Investigating the Effect of Seat Suspension System and Cushion on the Dynamic Behavior of the 214 Helicopter Pilot's Body

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Abstract:

In this study, the influence of the stiffness and damping characteristics of the pilot seat cushion in the Bell 214 helicopter is investigated as two critical factors in reducing the transmission of vibrations to the pilot's body. Since the dominant vibrations in the Bell 214 occur at low frequencies typically between 2 and 15 Hz the seat suspension system is not effective in this range, and the seat cushion plays a more significant role in vibration isolation. First, the biodynamic response of the human body is validated by comparing the results with existing experimental and analytical data related to helicopter vibration exposure. Then, the biodynamic equations of motion are analyzed using a four-degree-of-freedom seated human model under various configurations, with and without suspension and seat cushioning. Finally, the frequency domain response is examined through three-dimensional plots to evaluate the effect of different cushion stiffness and damping values. The results indicate that selecting optimal mechanical properties for the cushion can significantly enhance pilot comfort and play an effective role in reducing vibration-induced physical injuries.

Keywords:

Ride comfort, cushion, Transmissibility, Mechanical Impedance, Apparent Mass.

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1- Introduction

Helicopters are utilized in a variety of military and civilian applications due to their unique capabilities. However, the high levels of vibration transmitted to the crew often lead to a range of health issues, including fatigue and long-term physical injuries. These vibrations primarily result from the operation of the rotors, gearbox, and engine, in addition to structural vibrations caused by the helicopter's unstable aerodynamics. Typically, these vibrations are transmitted to the pilot and passengers through the cabin floor and seats [1-5].

In efforts to understand the impact of vibrations on the human body, numerous researchers have explored multi-degree-of-freedom models of the seated human body [6]. These models range from single-degree-of-freedom models to more complex 11-degree-of-freedom models [7-18]. Notably, Griffin and Lewis, in 2001, replaced parts of the passenger's body with solid masses and investigated two- and three-degree-of-freedom models [10]. Suleiman et al. utilized a control method to manage the suspension system of a vehicle, with the vehicle modeled using a seven-degree-of-freedom (DOF) system, discussing various control strategies [17]. Among multi-degree-of-freedom models, the Wan-Schimmels model stands out for its excellent alignment with experimental tests [19]. Marzban Rad et al. further explored the biomechanical model of a seated human body to calculate vertical vibration transmissibility, using a genetic algorithm to introduce models with and without back support at specific frequencies [20].

Several studies have specifically focused on seat suspension systems and their effectiveness in reducing vibrations transmitted to the human body [21-32]. For example, Heydarian concluded that active suspension systems are more effective than semi-active systems in reducing vibrations at low frequencies [27]. Rakhja et al. (2010) conducted a review study examining the relationship between body vibrations and apparent mass, comparing it with the ISO 5982 standard [32]. Khanlo et al. (2024) found that modeling the seat cushion as a parallel spring and damper (Kelvin-Woigt model) yields more accurate results in predicting passenger comfort compared to the Maxwell model [33].

The earliest research on seat cushions and their role in vibration transmission dates back to 1989, when Corbridge and Griffin investigated the effects of vibration characteristics, seat type, and seating position on the transmission of vertical vibrations through seat cushions. They concluded that seat characteristics have a greater impact than other parameters. Their study of 10 different seat cushions found that the cushion's quality, particularly its durability, has a more significant effect on vibration transmission than factors like seating position and support [34]. In 1990, Gerard Gouw conducted experimental studies on the hardness of various cushions with different materials [35]. In 1997, Smith explored the effects of cushion thickness and hardness in reducing vibrations transmitted to the passenger's body [36]. Later, in 2004, Jian Pang compared the vibration transmissibility of two types of car seats, examining the non-linear interaction between the passenger and the seat [36]. In 1997, Wan et al. studied the optimal design of seat suspension systems, finding that the optimal damping coefficient for seats and cushions could vary by up to 65%, depending on cushion stiffness, in order to minimize vibration exposure to the passenger [37].

As previously discussed, while many studies have focused on whole-body vibration (WBV), there remains a gap in research regarding the effects of varying cushion stiffness and damping across different frequency spectrums. Therefore, this study aims to fill this gap by first introducing a biodynamic model of the human body under vibration. Next, a four-degree-of-freedom model of the seated human body is compared with previous experimental and analytical models. The biodynamic equations of motion are then explored for various configurations, both with and without seat suspension and cushioning. Finally, the effects of cushion stiffness and damping are analyzed in the frequency domain, with results presented through 3D diagrams for comparison.

2- Biodynamic mathematical modeling

The human body mass was assumed to be a specific value based on commonly accepted standards in biomechanical modeling. The mass value used in this research is 82 kg, which is a typical reference value for a standard adult human body, often used in similar studies in the literature (e.g., in biomechanical and biodynamic analyses). This assumption is in line with international standards and practices, such as those outlined by ISO 2631-1 for human body modeling.

As for the range of body mass, the stiffness and damping values concluded in this study are valid for a typical range of human body masses, which can vary between 50 kg to 100 kg. However, the results can be adjusted for individuals outside this range by scaling the parameters accordingly, but for this study, the assumption of an 82 kg body mass was deemed appropriate for the analysis of seat cushion properties.

As previously mentioned, numerous models with varying degrees of freedom have been introduced to study the human body's response to chair vibrations. Among these models, the Wan-Schimmels 4DOF model aligns well with Boileau's experimental model. Therefore, this study focuses on the mathematical description of the Wan-Schimmels 4DOF model [7]. In this model, the four degrees of freedom are used to represent the human body's vibration in the **Z**-direction. Fig. 1 illustrates the human body model, where body parts are depicted as masses connected by linear springs and linear dampers.

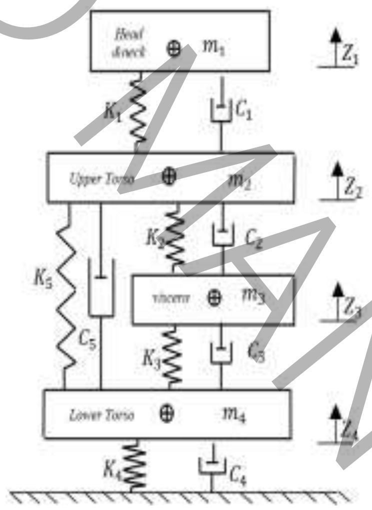


Fig. 1. 4 DOF wan – schimmels model [10].

In Fig. 1, ($\boldsymbol{m_1}$) represents head mass, ($\boldsymbol{m_2}$) upper body mass, ($\boldsymbol{m_3}$) small and accessory mass, and ($\boldsymbol{m_4}$) seat and thigh mass, and ***c*** and ***k*** represent, damping and stiffness.

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

The differential equations corresponding to each degree of freedom can generally be represented by Eq. (1) In this equation, the matrices [**M**], [**C**], [**K**], and {**P**} denote the mass matrix, the damping matrix, the stiffness matrix, and the matrix of external excitation forces, respectively. Eqs. (2) - (5) present the differential equations governing the 4 DOF system of a seated human body.

$$m_1\ddot{z}_1 + c_1\dot{z}_1 - c_1\dot{z}_2 + k_1 z_1 - k_1 z_2 = 0 \tag{2}$$

$$m_2\ddot{z}_2 - c_1\dot{z}_1 + c_1\dot{z}_2 + c_2\dot{z}_2 + c_5\dot{z}_2 - c_2\dot{z}_3 - c_5\dot{z}_4 - k_1 z_1 + k_1 z_2 + k_2 z_2 + k_5 z_2 - k_2 z_3 - k_5 z_4 = 0 \tag{3}$$

$$m_3\ddot{z}_3 - c_2\dot{z}_2 + c_2\dot{z}_3 + c_3\dot{z}_3 - c_3\dot{z}_4 - k_2 z_2 + k_2 z_3 + k_3 z_3 - k_3 z_4 = 0 \tag{4}$$

$$m_4\ddot{z}_4 - c_5\dot{z}_2 - c_3\dot{z}_3 + c_3\dot{z}_4 + c_4\dot{z}_4 + c_5\dot{z}_4 - c_4\dot{z}_5 - k_5 z_2 - k_3 z_3 + k_3 z_4 + k_4 z_4 + k_5 z_4 - k_4 z_5 = 0 \tag{5}$$

In Eq. (2) to 5, $\ddot{z}, \dot{z}, z$ represent acceleration, velocity and displacement vectors, respectively. m_i represents the mass and k_i represents the stiffness of the spring and C_i represents the damping.

In Fig. 2, the Kelvin-Woigt model was used in cushion modeling [33]. therefore, (m_c) is the mass of the seat cushion and C_c and k_c represent the damping and stiffness of the seat cushion. Therefore, the equation of motion related to the seat cushion is expressed as Eq. (6) and its motion is in the **Z** direction.

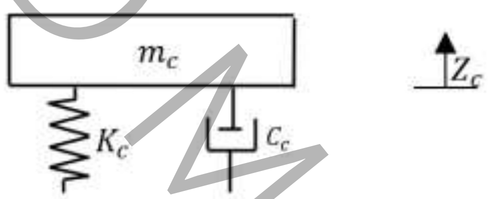


Fig. 2. Seat Cushion Mass-Spring Model

$$m_c\ddot{z}_c - c_4\dot{z}_4 + c_4\dot{z}_5 + c_c\dot{z}_c - c_c\dot{z}_s - k_4 z_4 + k_4 z_5 + k_c z_c - k_c z_s = 0 \tag{6}$$

In Fig. 3, m_s, the mass of the seat cushion and C_s and k_s represent the damping and stiffness of the seat suspension, respectively. Therefore, the equation of motion related to the seat suspension system is expressed as Eq. (7) and its motion is in the **Z** direction.

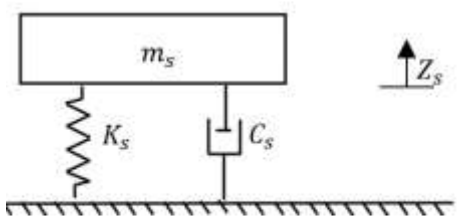


Fig. 3. Seat suspension mass-spring model

$$m_s\ddot{z}_s - c_5\dot{z}_c + c_c\dot{z}_s + c_s\dot{z}_s - k_5 z_c + k_c z_s + k_s z_s = 0 \tag{7}$$

Thus, the equations of motion governing the interaction between the human body, seat, and cushion form a system with six degrees of freedom. Four of these degrees of freedom are associated with the human body, while the remaining two pertain to the seat and cushion, as presented in Eqs. (2)- (7).

In examining the effects of vibrations on the body, the biodynamic equations of motion are typically analyzed, focusing on three key parameters: transmissibility, mechanical impedance at the moving point, and apparent mass. The transmissibility of the seat to the head, the resistance to movement, and the apparent mass equation are illustrated in Eqs. (8)-(10).

$$\mathrm{STHT}(j\omega) = \frac{X_n(j\omega)}{X_1(j\omega)} \tag{8}$$

$$\mathrm{DPMI}(j\omega) = \frac{F(j\omega)}{V(j\omega)} \tag{9}$$

$$\mathrm{APMS}(j\omega) = \frac{F(j\omega)}{a(j\omega)} \tag{10}$$

In Eqs. (1)-(8), STHT denotes transmissibility, which represents the ratio of seat-to-head displacement. DPMI stands for driving point mechanical impedance, defined as the ratio of excitation force to velocity, while APM refers to the apparent mass, which is the ratio of excitation force to acceleration. Additionally, jω represents the complex vector of the Fourier transform [13].

3- solution method

The solution approach involves using a four-degree-of-freedom biodynamic model of a seated human body to simulate the effects of seat cushion stiffness and damping on vibration transmission to the body. This analysis is performed both with and without the suspension system, focusing on low-frequency vibrations in the range of 2 to 15 Hz, which are typical for the Bell 214 helicopter.

In this section, the experimental and analytical results related to the biodynamic parameters of movement in different cases are calculated and compared with each other.

The model used solves the biodynamic equations of motion derived from the principles of human body dynamics under vibration exposure. These equations account for the interaction between the human body, seat cushion, and suspension system, and are solved numerically to evaluate the effect of different cushion properties (stiffness and damping) on comfort and vibration reduction.

3-1- Case study

In this part, results of the biodynamic equations of the human body under WBV, with and without seat suspension and cushion are compared. And in the next section, the effect of cushion stiffness and damping ratio in the various frequency domain in 3D diagrams, investigated. The characteristics of the 4 DOF model of human and seat and cushion, according to the mass of different parts and stiffness and damping of connections are shown in Table 1 and since this study was done mostly to investigate the vibrations on the helicopter pilot's body, therefore the frequency range of excitation is within the range of helicopter vibrations. The specifications related to the excitation frequency and the stiffness and damping of the seat suspension system and cushion, are listed in Table 2 and 3 [39-40].

The damping and stiffness properties of a seat cushion, such as those of a polyurethane cushion, are critical in understanding its performance under dynamic loading. Damping values, such as the damping ratio, are typically determined through experimental methods like vibration testing, where the cushion is subjected to controlled dynamic excitation, and its response is measured. Techniques such as free vibration decay, frequency response analysis, or shock testing are commonly employed to extract these characteristics. Previous studies and available experimental data often report damping ratios for polyurethane cushions in the range of 0.05 to 0.1. Given the limitations in experimental facilities for this study, the damping ratio was selected based on relevant literature, engineering assumptions for the Bell 214 helicopter's low-frequency vibration conditions, and references [38] and [39], which provide appropriate values for similar seat cushions.

The stiffness of the seat cushion, particularly in a helicopter setting, can also be determined through experimental testing or by referencing existing literature and material specifications. Stiffness is typically measured by applying a known static or dynamic load to the seat and calculating the deformation, with the stiffness value being the ratio of applied force to displacement ($k = F/x$). In the absence of experimental testing, as in the present study, stiffness values were estimated based on the standard mechanical properties of polyurethane and reliable engineering data from references [38] and [39], which outline the stiffness of similar materials under comparable loading conditions. These values were used to ensure the accuracy of the seat cushion's stiffness properties in the study.

It is important to note that in this model, all spring stiffness and damping are assumed to be linear. Additionally, the weight of a seated person is approximately 73% of their total body weight, excluding the weight of the lower thigh [14]. Based on this, the model represents a person with an average weight of around 82 kg, with a seated weight of approximately 60 kg (73% of 82 kg), excluding the weight of the lower body parts [12]. Following this, the biodynamic motion parameters derived from the four-degree-of-freedom model for human exposure to whole-body vibration (WBV), as illustrated in Fig. 4, have been compared and validated against previous experimental and analytical results [11]

Table 1. Characteristics of the mass, stiffness and damping of the human body [12]

Parameter	Value	Parameter	Value
m_1	6.5 (kg)	K_3	162 (kN/m)
m_2	30.5 (kg)	K_4	90 (kN/m)
m_3	9.5 (kg)	C_1	400 (N.s/m)
m_4	13.5 (kg)	C_2	4750 (N.s/m)
K_1	310 (kN/m)	C_3	4600 (N.s/m)
K_2	183 (kN/m)	C_4	2100 (N.s/m)

3-2- Verification

To ensure the validity of the analytical procedure employed in this study, a comprehensive comparison has been made between the analytical results of a four-degree-of-freedom seated human model and the well-established results from previous experimental studies by Boileau [11] and analytical studies by Wan-Schimmels [10]. Fig. 4 illustrates this validation process, showing that the mass, stiffness, and damping parameters used in the analysis were directly adopted from references [10, 12], resulting in a perfect match between the analytical outcomes of the present study and those reported by Wan-Schimmels. In particular, the natural frequency of the occupant's body, as seen in Fig. 4(a), is in the range of 5 Hz in the experimental results, while the previous analytical results showed a peak frequency between 5 and 6 Hz. The trend of changes related to transmissibility in the current analytical model aligns very well with the prior experimental and analytical results, confirming the accuracy of the calculations and the biodynamic modeling approach used in this study.

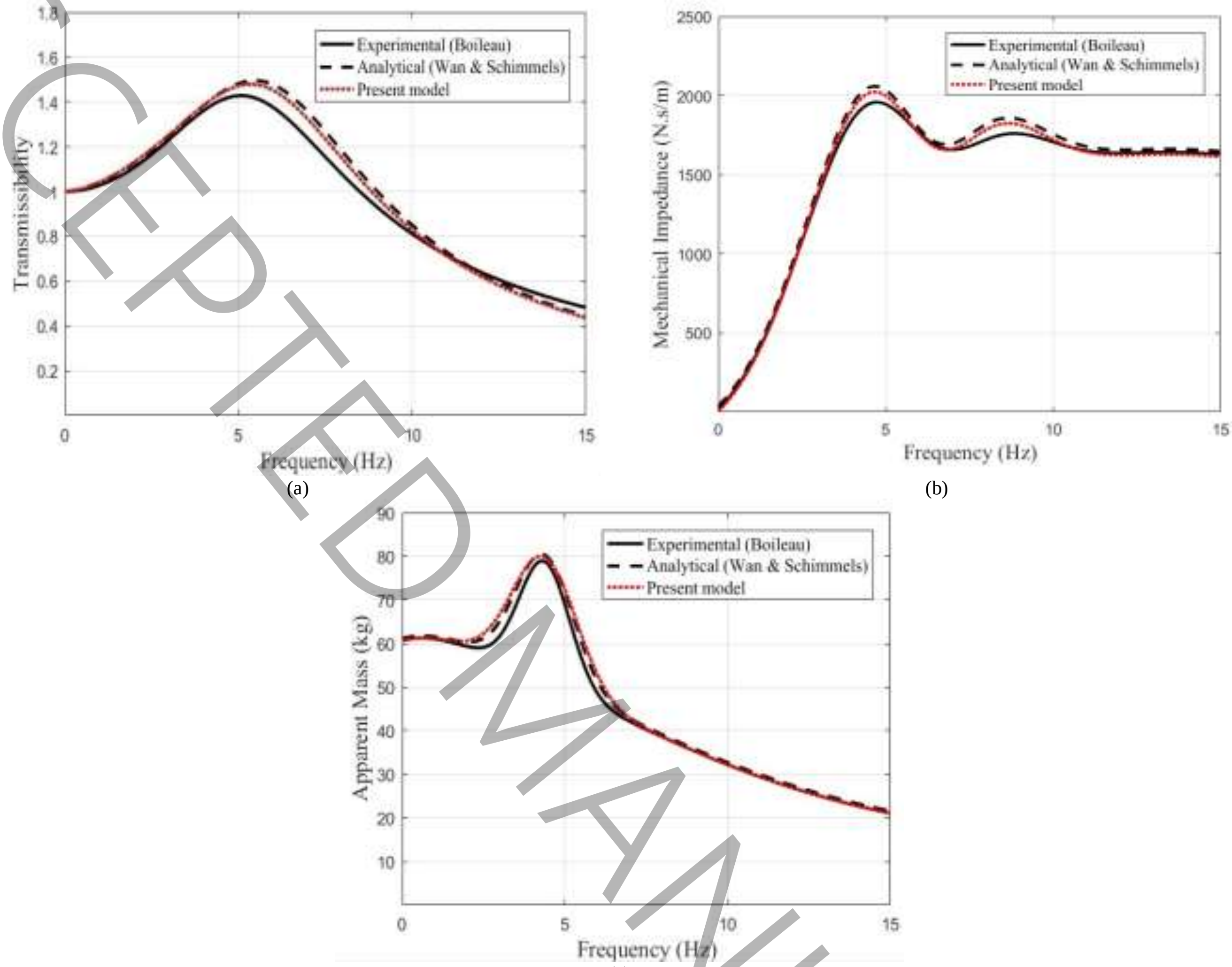


Fig. 4. (a) Validation of the analytical model of seat-to-head transmissibility by comparison with experimental and analytical model. (b) Validation of the mechanical impedance. (c) Validation of the apparent mass.

Furthermore, Figs. 4(b) and 4(c), which correspond to mechanical impedance and apparent mass, respectively, demonstrate a strong agreement between the previous experimental and analytical results and the current analytical findings. These agreements validate the correctness of the work and provide confidence in the modeling process. The discrepancies observed between the analytical and experimental results, as shown in Fig. 4, are consistent with those documented in previous studies [10-12, 14], further supporting the reliability of the results.

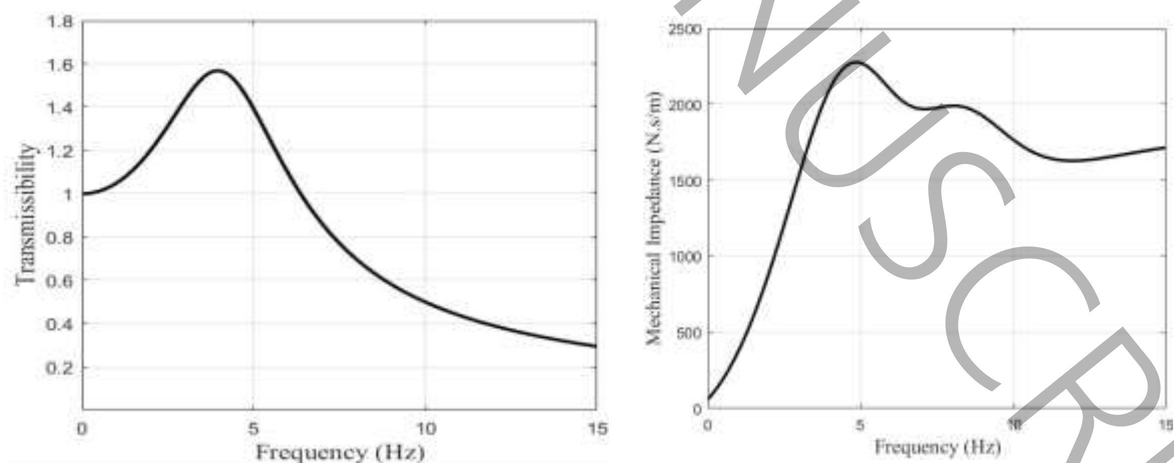
In addition, the four-degree-of-freedom model used in this study provides superior accuracy when compared to other multi-degree-of-freedom models, such as those with seven or nine degrees of freedom, as noted in reference [19]. Following the verification process, the study proceeds to calculate and compare the parameters such as transmissibility, mechanical impedance, and apparent mass in four different positions both with and without seat cushions and suspension systems—highlighting the significant impact of these factors on the overall biodynamic response.

After the verification, in the following, in four different positions, human with and without seat cushions and suspension system, parameters such as transmissibility, mechanical impedance and apparent mass are calculated and compared in various cases.

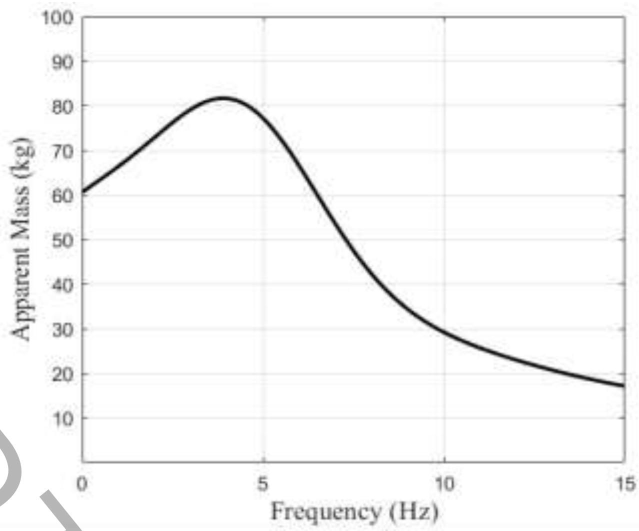
3-3- Human seated without suspension system and cushion

It is considered as the first position of human being sitting on a seat, without seat suspension system and cushion as shown in Fig. 1 and all the excitation force enters the human body. The characteristics related to the mass of each part of the body and the stiffness and damping of the joints are listed in Table 1.

In Fig. 5, the seat-to-head transmissibility, mechanical impedance, and apparent mass diagrams are presented, with the excitation frequency range spanning from 0 to 15 Hz.



(a) (b)



(c)

Fig. 5. (a) seat-to-head transmissibility of a human seated without considering the effect of the seat suspension and cushion. (b) Mechanical impedance of a human seated without considering the effect of the seat suspension and cushion. (c) Apparent Mass of a human seated without considering the effect of the seat suspension and cushion.

In Fig. 5(a), the peak of the transmissibility diagram occurs at 1.58 and at a frequency of 4 Hz. In Fig. 5(b), the first peak in the mechanical impedance diagram is observed at a value of 2300 N·s/m and a frequency of 4.5 Hz. In Fig. 5(c), the peak of the apparent mass diagram is observed at a value of 81 kg and a frequency of 4 Hz.

3-4- Human on seat with suspension system and without cushion

In this case, as shown in Fig. 6, a 5 DOF system is considered, where the human is sitting on a seat that only has a suspension system and no cushion.

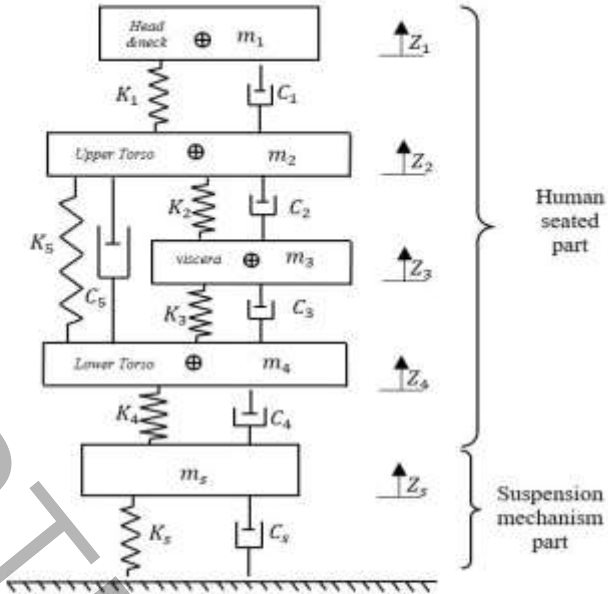


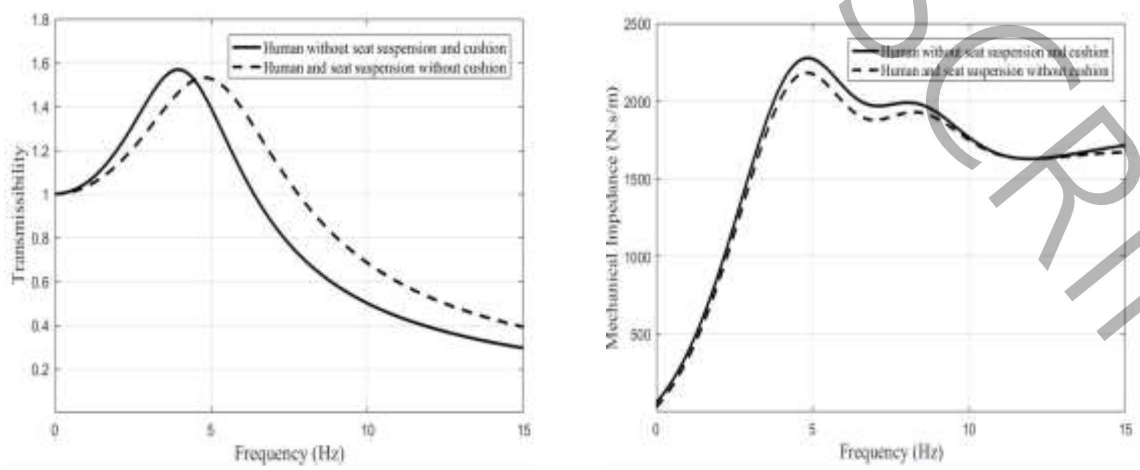
Fig. 6. The 5 DOF human and seat model

The characteristics of the human body are listed in Table 1, while the characteristics of the seat suspension system are provided in Table 2. An excitation frequency in the range of 0-15 Hz is applied at the bottom of the seat.

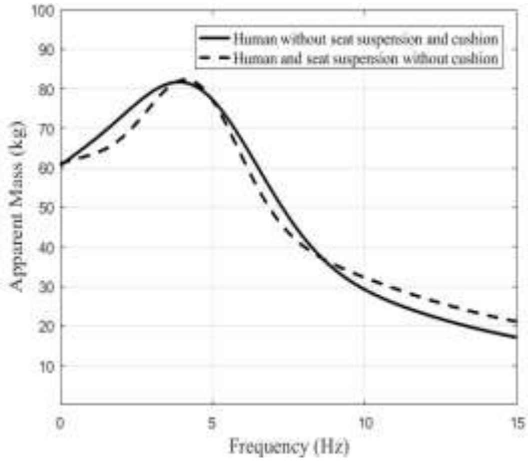
Table 2. Characteristics of seat suspension system [38-39]

Parameter	Value
Excitation frequency	0-15 (Hz)
m_s	45(kg)
k_s	250 (kN/m)
c_s	150 (N.s/m)

Some of the excitation force is absorbed by the seat suspension system, while the remainder is transmitted to the human body. The resulting biodynamic motion parameters are depicted in Fig. 6.



(a) (b)



(c)

Fig. 7. (a) Comparison of the seat-to-head transmissibility of a human seated in two situations with and without a seat suspension. (b) Comparison of the Mechanical Impedance. (c) Comparison of the Apparent mass

As illustrated in Fig. 7(a), in the absence of both a seat cushion and a suspension system, the peak of the transmissibility curve is observed at a value of 1.58 and a frequency of 4 Hz. However, when a seat suspension system is introduced, the peak of the transmissibility curve shifts to a value of 1.52 and a frequency of 4.85 Hz. This indicates that the presence of the seat suspension system results in a reduction of transmissibility by 0.06 units, or 3.7% at the peak, signifying a more favorable condition for the occupant.

In Fig. 7(b), when neither a seat cushion nor a suspension system is present, the first peak in the mechanical impedance diagram is recorded at 2300 N·s/m and a frequency of 4.5 Hz. In contrast, with the addition of a seat suspension system, a significant reduction in the magnitude of both the first and second peaks is observed. Specifically, the mechanical impedance at the first peak decreases by 100 N·s/m, reaching a value of 2200 N·s/m. Notably, in both configurations, there are minimal changes in frequency, with the primary effect being a decrease in the mechanical impedance value at the peak, which in turn leads to a 4.3% improvement in human comfort.

In Fig. 7(c), the position of the peak in the apparent mass diagram remains largely unchanged, both with and without the seat suspension system, occurring at a value of 81 kg and a frequency of 4 Hz. This suggests that the presence of the seat suspension system alone does not significantly affect the reduction of the apparent mass at the peak frequency.

3-5- Human on seat with cushion but without suspension system

In this case, as depicted in Fig. 8(a), 5DOF system is considered, where the human occupant is seated on a seat that consists solely of a cushion, with no suspension system.

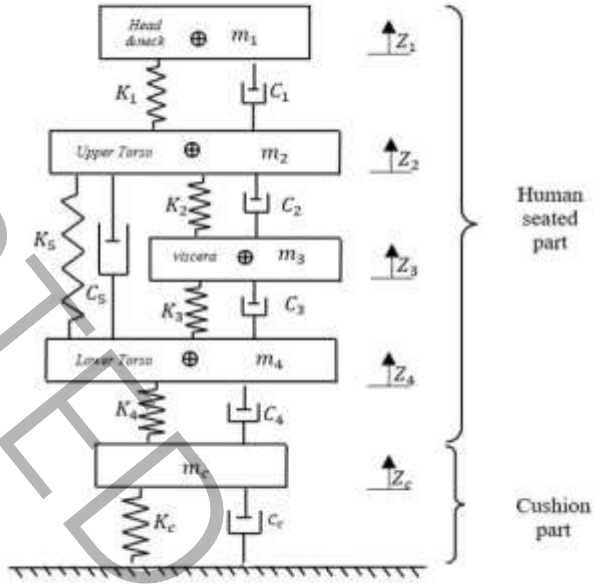


Fig. 8. 5 DOF human and cushion model

The characteristics of the human occupant are provided in Table 1, while the properties of the cushion are detailed in Table 3. An excitation frequency ranging from 0 to 15 Hz is applied at the bottom of the cushion.

Table 3. Characteristics of seat cushion [38-39]

Parameter	Value
Excitation frequency	0-15 (Hz)
m_c	0.7(kg)
k_c	20 (kN/m)
c_c	200 (N.s/m)

A portion of the excitation force is absorbed and damped by the cushion, while the remaining force is transmitted to the human body. The results of this interaction are presented in Fig. 9.

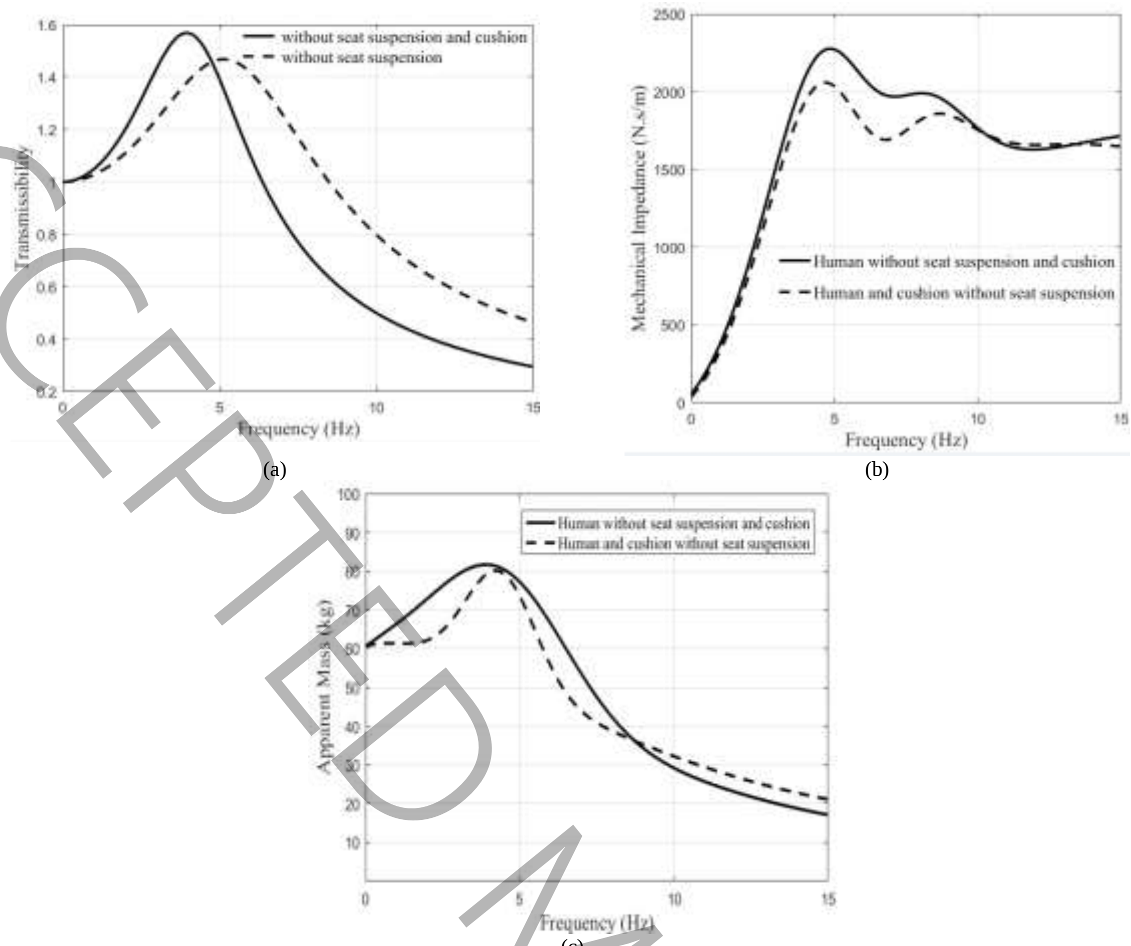


Fig. 9. (a) Comparison of the seat-to-head transmissibility of a human seated in two situations with and without a cushion. (b) Comparison of the Mechanical Impedance. (c) Comparison of the Apparent mass

As shown in Fig. 9(a), demonstrates that in the absence of both a seat cushion and a suspension system, the peak of the transmissibility curve occurs at a value of 1.58 and a frequency of 4 Hz. In contrast, when a seat suspension system is introduced, the peak of the transmissibility curve shifts to 1.43 at a frequency of 5 Hz. This indicates that, due to the presence of the cushion, the transmissibility at the peak is reduced by 0.15 units, or 9.4%, reflecting a significant improvement in comfort conditions for the occupant.

In Fig. 9(b), when there is no seat cushion or suspension system, the first peak in the mechanical impedance diagram occurs at 2300 N·s/m and at a frequency of 4.5 Hz. However, with the addition of a seat suspension system, both the first and second peak values decrease considerably. Specifically, the mechanical impedance at the first peak decreases by 250 N·s/m, reaching a value of 2050 N·s/m. Notably, in both configurations, the frequency remains relatively unchanged, with the primary alteration being a reduction in the mechanical impedance at the peak. This change leads to a 10.8% increase in human comfort.

In Fig. 9(c), the presence of the seat cushion results in a decrease in apparent mass both before and after the peak. However, the peak position of the apparent mass diagram, both with and without the seat suspension system, remains virtually unchanged, occurring at a value of 81 kg and at a frequency of 4 Hz. This observation suggests that the seat suspension system alone has a minimal effect on reducing the apparent mass at the peak frequency.

3-6- Human on seat with suspension system and cushion

The next system, as illustrated in Fig. 10, comprises a 6-degree-of-freedom (DOF) system, with the human occupant seated on a seat equipped with both a cushion and a seat suspension system. The characteristics of the human occupant are provided in Table 1, while the properties of the seat and cushion are detailed in Tables 2 and 3, respectively.

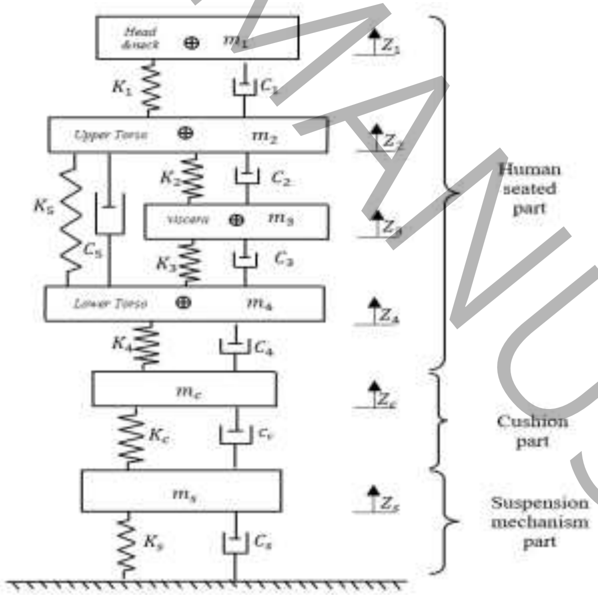


Fig. 10. 6 DOF human and seat suspension and cushion model

The excitation frequency is applied at the bottom of the seat. As a result, a portion of the excitation force is absorbed and damped by the seat suspension system and cushion, while the remaining force is transmitted to the human body. The outcomes of this interaction are presented in Fig. 11.

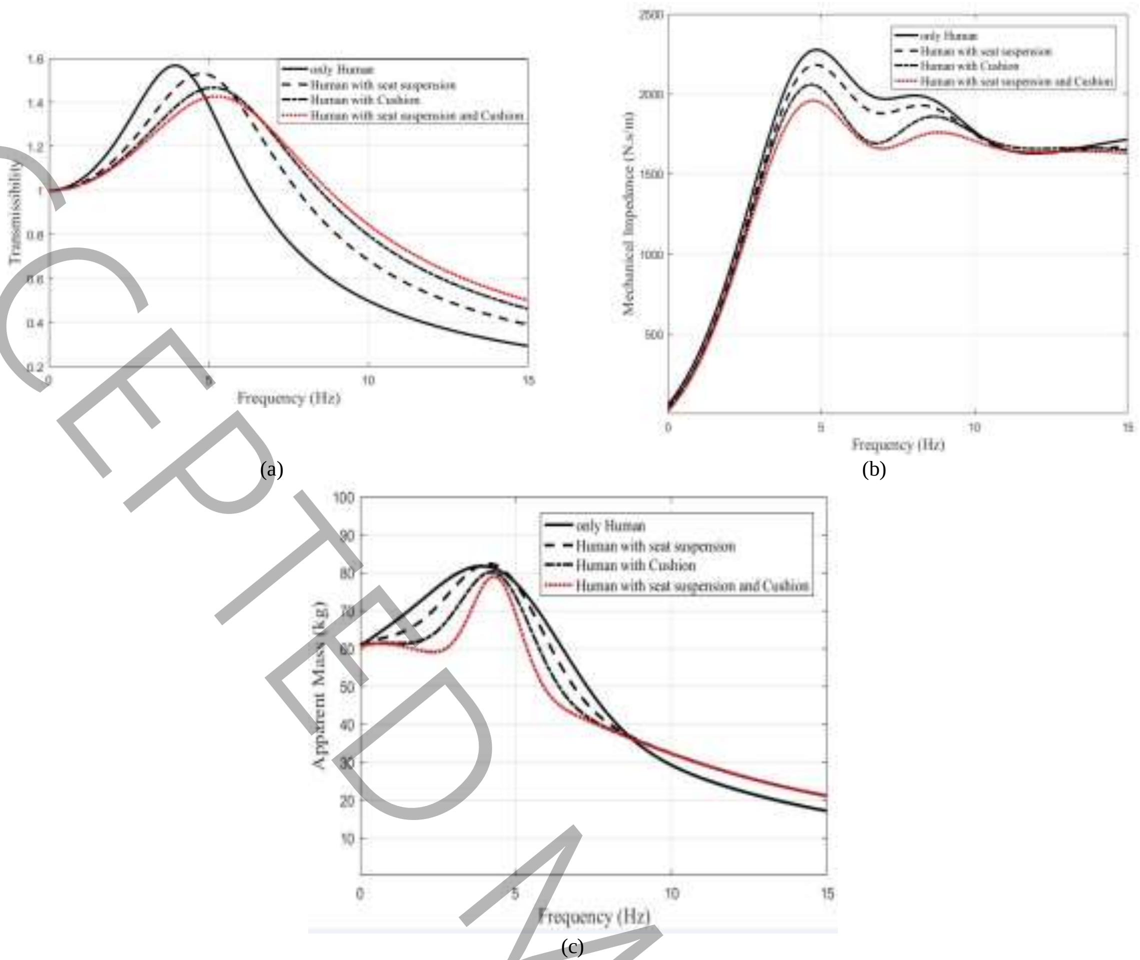


Fig. 11. (a) Investigating the effect of seat cushion and suspension system on the seat-to-head transmissibility of a human seated (b) Investigating the effect of the Mechanical Impedance. (c) Investigating the effect of the Apparent mass

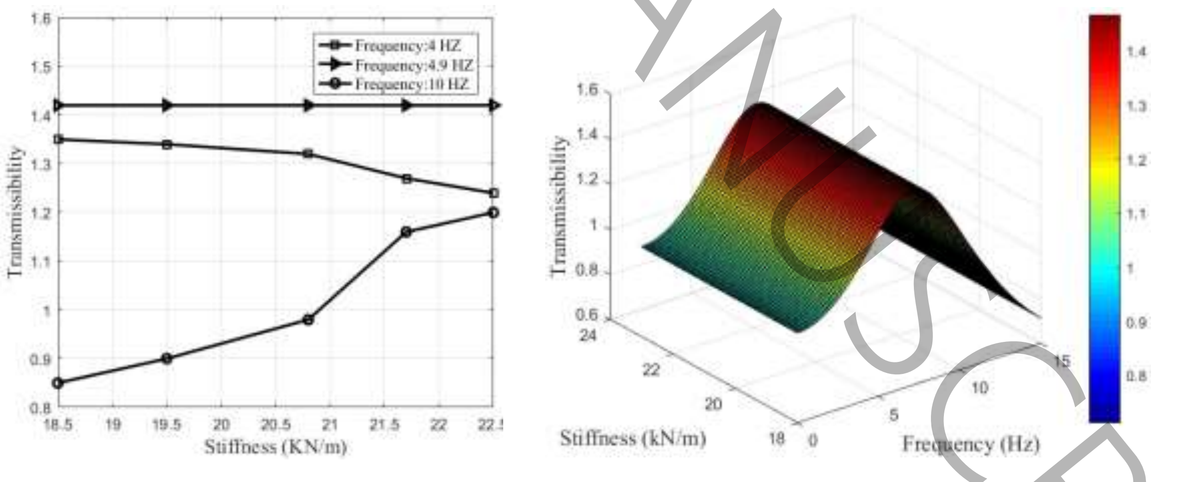
In accordance with Fig. 11, part (a) illustrates that in the absence of both a seat cushion and suspension system, the peak of the transmissibility diagram occurs at a value of 1.58 and a frequency of 4 Hz. However, when both the seat suspension system and cushion are present, the peak of the transmissibility diagram shifts to 1.41 at a frequency of 5.5 Hz. This indicates that, due to the addition of the cushion and seat suspension, the transmissibility value at the peak is reduced by 0.17 units, or 10.7%, reflecting more favorable conditions for the pilot.

In part (b), when there is no seat cushion or suspension system, the first peak in the mechanical impedance diagram is recorded at 2300 N·s/m and a frequency of 4.5 Hz. In contrast, when both the seat suspension system and cushion are added, the peak values at both the first and second peaks are significantly reduced. Specifically, the mechanical impedance at the first peak decreases by 350 N·s/m, reaching a value of 1950 N·s/m. Notably, in all cases, frequency changes are minimal, with the primary change being a reduction in the mechanical impedance at the peak. This reduction leads to a 15.2% increase in human comfort.

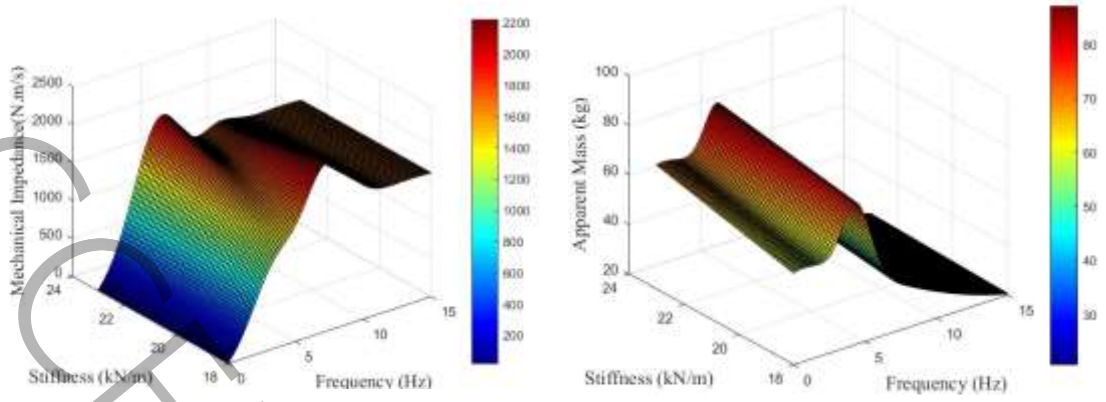
In part (c), when both the seat suspension and cushion are present, there is a decrease in the apparent mass before and after the peak. The value of apparent mass slightly decreases, reaching 78 kg at the frequency of 4 Hz. This indicates that the presence of the seat suspension and cushion contributes to a reduction in the apparent mass by 3.6%.

3-7- Investigating the effect of cushion stiffness

In this section, the effect of cushions with varying stiffnesses on the biodynamic equations of motion is examined. The model under consideration comprises a 6-degree-of-freedom system, as illustrated in Fig. 12. The characteristics of the masses, spring stiffnesses, and damping properties of both the human and seat are presented in Tables 1 and 2. The mass of the cushion is 0.7 kg, its damping coefficient is 210 N·s/m, and its stiffness varies between 18.5 and 22.5 kN/m [39]. The results of the biodynamic parameters, including seat-to-head transmissibility, mechanical impedance, and apparent mass, are shown in Fig. 12.



(a) (b)



(c) (d)

Fig. 12. (a)2D Investigating the effect of seat cushion and suspension system on the seat-to-head transmissibility of a human seated (b) Investigating of the seat-to-head transmissibility for human seated by considering the effect of cushions with different stiffness (c) Investigating of the mechanical impedance. (d) Investigating of the apparent mass

In accordance with Fig. 12, part (a) compares the transmissibility diagram across various cushion stiffnesses at three different excitation frequencies. At a frequency of 4 Hz (the frequency before the peak), the graph exhibits an increasing trend, indicating that as the stiffness of the cushion increases, the transmissibility rises by 33%, which leads to a reduction in human comfort. At a frequency of 4.9 Hz (the peak frequency), the graph remains relatively constant, showing that changes in cushion stiffness do not significantly affect the transmissibility at this frequency. At a frequency of 10 Hz (the frequency after the peak), the graph shows a decreasing trend, where the transmissibility decreases by 7% with an increase in cushion stiffness, thereby improving human comfort.

In Fig. 12(b), the changes in transmissibility are illustrated for different excitation frequencies and cushion stiffnesses. The graph demonstrates that increasing or decreasing the stiffness of the cushion directly influences the displacement of the peak of the transmissibility diagram by up to 5%. Furthermore, in Fig. 12(c), it is shown that by increasing the stiffness of the cushion from 18.5 kN/m to 22.5 kN/m, the mechanical impedance increases by 17%. In Fig. 12(d), it is evident that with an increase in the cushion stiffness from 18.5 kN/m to 22.5 kN/m, the apparent mass increases by 3.7%.

3-8- Investigating the effect of cushion damping

In this case, the effect of cushions with varying damping ratios on the biodynamic equations of motion is examined. The model comprises a 6-degree-of-freedom system, as shown in Fig. 13. The characteristics of the masses, spring stiffnesses, and damping properties of the human body and seat are provided in Tables 1 and 2. The mass of the cushion is 0.7 kg, its stiffness is 20 kN/m, and its damping ratio varies between 0.06 and 0.1 [39]. The results of the biodynamic parameters, including seat-to-head transmissibility, mechanical impedance, and apparent mass, are presented in Fig. 13.

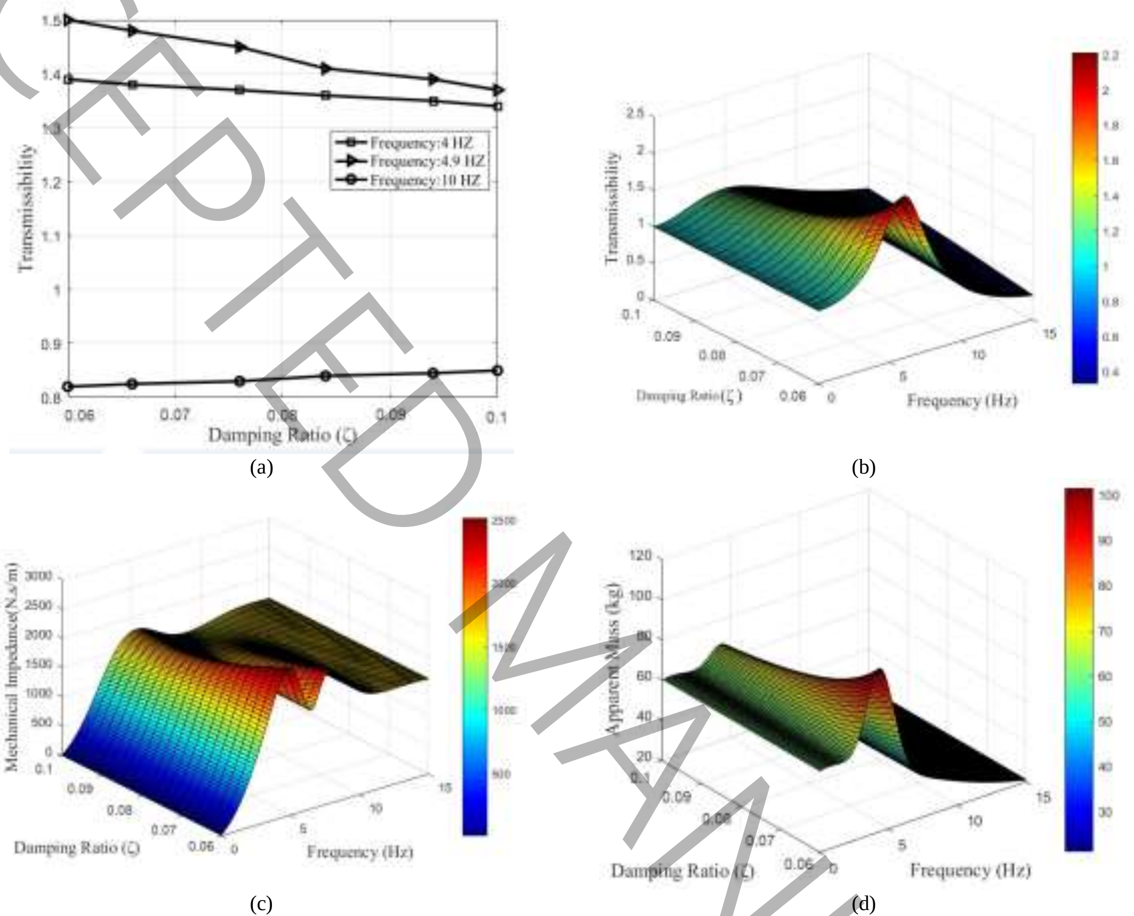


Fig. 13. (a) 2D Investigating of the seat-to-head transmissibility for human seated by considering the effect of cushions with different damping ratio (b) Investigating of the seat-to-head transmissibility for human seated by considering the effect of cushions with different damping ratio (c) Investigating of the mechanical impedance. (d) Investigating of the apparent mass

In accordance with Fig.13 in part (a) the transmissibility diagram vs various cushion damping ratio and in three different frequencies has been compared. By increasing cushion damping ratio at 4 Hz frequency and also at 4.9 Hz frequency, the peak of the transmissibility graph decreases 3-5%. In Fig. 13(b), the transmissibility changes are shown for different excitation frequencies and different cushion damping ratio, this shows that, increase of the cushion damping ratio, causing a decrease 0.5 unit or 29.4% in peak height. And also in Fig. 12(c), By increasing the damping ratio of the cushion from 0.06

to 0.1, the mechanical impedance has decreased up to 17% in 5 Hz frequency. In Fig. 12(d), By increasing the cushion damping ratio, the apparent mass has decreased 16%.

4- Conclusion

In this research, the effect of the helicopter's seat suspension system and cushion with various stiffness and damping exposed to helicopter vibrations is investigated. First, the results of the biodynamic equation of motion of the human body have been verified with previous experimental and analytical results. Next, the biodynamic equations of motion in different cases of the 4-degree-of-freedom model of the seated human body with and without seat suspension and cushion were compared. In the final section, the effect of cushion stiffness and damping in the frequency domain is investigated in 3D diagrams. The obtained results are summarized as follows:

• According to this study, the effect of the seat cushion is more effective than the seat suspension system in reducing the range of vibrations to the sitting human body.

• In the frequency range of 0 to 15 Hz, with the increase of cushion damping ratio from 0.06 to 0.1, the peak of the biodynamic equation of motion decreases significantly, and this is effective in increasing between 16-29.4% in the pilot's comfort during flight.

• Changes in the stiffness of the cushion affect the biodynamic equations of motion such as transmissibility, mechanical impedance, and apparent mass diagrams, so that with the increase of the stiffness of the cushion from 18.5 kN/m to 22.5 kN/m, human comfort decreases between 17-33%.

• The effects of damping and stiffness of the cushion conflict with each other, so according to this study, to have an optimal value of stiffness and damping of the cushion, the amount of damping ratio should be between 0.068-0.089 and the amount of stiffness of the cushion should be between 19.7-21.3 kN/m.

• For future research, it is recommended to investigate the effect of using nonlinear viscoelastic or auxetic materials for the cushion, which may offer improved vibration isolation performance. Additionally, incorporating time-domain analysis and considering the effects of multi-directional vibrations or pilot posture changes can lead to a more comprehensive understanding of pilot comfort in real flight conditions

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