame model is turned into a flexible body. A rigid flexible coupling model is established with suspensions, tires and the flexible frame body. The dynamic response of the model is analyzed in class D road at a uniform motion, the load-time histories of the connected locations are extracted; The S-N curve of material is fitted through the tensile strength limit; combined with the stress distribution of unique load, the dangerous positions and theirs fatigue life are predicted. The result shows that the frame damage hotspots are mainly distributed in the bracket connected with the body, frequency domain energy of connected location load is within 18Hz, the minimum fatigue life mileage is km. The result has a guiding significance for the research of the fatigue durability of vehicle frame.

**Keywords** Fatigue durability · Rigid flexible coupling · Multi body dynamics · Frame · Model analysis

## 4.1 Introduction

The pavement conditions Sport Utility Vehicle (SUV) suffered are more complex than passenger car's, subject to bending, torsion, impact and combinations of them etc. variety of loads, which had a abrupt change and large amplitude during the sport driving. The frame of the body-on-frame absorbs directly impacts came from the road surface, dynamic and alternating loads which are lower than the yield

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strength of the material. Fatigue damage would be also caused by experienced certain loads which are lower than yield strength of material. The frame under such load conditions requires not only rigidity and strength to guarantee driving safety, comfort, usage of parts installed in, but also fatigue life to ensure that fatigue failure does not occur when it is subjected to dynamic loads below the yield strength in the life time. The fatigue durability test has a long cycle, a high cost, and need to be carried out after the assembly of prototype. The cost of rectification is high when bugs expose at the prototype. It is necessary to analyze the fatigue life of products for the specific pavement spectrum it works at the beginning of design.

Computer aided design has been widely used in structural strength and stiffness analysis, modal analysis and collision analysis, and the results are in good agreement with experiment. Fan [1] conduced a topology optimization of multi stiffness for a bus frame owned three section; Yang [2] improved the elastic modal frequencies at low order and mode shapes of a box beam frame in the heavy mining truck with finite element method; Liu [3] used compromise programming method to establish a topology optimization model for multi-objective to optimize the stiffness and vibration frequency of a frame; Li [4] conducted a 100% positive impact simulation with the frame filled of aluminum foam materials, and improved the collision performance of the frame; Xin [5] conducted a 40% offset impact simulation of a frame and optimized its collision safety. Many scholars also introduced the actual traffic conditions as the boundary into computer design. Chen [6] measured the strain time history of corn harvester at the chassis frame on the working conditions; Huang [7] predicted the fatigue life of all terrain SUV frame in rigid flexible coupling with the cobbles as the boundary conditions. The fatigue prediction of a special vehicle frame under impact load was studied by Wang et al. [8]. However, the obtain of the actual road conditions and the working circumstance are more difficult, all conditions in the life time can not be encountered even in the professional test field. Therefore, it is necessary to apply the virtual technology of fatigue durability analysis to the frame.

The substructure mode synthesis method is used to established the frame of rigid flexible coupling model, the stress spectrum of connected location is obtained though the excitation of the surface roughness spectrum at the multi body dynamics model; combined with the stress distribution of unit load and fitting S-N curve, fatigue life of the frame is predicted.

## 4.2 Establishment and Validation of Finite Element Model of Frame

### *4.2.1 The Establishment of the Finite Element Model of Frame*

Frame adopts edge beam edge structure, the section of side rails is square welded by two slot shaped opening, in the anterior segment of the side rails is design with reinforcing plate in order to increase the strength and stiffness. Frame adopts SAPH440 steel plate, connected by welded and bolt. The elastic modulus E of the material is 2.1 × 105 MPa, Poisson's ratio μ is 0.3, the density is $7.85 \times 10^{-6}$ kg/mm$^3$, yield strength $\sigma_s$ is 305 Mpa, tensile strength $\sigma b$ is 440 MPa. Frame was meshed by the quadrilateral shell element, with the size 10 mm. The welding and bolt connection are simulated by Rbe2 element. The total number of element is 126 239, the total number of nodes is 128 891 and the finite element model of the frame is shown in Fig. 4.1.

### *4.2.2 Modal Analysis and Verification of Vehicle Frame*

Modal analysis is the process of distinguishing the characteristics of the structure, that is to identify the inherent characteristics, frequency, damping and mode. Those can be obtained by calculating modal analysis and experimental modal analysis.

Calculating modal analysis is done by solving the motion equations of a multi degree system

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{\mathrm{x}\} = \{F\} \tag{4.1}$$

where $\{\ddot{x}\}$ is the acceleration, $\{\dot{x}\}$ is the velocity, $\{x\}$ is the displacement; [M] is the mass matrix, [C] is the damping matrix, [K] is the stiffness matrix, {F} for external load vector.

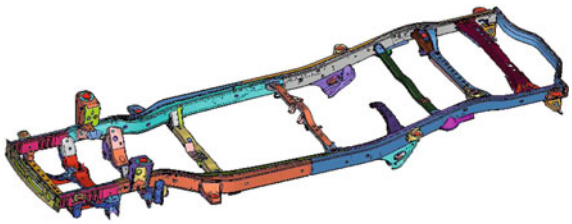


**Fig. 4.1** Finite element model of frame

When the system damping is very small, it can be ignored; thus, it is equivalent to the solution of the characteristic equation:

$$\left([K] - \omega^2[M]\right)\{\varphi\} = 0 \quad (4.2)$$

The Block Lanszos method combines the transform method and the tracing method, with a set of vectors to achieve the recursive calculation, is very efficiency for solving the generalized eigenvalue of large sparse matrix. The Block Lanszos is used to calculate the first 5 order modal frequencies for non rigid.

The modal test uses the free mode, frame are connected by four rubber ropes in the nodes, hung in the customized cradle. The frame is in free state and maintained level, as shown in Fig. 4.2. The natural frequency of the rubber rope suspension system is within 2 Hz. The measuring point is selected to avoid the node of the frame, which is symmetrical and evenly distributed. The modal test uses the free modal test method combined from excitation from single point and picking vibration up from multi-points. That is, An excitation position is selected to apply excitation at frame by a test hammer. At the same time, sensors are used to measure the frequency response information at one excitation. During the measurement, observed the coherence function and frequency response function synchronous, remeasure the points when the coherence function disordered.

As shown in Table 4.1 and Fig. 4.3, the natural frequency of the frame modal test mode and finite element calculated mode are in good agreement, the first 5 order natural frequency error value is less than 5%, the two modes are basically the same, there is no obvious difference in the amplitude, visible in the finite element model has a high precision, can be used as a follow-up study.

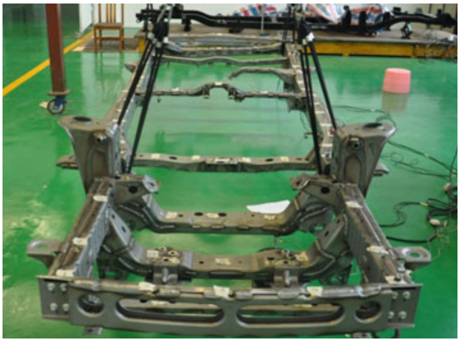
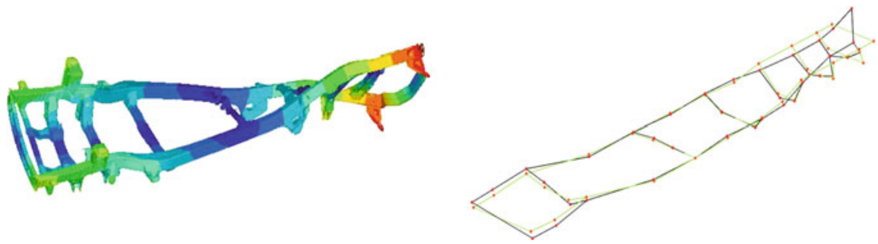


**Fig. 4.2** Hung of frame



a) First order calculated modal b) First order experimental modal

**Fig. 4.3** Comparison of frame first order calculated modal and experimental modal

**Table 4.1** Comparison of calculated and experimental results

| Modal order | Modal description | Calculated (Hz) | Experimental (Hz) | Deviation (%) |
|---|---|---|---|---|
| 1 | First order torsion | 23.39 | 24.37 | 4.02 |
| 2 | First order bend | 29.02 | 30.19 | 3.88 |
| 3 | First order cross bend | 34.67 | 35.06 | 1.11 |
| 4 | Second order torsion | 48.99 | 50.75 | 3.47 |
| 5 | Local deformation | 52.73 | 55.31 | 4.67 |

## 4.3 Vehicle Dynamics Analysis

### *4.3.1 The Frame of the Rigid Flexible Coupling Multi-Body Dynamics Model*

The frame suffer most of loads came from the engine, gearbox, clutch, suspensions and bridge which installed on it as a non bearing body, complexity of force amplitude in space and frequency domain decided it is need to be studied as a flexible body.

Considering the effect of elastic and using the Component Model Synthesis, the frame is tuned into a Flexible body. Then establishes the multi-body rigid flexible coupling model of frame-mass-suspension. It considers the connection points between frame and suspensions and body as out nodes, determined the modal degrees of freedom after reduced, according to the Component Mode Synthesis method, used the modal neutral file which contained the frame modal information to generated the frame into a flexible body. Front suspension is double wishbone suspension, rear suspension is five link non independent suspension, the coefficient of suspension mass distribution $\varepsilon = 1$, the body mass is simplified to the 8 connection positions between frame and the body. The vehicle model fitted with a tire model is shown in Fig. 4.4.

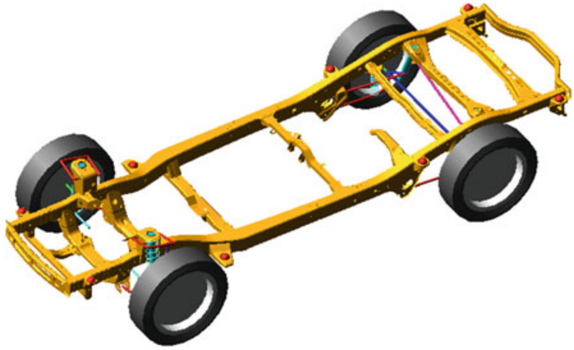


**Fig. 4.4** Vehicle dynamics model

## 4.3.2 *Establishment of Road Spectrum*

The evenness of the road surface generates incentives to the driving vehicle, creates dynamic loads along the tire and suspension system to the frame. The reason for the excitation is the deviation between the ideal reference plane and the road surface in the direction of travel, which is usually called the road roughness function, and can be described by the power spectral density $G_q(n)$ of the vertical displacement.

$$G_q(n) = G_q(n_0)\left(\frac{n}{n_0}\right)^{-w} \tag{4.3}$$

where, n is the spatial frequency, $n = 1/\lambda$, $n_0$ is the relative spatial frequency, $n_0 = 0.1\ \mathrm{m}^{-1}$; $G_q(n_0)$ is the coefficient of road roughness; W is frequency index.

Pavement roughness can be divided into 8 stages according to the vertical displacement power spectral density when $G_q(n)$ frequency index W = 2, as shown in Table 4.2. In order to make the research more general, this paper establishes the virtual pavement spectrum according to the D level road surface standard.

**Table 4.2** Standard for classification of road roughness

| Pavement class | $G_q(n_0)/(10^{-6}\ \mathrm{m}^3)$ | $\sigma_q/(10^{-3}\mathrm{m}^3)(0.011\mathrm{m}^{-1} < n < 2.83\mathrm{m}^{-1})$ |
|---|---|---|
| | Geometric mean | Geometric mean |
| A | 16 | 3.81 |
| B | 64 | 7.61 |
| C | 256 | 15.23 |
| D | 1024 | 30.45 |
| E | 4096 | 60.90 |
| F | 16,384 | 121.80 |
| G | 65,536 | 243.61 |
| H | 262,144 | 487.22 |

### 4.3.3 Vehicle Dynamic Response

Static equilibrium of rigid flexible coupling dynamic model is analyzed firstly, by setting in gravity only to reach a stationary equilibrium. Analyzed the dynamic response in the road roughness of 0, that is the ideal flat surface. The speed of vehicle is 36 km/h. The vehicle can not be immediately changed from static to uniform motion state, there is a transition process during the acceleration, that is, uniform acceleration movement of the stage. The change of vehicle speed in 0–30 s is shown in Fig. 4.5. The speed of vehicle had achieved stability within 2.5 s. According to the left spring force of front suspension showed in Fig. 4.6, the dynamic force of the components achieved stability when the time is 7 s. The accelerated process will affect the dynamic response between 0 and 7 s. Therefore, the analysis of the dynamic load of the frame under this condition should be carried out after 7 s when the response got stable. In this paper, the dynamic loads of some interesting locations is analyzed in 10–30 s.

The force between the parts and the frame are three force components: $F_x$, $F_y$ and $F_z$ as well as three torque components: $T_x$, $T_y$ and $T_z$, that is, the force and

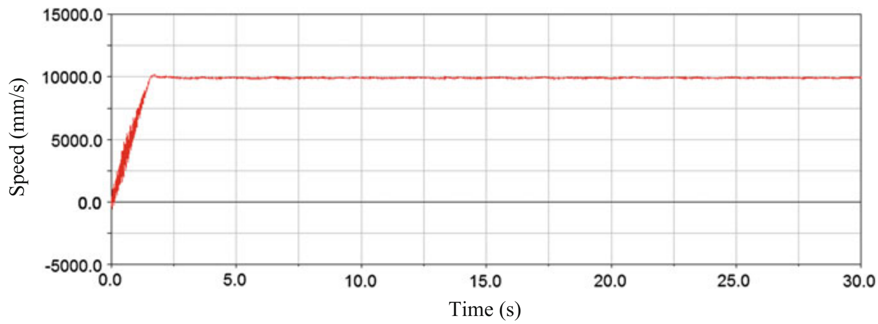



**Fig. 4.5** Vehicle speed

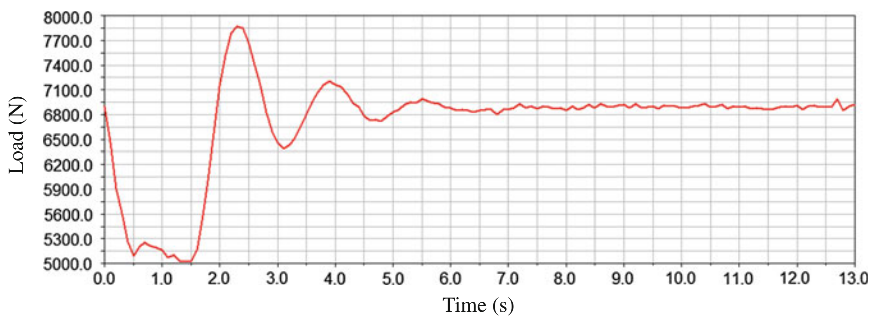



**Fig. 4.6** Z direction force at the top of *left* spring of front suspension on ideal road

**Table 4.3** Input load components for fatigue analysis

| Connected location | Number | Load component | Total load |
|---|---|---|---|
| To BIW | 8 | $Fz$ | 8 |
| To spring damper of front suspension | 2 | $Fz$ | 2 |
| To rear suspension spring | 2 | $Fz$ | 2 |
| To rear suspension damper | 2 | $Fy$, $Fz$ | 4 |

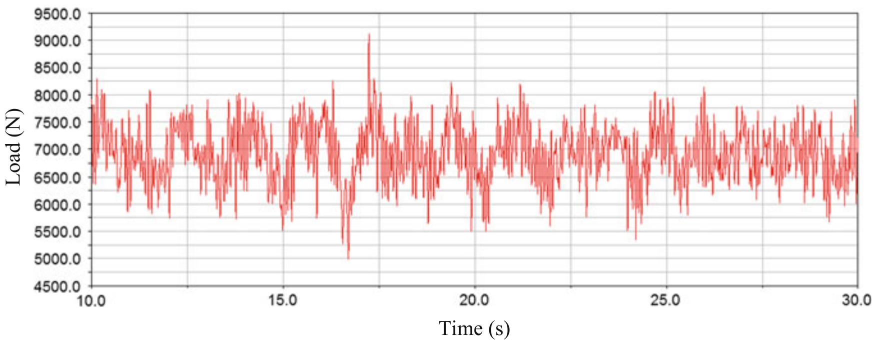



**Fig. 4.7** The Z direction load at the *top* of left spring damper in the front suspension

torque in the direction of the three coordinate axes. Because the torque component are smaller compared with the force component, which are bigger obviously and whose influence on the durability were greater. In order to improve the computational efficiency without sacrificing the accuracy of the simulation, the effect of the torque component which had smaller amplitude on the frame fatigue can be neglected. 16 load components are selected as the load used for calculating the fatigue durability of the frame, as shown in Table 4.3, which suffered more severe condition.

The Z direction load at the top of left spring damper in the front suspension in 10–30 s is shown in Fig. 4.7.

## 4.4 Fatigue Life Analysis

### *4.4.1 Fatigue Analysis Method*

According to the difference between structure natural frequency and the excitation frequency, fatigue life analysis methods can be divided into three kinds, respectively is static or quasi-static fatigue analysis method, dynamic fatigue analysis method and vibration fatigue analysis method. When the first order natural frequency of the structure is more than 3 times of the excitation frequency, the static (or quasi-static) fatigue analysis method can get a reasonable result.

According to analyzing the power spectral density of the connection point of the frame, the range of energy distribution of the loads can be obtained. Take the Z direction load at the top of spring shock absorber in the front and left suspension for example, as shown in Fig. 4.8. The frequency is mainly distributed within 0–18 Hz, and the most are the low frequencies of 1 Hz, and first-order natural frequency of the frame is 23.39 Hz. Therefore the (quasi) static fatigue analysis method must be choose to predict the fatigue life of the frame.

The fatigue characteristic curve of material is the basis of the research on fatigue life, and it is generally classified into E-N and S-N curve. The S-N curve is needed to be used to predict the fatigue life for Static (quasi) fatigue analysis methods.

The frame materials is SAPH440, and its physical character is as follow: tensile strength of $\sigma_u = 440$ MPa, stress ratio $r = -1$, the logarithmic standard deviation is 0.1. The S-N curve of SAPH440 is fitted as shown in Fig. 4.9.

## *4.4.2 Fatigue Life Analysis*

Under normal circumstances, the S-N curve is measured under symmetrical cyclic stress(whose $r = -1, \sigma_m = 0$), but in fact most force of the parts in structure does not meet the condition, the amplitude of the load are random values owned statistical features usually. Experiments show that the average and amplitude of the stress are related to the cycle times N, and this relationship is first proposed by Haigh. Under the same stress amplitude, the fatigue life of components or materials decreases when the average stress is tensile, and increases when the average stress is compression. The stress cyclic block of a non-zero mean value can be obtained

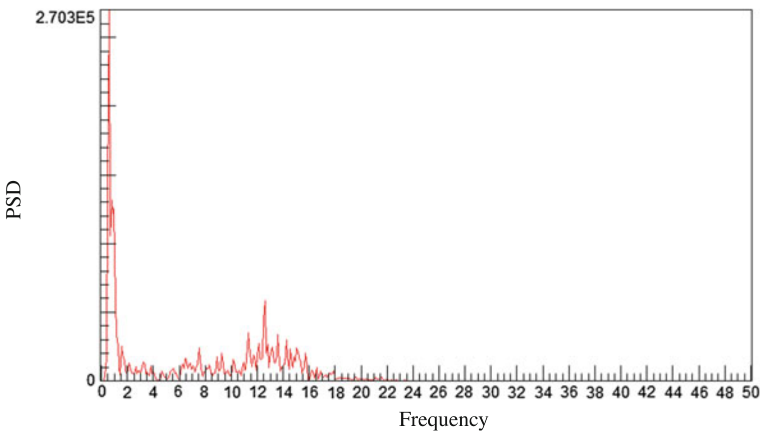



**Fig. 4.8** Front suspension *left* spring damper Z to load power spectrum density

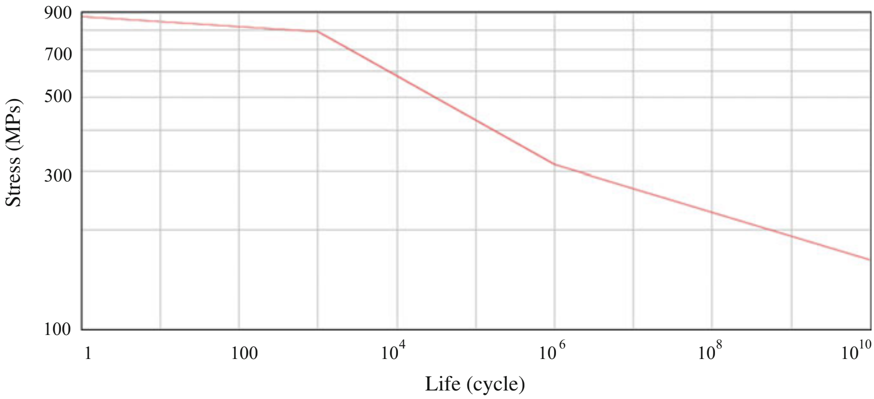



**Fig. 4.9** The S-N curve of SAPH440

by counting the load spectrum. But the S-N curve is obtained in the experiment of zero stress average. So the blocks must be converted to the stress cycle which the average is zero and the stress ratio is minus 1 to calculate the fatigue life with the S-N curve. In this paper, Good-man curve is selected to modify the mean of stress.

Miner linear cumulative damage theory is used to accumulate the damage caused by the stress cycle at all levels, and the fatigue life of each grids is obtained with the failure criterion. According to the Miner fatigue cumulative damage theory, under the action of a constant amplitude stress level S, the number of cycle until failure is N, the damage caused by N cycles is calculated by the follow:

$$\mathrm{D} = \mathrm{n}/\mathrm{N} \tag{4.4}$$

Under the action of different stress level $\mathrm{S_i}$, the damage of $\mathrm{n_i}$ cycle is

$$\mathrm{D_i} = \mathrm{n_i}/\mathrm{N_i} \tag{4.5}$$

When the cumulative damage is reached to 1, that is, the

$$D = \sum_{i=1}^{l} \frac{n_i}{N_i} = 1 \tag{4.6}$$

Structure is considered to be destroyed by fatigue. Where $\mathrm{n_i}$ is the number of cycles under $\mathrm{S_i}$, which is determined by the load spectrum; $\mathrm{N_i}$ is the cycle to the destruction of the life cycle under the action of $\mathrm{S_i}$.

The vehicle structure reliability is related to the safety of passengers, so the survival rate is set to 99%. After simulation, the fatigue life cycle of the frame is obtained, and the locations of the dangerous positions owned the least fatigue life are predicted, as shown in Fig. 4.10.

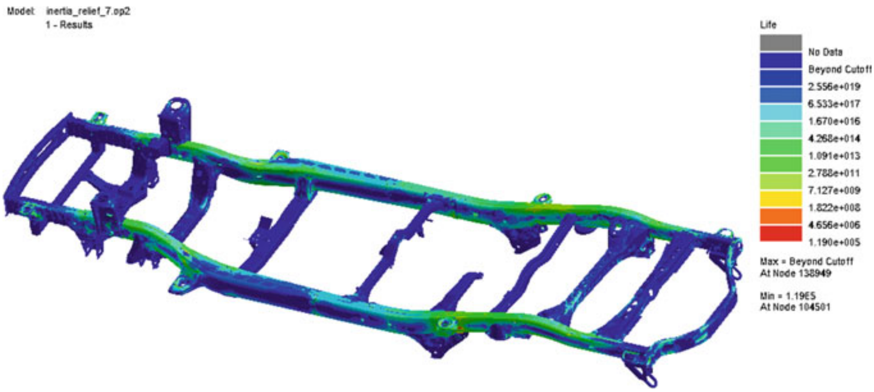



**Fig. 4.10** Logarithmic fatigue life of vehicle frame

Fatigue damage is calculated as the number of cycles to calculate the fatigue

$$D_i = v \times t \times n_i \tag{4.7}$$

where, $D_i$ is the fatigue mileage of node, v is the speed, t is time; $n_i$ is the number of cycles.

**Table 4.4** Fatigue endurance performance analysis of frame structure

| Serial | Node | Location | Damage $10^6$ | Cycle times/times $10^5$ | Fatigue mileages/$10^4$ km |
|---|---|---|---|---|---|
| 1 | 104501 | Nub. 3 rear junction of right side rail and body | 8.40 | 1.19 | 2.38 |
| 2 | 104500 | Nub. 3 front junction of right side rail and body | 5.64 | 1.77 | 3.55 |
| 3 | 103734 | Nub. 3 rear junction of left side rail and body | 2.28 | 4.39 | 8.79 |
| 4 | 47300 | The right of rear cross member | 1.50 | 6.68 | 13.35 |
| 5 | 31074 | Shackle at the junction of left side rail and body | 1.50 | 6.68 | 13.37 |
| 6 | 47303 | The right of rear cross member | 1.42 | 7.04 | 14.09 |
| 7 | 103730 | Nub. 3 front junction of left side rail and body | 0.352 | 0.284 | 56.88 |
| 8 | 83902 | The front at junction of right side rail and first cross member | 0.073 | 0.0137 | 273.80 |
| 9 | 83977 | The rear at junction of right side rail and first cross member | 0.0444 | 0.0225 | 450.80 |
| 10 | 84279 | Nub. 2 front junction of right side rail and body | 0.0321 | 0.0312 | 623.20 |

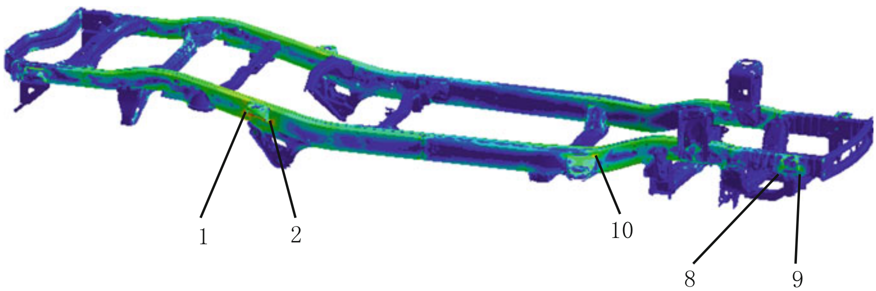



**Fig. 4.11** Distribution of partial dangerous location

The load spectrums are recorded at the location above at the constant 10 m/s speed during 20 s the vehicle drove in the computer analysis, fatigue damage cycle times of node 104501 at the largest region is $1.19 \times 10^5$, the fatigue mileage of dangerous zones is calculated for $2.38 \times 10^4$ km according to formula (4.7). The fatigue mileages of the largest 10 hot spots in fatigue damage can be seen at Table 4.4, some location of those 10 spots are shown at Fig. 4.11. According to the life-cycle analysis results can be found, the less dangerous location in fatigue life are basically lay out at the junctions of trails and the body. The shape changes abruptly and greatly at the junctions of two components, and the stress concentration is prone to appeared there; the junctions of trails and the body bear the loading came from body directly. The stress and fatigue life results of the danger locations can be considered as a guidance for structure and lightweight design in the future.

## 4.5 Conclusion

(1) The finite element model of a SUV frame is established, the first 5 order natural modal frequencies are calculated and compared with the modal test, the vibrations mode are the same, and the deviation of frequencies is acceptable. The accuracy of the finite element model is verified.

(2) According to the dynamics of a rigid flexible coupling system, the frame is translated into a flexible body, multi-body dynamics system coupling with rigid and flexible is established with the combination of the simplified body quality, suspension model and the tire model. Load history of the connected locations are acquired under the excitation of road surface spectrum. This technical method has certain norms and versatility, can be applied further to study the fatigue durability of the frame under different spectrums of road surface.

(3) The corresponding fatigue mileage at the most area of two side rails can arrive at 5 million km according to the fatigue durability analysis. Weight can be

reduced by adjusting the frame structure and reducing the material thickness of the frame in a prescribed mileage. The method above has a certain guiding significance for lightweight frame.

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