Stiffness Parameter Design of Suspension Element of Under-Chassis-Equipment for A Rail Vehicle

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Abstract. According to the frequency configuration requirements of the vibration of railway under-chassis-equipment, the three- dimension stiffness of the suspension elements of under-chassis-equipment is designed based on the static principle and dynamics principle. The design results of the concrete engineering case show that, compared with the design method based on the static principle, the three- dimension stiffness of the suspension elements designed by the dynamic principle design method is more uniform. The frequency and decoupling degree analysis show that the calculation frequency of under-chassis-equipment under the two design methods is basically the same as the predetermined frequency. Compared with the design method based on the static principle, the design method based on the dynamic principle is adopted. The decoupling degree can be kept high, and the coupling vibration of the corresponding vibration mode can be reduced effectively, which can effectively reduce the fatigue damage of the key parts of the hanging element.

1. Introduction

At present, in the process of stiffness design of railway vehicles, the vibration decoupling between the under-chassis-equipment and the vehicle is mainly carried out. Zhang et al [1] found that the under-chassis-equipment operation under severe force, based on vehicle-track coupling dynamics model and mode calculation, select the suspension parameters of suspension elements, the stiffness and damping parameters of the suspended elements are determined according to the characteristics of the rubber and its installation space. Gong et al. [2-4] established a vertical rigid-flexible coupled dynamic model of the railway vehicle including the under-chassis-equipment, designed the parameters of the elements, and analyzed the parameters of the suspension elements based on vibration isolation theory, Running stability and the vibration of the under-chassis-equipment itself are analyzed for different equipment mass and suspension position on the vehicle. However, in the design of the under-chassis-equipment parameters of the railway vehicle, the under-chassis-equipment is regarded as the single-degree-of-freedom mass unit. This kind of design method does not take into account the coupling vibration of the under-chassis-equipment due to the equipment eccentricity. In fact, the under-chassis-equipment is generally a large box, or a box containing multiple equipment, so the eccentric state of the under-chassis-equipment is in general. Based on the actual conditions of the project, this paper considers the eccentric state of the under-chassis-equipment in detail, combining with the engineering case and design the three- dimension of the suspension element based on the static and dynamic principles

respectively. The design method can be widely applied to the stiffness design of the under-chassis-equipment of railway vehicles.

2. Design method based on static principle

In the statics, when the under-chassis-equipment is in the eccentric state, still meet the horizontal and vertical torque reciprocal theorem. The eccentric load due to the eccentric state can be calculated reversely, and the stiffness of the suspension element is designed. Fig. 1 is a top view of a under-chassis-equipment and suspension elements, and a positive direction is defined as a bit having a small number and a L-side being a 1-bit side.
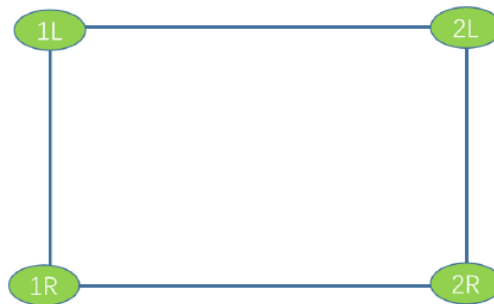

Figure 1. Under-chassis-equipment and its suspension elements of the top view

Based on the principle of static design method to design the three- dimension stiffness of the suspension elements, the general requirements with the following design parameters: Equipment total mass m_T, the number of suspension elements n, equipment vertical vibration natural frequency f_z, static and dynamic stiffness ratio R_z, suspension elements from the device centroid vertical distance x_{e1} and x_{e2} (Engineering parameters are generally in the form of longitudinal eccentricity and longitudinal span, can be converted to x_{e1} and x_{e2}), Lateral eccentricity y_e and lateral span y_L, Combined with the characteristics of the rubber element, the lateral stiffness is designed according to 1/3 of the vertical stiffness. Considering the general existence of the stopper device in the longitudinal direction of the under-chassis-equipment, the longitudinal stiffness is designed according to double of the vertical stiffness.

According to the stiffness design method of the traditional rubber element, the natural frequency of the vertical vibration of the equipment is related to the vertical dynamic and static stiffness of each suspension element as follows.

$$k_{dz} = m_T (2\pi f_z)^2 / n \tag{1}$$

$$k_{stz} = k_{dz} / R_z \tag{2}$$

The vertical average static load of each suspension component is:

$$F_{stz} = m_T * 9.81 / n \tag{3}$$

When the under-chassis-equipment is under the eccentric state, as a result of vertical eccentric and lateral eccentric load are:

$$F_{exz} = \frac{abs(abs(x_{e1}) - abs(x_{e2}))}{abs(x_{e1}) + abs(x_{e2})} * F_{stz} \tag{4}$$

$$F_{eyz} = \frac{abs(y_e)}{abs(y_L)} * F_{stz} * 2 \tag{5}$$

Accordingly, the suspension components of the under-chassis-equipment actually bear the vertical static load are:

$$\begin{cases} F_{1Lz} = F_{stz} + F_{exz} - F_{eyz} \\ F_{1Rz} = F_{stz} + F_{exz} + F_{eyz} \\ F_{2Lz} = F_{stz} - F_{exz} - F_{eyz} \\ F_{2Rz} = F_{stz} - F_{exz} + F_{eyz} \end{cases} \tag{6}$$

When the suspension element 1L to withstand the vertical static load is F_{1Lz}, the vertical dynamic stiffness is:

$$k_{dz1L} = F_{1Lz} / F_{stz} \square k_{dz} \tag{7}$$

In the same way, the vertical dynamic stiffness of suspension elements 1R, 2L, 2R satisfying the reciprocal theorem of moment can be obtained. The lateral and longitudinal dynamic stiffness k_{dz1R}, k_{dz2L}, k_{dz2R} of each suspension element are designed according to 1/3 and 2 times of its vertical stiffness respectively.

3. Design method based on dynamic principle

According to the physical model of the under-chassis-equipment studied in this paper, the dynamic model of the under-chassis-equipment can be established as shown in Fig.2. Considering the under-chassis-equipment as a rigid body with 6 degrees of freedom, the suspension element is simplified as a spring with three-dimensional stiffness and damping, set the mass center of the under-chassis-equipment as the coordinate origin, where the positive direction of x and y are defined as 1-bit terminal and 1-bit side.
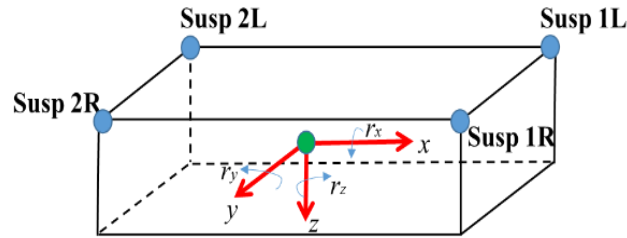

Figure 2. Dynamic model of the under-chassis-equipment

According to the dynamic model of the under-chassis-equipment, the sixth order mode vector and the modal frequency are solved.

$$\boldsymbol{KT}_j = \lambda_j \boldsymbol{MT}_j, \quad j = 1, 2, 3, 4, 5, 6 \tag{8}$$

The j_{th} vibration energy distribution matrix of under-chassis-equipment is defined as [5-6]:

$$\boldsymbol{EM}(p,k) = f_j^2 \boldsymbol{M}(p,k) \boldsymbol{T}_j^p \boldsymbol{T}_j^k / 2 \tag{9}$$

The energy allocation of the j_{th} mode is:

$$\boldsymbol{EP}(k,j) = \frac{\sum_{p=1}^{6} \boldsymbol{EM}(p,k)}{\sum_{k=1}^{6} \sum_{p=1}^{6} \boldsymbol{EM}(p,k)} \tag{10}$$

The basic principle of the design method based on the dynamic principle is to make the vibration of the under-chassis-equipment independent of the design frequency. For any group of vectors $X = \begin{bmatrix} k_{x1} & k_{x2} & k_{x3} & k_{x4} & k_{y1} & k_{y2} & k_{y3} & k_{y4} & k_{z1} & k_{z2} & k_{z3} & k_{z4} \end{bmatrix}^{\mathrm{T}}$ composed of the three-dimensional stiffness of the rubber elements, the decoupling degree and the modal frequency can be calculated

$$\min f(X) = \sum_{j=1}^{6} \alpha_j (1 - \max_{k=1,2,\ldots 6} EP(k,j))^2 + \sum_{j=1}^{6} \beta_j ((f_j(X) - f_{jo}) / f_{jo})^2$$

$$s.t. \ X^L \le X \le X^U$$

$$f_j^L \le f_j(X) \le f_j^U \ , \qquad j = 1, 2, \ldots 6$$

$$(11)$$

As genetic algorithm has the characteristics of strong searching ability and good ergodicity [7-8], this paper will use genetic algorithm to solve (11).

4. Case study

According to a factory to provide under-chassis-equipment parameters and frequency configuration requirements, the under-chassis-equipment is suspended by four suspension elements, vertical vibration design frequency is 8Hz, based on the principle of static design, the three-dimensional dynamic stiffness of the suspension elements is shown in Table 1. The lateral stiffness is designed according to 1/3 of the vertical stiffness, and the longitudinal stiffness is twice as high as the vertical stiffness.

Table 1. The three-dimension dynamic stiffness of the suspension elements designed according to static principle

Suspension element location	three- dimension stiffness		
	Kx(N/mm)	Ky (N/mm)	Kz (N/mm)
1L	1304	217	652
1R	1410	235	705
2L	1022	170	511
2R	1128	188	564

Compared with the design method of the static principle, the design method based on the dynamic principle supplements the three-direction rotational inertia of the under-chassis-equipment. The coordinate parameter of the suspension element changes from two-dimensional to three-dimension, and the stiffness ratio is to give a certain numerical range rather than specific values. Table 2 is the three-dimensional dynamic stiffness of the suspension elements designed based on the dynamic principle. From the numerical results, it can be seen that, compared with the design method based on the static principle, the three-way dynamic stiffness of the suspension elements designed by the dynamic principle is more uniform. The next section of this paper will analyze the coupling vibration of the under-chassis-equipment when the two design methods were taken.

Table 2. The three-dimension dynamic stiffness of the suspension elements designed according to dynamic principle

Suspension element location	three- dimension stiffness		
	Kx (N/m)	Ky (N/m)	Kz (N/m)
1L	5.419e5	2.877e5	6.706e5
1R	5.373e5	2.024e5	6.868e5
2L	5.417e5	2.099e5	5.737e5
2R	5.511e5	1.885e5	5.690e5

5. Coupling vibration evaluation of the under-chassis-equipment

According to the analysis of the decoupling degree, the results of Modal frequency and decoupling degree based on two principles are shown in Table 3 and Table 4, respectively. From Table 3 and Table 4 can be seen the float vibration mode frequencies were 8.00Hz and 8.09Hz, and calculation results are in accordance with the design frequency. From Table 3, it can be seen that the design method based on the principle of static does not fully consider the coupling vibration of each degree of freedom of the under-chassis-equipment. The decoupling degree of the second and fourth order modes is 68.8 %, Nodal and telescopic vibration are serious. From Table 4, it can be seen that the decoupling degree of the vibration modes of the second and fourth order increases by 86.6% when the design method based on the dynamic principle is adopted, and the nodal and telescopic vibration are effectively reduced, thereby effectively reducing the key parts of the suspension elements of fatigue damage.

Table 3. Modal frequencies and decoupling degree based on static principle

Order		1	2	3	4	5	6
Natural frequency/Hz		4.56	7.10	8.00	17.28	21.00	29.72
Distribution of the energy %	x	0.0	**68.8**	0.0	31.2	0.0	0.0
	y	**99.9**	0.0	0.0	0.0	0.1	0.0
	z	0.0	0.0	**100.0**	0.0	0.0	0.0
	α	0.1	0.0	0.0	0.0	**99.9**	0.0
	β	0.0	31.2	0.0	**68.8**	0.0	0.0
	γ	0.4	0.0	0.0	0.0	0.0	**100.0**

Table 4. Modal frequencies and decoupling degree based on dynamic principle

Order		1	2	3	4	5	6
Natural frequency/Hz		4.77	6.08	8.09	13.71	20.25	21.34
Distribution of the energy %	x	0.0	**86.6**	0.2	13.3	0.0	0.0
	y	**99.9**	0.0	0.0	0.0	0.0	0.1
	z	0.0	0.1	**99.8**	0.0	0.0	0.0
	α	0.1	0.0	0.0	0.0	0.0	**99.9**
	β	0.0	13.3	0.0	**86.7**	0.1	0.0
	γ	0.4	0.0	0.0	0.0	**99.9**	0.0

6. Conclusions

(1) Based on the principle of static mechanics and dynamics, according to the design requirements, respectively, to design the three-dimension stiffness of the suspension elements of the under-chassis-equipment. From the numerical results, the two design methods can be applied to specific engineering

cases, compared with the design method based on the static principle, the three-dimension Stiffness is more uniform when dynamic principle is adopted.

(2) Based on the dynamic equation of the under-chassis-equipment and the calculation of the vibration frequencies and the decoupling degree, the coupling vibration of the under-chassis-equipment of two design methods is evaluated. The results show that the vibration frequencies of the under-chassis-equipment are close to the design frequencies. Compared with the design method based on the static principle, the design method based on the dynamic principle is adopted. The coupling vibration of the corresponding vibration mode can be reduced effectively, and the fatigue damage of the key parts of the suspension elements can be effectively reduced.

7. References

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