Research Article

Vibration Characteristics Analysis of Convalescent-Wheelchair Robots Equipped with Dynamic Absorbers

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To provide a theoretical guidance for the vibration isolation design, the vibration responses characteristics for the convalescent-wheelchair robot with DA (dynamic absorber) undergoing the random road were revealed. Firstly, the vibration model of the convalescent-wheelchair system with DA was created. The frequency response functions of the road excitation velocity to the convalescent acceleration, the wheelchair body acceleration, and the tire dynamic deflection were deduced. Then, the numerical calculation method of the PSD (power spectral density) and the RMS (root mean square) responses were proposed. Thirdly, the vibration isolation performances of the wheelchair robot with DA and without DA were compared. Finally, the sensitivity analysis of the vibration responses to the mass ratios, the damping ratios, and the natural frequencies was carried out to reveal the effects of the parameters on the vibration responses. The results show that the DA can partly suppress the vibration of the convalescent and the wheelchair body, especially in the resonance area of the wheelchair body. However, the DA cannot successfully improve the tire contact behavior.

1. Introduction

Convalescent-wheelchair robots belong to the rehabilitation medical machines [1]. They are important travel tools for convalescents. With the development of the society and the improvement of the living standard, the demand for them is increasing, for example, their ride comfort, their operational reliability, and so on [2]. When a wheelchair robot runs on the random road, its vibration caused by the road irregularity is inevitable. The harmful vibration produces adverse effects for the wheelchair and its users [3–5].

In recent years, many scholars have studied the related vibration problems of wheelchair robots. The damping characteristics of the wheelchair cushion were studied on the basis of the standard ISO 16840-2:2007 in [6]. A new dynamic model for the wheelchair coupled with the user was developed and validated in [7]. The vibration transmission characteristics of wheelchairs and their effects on patients were analyzed in [8]. Hikmawan and Nugraha [9] simulated the vibration of a wheelchair under the sinusoidal excitation

based on a 6 degree of freedom model of the wheelchair user. Wu et al. [10] proposed a new method of evaluating the ride comfort of wheelchair robots based on the Q-learning algorithm, which has a certain reference value for the vibration evaluation of wheelchair robots. Wang et al. [11] established a mathematical model of the wheelchair cushion comfortableness, which provides a reference for the wheelchair design. Miyawaki and Takahashi [12] designed a new wheelchair and analyzed the whole-body vibration of the user sitting on the wheelchair. Su et al. [13] designed a magnetic suspension vibrator to improve the vibration attenuation performance of electric wheelchairs. Ababou et al. [14] developed a test bench to provide harmful vibrations for wheelchairs and the test bench can be applied to analyze the wheelchair transmissibility. Kundu et al. [15] designed a four wheel-driven wheelchair and investigated its sliding and vibration characteristics. Garcia-Mendez et al. [16] estimated the whole-body vibration of the wheelchair users in their communities. Requejo et al. [17] studied the effect of hand-rim wheelchairs equipped with rear

suspension on seat forces. The influences of the user anatomy and wheelchair cushion type on the tissue deformation were analyzed in [18]. The aforementioned studies provide useful references for improving the vibration isolation performances of wheelchair robots. However, there are few studies on the comprehensive performance of wheelchair robots undergoing the random road.

Most of convalescent-wheelchair robots do not have suspension systems due to the limited installation space. Their users' vibration caused by the road excitation is mainly reduced by the tires and the cushion. The DA is a vibration reducing device connected to the vibration system by elastic and damping elements [19, 20]. It occupies a small space and provides a cheap and effective way to attenuate the wheelchair robots vibration.

In the previous work [21], the impact responses and parameters sensitivity of electric wheelchairs were analyzed. In this paper, to provide a theoretical support for the design of convalescent-wheelchair robots, the vibration responses characteristics for the convalescent-wheelchair robot with DA undergoing the random road were revealed.

2. The Vibration Model of the Convalescent-Wheelchair System with DA

In the initial design stage of convalescent-wheelchair robots, the parameters needed to establish accurate vibration models are often unknown. Therefore, to easily guide their design in theory, the convalescent-wheelchair system should be simplified.

In this paper, the cushion and tires are equivalent to a spring-damper, respectively. Moreover, the convalescent is regarded as a rigid body mass. The vibration model of the convalescent-wheelchair system with DA is created, as shown in Figure 1 [21]. In Figure 1, m_2 , m_1 , and m_0 represent the convalescent mass, the wheelchair body mass, and the DA mass, respectively; K_2 , K_1 , and K_0 represent the cushion stiffness, the tire stiffness, and the DA stiffness, respectively; C_2 , C_1 , and C_0 represent the cushion damping, the tire damping, and the DA damping, respectively; z_2 , z_1 , and z_0 represent the convalescent vertical displacement, the wheelchair body vertical displacement, and the DA vertical displacement, respectively; and q represents the road displacement input. To facilitate the comparative analysis, the vibration model of the convalescent-wheelchair system without DA is also given in Figure 2.

According to Newton's second law, the motion equation for the vibration model of the convalescent-wheelchair system without DA can be expressed as

$$\begin{cases} m_2 \ddot{z}_2 + C_2 (\dot{z}_2 - \dot{z}_1) + K_2 (z_2 - z_1) = 0, \\ m_1 \ddot{z}_1 + C_2 (\dot{z}_1 - \dot{z}_2) + K_2 (z_1 - z_2) + C_1 (\dot{z}_1 - \dot{q}) + K_1 (z_1 - q) = 0. \end{cases} \quad (1)$$

Analogously, the motion equation for the vibration model of the convalescent-wheelchair system with DA can be expressed as

FIGURE 1: The vibration model of the convalescent-wheelchair system with DA.

FIGURE 2: The vibration model of the convalescent-wheelchair system without DA.

$$\begin{cases} m_2 \ddot{z}_2 + C_2 (\dot{z}_2 - \dot{z}_1) + K_2 (z_2 - z_1) = 0, \\ m_0 \ddot{z}_0 + C_0 (\dot{z}_0 - \dot{z}_1) + K_0 (z_0 - z_1) = 0, \\ m_1 \ddot{z}_1 + C_2 (\dot{z}_1 - \dot{z}_2) + K_2 (z_1 - z_2) + C_0 (\dot{z}_1 - \dot{z}_0) \\ \quad + K_0 (z_1 - z_0) + C_1 (\dot{z}_1 - \dot{q}) + K_1 (z_1 - q) = 0. \end{cases} \quad (2)$$

By comparing Equation (1) with Equation (2), when m_0 equals zero, Equation (2) changes into Equation (1).

To easily facilitate the wheelchair robots design, the following auxiliary variables are introduced:

$$r_0 = \frac{m_0}{m_1},$$

$$r_2 = \frac{m_2}{m_1},$$

$$f_2 = \frac{1}{2\pi} \sqrt{\frac{K_2}{m_2}},$$

$$f_1 = \frac{1}{2\pi} \sqrt{\frac{K_1}{m_1}},$$

$$\begin{aligned}
f_0 &= \frac{1}{2\pi} \sqrt{\frac{K_0}{m_0}}, \\
\xi_2 &= \frac{C_2}{2\sqrt{m_2 K_2}}, \\
\xi_1 &= \frac{C_1}{2\sqrt{m_1 K_1}}, \\
\xi_0 &= \frac{C_0}{2\sqrt{m_0 K_0}},
\end{aligned} \quad (3)$$

where, r_0 and r_2 are the mass ratios; f_0 is defined as the DA natural frequency; f_1 is defined as the wheelchair body natural frequency; f_2 is defined as the convalescent natural frequency; and ξ_0 , ξ_1 , and ξ_2 are the damping ratios of the DA system, the tire system, and the cushion system.

Based on the introduced variables, Equation (2) can be expressed as

$$\begin{cases}
m_2 \ddot{z}_2 + 2m_2 \xi_2 \omega_2 (\dot{z}_2 - \dot{z}_1) + m_2 \omega_2^2 (z_2 - z_1) = 0, \\
m_0 \ddot{z}_0 + 2m_0 \xi_0 \omega_0 (\dot{z}_0 - \dot{z}_1) + m_0 \omega_0^2 (z_0 - z_1) = 0, \\
m_1 \ddot{z}_1 + 2m_2 \xi_2 \omega_2 (\dot{z}_1 - \dot{z}_2) + m_2 \omega_2^2 (z_1 - z_2) + 2m_0 \xi_0 \omega_0 (\dot{z}_1 - \dot{z}_0) \\
+ m_0 \omega_0^2 (z_1 - z_0) + 2m_1 \xi_1 \omega_1 (\dot{z}_1 - \dot{q}) + m_1 \omega_1^2 (z_1 - q) = 0,
\end{cases} \quad (4)$$

where, $\omega_0 = 2\pi f_0$, $\omega_1 = 2\pi f_1$, and $\omega_2 = 2\pi f_2$.

Equation (4) can be simplified as

$$\begin{cases}
\ddot{z}_2 + 2\xi_2 \omega_2 (\dot{z}_2 - \dot{z}_1) + \omega_2^2 (z_2 - z_1) = 0, \\
\ddot{z}_0 + 2\xi_0 \omega_0 (\dot{z}_0 - \dot{z}_1) + \omega_0^2 (z_0 - z_1) = 0, \\
\ddot{z}_1 + 2r_2 \xi_2 \omega_2 (\dot{z}_1 - \dot{z}_2) + r_2 \omega_2^2 (z_1 - z_2) + 2r_0 \xi_0 \omega_0 (\dot{z}_1 - \dot{z}_0) \\
+ r_0 \omega_0^2 (z_1 - z_0) + 2\xi_1 \omega_1 (\dot{z}_1 - \dot{q}) + \omega_1^2 (z_1 - q) = 0.
\end{cases} \quad (5)$$

3. System Amplitude Frequency Characteristics

For the free vibration of the convalescent-wheelchair system with DA, each mass vibrates at the same circular frequency ω and the same phase angle φ . Moreover, the corresponding amplitudes are z_{00} , z_{10} , and z_{20} , respectively. Thus, the solution of the free vibration equation can be expressed as

$$\begin{cases}
z_0 = z_{00} e^{j(\omega t + \varphi)}, \\
z_1 = z_{10} e^{j(\omega t + \varphi)}, \\
z_2 = z_{20} e^{j(\omega t + \varphi)}.
\end{cases} \quad (6)$$

Substituting Equation (6) to Equation (5) produces

$$\begin{cases}
z_2 (-\omega^2 + 2\xi_2 \omega_2 \omega j + \omega_2^2) = z_1 (2\xi_2 \omega_2 \omega j + \omega_2^2), \\
z_0 (-\omega^2 + 2\xi_0 \omega_0 \omega j + \omega_0^2) = z_1 (2\xi_0 \omega_0 \omega j + \omega_0^2), \\
z_1 [-\omega^2 + (2r_0 \xi_0 \omega_0 + 2\xi_1 \omega_1) \omega j + r_2 \omega_2^2 + r_0 \omega_0^2 + \omega_1^2] \\
= z_2 (2r_2 \xi_2 \omega_2 \omega j + r_2 \omega_2^2) + z_0 (2r_0 \xi_0 \omega_0 \omega j + r_0 \omega_0^2) \\
+ q (2\xi_1 \omega_1 \omega j + \omega_1^2).
\end{cases} \quad (7)$$

Equation (7) can be further expressed as

$$\begin{cases}
A_1 z_2 = A_2 z_1, \\
A_3 z_0 = A_4 z_1, \\
A_5 z_1 = A_6 z_2 + A_7 z_0 + A_8 q.
\end{cases} \quad (8)$$

where

$$\begin{aligned}
A_1 &= -\omega^2 + 2\xi_2 \omega_2 \omega j + \omega_2^2, \\
A_2 &= 2\xi_2 \omega_2 \omega j + \omega_2^2, \\
A_3 &= -\omega^2 + 2\xi_0 \omega_0 \omega j + \omega_0^2, \\
A_4 &= 2\xi_0 \omega_0 \omega j + \omega_0^2, \\
A_5 &= -\omega^2 + (2r_0 \xi_0 \omega_0 + 2r_2 \xi_2 \omega_2 + 2\xi_1 \omega_1) \omega j \\
&\quad + r_2 \omega_2^2 + r_0 \omega_0^2 + \omega_1^2, \\
A_6 &= 2r_2 \xi_2 \omega_2 \omega j + r_2 \omega_2^2, \\
A_7 &= 2r_0 \xi_0 \omega_0 \omega j + r_0 \omega_0^2, \\
A_8 &= 2\xi_1 \omega_1 \omega j + \omega_1^2.
\end{aligned} \quad (9)$$

From Equation (8), obtain the following:

$$\frac{z_2}{q} = \frac{A_2 A_3 A_8}{A_1 A_3 A_5 - A_2 A_3 A_6 - A_1 A_4 A_7}, \quad (10)$$

$$\frac{z_1}{q} = \frac{A_1 A_3 A_8}{A_1 A_3 A_5 - A_2 A_3 A_6 - A_1 A_4 A_7}. \quad (11)$$

The amplitude frequency characteristics function J_1 between the road velocity input $\dot{q}$ and the wheelchair body acceleration $\ddot{z}_1$ can be expressed as

$$J_1 = |H(j\omega)|_{\ddot{z}_1 \sim \dot{q}} = \left| \frac{\ddot{z}_1}{\dot{q}} \right| = \omega \left| \frac{z_1}{q} \right|. \quad (12)$$

The amplitude frequency characteristics function J_2 between the road velocity input $\dot{q}$ and the convalescent acceleration $\ddot{z}_2$ can be expressed as

$$J_2 = |H(j\omega)|_{\ddot{z}_2 \sim \dot{q}} = \left| \frac{\ddot{z}_2}{\dot{q}} \right| = \omega \left| \frac{z_2}{q} \right|. \quad (13)$$

The amplitude frequency characteristics function J_3 between the road velocity input $\dot{q}$ and the TDD (tire dynamic deflection) $f_d = z_1 - q$ can be expressed as

$$J_3 = |H(j\omega)|_{f_d \sim q} = \left| \frac{f_d}{\dot{q}} \right| = \left| \frac{z_1 - q}{\dot{q}} \right| = \frac{1}{\omega} \left| \frac{z_1 - q}{q} \right|. \quad (14)$$

Substituting Equation (11) in Equation (12) produces

$$J_1 = \omega \left| \frac{A_1 A_3 A_8}{A_1 A_3 A_5 - A_2 A_3 A_6 - A_1 A_4 A_7} \right|. \quad (15)$$

Substituting Equation (10) in Equation (13) produces

$$J_2 = \omega \left| \frac{A_2 A_3 A_8}{A_1 A_3 A_5 - A_2 A_3 A_6 - A_1 A_4 A_7} \right|. \quad (16)$$

Substituting Equation (11) in Equation (15) produces

$$J_3 = \frac{1}{\omega} \left| \frac{A_1 A_3 A_8 - A_1 A_3 A_5 + A_2 A_3 A_6 + A_1 A_4 A_7}{A_1 A_3 A_5 - A_2 A_3 A_6 - A_1 A_4 A_7} \right|. \quad (17)$$

4. The PSD and RMS Responses under the Random Road

4.1. Random Road Model. The spatial frequency PSD $G_q(n)$ of the random road roughness q can be expressed as [22]

$$G_q(n) = G_q(n_0) \left(\frac{n}{n_0} \right)^{-w}, \quad (18)$$

where, n is the spatial frequency; n_0 is the spatial reference frequency and $n_0 = 0.1 \text{ m}^{-1}$; $G_q(n_0)$ is the road roughness coefficient; and w is the frequency index.

When a wheelchair robot runs on the random road at a certain speed u , the time frequency of the road excitation is $f = un$. Thus, the time frequency PSD $G_q(f)$ is

$$G_q(f) = \frac{1}{u} G_q(n). \quad (19)$$

When $w = 2$, the time frequency PSD $G_q(f)$ can be expressed as

$$G_q(f) = \frac{1}{u} G_q(n_0) \left(\frac{n}{n_0} \right)^{-2} = G_q(n_0) n_0^2 \frac{u}{f^2}. \quad (20)$$

The relationship between the time frequency PSD $G_q(f)$ of the random road roughness q and the time frequency PSD $G_{\dot{q}}(f)$ of the random road excitation velocity $\dot{q}$ is given by

$$G_{\dot{q}}(f) = (2\pi f)^2 G_q(f) = 4\pi^2 G_q(n_0) n_0^2 u. \quad (21)$$

4.2. The PSD and RMS Responses Calculation. The auto-correlation function $R_{\dot{q}}(\tau)$ of $\dot{q}(t)$ is

$$R_{\dot{q}}(\tau) = \lim_{T \rightarrow \infty} \frac{1}{T} \int_{-T/2}^{+T/2} \dot{q}(t) \dot{q}(t + \tau) dt. \quad (22)$$

The convalescent acceleration $\ddot{z}_2(t)$ can be expressed as

$$\ddot{z}_2(t) = \int_0^{+\infty} \dot{q}(\tau) h(t - \tau) d\tau, \quad (23)$$

where $h(t)$ is impulse response function.

The PSD $G_{\ddot{z}_2}(\omega)$ of the convalescent acceleration $\ddot{z}_2(t)$ can be expressed as

TABLE 1: The values of the parameters for the convalescent-wheelchair system.

Parameter	Value
m_2 (kg)	75
m_1 (kg)	25
m_0 (kg)	2.5
C_2 (Ns/m)	377
C_1 (Ns/m)	314
C_0 (Ns/m)	63
K_2 (N/mm)	11.84
K_1 (N/mm)	98.70
K_0 (N/mm)	9.87

$$G_{\ddot{z}_2}(\omega) = E[\ddot{z}_2(t) \ddot{z}_2(t + \tau)] = H(j\omega) G_{\dot{q}}(\omega) H(j\omega) = J^2 G_{\dot{q}}(\omega). \quad (24)$$

Substituting Equation (21) in Equation (24) produces

$$G_{\ddot{z}_2}(\omega) = 4\pi^2 G_q(n_0) n_0^2 u J_2^2. \quad (25)$$

The convalescent RMS acceleration $\sigma_{\ddot{z}_2}$ can be expressed as

$$\sigma_{\ddot{z}_2} = \sqrt{\frac{1}{2\pi} \int_0^{+\infty} G_{\ddot{z}_2}(\omega) d\omega}. \quad (26)$$

Substituting Equation (25) in Equation (26) produces

$$\sigma_{\ddot{z}_2} = \sqrt{2\pi G_q(n_0) n_0^2 u \int_0^{+\infty} J_2^2(\omega) d\omega}. \quad (27)$$

Analogously, the wheelchair body RMS acceleration $\sigma_{\ddot{z}_1}$ can be expressed as

$$\sigma_{\ddot{z}_1} = \sqrt{2\pi G_q(n_0) n_0^2 u \int_0^{+\infty} J_1^2(\omega) d\omega}. \quad (28)$$

The TDD RMS can be expressed as

$$\sigma_{f_d} = \sqrt{2\pi G_q(n_0) n_0^2 u \int_0^{+\infty} J_3^2(\omega) d\omega}. \quad (29)$$

It is difficult to analytically calculate the Equations (27)–(29). The numerical integration method is adopted as

$$\begin{aligned} \sigma_{\ddot{z}_2} &= \sqrt{2\pi G_q(n_0) n_0^2 u \sum_{i=1}^N [J_2^2(2\pi i \Delta f)] \times (2\pi i \Delta f)}, \\ \sigma_{\ddot{z}_1} &= \sqrt{2\pi G_q(n_0) n_0^2 u \sum_{i=1}^N [J_1^2(2\pi i \Delta f)] \times (2\pi i \Delta f)}, \\ \sigma_{f_d} &= \sqrt{2\pi G_q(n_0) n_0^2 u \sum_{i=1}^N [J_3^2(2\pi i \Delta f)] \times (2\pi i \Delta f)}, \end{aligned} \quad (30)$$

where N is the number of the discrete frequency value; Δf is the frequency bandwidth; and $i = 1, 2, 3, \dots, N$.

FIGURE 3: A comparison of the dynamic responses calculated from the vibration models of the convalescent-wheelchair with DA and without DA: (a) the wheelchair body RMS acceleration $\sigma_{\ddot{z}_1}$ versus the running speed u ; (b) the convalescent RMS acceleration $\sigma_{\ddot{z}_2}$ versus the running speed u ; (c) the TDD RMS σ_{f_d} versus the running speed u ; (d) the wheelchair body acceleration PSD $G_{\ddot{z}_1}$ versus the frequency f ; (e) the convalescent acceleration PSD $G_{\ddot{z}_2}$ versus the frequency f ; (f) the TDD PSD G_{f_d} versus the frequency f .

TABLE 2: The values of the system parameters for the analysis.

Parameter	Baseline	+100%	-50%
r_2	3.0	6.0	1.5
r_0	0.10	0.05	0.20
ξ_2	0.25	0.5	0.125
ξ_1	0.25	0.5	0.125
ξ_0	0.25	0.5	0.125
f_2	2.0	4.0	1.0
f_1	10.0	20.0	5.0
f_0	10.0	20.0	5.0

FIGURE 4: Influences of r_2 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d : (a) the amplitude frequency characteristics J_1 at different values of r_2 ; (b) the amplitude frequency characteristics J_2 at different values of r_2 ; (c) the amplitude frequency characteristics J_3 at different values of r_2 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus r_2 .

FIGURE 5: Influences of r_0 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d : (a) the amplitude frequency characteristics J_1 at different values of r_0 ; (b) the amplitude frequency characteristics J_2 at different values of r_0 ; (c) the amplitude frequency characteristics J_3 at different values of r_0 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus r_0 .

5. Vibration Characteristics Comparison

In order to facilitate the vibration characteristics analysis of the convalescent-wheelchair system with DA, the convalescent-wheelchair system without DA is taken as a comparison object. Based on a typical commercially available wheelchair and the selection method of the DA [19], the values of the parameters for the convalescent-wheelchair system is given in Table 1 [21]. The simulation condition is set as: $u = 0.5 \sim 2.5$ m/s, the C grade road with $G_q(n_0) = 256 \times 10^{-6} \text{ m}^{-3}$. For the numerical calculation, $\Delta f = 0.1$ Hz, and the upper limit frequency is set as 30 Hz. Based on the vibration models of the convalescent-wheelchair system with DA and without DA, the values of $\sigma_{\ddot{z}_2}$, $\sigma_{\ddot{z}_1}$, σ_{f_d} , $G_{\ddot{z}_2}$, $G_{\ddot{z}_1}$, and G_{f_d} were numerically

calculated, respectively. The calculated results are shown in Figure 3.

Figures 3(a)–3(c) show that the DA can effectively reduce the wheelchair body RMS acceleration $\sigma_{\ddot{z}_1}$ and the convalescent RMS acceleration $\sigma_{\ddot{z}_2}$, and the TDD RMS σ_{f_d} is not improved obviously. Figure 3(d) shows that the peak value of the wheelchair body acceleration PSD $G_{\ddot{z}_1}$ is decreased by 40.2% by using the DA. Figure 3(e) illustrates that the DA reduces the convalescent acceleration PSD $G_{\ddot{z}_2}$ around 10 Hz but causes a slight increase in the range of 4~7 Hz. Figure 3(f) depicts that the DA makes the peak value of the TDD PSD G_{f_d} shift to the left. The comparison results show that the DA partly suppresses the vibration of the convalescent and the wheelchair body, especially in the resonance area of the wheelchair body. However, the

FIGURE 6: Influence of ξ_2 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d : (a) the amplitude frequency characteristics J_1 at different values of ξ_2 ; (b) the amplitude frequency characteristics J_2 at different values of ξ_2 ; (c) the amplitude frequency characteristics J_3 at different values of ξ_2 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus ξ_2 .

DA does not successfully improve the tire contact behavior.

6. Sensitivity Analysis of Vibration Responses to System Parameters

In order to analyze the effects of r_2 , r_0 , ξ_1 , ξ_2 , ξ_0 , f_0 , f_1 , and f_2 on $\sigma_{\ddot{z}_2}$, $\sigma_{\ddot{z}_1}$, and σ_{f_d} , the simulation calculation was carried out based on the vibration model of the convalescent-wheelchair system with DA. The simulation condition is set as $u=0.1$ m/s, the C grade road with $G_q(n_0)=256 \times 10^{-6} \text{ m}^{-3}$. To represent the curve changes in the same

diagram, the values of $\sigma_{\ddot{z}_2}$, $\sigma_{\ddot{z}_1}$, and σ_{f_d} are normalized on the basis of the baseline RMS values, respectively. The processing method is as follows: $\sigma_2 = \sigma_{\ddot{z}_2} / |\sigma_{\ddot{z}_2}|_b$, $\sigma_1 = \sigma_{\ddot{z}_1} / |\sigma_{\ddot{z}_1}|_b$, $\sigma_3 = \sigma_{f_d} / |\sigma_{f_d}|_b$, where the subscript “b” represents “baseline.” In addition, to observe the performance changes in the local frequency band, the amplitude frequency characteristics functions J_1 , J_2 , and J_3 are also calculated.

When the influence of one of the eight parameters is analyzed, the baseline value is increased by 100% or decreased by 50%, and the other parameters are fixed. The values of the system parameters for the analysis are given in Table 2 [21]. The baseline values are converted from Table 1.

FIGURE 7: Influences of ξ_1 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d : (a) the amplitude frequency characteristics J_1 at different values of ξ_1 ; (b) the amplitude frequency characteristics J_2 at different values of ξ_1 ; (c) the amplitude frequency characteristics J_3 at different values of ξ_1 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus ξ_1 .

6.1. The Influences of the Mass Ratios

6.1.1. The Influences of the Mass Ratio r_2 . Figure 4 depicts the influences of r_2 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . From Figure 4(a), it can be seen that increasing r_2 can almost reduce the amplitude of J_1 in the whole frequency domain. Therefore, increasing r_2 is conducive to reducing the wheelchair body acceleration. Figure 4(b) illustrates that increasing r_2 can reduce the amplitude of J_2 in the range of 5~15 Hz, but it causes the amplitude to increase around the low frequency resonance peak. From Figure 4(c), it can be seen that with the increase of r_2 , the low frequency resonance peak of J_3 increases sharply, while the high frequency formant decreases. Therefore, too large r_2 is not conducive to improving the tire grounding safety. As shown in Figure 4(d), σ_1 and σ_2 are in inverse proportion to r_2 , and σ_3 has a minimum value for r_2

in the range of 2.5~3.5. Thus, the increase of r_2 is mainly to reduce the values of σ_1 and σ_2 .

6.1.2. The Influences of the Mass Ratio r_0 . Figure 5 shows the influences of r_0 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . From Figure 5(a), it can be seen that increasing r_0 can effectively reduce the peak value of J_1 . Figure 5(b) shows that increasing r_0 can reduce the amplitude of J_2 from 7 Hz to 15 Hz, but the amplitude in the range of 4~7 Hz increases slightly. From Figure 5(c), it can be seen that the larger the value of r_0 is, the higher the formant of J_3 is. As shown in Figure 5(d), σ_1 and σ_2 are in inverse proportion to r_0 , σ_3 is proportional to r_0 , and σ_1 is most sensitive to the change of r_0 . Thus, the use of the DA with the larger mass helps to reduce the vibration of the convalescent and the wheelchair.

FIGURE 8: Influences of ξ_0 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d : (a) the amplitude frequency characteristics J_1 at different values of ξ_0 ; (b) the amplitude frequency characteristics J_2 at different values of ξ_0 ; (c) the amplitude frequency characteristics J_3 at different values of ξ_0 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus ξ_0 .

6.2. The Influences of the Damping Ratios

6.2.1. The Influences of the Damping Ratio ξ_2 . Figure 6 depicts the influences of ξ_2 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . As shown in Figure 6(a), increasing ξ_2 can help to reduce the peak value of J_1 . From Figure 6(b), it can be seen that increasing ξ_2 can effectively reduce the low frequency resonance peak of J_2 , but the amplitude after 3.0 Hz increases. Figure 6(c) shows that the larger the value of ξ_2 is, the smaller the peak values of J_3 are, but the amplitude between the two formants increases obviously. Figure 5(d) illustrates that σ_1 is almost in inverse proportional to ξ_2 , there is a minimum value of σ_2 for the value of ξ_2 between 0.15~0.25, and σ_3 has a minimum value when the value of ξ_2 belongs to 0.15~0.25. The ride comfort requires that the cushion system has the

smaller value of ξ_2 , however, the tire grounding safety and the operational reliability need the larger value of ξ_2 . Thus, taking the smaller value of ξ_2 , about 0.2, is helpful to improve comfort.

6.2.2. The Influences of the Damping Ratio ξ_1 . Figure 7 depicts the influences of ξ_1 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . From Figure 7(a), it can be seen that increasing ξ_1 can effectively suppress the peak value of J_1 , but the amplitude after 15 Hz increases. As shown in Figure 7(b), the increase of ξ_1 can effectively suppress the low frequency and the high frequency resonance peak values of J_2 , and the suppression effectiveness around the high frequency resonance peak is more obvious. However, the amplitude after 15 Hz

FIGURE 9: Influences of f_2 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d : (a) the amplitude frequency characteristics J_1 at different values of f_2 ; (b) the amplitude frequency characteristics J_2 at different values of f_2 ; (c) the amplitude frequency characteristics J_3 at different values of f_2 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus f_2 .

also increases. Figure 7(c) clearly shows that increasing ξ_1 can almost reduce the amplitude of J_3 in the full frequency domain. From Figure 7(d), it can be seen that both σ_2 and σ_3 are approximately in inverse proportion to ξ_1 , and when the value of ξ_1 belongs to 0.1~0.2, a minimum value of σ_1 exists. Thus, the ride comfort and the tire grounding safety require that the tire system has the larger value of ξ_1 , however, the operational reliability needs the smaller value of ξ_1 .

6.2.3. The Influences of the Damping Ratio ξ_0 . Figure 8 depicts the influences of ξ_0 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . According to Figure 8(a), the amplitude of J_1 in the region of 8~13 Hz increases with the increase of ξ_0 , and the amplitude decreases after 13 Hz. Figure 8(b) shows that increasing ξ_0 can

effectively suppress the high frequency resonance peak value of J_2 . According to Figure 8(c), ξ_0 has obvious effect on the high frequency resonance peak value of J_3 . From Figure 8(d), it can be seen that, with the increase of ξ_0 , the value of σ_1 decreases. When ξ_0 belongs to 0.1~0.2, σ_3 has a minimum value, and when ξ_0 belongs to 0.15~0.25, σ_2 has a minimum value. When the value of ξ_0 equals 0.2, the DA system can well give consideration to the three indexes: the ride comfort, the tire grounding safety, and the operational reliability.

6.3. The Influences of the Natural Frequencies

6.3.1. The Influences of the Natural Frequency f_2 . Figure 9 depicts the influences of f_2 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . From Figure 9(a), it can be seen that increasing f_2 is beneficial to

FIGURE 10: Influences of f_1 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d : (a) the amplitude frequency characteristics J_1 at different values of f_1 ; (b) the amplitude frequency characteristics J_2 at different values of f_1 ; (c) the amplitude frequency characteristics J_3 at different values of f_1 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus f_1 .

suppress the resonance peak value of J_1 . Figures 9(b) and 9(c) show that increasing f_2 makes the low frequency formants of J_2 and J_3 increase sharply and shift rightward. As shown in Figure 9(d), σ_1 is almost in inverse proportion to f_2 , σ_2 is proportional to f_2 and σ_3 is most sensitive to the change of f_2 , and when f_2 belongs to 1.5~2.5 Hz, σ_3 has a minimum value. Therefore, adopting a soft cushion to reduce f_2 is an important measure to improve ride comfort.

6.3.2. The Influences of the Natural Frequency f_1 . Figure 10 depicts the influences of f_1 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . Figure 10(a) shows that with the increase of f_1 , the resonance peak value of J_1 increases sharply and moves to the right. Figure 10(b) illustrates that increasing f_1 is beneficial

to reduce the peak value of the low frequency resonance peak value of J_2 , but the amplitude after 10 Hz increases significantly. Figure 10(c) shows that the increase of f_1 is conducive to significantly reducing the amplitude of J_3 from 0 Hz to 18 Hz, although the amplitude increases slightly after 18 Hz. Figure 10(d) shows that σ_1 and σ_2 are proportional to f_1 , and σ_3 nonlinearly decreases as f_1 increases. Thus, the adoption of soft tires is conducive to improving ride comfort and operational reliability.

6.3.3. The Influences of the Natural Frequency f_0 . Figure 11 depicts the influences of f_0 on $\ddot{z}_1$, $\ddot{z}_2$, and f_d . Figure 11(a) shows that the increase of f_0 has little effect on J_1 in the low frequency region 0~5 Hz and the high

FIGURE 11: Influences of f_0 on $\ddot{z}_1$, $\ddot{z}_2$, and f_4 : (a) the amplitude frequency characteristics J_1 at different values of f_0 ; (b) the amplitude frequency characteristics J_2 at different values of f_0 ; (c) the amplitude frequency characteristics J_3 at different values of f_0 ; (d) the RMS σ_1 , σ_2 , and σ_3 versus f_0 .

TABLE 3: The sensitivity values of each response.

System parameter	$s_{\ddot{z}_1}$	$s_{\ddot{z}_2}$	s_{f_4}
r_2	0.362	0.324	0.330
r_0	0.198	0.107	0.099
ξ_2	0.367	0.460	0.269
ξ_1	0.498	0.242	0.505
ξ_0	0.110	0.057	0.092
f_2	0.376	1.095	0.511
f_1	1.247	0.560	0.912
f_0	0.173	0.113	0.184

frequency 25~30 Hz. In the mid frequency region 5~25 Hz, increasing f_0 is beneficial to the amplitude reduction of J_1 . Figure 11(b) shows that f_0 only obviously affects the

amplitude of J_1 from 5 Hz to 20 Hz. Figure 11(c) clearly states that f_0 has a great influence on the high frequency resonance peak value of J_3 . Figure 11(d) illustrates that σ_1 nonlinearly decreases as f_0 increases, when the value of f_0 belongs to 8~12, σ_2 has a minimum value, and when the value of f_0 belongs to 6~8, σ_3 has a minimum value. Thus, when f_0 approximately equals to f_1 , the DA system can well give consideration to the comprehensive isolation performance.

6.4. Sensitivity Calculation. Based on the influence trends analysis of the system parameters on the RMS responses in Sections 6.1 to 6.3, sensitivity calculation was carried out by

FIGURE 12: The vibration response sensitivity: (a) the wheelchair body RMS acceleration sensitivity $s_{\ddot{z}_1}$; (b) the convalescent RMS acceleration sensitivity $s_{\ddot{z}_2}$; (c) the TDD RMS sensitivity s_{f_d} .

using the perturbation method. The calculation formulae are as follows:

$$s_{xi} = \frac{\sigma(U_{xi} + \Delta U_{xi}) - \sigma(U_{xi}) / \sigma(U_{xi})}{\Delta U_{xi} / U_{xi}}, \quad (31)$$

$$s_x = \sqrt{\frac{1}{n} \sum_{i=1}^n |s_{xi}|^2}, \quad (32)$$

where n is the number of the intervals, i represents the i th interval, ΔU_{xi} refers to the incremental value of the system parameter x in the i th interval, and s_x is the response sensitivity to the system parameter x .

According to the calculated RMS responses based on the system parameters in Table 2, the sensitivity values of each response are obtained by using Equations (31) and (32), as shown in Table 3. To easily observe the differences in sensitivity, the sensitivity values were reordered, as shown in Figure 12. Figure 12(a) shows that f_1 is the most important influence factor on $\sigma_{\ddot{z}_1}$, and ξ_0 is the weakest influence factor. Other factors are $\xi_1, f_2, \xi_2, r_2, r_0$, and f_0 from strong to weak. From Figure 12(b), it can be seen that $\sigma_{\ddot{z}_2}$ is most sensitive to f_2 , and the sensitivity is as high as 1.095. $\sigma_{\ddot{z}_2}$ is most insensitive to ξ_0 . Other factors are $f_1, \xi_2, r_2, \xi_1, f_0$, and r_0 from strong to weak. Figure 12(c) illustrates that the greatest influence factor on σ_{f_d} is f_1 , and the minimum influence factor is ξ_0 . Other factors affecting σ_{f_d} in an order

from intensely to slightly are f_2 , ξ_1 , r_2 , ξ_2 , f_0 , and r_0 . Thus, for the DA, ξ_0 has the smallest influence on each RMS response, and f_0 and r_0 have relatively greater influences.

7. Conclusions

This paper compared the vibration isolation performances of the convalescent-wheelchair robot with DA and without DA and conducted the sensitivity analysis of the vibration responses to the important dynamic parameters. The parameters include the following: the mass ratio r_0 of the DA mass to the wheelchair body mass, the mass ratio r_2 of the convalescent mass to the wheelchair body mass, the DA natural frequency f_0 , the wheelchair body natural frequency f_1 , the convalescent natural frequency f_2 , the DA system damping ratio ξ_0 , the tire system damping ratio ξ_1 , and the cushion system damping ratio ξ_2 . Some useful conclusions are as follows:

- (1) The DA can partly suppress the vibration of the convalescent and the wheelchair body, especially in the resonance area of the wheelchair body. However, the DA cannot successfully improve the tire contact behavior.
- (2) The increase of r_2 is mainly to reduce the values of the wheelchair body RMS acceleration $\sigma_{\ddot{z}_1}$ and the convalescent RMS acceleration $\sigma_{\ddot{z}_2}$. The use of the DA with the larger mass can help to reduce the vibration of the convalescent and the wheelchair. Taking the smaller value of ξ_2 , about 0.2, is helpful to improve comfort. The ride comfort and the tire grounding safety require that the tire system has the larger value of ξ_1 ; however, the operational reliability needs the smaller value of ξ_1 . When the value of ξ_0 equals 0.2, the DA system can well give consideration to the comprehensive performance. Adopting a soft cushion to reduce f_2 is an important measure to improve ride comfort.
- (3) f_1 is the most important influence factor on $\sigma_{\ddot{z}_1}$, and ξ_0 is the weakest influence factor. Other factors are ξ_1 , f_2 , ξ_2 , r_2 , r_0 , and f_0 from strong to weak. $\sigma_{\ddot{z}_2}$ is most sensitive to f_2 . The greatest influence factor on $\sigma_{\ddot{z}_d}$ is f_1 . For the DA, ξ_0 has the smallest influence on each RMS response, and f_0 and r_0 have relatively greater influences.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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