International Journal of Research in Engineering and Science (IJRES)
ISSN (Online): 2320-9364, ISSN (Print): 2320-9356
www.ijres.org Volume 3 Issue 11 ǁ November. 2015 ǁ PP.12-17

Study on Vibration Reduction Effect of the Rail Vehicle with Axle Dynamic Vibration Absorber

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ABSTRACT*:* *To reduce the vertical vibration of the vehicle, the vehicle model including the axle dynamic vibration absorber is established, and the vertical vibration acceleration spectrum of the wheel, the steering frame and the body is obtained by numerical simulation. The vibration reduction effect of the dynamic vibration absorber is analyzed. The results show that the axle dynamic vibration absorber not only has a good vibration reduction effect on the wheel, but also has an important effect on the vertical vibration of the bogie and the vehicle body.*
Keywords*:* *Dynamic vibration absorber, Axle, Vertical vibration, Mass ratio, Damping*

I. INTRODUCTION

The vibration of rail vehicle may cause wheel, bogie and other components cracks and other damages, which may affect the safe operation of the vehicle. V The wheel-rail interaction leads to vehicle vibration and passes primary and second suspension system to the bogie and the car body. Therefore, the research on vehicle vibration damping device is very necessary.

At present, many scholars have studied the vehicle model, Zhai Wan-ming[1] established a vehicle/track coupled mode. The vehicle was simplified as a multi-degree of freedom model, the track is considered as a Timoshenko beam, Vehicle and track model are considered more complete; Zhao Honglun [2] established a finite element model of vehicle systems for analysis, and provided a reference for the development of rail vehicles. In addition, in order to reduce the vertical vibration of the vehicle, many scholars have established a vehicle model with dynamic vibration absorber, Foo [3] proposed a dynamic vibration absorber which is installed at the bottom of the car body, The results show that vibration can be reduced in some parts of the vehicle body; Zeng Jing [4] established the vehicle model installation dynamic vibration absorber and analysis its damping, The research results show that the damping performance of the vibration absorbers is very good in the low frequency and at speed greater than 150km/h; Zhou Jingsong [5-6]established the vehicle model with dynamic vibration absorber, and introduced the vehicle running stability index to discuss the damping performance of vehicle dynamic vibration absorber, obtain that dynamic vibration absorber can make the running stability of vehicles reached excellent, and the higher its quality the better its damping performance; Tomioka [7]considered the bogie as a dynamic vibration absorber. The research results show that it can significantly reduce the bending vibration of car body. In general, the vehicle model is becoming more and more complex, and the results are more realistic. Moreover, the dynamic vibration absorber is seemed as a an effective device for vehicle vibration reduction.

Therefore, according to Lagrange equation, in order to reduce the vertical vibration of the vehicle, we establish vehicle dynamics model with an axle dynamic vibration absorber, in which the dynamic vibration absorber mounted on the axle portion of the system by reducing wheel vibration throughout the vehicle to reduce the effect of vibration. According to the obtained vehicle acceleration power spectral density of the components, vehicle dynamic vibration absorber damping effect is analysis carefully.

II. WHEEL MODEL

The dynamic vibration absorber is mounted on the axle, as shown in Fig. 1, the axle wrap a layer of shock-absorbing materials and vibration absorber simplified model to simplify mass, spring and damping is shown in Fig. 2. Model has a body of ups and downs and nod (two degrees of freedom), front and rear bogies of ups and downs and nod (four degrees of freedom), four wheels drifting (four degrees of freedom) and dynamic vibration absorber ups and downs (one degree of freedom) simplified model fourteen degrees of freedom.

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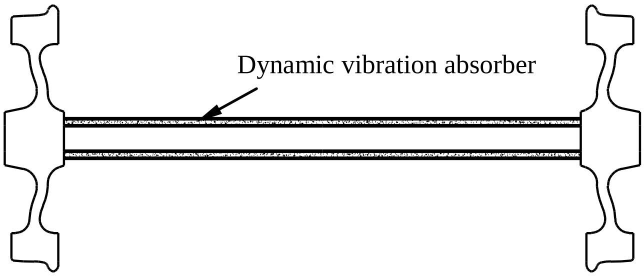


Fig.1 Wheel of dynamic vibration absorber schematic

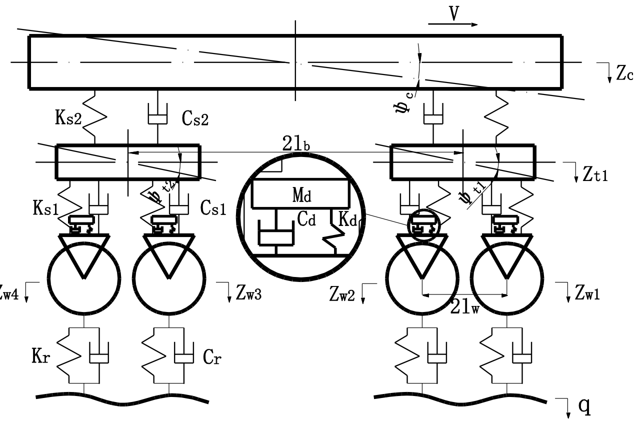


Fig.2 Vehicle model including the dynamic vibration absorber of a wheel

In Fig. 2, the vertical displacement of the vehicle to the right, ， Z_c、 Z_{t1}、 Z_{t2} respectively body, front and rear bogie vertical displacement, φ_c、 φ_{t1}、 φ_{t2} respectively front and rear bogie nod corner, Z_{w1}、 Z_{w2}、 Z_{w3}、 Z_{w4} respectively for the first one, two, three and four of the vertical displacement.

According to Lagrange equation for vehicle dynamics model is derived and its kinetic equation written in compact form as follows:

$$[M]\{\ddot{Z}\}+[C]\{\dot{Z}\}+[K]\{Z\}=[F], \tag{1}$$

In the formula: $[Z]$ is the displacement matrix, $[M]$ is the mass matrix, $[K]$ is the stiffness matrix, $[C]$ is the damping matrix, $[F]$ is the force matrix for the wheel. In the matrix $\{Z\}$, Z_{d1}、 Z_{d2}、 Z_{d3}、 Z_{d4} are the vertical displacement of the dynamic vibration absorber corresponding to the corresponding wheel is represented.

$$[F]=[C_r][\dot{q}]+[K_F][q], \tag{2}$$

$$[C_r]=\begin{bmatrix} 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ C_r&0&0&0\\ 0&C_r&0&0\\ 0&0&C_r&0\\ 0&0&0&C_r\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0 \end{bmatrix} [K_r]=\begin{bmatrix} 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ K_r&0&0&0\\ 0&K_r&0&0\\ 0&0&K_r&0\\ 0&0&0&K_r\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0\\ 0&0&0&0 \end{bmatrix}, \tag{3}$$

III. MODEL SOLVING

The Fourier transform (1) is performed in the frequency domain:

$$\left[H(\omega)\right]=\left(-\omega^2[M]+j\omega[C]+[K]\right)^{-1}[F], \tag{4}$$

Since the excitation of each wheel has a certain delay, the input of the three wheel is considered as the input of the first wheel.
Wheels by incentive matrix:

$$\left[q_1(\omega) \quad q_2(\omega) \quad q_3(\omega) \quad q_4(\omega)\right]^T=[Q]q_1(\omega), \tag{5}$$

In the formula: $q_1(\omega)$ 、 $q_2(\omega)$ 、 $q_3(\omega)$ 、 $q_4(\omega)$ respectively, the excitation of the four wheels in the frequency domain.
Including:

$$[Q]=\left[1 \quad e^{-j\omega T_2} \quad e^{-j\omega T_3} \quad e^{-j\omega T_4}\right]^T, \tag{6}$$

In the formula $T_2=2l_t/u$， $T_3=2l_c/u$， $T_4=2(l_t+l_c)/u$.
It can be obtained that the frequency response matrix of each part of the vehicle is not smooth, and each element of the matrix is the frequency response function of the displacement of the vehicle.

$$\left[h(\omega)\right]=\left[H(\omega)\right][Q]=\left[h_1(\omega) \quad \cdots \quad h_{14}(\omega)\right]^T, \tag{7}$$

According to the frequency response function of the displacement, the acceleration response function can be obtained.

$$h(\omega)_{\ddot{Z}_i-q_1}=-\omega^2\frac{Z_i(\omega)}{q_1(\omega)}=-\omega^2 h_i(\omega), \tag{8}$$

$$G(\omega)_{\ddot{Z}_i-q_1}=\left|h(\omega)\right|^2_{\ddot{Z}_i-q_1} q_1(\omega), \tag{9}$$

In the formula: $h(\omega)_{\ddot{Z}_i-q_1}$ is the acceleration frequency response function of the corresponding component relative to the irregularity excitation is expressed, $G(\omega)_{\ddot{Z}_i-q_1}$ is the acceleration power spectral density of the corresponding components for the irregularity excitation is indicated.
Although the acceleration power spectral density can be very convenient to analyze the frequency characteristics of vertical vibration of vehicle system, the acceleration of vertical vibration of the vehicle in the frequency range of ω_1 to ω_n frequency range is required:

$$G_i=\left[\int_{\omega_1}^{\omega_n} G_{\ddot{Z}_i}(\omega)d(\omega)\right]^{\frac{1}{2}}, \tag{10}$$

IV. SIMULATION RESULTS

4.1 MODEL PARAMETERS

Based on one type of vehicle as an example, According to the dynamic vibration absorber principle can determine the dynamic vibration absorber absorbing quality and stiffness, the parameters are shown in Table 1, With American class six track irregularity spectrum as incentive of the vehicle vertical vibration model.

Table1 A certain type of vehicle parameters and track parameters

Meaning	Parameter symbol	Value
Mass of vehicle	M_b/kg	4.18×10^4
Car body pitch inertia	J_b/(kg·m2)	1.686×10^6
Mass of bogie	M_t/kg	2.8×10^3
Bogie pitch inertia	J_t/(kg·m2)	1.93×10^3
Secondary spring stiffness per bigie	K_{s2}/(kN·m-1)	6×10^5
Secondary damping per bogie	C_{s2}/(kN·s·m-1)	1.1×10^4
Primary spring stiffness per axle	K_{s1}/(kN·m-1)	1.11×10^6
Primary spring damping per axle	C_{s1}/(kN·s·m-1)	2.4×10^4
Half of bogie centers	l_b/m	9
Half of bogie wheel base	l_w/m	1.25
Mass of Dynamic vibration absorber	M_d/kg	140
Dynamic vibration absorber stiffness	K_d/(kN·m-1)	4×10^5
Dynamic vibration absorber damping	C_d/(kN·s·m-1)	50

4.2 PERFORMANCE ANALYSIS OF DYNAMIC VIBRATION ABSORBER

To simulate the model and obtain the components of the vehicle vertical vibration acceleration power spectral density.

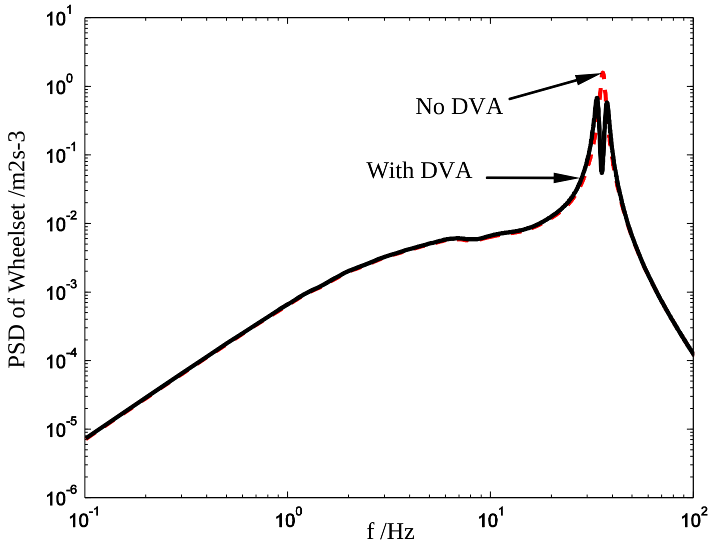
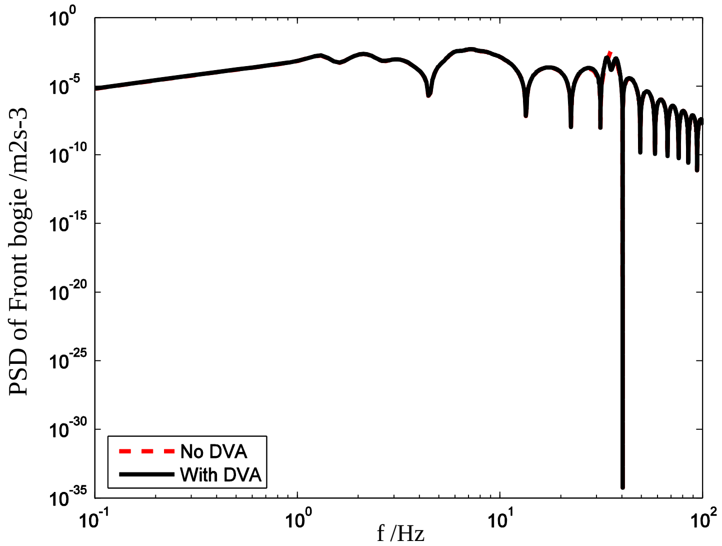


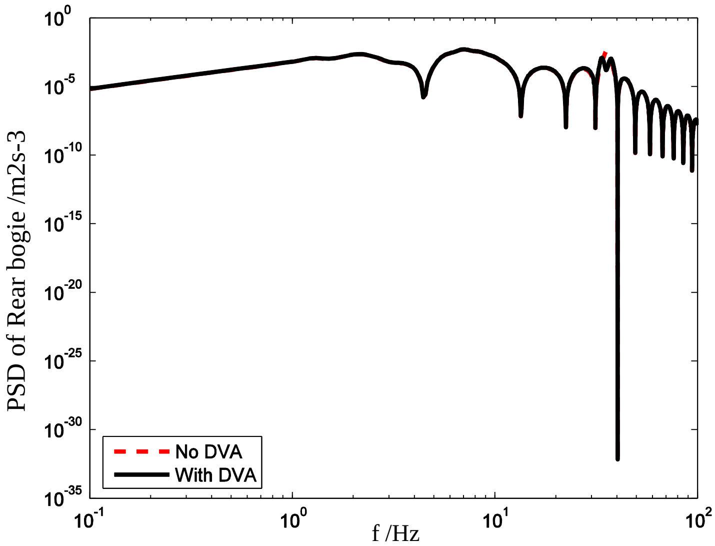
Fig.3 Effect of dynamic vibration absorber on the vibration of wheel set

The wheel vertical acceleration power spectral density curves of vibration is as shown in Fig. 3, It can be seen from Fig. 3 that the wheel's vertical vibration at 40Hz with excitation frequency resonance and reached peak. Therefore, the wheel vibration is most intense, resulting in damage to the wheel strength, seriously affecting the safe operation of the vehicle. After installing the dynamic vibration absorber, vertical vibration of the wheel has been significantly reduced in the resonance frequency.

Wheel vertical vibration is passed primary suspension up to the bogie, The wheel vibration directly affects the vibration situation of the bogie, the Front and rear bogie vertical acceleration power spectral density curves of vibration is as shown in Figure 4, It can be seen from Fig. 4 thatdue to the wheels produce resonance in 40Hz, leading to the bogie at 40Hz also produces vibration peak, dynamic vibration absorber reduce the vertical vibration of wheel at the resonant frequency, Vertical vibration transmitted to the bogie has been weakened, Therefore, the axle dynamic vibration absorber can not only reduce the wheel vertical vibration can also weaken the vertical vibration of the bogie.



(a) Front bogie



(a) Rear bogie

Fig.4 The influence of dynamic vibration absorber on the vibration of bogie

The car body vertical acceleration power spectral density curves of vibration is as shown in Fig. 5, It can be seen from Fig.5 that due to bogie vertical vibration in the vicinity of 40Hz is weakened, so the vertical vibration of the vehicle body vibration at this frequency is also weakened. However, the body's natural frequency of about 1.5Hz, body vertical vibration of the energy is concentrated at low frequencies near the natural frequency. The vibration energy is smaller in the near of 40Hz frequency. Therefore, although the dynamic vibration absorber can reduce vibration here, the effect of reduce the vehicle's vertical vibration is limited

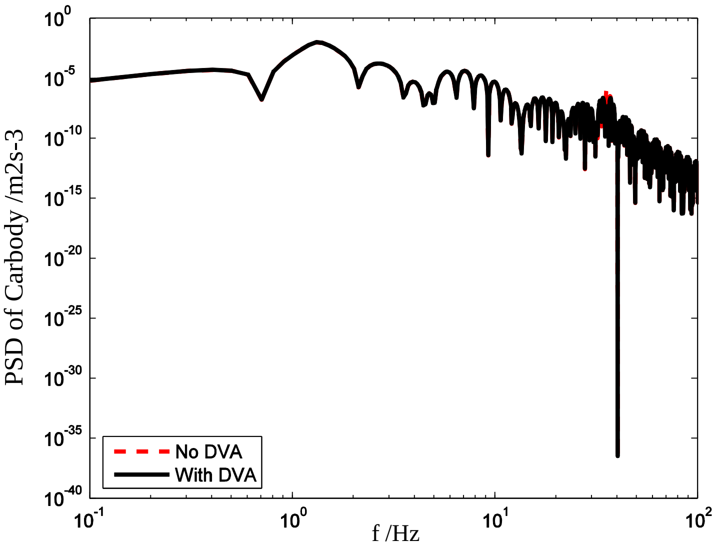


Fig.5 Effect of dynamic vibration absorber on the vibration of car body

The wheel has a good vertical vibration damping effect on the wheel. In order to improve the vibration performance, the mass ratio of the dynamic vibration absorber is improved. Fig. 6 shows the influence of the mass ratio of the dynamic vibration absorber on the vibration absorption performance of the dynamic absorber. The mass ratio of the dynamic vibration absorber can obviously improve the vibration reduction performance. Moreover, the vibration frequency band of the dynamic vibration absorber is gradually widened.

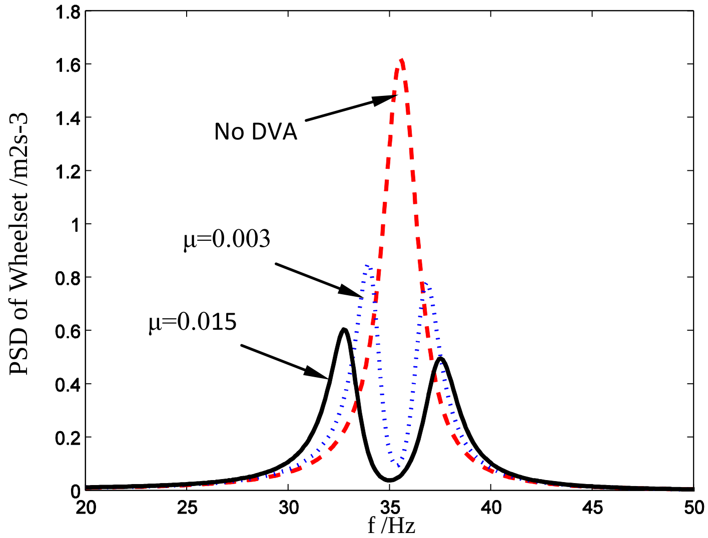


Fig.6 Influence of mass ratio on the vibration reduction effect of dynamic vibration absorber

V. CONCLUSIONS

Axle vibration absorber for vertical vibration of the wheel not only has a good inhibition, but also on the bogie and the body has a important damping effect. For the vertical vibration of the vehicle body is mainly concentrated in the low frequencies, therefore, the axle dynamic vibration damping effect on the body is limited. The vibration reduction effect can be improved by increasing the mass ratio of the axle dynamic vibration absorber. The vibration of the wheel is radiated from the outside; therefore, the vibration of the axle can be controlled by reducing the vertical vibration of the wheel.

VI. ACKNOWLEDGEMENTS

The authors would like to acknowledge the support of National Natural Science Foundation of China (Grant No. 11472176), Natural Science Foundation of Shanghai (Grant No. 15ZR1419200) and Discipline Construction Project for Transportation Engineering (Grant No. 13SC002).

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