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A BIODYNAMIC MODEL OF WHEELCHAIR WITH CHANGEABLE SEAT CUSHIONS SUBJECTED TO VERTICAL VIBRATIONS

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Abstract. *This paper discusses the investigation of transfer functions outputted by a mathematical model of a wheelchair and occupant system in order to obtain insights into the behavior of such a system when it is subjected to vibrations from typical ground traffic surfaces. The wheelchair is modeled with seven types of seat cushion having different linear stiffness and damping characteristics. A matrix of equations of motion in nine degrees of freedom is employed to define the dynamic behavior of vibration responses from the wheelchair and its occupant. Laplace transforms and frequency response functions are applied to solving the EOMs. Model outputs, expressed as magnitudes and phase angles of transfer functions, are validated to a level of confidence against other experimental data from published literature. The model employed for this study should be useful in the area of wheelchair suspension design for helping to predict the behavior of detrimental vibrations.*
Keywords: Mathematical model, Transfer functions, Wheelchair cushion, Vibrations

1. **Introduction.** Prolonged exposure to vibration not only causes discomfort to the wheelchair occupant, but can also have undesirable impacts on the rider's health [1-3]. Rider's organs, such as the brain, may suffer internal hemorrhage as a result of jarring movements within the skull. Soft tissues in the intestines and rectum can be damaged, and tendons and joints can become strained. Long-term exposure to vibrations has been shown to cause neurological and physiological effects, including arthritic joints, lower back pain and other musculo-skeletal syndromes [2-7].

Accordingly, a number of researchers have attempted to mitigate the effects of vibrations on the manual wheelchair through design improvements whereby mathematical models of the wheelchair system and its response to vibrations are studied for the purpose. Marzbanrad and Afkar [8] employed Laplace transform to solve a 7-DOF linear lumped parameter model to develop vibrational control methods for wheelchair seat systems in order to achieve a greater level of rider comfort. Hassaan and Mohammed [9] employed frequency response functions to examine a 10-DOF full-car model for a similar purpose. Vibrations in the frequency range of 0.5 to 11 Hz, although quite low, were found to contain sufficient energies to cause discomfort to passengers in a typical vehicle. The same range of frequencies has also been found in other studies to be the offending

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vibrations [10, 11]. Garcia-Mendez et al. [4] applied 1-DOF and 2-DOF models to assessing the vibration transmissibility in a wheelchair system with changeable seat cushions. According to their findings, air-based cushions fare better than the gel- and foam-based options and thus should be considered when selecting a cushion to help reduce vibration transmissibility.

Changing to an improved seat cushion as an option to reduce vibration transmissibility is sensible cost-wise [12]. It is more pragmatic and economical than replacing an entire wheelchair. Simulations from this study, as well as those by Wang et al. [13], Wicaksono et al. [14] and Lariviere et al. [15], have confirmed the effectiveness of a suitable cushion in helping to absorb much of the transmitted vibrations. Design engineers should therefore develop an insight into relevant simulation methods for determining the optimum shock absorption properties of various seat cushion brands on offer including air-based, foam-based and gel-based types. Current studies on such simulation techniques are available in industry literature [14, 16, 17].

The rationale above inspired us to attempt a model for predicting vibrational response using transfer functions. After a careful review of existing literature, we chose to develop a model with a sufficient number of biodynamic parts to ensure high reliability, as well as the flexibility for testing a wide range of seat cushions. We therefore extended the work of Weerapong et al. [18], which employed a 9-DOF wheelchair-occupant system for determining vibration transmissibility in the vertical plane, in order to focus more on studying behavior of vibration responses of transfer function from excitation to wheelchair occupant's head as well as the vibration absorption efficacy of different cushion types. The seven cushions simulated in this study belong in three categories: air-based, comprising Roho High Profile, Roho Low Profile, Vector with Vicair Technology; foam-based, comprising Zoombang Protective Gear with Foam, Meridian Wave and Comfort Mate Foam; and gel-based, comprising Jay J2 Deep Contour. The stiffness and damping parameters of the cushions were sourced from published information of international institutions that promote assistive technology.

This paper is organized in the following sections. Section 2 analyzes the wheelchair and occupant regarding properties of human tissue, spring and damper, and defines the force vectors upon the masses in the free body diagrams. Section 3 has the model's EOMs arranged in Laplace-transformed matrices containing equations and frequency response functions. Section 4 concerns the assessment and validation of the biodynamic model and seat cushions, and application of the transfer functions to obtain output magnitudes and phases. Section 5 delivers concluding remarks.

2. **Analysis of Wheelchair-Occupant's Vibrational Response.** Figure 1 shows the free body diagram, for modeling the wheelchair-occupant system. The model, with seven DOFs representing the human body and two DOFs, the wheelchair frame, features altogether nine DOFs, which are denoted as nine masses, numbered 1 to 9, comprising chassis & tire, seat, pelvis, abdomen, diaphragm, thorax, torso, back and head [18, 19]. In mechanical terms, these are like springs and dampers and connecting masses similar to the models in [10, 20]. Upon inputs of sinusoidal excitation forces, the model would respond and exhibit steady-state response functions. Vertical vibrations are inputted as the forces acting on each of the nine masses which would result in displacements in the direction of positive y. In order to determine such displacements, the positions of the nine masses must be determined beforehand. Table 1 lists the parameters employed for simulating the wheelchair frame, consisting of weight masses, spring and damper constants, and related properties of commercially-available wheelchairs chosen from the range suggested by the Wheelchair & Seating Clinic of the Center for Assistive Technology, Pittsburgh Healthcare System, U.S. Department of Veteran Affairs. In addition, occupant parameters were obtained from previous studies of characteristics of subsystems by Patil and

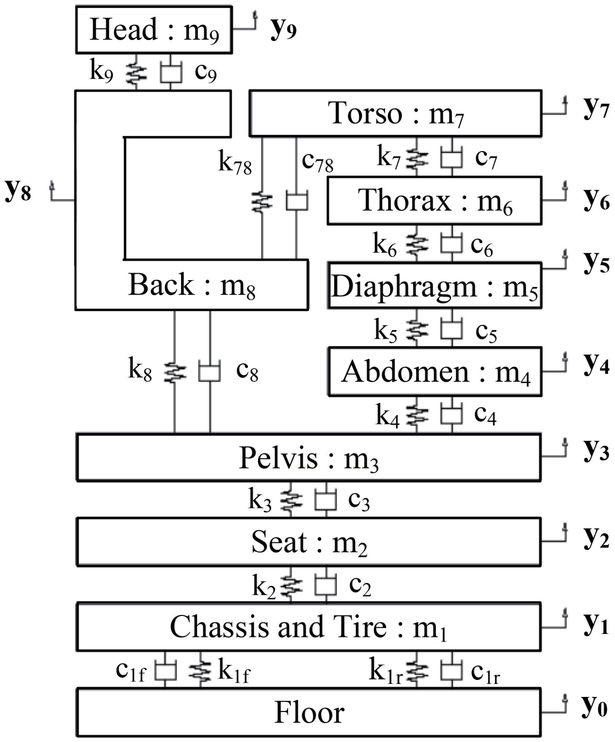

Figure 1. The nine-DOF biomechanical model of seated human body in wheelchair subjected to vertical vibrations [18, 19]

Table 1. Parameter values of manual wheelchair and seat cushions [4, 5]

Mass $[M]$ (kg)	Damping constant $[C]$ (N/m/sec)	Spring constant $[K]$ (N/m)
$m_1 = 16.5$ $m_2 = 1.5$	$c_{1f^*} = 500$, $c_{1r^*} = 500$ $c_{2a} = 840$, $c_{2b} = 397$ $c_{2c} = 834$, $c_{2d} = 571$ $c_{2e} = 1015$, $c_{2f} = 1507$ $c_{2g} = 1689$	$k_{1f^*} = 60000$, $k_{1r^*} = 6000$ $k_{2a} = 95040$, $k_{2b} = 76010$ $k_{2c} = 94220$, $k_{2d} = 39970$ $k_{2e} = 68600$, $k_{2f} = 174900$ $k_{2g} = 183200$
input magnitude vibration, $Y_0 = 5$ mm.		

* represents the parameter value for two tires (front or back, as the case may be). Subscript a represents the parameter value for Vector with Vicair; b, for Meridian Wave; c, for Roho High Profile; d, for Jay J2 Deep Contour; e, for Roho Low Profile; f, for Zoombang Protective Gear with Foam; and g for Comfort Mate Foam.

Palanichamy [10] and Liang and Chiang [20] with valid seat-to-head acceleration ratios. These are listed in Table 2.

3. **Deriving Matrix of Equations of Motion (EOMs).** The derivation of EOMs of the lumped-parameter model is analyzed from Figure 1 which is the composite model of the occupant and wheelchair [18, 19]. From the FBD, system EOMs can be determined and expressed in matrix form which are solvable with the application of the Laplace transform and frequency response functions.

3.1. **Description of the nine DOFs.** The differential equations of the MDOF system to be analyzed are arranged in matrix form and shown below. In notation, the system consists of mass m_i, $(i = 1, \ldots, 9)$ in matrix $[M]$ which is connected to ground by a spring of stiffness k_i in $[K]$ plus a damper with damping coefficient c_i in $[C]$. Since the masses of $[M]$ can only move in the y direction, the displacement, velocity and acceleration of

TABLE 2. Parameter values of biodynamic model [10,20]

Mass $[M]$ (kg)	Damping constant $[C]$ (N/m/sec)	Spring constant $[K]$ (N/m)
$m_3 = 27.7$	$c_3 = 378$	$k_3 = 25500$
$m_4 = 6.02$	$c_4 = 298$	$k_4 = 894.1$
$m_5 = 0.46$	$c_5 = 298$	$k_5 = 894.1$
$m_6 = 1.38$	$c_6 = 298$	$k_6 = 894.1$
$m_7 = 33.33$	$c_7 = 298$	$k_7 = 894.1$
	$c_{78} = 3651$	$k_{78} = 53640$
$m_8 = 6.94$	$c_8 = 3651$	$k_8 = 53640$
$m_9 = 5.5$	$c_9 = 3651$	$k_9 = 53640$

each DOF – expressed as $y_i(t)$, $\dot{y}_i(t)$, $\ddot{y}_i(t)$ in vector of $\{y(t)\}$, $\{\dot{y}(t)\}$ and $\{\ddot{y}(t)\}$ – and input displacement, velocity and acceleration y_0, $\dot{y}_0$ and $\ddot{y}_0$ are all confined in the same direction. They are thus sufficient for defining the system configuration. The excitation force, $f(t)$, is to be applied to the masses. The EOMs for the system are expressed by

$$[M]\{\ddot{y}(t)\} + [C]\{\dot{y}(t)\} + [K]\{y(t)\} = \{f(t)\}. \quad (1)$$

3.1.1. *Laplace transform solutions.* Let $\mathcal{L}\{\ddot{y}(t)\} = s^2Y(s)$, $\mathcal{L}\{\dot{y}(t)\} = sY(s)$, $\mathcal{L}\{y(t)\} = Y(s)$, $\mathcal{L}\{\dot{y_0}\} = sY_0(s)$ and $\mathcal{L}\{y_0\} = Y_0(s)$, $F_0(s)$ represent the Laplace transform of $f(t)$. Then, setting the initial conditions to zero, i.e., $\{\dot{y}(0)\} = 0$ and $\{y(0)\} = 0$, allows us to take the transfer function of the system's steady-state response to sinusoidal input [21]. Upon taking Laplace transform of Equation (1) and solving for algebraic equation in function of Laplace variable for $\{Y(s)\}$ we obtain the Laplace transform of the MDOF equations of motion on Equation (1), as

$$\left[[M]\left\{s^2\right\} + [C]\{s\} + [K]\right]\{Y(s)\} = \{F_0(s)\}. \quad (2)$$

$$\begin{bmatrix} A1 & A2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ B1 & B2 & B3 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & C1 & C2 & C3 & 0 & 0 & 0 & C4 & 0 \\ 0 & 0 & D1 & D2 & D3 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & E1 & E2 & E3 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & F1 & F2 & F3 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & G1 & G2 & G3 & 0 \\ 0 & 0 & H1 & 0 & 0 & 0 & H2 & H3 & H4 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & J1 & J2 \end{bmatrix} \begin{Bmatrix} Y_1(s) \\ Y_2(s) \\ Y_3(s) \\ Y_4(s) \\ Y_5(s) \\ Y_6(s) \\ Y_7(s) \\ Y_8(s) \\ Y_9(s) \end{Bmatrix} = \begin{Bmatrix} J3 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{Bmatrix} Y_0(s). \quad (3)$$

Here

$A1 = m_1s^2 + (c_2 + c_{1f} + c_{1r})s + k_2 + k_{1f} + k_{1r}$, $A2 = -c_2s - k_2$,
$B2 = m_2s^2 + (c_2 + c_3)s + k_2 + k_3$, $B1 = -c_2s - k_2$,
$B3 = -k_3 - c_3s$, $C1 = -c_3s - k_3$,
$C2 = m_3s^2 + (c_3 + c_4 + c_8)s + k_3 + k_4 + k_8$, $C3 = -c_4s - k_4$,
$C4 = -k_8 - c_8s$, $D1 = -c_4s - k_4$,
$D2 = m_4s^2 + (c_4 + c_5)s + k_4 + k_5$, $D3 = -c_5s - k_5$,
$E2 = m_5s^2 + (c_5 + c_6)s + k_5 + k_6$, $E1 = -c_5s - k_5$,
$E3 = -c_6s - k_6$, $F1 = -c_6s - k_6$,
$F2 = m_6s^2 + (c_6 + c_7)s + k_6 + k_7$, $F3 = -k_7 - c_7s$,

$G2 = m_7s^2 + (c_7 + c_{78})s + k_7 + k_{78}$, $G1 = -c_7s - k_7$,
$G3 = -c_{78}s - k_{78}$, $H1 = -c_8s - k_8$,
$H3 = -m_8s^2 + (c_8 + c_9 + c_{78})s + k_8 + k_9 + k_{78}$, $H2 = -c_{78}s - k_{78}$,
$H4 = -c_9s - k_9$, $J1 = -c_9s - k_9$,
$J2 = m_9s^2 + c_9s + k_9$, $J3 = (c_{1f} + c_{1r})s + (k_{1f} + k_{1r})$.

3.1.2. *Transfer functions of MDOF systems complex function.* By substituting $j\omega$ for s it is possible to calculate the frequency response, where j is the imaginary operator [21] in Equation (4)

$$\frac{\{Y(s)\}}{\{F_0(s)\}} = \frac{1}{[[M]\{s^2\} + [C]\{s\} + [K]]}. \tag{4}$$

Equations of motion of the 9-DOF system are expressed in complex terms in Laplace domain of the transfer function in Equation (4) thereby yielding the algebraic equation below:

$$\frac{\{Y(j\omega)\}}{\{F_0(j\omega)\}} = \frac{1}{[-\omega^2[M] + j\omega[C] + [K]]}. \tag{5}$$

Next, we examine Equation (5). The $[-\omega^2[M] + j\omega[C] + [K]]$ term therein is impedance matrix for assessing mechanical responses from the human and vehicle frames, and it may be expressed in transfer function matrices as shown below.

$\{Y(j\omega)\}$ and $\{F_0(j\omega)\}$ are the corresponding complex Fourier transform vectors of $Y_i(j\omega)$ and $F_j(j\omega)$ respectively and ω, the excitation frequency. Upon substitution in Equation (5), we obtain the following:

$$\frac{Y_i(j\omega)}{F_j(j\omega)} = \frac{1}{-\omega^2 m_{ij} + j\omega c_{ij} + k_{ij}} = H_{ij}(j\omega), \quad (i, j = 1, \ldots, 9), \tag{6}$$

where H_{ij} is response at mass of i per unit force excitation at j.

The $[-\omega^2[M] + j\omega[C] + [K]]$ portion has been generated by the equations of mass, damping and stiffness matrices. Its inverse, as shown in Equation (7), becomes a transfer function which is $[H(j\omega)]$. This gives us the set of matrix equations below:

$$\begin{bmatrix} H_{11}(j\omega) & H_{12}(j\omega) & H_{13}(j\omega) & \ldots & H_{19}(j\omega) \\ H_{21}(j\omega) & H_{22}(j\omega) & H_{23}(j\omega) & \ldots & H_{29}(j\omega) \\ H_{31}(j\omega) & H_{32}(j\omega) & H_{33}(j\omega) & \ldots & H_{39}(j\omega) \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ H_{91}(j\omega) & H_{92}(j\omega) & H_{93}(j\omega) & \ldots & H_{99}(j\omega) \end{bmatrix} = \begin{Bmatrix} Y_1(j\omega) \\ Y_2(j\omega) \\ Y_3(j\omega) \\ \vdots \\ Y_9(j\omega) \end{Bmatrix} \begin{Bmatrix} F_1(j\omega) \\ F_2(j\omega) \\ F_3(j\omega) \\ \vdots \\ F_9(j\omega) \end{Bmatrix}^{-1}. \tag{7}$$

As laid out above, the $[H_{ij}(j\omega)]$, $(i, j = 1, \ldots, 9)$ is 9 by 9 matrix of transfer functions with 81 possible contributions. By using $Y_i(j\omega)$, $(i = 1, \ldots, 9)$ and setting the input force vectors $F_j(j\omega) = 0$, $(j = 2, \ldots, 9)$, we obtain the displacement values of each DOF of the body segments and wheelchair components from Equation (7).

3.2. Magnitude and phase angle of the transfer function in complex terms.

$$|H_{ij}(j\omega)| = \frac{|Y_i(j\omega)|}{|F_j(j\omega)|}, \quad (i, j = 1, \ldots, 9), \tag{8}$$

$$\phi_{ij}(H_{ij}(j\omega)) = \tan^{-1}\left[\frac{\Im\{H_{ij}(j\omega)\}}{\Re\{H_{ij}(j\omega)\}}\right], \quad (i, j = 1, \ldots, 9). \tag{9}$$

Here $|H_{ij}(j\omega)|$ and $\phi_{ij}(H_{ij}(j\omega))$ are the magnitude and phase angle (radians) in complex terms of the transfer function associated with response displacement and force excitation, where $\Re$ is the real part and $\Im$ is the imaginary part of $H_{ij}(j\omega)$. Assuming that $Y_i(j\omega) =$

$a_i + b_i i$ and $F_j(j\omega) = c_j + d_j i$ are their complex values, plugging them into Equations (8) and (9) yields the following:

$$|H_{ij}(j\omega)| = \frac{|a_i + b_i i|}{|c_j + d_j i|} = \frac{\sqrt{a_i^2 + b_i^2}}{\sqrt{c_j^2 + d_j^2}}, \tag{10}$$

$$\phi_{ij}(H_{ij}(j\omega)) = \tan^{-1}\left(\frac{a_i + b_i i}{c_j + d_j i}\right) = \tan^{-1}(b_i/a_i) - \tan^{-1}(d_j/c_j). \tag{11}$$

4. **Assessment of the Biodynamic Model and Seat Cushions.**

4.1. **Validation of linear biodynamic model of wheelchair.** Figure 2 illustrates a measure of confidence for the model based on comparison with Patil & Palanichamy as well as the parameters in Tables 1 and 2. Over the frequency range of 0.5 to 11 Hz, the head-to-pelvis acceleration ratios yielded by the model exhibit a peak at approximately 3 Hz. The graph was derived from sinusoidal inputs with controlled confidence in steady-state and comparing the model results with those cited in the referenced articles. The test results indicate it is a quality composite model since good agreement is found between it and the corresponding models in [10].

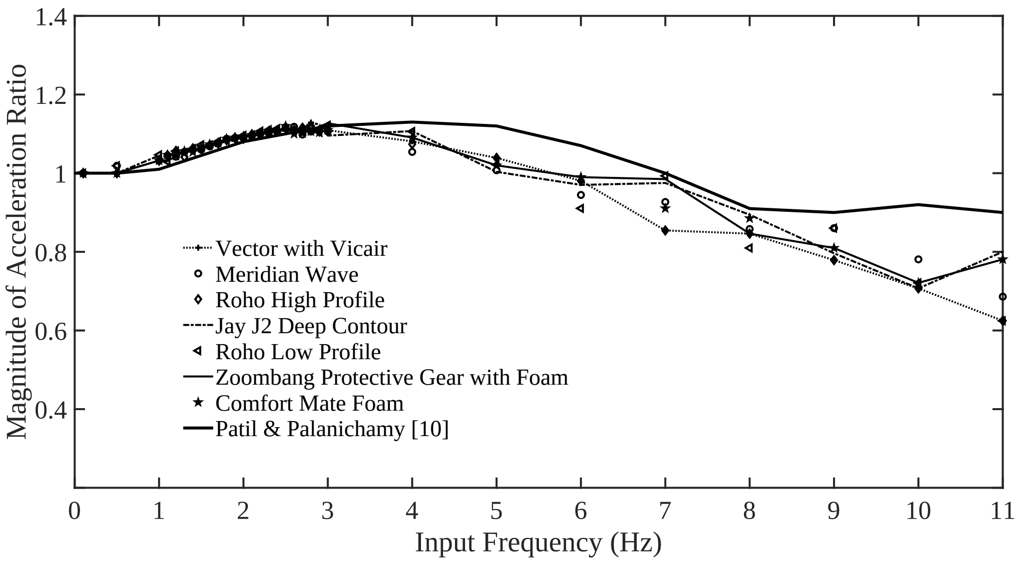

FIGURE 2. Comparison of output acceleration ratios obtained from seven seat cushion types

4.2. **Transfer functions for 9-DOF linear model.** Response displacements are obtained for each of the seven cushion seats in order to study the effects of vibration on the model's head and wheelchair seat. At peak magnitudes of transfer function from the head mass $|H_{91}(j\omega)|$ it is found that among all the cushion types tested, Meridian Wave has the highest response magnitude at frequency 2 Hz, while Comfort Mate Foam the lowest at frequency 2.1 Hz.

When results deriving from all the cushion types are listed in descending order, it is found that the types with lower damping constants would yield higher transfer function values and, conversely, those with higher damping constants would have smaller transfer functions as shown in Figure 3.

Figure 3 plots the magnitudes of the transfer functions. The model in this study has nine resonance frequencies and an absolute maximum transfer function. 1) We consider the magnitude of frequency response that should change across the frequency range upon

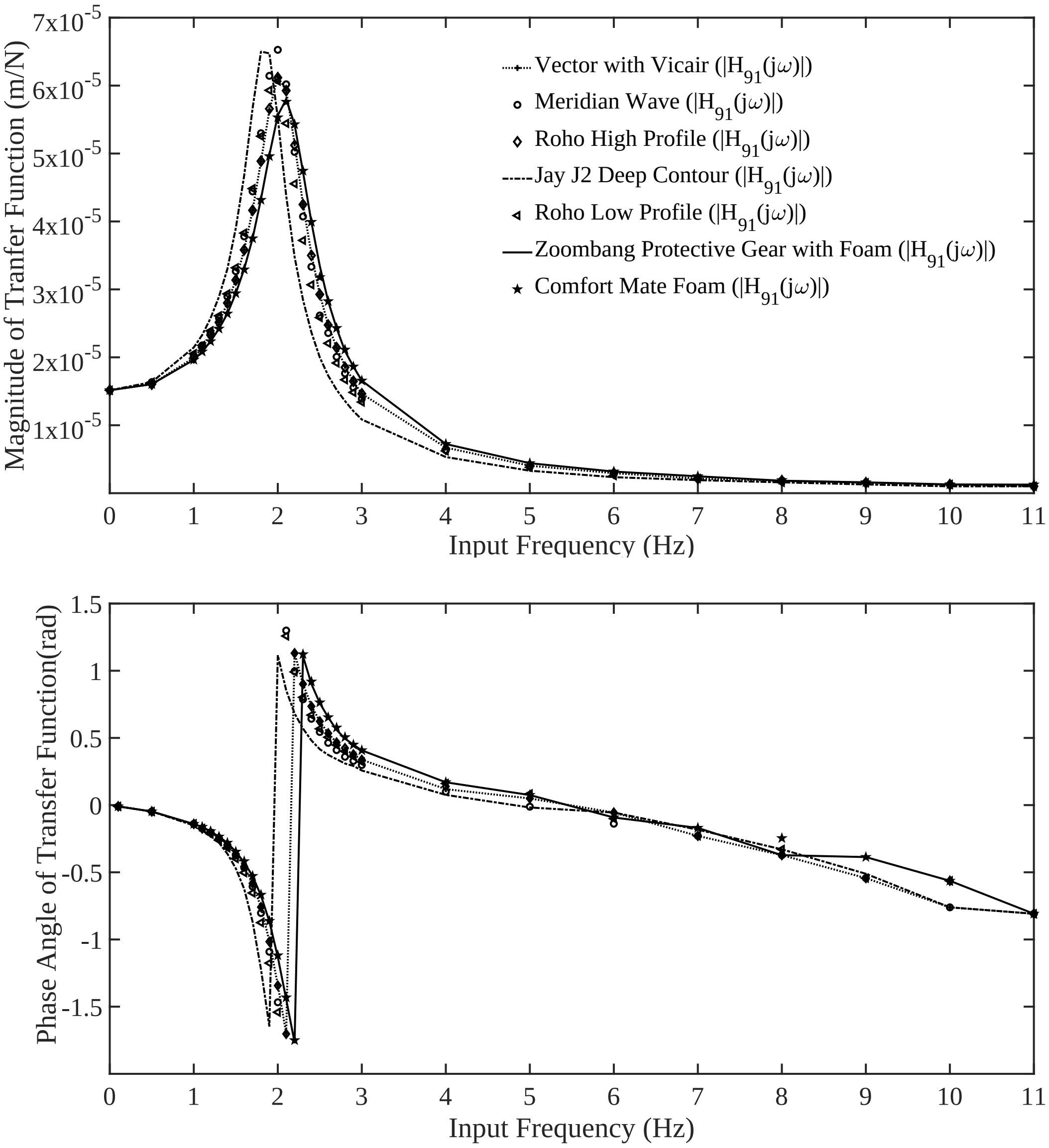

Figure 3. Comparison of magnitude and phase angle of transfer function from excitation to wheelchair occupant's head for tests on seven seat cushion types

any change in seat parameters. It should be noted that at the low end of excitation frequency, the response exhibited is roughly 1/k. So if we want to reduce this magnitude of frequency response then we would need to increase the stiffness constant (making k larger). 2) Whenever a peak reduction is desired, increases in the damping constant should thus be considered. Or damping control is key to controlling the resonance frequency.

In Figure 3, the vibration behavior of the system is related to the phase angle of the transfer function from each cushion seat according to input excitation and resulting displacement of the masses of the model – in this case the occupant's head is the component of interest. At the low end of excitation frequency, all seven types of cushion exhibit a phase angle approaching zero radians. This indicates that both the vectors of excitation and the head mass are pointing in the same direction, or being "in-phase". Towards higher frequencies, however, the phase angle becomes larger than zero radians. This means the vectors of input excitation and model response are moving at different speeds and directions, or being "out of phase".

As the excitation frequency approaches 1.8 to 2.2 Hz – the range of resonance frequency for all seven cushion types – the phase angle approaches $\pi/2$ radians. This means the excitation frequency is approaching the natural frequency of the wheelchair system, and the vectors of the input force and system response are vibrating in phase. As the excitation frequency increases well past the wheelchair's natural frequency, the phase angle will reach π radians. This indicates that the vectors of the input force and system response are vibrating out of phase.

5. **Conclusions.** This study provides a systematic approach for creating mass damper and stiffness matrices for determining transfer functions from the equations of motion of a 9-DOF model. The ordinary linear differential equations governing the model were solved using the method of Laplace transform and frequency function matrix, which enabled the determination of frequency response from the model which was simulated for seven types of seat cushion with differing properties of stiffness and damping. These two properties were found to be key determinants of the resulting frequency response of the system to input vibrations. The study illustrates a process through which mathematical modeling is carried out to reach specific findings and conclusions, a process that should prove useful for future engineering of the manual wheelchair. For future work, we design a method in consideration of the transient response of a system defined by linear ordinary differential equations that have initial conditions.

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REFERENCES

[1] S. Maeda, M. Futatsuka, J. Yonesaki and M. Ikeda, Relationship between questionnaire survey results of vibration complaints of wheelchair users and vibration transmissibility of manual wheelchair, *Environ. Health Prev. Med.*, vol.8, no.3, pp.82-89, 2003.
[2] T. H. Kim, Y. T. Kim and Y. S. Yoon, Development of a biomechanical model of the human body in a sitting posture with vibration transmissibility in the vertical direction, *Int. J. Ind. Ergon.*, vol.35, pp.817-829, 2006.
[3] C.-C. Ou and T.-C. Chen, Power-assisted wheelchair design based on a Lyapunov torque observer, *International Journal of Innovative Computing, Information and Control*, vol.8, no.12, pp.8089-8102, 2012.
[4] Y. Garcia-Mendez, J. L. Pearlman, R. A. Cooper and M. L. Boninger, Dynamic stiffness and transmissibility of commercially available wheelchair cushions using a laboratory test method, *Journal of Rehabilitation Research & Development*, vol.49, no.1, pp.7-22, 2012.
[5] M. F. Hikmawan and A. S. Nugraha, Analysis of electric wheelchair passenger comfort with a half car model approach, *ICSEEA 2016 International Conference on Sustainable Energy Engineering and Application*, pp.76-80, 2016.
[6] Y. Matsuoka, K. Kawai and R. Sato, Vibration simulation model of passenger-wheelchair system in wheelchair-accessible vehicle, *Journal of Mechanical Design*, vol.125, pp.779-785, 2003.
[7] W. Budihartoi, Intelligent controller of electric wheelchair from manual wheelchair for disabled person using 3-axis joystick, *ICIC Express Letters, Part B: Applications*, vol.12, no.2, pp.201-206, 2021.
[8] J. Marzbanrad and A. Afkar, A biomechanical model as a seated human body for calculation of vertical vibration transmissibility using a genetic algorithm, *Journal of Mechanics in Medicine and Biology*, vol.13, pp.1350053-1-1350053-18, 2013.
[9] G. A. Hassaan and N. Mohammed, Frequency response of 10 degrees of freedom full-car model for ride comfort, *International Journal of Scientific Research Engineering*, vol.4, no.1, pp.43-49, 2015.
[10] M. K. Patil and M. S. Palanichamy, A mathematical model of tractor-occupant system with a new seat suspension for minimization of vibration response, *Applied Mathematical Modelling*, vol.12, pp.63-71, 1988.
[11] A. Sezgin and Y. Z. Arslan, Analysis of the vertical vibration effects on ride comfort of vehicle driver, *Journal of Vibroengineering*, vol.14, no.2, pp.559-571, 2012.

[12] S. Y. Kim, J. R. An and K. S. Kim, A study on the stiffness characteristics according to the body pressure on the seat cushion for vehicle, *Indian Journal of Science and Technology*, vol.9, pp.1-6, 2016.

[13] S. Wang, L. Zhao, Y. Hu and F. Yang, Vibration characteristics analysis of convalescent-wheelchair robots equipped with dynamic absorbers, *Shock and Vibration*, vol.2, pp.1-16, 2018.

[14] Y. A. Wicaksono, L. Herdiman and S. Susmartini, Cushion seat design on manual wheelchair for people with paralysis using value engineering method to improve activity comfort: A preliminary study, *Journal of Physics: Conference Series*, vol.1450, no.1, pp.1-9, 2020.

[15] O. Lariviere, D. Chadefaux, C. Sauret and P. Thoreux, Vibration transmission during manual wheelchair propulsion: A systematic review, *Vibration*, vol.4, pp.444-481, 2021.

[16] B. Cvetanovic, D. Cvetkovic, M. Prascevic, M. Cvetkovic and M. Pavlovic, An analysis of the impact of agricultural tractor seat cushion materials to the level of exposure to vibration, *Journal of Low Frequency Noise, Vibration and Active Control*, vol.36, no.2, pp.116-123, 2017.

[17] K. Kohyama, K. Arisawa, Y. Arisawa and Y. Morishima, Burn on the buttocks caused by gel wheelchair cushion left on driver's seat of hand control car, *The Journal of Spinal Cord Medicine*, vol.42, no.6, pp.797-799, 2019.

[18] P. Weerapong, K. Hashikura, M. A. S. Kamal and K. Yamada, A model for the response of an occupant and wheelchair system subjected to vertical vibrations, *International Journal of Innovative Computing, Information and Control*, vol.17, no.6, pp.1823-1841, 2021.

[19] P. Weerapong, K. Hashikura, M. A. S. Kamal and K. Yamada, Mathematical modelling of an occupant and wheelchair system subjected to vertical vibrations, *2021 2nd International Symposium on Instrumentation, Control, Artificial Intelligence, and Robotics (ICA-SYMP)*, pp.63-66, 2021.

[20] C. C. Liang and C. F. Chiang, A study on biodynamic models of seated human subjects exposed to vertical vibration, *International Journal of Industrial Ergonomics*, vol.36, pp.869-890, 2006.

[21] M. R. Hatch, *Vibration Simulation Using MATLAB and ANSYS*, Chapman and Hall/CRC, 2001.