Design, Analysis, and Implementation of a Four-DoF Chair Motion Mechanism

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ABSTRACT In this study, the proposed design of a motion chair with four degrees of freedom (DoF) provides the pilot with the tactile sensations required by the human body to increase the immersion of flight simulation training. The chair includes a seat pan, a back pad, and a seat height. For the development of the proposed mechanism structure, the modular design is used to achieve the required displacement and acceleration. In addition, inverse kinematics is also constructed to meet the actuation needs. The control system can achieve real-time closed-loop control using an industrial computer, a motion control card, servo motors and drivers. This enables the structure to reach destinations with the shortest motion time and completes the required T-curve motion trajectory. The displacement, velocity, and acceleration of the designed structure are analyzed using SimWise4D software to verify the performance specifications of the structure. The experimental results indicate that the four-DoF control meets requirements of displacement, velocity, and acceleration.

INDEX TERMS Motion chair, mechatronics, kinematics, motion analysis.

I. INTRODUCTION The training of pilots worldwide is typically lengthy. Therefore, flight training simulators are in high demand and serve as essential tools in flight training. As indicated in [1], a complete high fidelity motion simulation system was first constructed in the 1950s. In 1954, the General Precision Corporation of United States developed a motion simulation system that performed $3^\circ$ pitching, rolling, and yawing. After continual improvements, the system performed $10^\circ$ pitching, rolling, and yawing by 1964. In 1965, Stewart proposed the six-axis parallel motion platform structure [2]. In 1969, the hydraulic driver of a civil aviation simulator controlled the motion of each axis, which meant that it possessed six DoF. Thus, the basic design framework of the simulator was established in [3]. The contact sensations of the human body differ from the spatial sensations generated by the six-DoF platform. Subsequently, many studies explored and improved the framework, including discussion on identifying the relationships of six-axis motion and each actuator through kinematics [4] and discussion on how the disturbance rate of the compensation load could enhance positioning precision [5].

However, the oppressive sensation generated by gravity on human body can be provided by contact with the motion chair. In a study conducted by Gum [6], the testing of a research model indicated the presence of a time delay between the applied force and the perception of various mechanisms. Intramuscular and pressure detection structures perceived applied force at a faster rate than the vestibular system did. In 1976, Cardullo and Harpursville [7] engaged in chair design with the hope of using a chair to stimulate the tactile sensory system. They hoped that the seat would independently produce the expected skeletal posture changes, simulate regions related to skin contact changes, and reflect variation in the muscular pressure gradient. They also aimed to simulate the effect of acceleration. The chair contains two airbags, which form the seat and Back pads. Therefore, the control method could be used to independently drive the airbags in order to manipulate the height, position, and shape of the support surface. In the design, the tactile sensation of the body surface is altered using changes in airbags. In 2007, Berger et al. performed a study on perceptions and visual sensations [8] and indicated that
qualitative matching between the perception and the size of visual acceleration could enhance the information transmission between them. The results indicated that the use of visual effects in the construction of a motion notification warning algorithm could enhance the linear acceleration motion effect of the simulator. In [9], Bruschetta et al. used the Stewart platform in conjunction with a seat in a motion platform. Specifically, the researchers created the feeling of continuous low-frequency acceleration by placing eight inflators on the chair. Four were placed on chair’s back, two on the seat pan, and one on either side of the safety belt. The devices created the feeling of continuous low-frequency acceleration through the sense of restraint imposed by the safety belt after the devices inflate. In [10], the researchers designed a single DoF seat simulator to examine the fatigue the body perceived after prolonged driving. The simulator comprised a one-DoF chair, steering wheel, and visual effects and tested the body’s level of fatigue through the adjustment of a surge axis on the back. In [11], a two-DoF (roll and pitch axis) seat was developed. The researchers used the seat rolling and the surge axis rotation to test whether automated cars would be able to predict the position and behaviors of other cars after 3 seconds. In [12], a shaker rig was designed based on linear servomotors for testing ride comfort drivers perceived.

The majority of simulators have been designed based on the six-DoF structure of the Stewart platform [13]–[15], which simulates and presents a six-DoF overall movement. However, the Stewart platform is unable to simulate the kinematic sensations that a pilot experiences when flying. Therefore, this work proposes a four-DoF chair system for motion simulation. The system consists of a gravity chair that provides the user with somatosensory sensations. The purpose of the system is to provide pilots with tactile sensations related to motion. The bodily sensations that pilots experience while flying are converted into data. This enables pilots to experience relevant tactile sensations and makes flight simulation a more immersive experience. The design, analysis, and control of the chair are conducted based on mathematical model construction. The kinematic models are employed to describe the relationship between the movement position of the simulator in space and the rotating angles of each joint, in structure analysis, and to understand the limitations of the motion space. The seat pan component consists of a heave axis structure and a roll axis structure. The back pad component comprises a sway axis structure and a surge axis structure. The seat height structure consists of a whole-chair adjustable vertical structure. Because each component has a single level of motion DoF, the researchers refer to the method used in [16]–[18] and identified inverse kinematic model based on the geometrical relationship between the connecting rods. By contrast, inverse kinematics involves obtaining the displacement or angles of each joint based on reverse engineering of the movement position and attitude of the simulator in space.

Generally speaking, it is very difficult to use the Stewart platform to train pilots as a simulator, especially for the sense of gravity in a short distance movement. The paper proposes a four-DoF chair as the gravity motion needed for flight. To the best of our knowledge, the ideas we mentioned have not been presented in the previously published papers [1]–[18]. As a result, this paper presents some new ideas in the implementation of motion chair control systems, including an adjustable speed control system and short distance stroke control system.

II. KINEMATIC MODEL

The structure of the motion chair contains three components shown in Fig. 1. The seat pan component is composed of the roll and heave structures. The chair’s back pad component is composed of the surge and sway structures. The seat height structure of the whole chair is composed of the whole-chair adjustable height structure. As shown in Figure 1, it includes a pilot and the direction of three coordinate axes.
relationship in Eqs. (1)-(4).

\[
\begin{align*}
    l_2 \cos \theta_2 + l_3 \cos \theta_3 - l_4 \cos \theta_4 - \omega_1 &= 0 \quad (1) \\
    l_2 \sin \theta_2 + l_3 \sin \theta_3 - l_4 \sin \theta_4 - h_1 &= 0 \quad (2) \\
    l_4 \cos \theta_4 - l_3 \cos \theta_3 + l_2 \cos \theta_5 + l_6 \cos \theta_6 - (\omega_0 - \omega_1) &= 0 \quad (3) \\
    l_4 \sin \theta_4 - l_3 \sin \theta_3 + l_5 \sin \theta_5 - l_6 \sin \theta_6 - (h_0 - h_1) &= 0 \quad (4)
\end{align*}
\]

For the inverse kinematics of the roll axis, the angles of each link from \( \theta_0 \) and the turning angle of the seating plate must be obtained. First, \( \theta_2 \) is determined and then followed by solving for \( l \) based on \( \theta_0 \) and \( l_0 \) as follows:

\[
\theta_0 = \tan^{-1} \frac{h_0}{\omega_0} \quad (5)
\]

\[
l_0 = \sqrt{h_0^2 + \omega_0^2} \quad (6)
\]

\[
l = \sqrt{l_2^2 + l_3^2 - 2l_2l_3 \cos(\theta_0 + \theta_3)} \quad (7)
\]

Subsequently, \( \theta_a \) can be obtained based on \( l \), \( l_0 \), and \( l_6 \)

\[
\frac{l_6}{\sin \theta_a} = \frac{l}{\sin(\theta_0 + \theta_3)} \quad (8)
\]

We can derive

\[
\theta_a = \sin^{-1} \left( \frac{l_6}{l} \sin(\theta_0 + \theta_3) \right) \quad (9)
\]

Based on \( l \), \( l_2 \), and \( l_3 \), \( \theta_b \) can be obtained as

\[
\theta_b = \cos^{-1} \left( \frac{l_2^2 + l_3^2 - l_6^2}{2l_2l_3} \right) \quad (10)
\]

\( \theta_2 \) can be calculated and given as

\[
\theta_2 = \theta_a + \theta_0 - \theta_b \quad (11)
\]

For the next step, \( \theta_3 \) and \( \theta_4 \) are solved. As shown in Figs. 3(a)-(b), there are two solutions for \( \theta_3 \) and \( \theta_4 \).

According to Fig. 2 and Fig. 3(a), the solution 1 can obtain \( \theta_3 \). Next, \( l_{34} \) must be first solved and expressed as

\[
\theta_1 = \tan^{-1} \frac{h_1}{\omega_1} \quad (12)
\]

\[
l_1 = \sqrt{h_1^2 + \omega_1^2} \quad (13)
\]

\[
l_{34} = \sqrt{l_1^2 + l_2^2 - 2l_1l_2 \cos(\theta_2 - \theta_1)} \quad (14)
\]

Then, \( \theta_f \) can be obtained from \( l_{34} \), \( l_3 \), and \( l_4 \),

\[
\theta_f = \cos^{-1} \frac{l_3^2 + l_{34}^2 - l_4^2}{2l_3l_{34}} \quad (15)
\]

According Fig. 2, \( \theta_e \) can be expressed as

\[
\theta_e = \tan^{-1} \frac{l_2 \sin \theta_2 - h_1}{\omega_1 - l_2 \cos \theta_2} \quad (16)
\]

From Eqs. (15)-(16), \( \theta_3 \) can be obtained

\[
\theta_3 = \theta_f - \theta_e \quad (17)
\]

Similarly, \( \theta_d \) can be obtained from \( l_{34} \), \( l_3 \), and \( l_4 \), and then \( \theta_4 \) can be calculated as

\[
\theta_d = \cos^{-1} \frac{l_3^2 + l_{34}^2 - l_4^2}{2l_3l_{34}} \quad (18)
\]

\[
\theta_4 = 180 - \theta_d - \theta_e \quad (19)
\]

Fig. 3(b) is the other solution for \( \theta_3 \) and \( \theta_4 \),

\[
\theta_3 = -\theta_f - \theta_e \quad (20)
\]

\[
\theta_4 = 180 + \theta_d - \theta_e \quad (21)
\]

The next step involves solving for \( \theta_5 \). Given that \( l_3 + l_4 < \omega_1 \), there is only one solution for \( \theta_2 \) and \( \theta_5 \). Based on the following figure, \( \theta_5 \) can be obtained based on \( l \), \( l_2 \), and \( l_3 \), whereas \( \theta_5 \) can be obtained based on \( \theta_e \) and \( \theta_2 \),

\[
\theta_5 = \cos^{-1} \frac{l_2^2 + l_3^2 - l_6^2}{2l_2l_3} \quad (22)
\]

\[
\theta_5 = 180 - (\theta_5 - \theta_2) \quad (23)
\]

According to Eqs. (11), (17), (20), (21) and (23), the link angles can be obtained as control commands and then achieve the seating plate control of the roll region.

2) INVERSE KINEMATICS OF THE HEAVE AXIS
The seating plate structure of the heave axis is designed as the slider crank mechanism shown in Fig. 4. The mechanism is similar to a press machine as [17], [18]. In Fig. 4, \( l_6 \) represents the driving link; \( l_7 \) denotes the initial height of the seating plate; and \( z \) stands for the seating plate’s displacement. The
displacement and the angles of each link must satisfy constraints of Eqs. (24)-(25).

\begin{align*}
l_7 - z &= l_8 \sin \theta_8 + l_9 \sin \theta_9 \tag{24} \\
l_8 \cos \theta_8 + l_9 \cos \theta_9 &= 0 \tag{25}
\end{align*}

Inverse kinematics of the heave axis and angles of each link from z-axis displacement of the seating plate must be first obtained. As shown in the Figs. 5(a) and (b), there are two solutions for this relationship.

**FIGURE 5. Two solutions of \( \theta_8 \) and \( \theta_9 \): (a) solution1 (2) solution2.**

According to Fig. 5(a), the solution 1 obtains \( \theta_g \) and \( \theta_h \), \( z \), \( l_8 \) and \( l_9 \) must be first solved and expressed as

\begin{align*}
\theta_g &= \cos^{-1} \frac{l_8^2 + l_9^2 - (l_7 - z)^2}{2l_8l_9} \\
\theta_h &= \cos^{-1} \frac{(l_7 - z)^2 + l_8^2 - l_9^2}{2l_8(l_7 - z)} \tag{26} \\
\theta_9 &= 90 - \theta_h \tag{28} \\
\theta_8 &= 180 - (\theta_g - \theta_9) \tag{29}
\end{align*}

Next, \( \theta_8 \) and \( \theta_9 \) can be obtained as

\begin{align*}
\theta_8 &= \theta_h + 90 \tag{30} \\
\theta_9 &= \theta_g - (180 - \theta_8) \tag{31}
\end{align*}

By combining (30) and (31), we can obtain

\begin{align*}
\theta_9 &= \theta_g + \theta_h - 90 \tag{32}
\end{align*}

From observation, the link angles (30), (32) can be obtained as control commands and further attain the seating place control of the heave region.

**FIGURE 6. The schematic diagram of the four-link mechanism.**

B. KINEMATICS OF THE BACK PAD

1) INVERSE KINEMATICS OF THE SURGE AXIS

As shown in Fig. 6, a four-link mechanism is used for the surge axis structure of the chair’s back. In Fig. 6, \( l_{11} \) represents the driving link and \( l_{13} \) represents the chair’s back. The angles of each link must satisfy Eqs. (33)-(34).

\begin{align*}
l_{11} \cos \theta_{11} + l_{12} \cos \theta_{12} - l_{13} \cos \theta_{13} - l_{10} &= 0 \tag{33} \\
l_{11} \cos \theta_{11} + l_{12} \sin \theta_{12} - l_{13} \sin \theta_{13} &= 0 \tag{34}
\end{align*}

The surge axis’s inverse kinematics involves obtaining \( \theta_{11} \) and \( \theta_{12} \) from the turning angle and \( \theta_{13} \) of the rear panel. In Figs. 7(a) and (b), there are two solutions with the collinear boundaries of links \( l_{11} \) and \( l_{12} \).

**FIGURE 7. Two solutions of \( \theta_{11} \) and \( \theta_{12} \): (a) solution1 (2) solution2.**

According to Fig. 7(a), the solution 1 obtaining from \( l_{11,12} \), \( l_{10} \), \( l_{13} \) and \( \theta_{13} \) must be first solved and expressed as

\begin{align*}
l_{11,12} &= l_{10}^2 + l_{13}^2 - 2l_{10}l_{13}(180 - \theta_{13}) \tag{35} \\
&= l_{10}^2 + l_{13}^2 + 2l_{10}l_{13}(\theta_{13})
\end{align*}

Next, \( \theta_{11} \) and \( \theta_{12} \) can be obtained based on \( \theta_i \), \( \theta_j \), and \( \theta_k \) as follows:

\begin{align*}
\theta_i &= \cos^{-1} \frac{l_{10}^2 + l_{11,12}^2 - l_{13}^2}{2l_{10}l_{11,12}} \tag{36} \\
\theta_j &= \cos^{-1} \frac{l_{13}^2 + l_{11,12}^2 - l_{12}^2}{2l_{13}l_{11,12}} \tag{37}
\end{align*}
The structure’s dimensions in the simulation are $l_2 = 10mm$, $l_3 = 180mm$, $l_4 = 3mm$, $l_5 = 10mm$, $l_6 = 22.5mm$, $h_0 = 16mm$, $h_1 = 4.5mm$ and $\omega_0 = \omega_1 = 22.5mm$. With regard to the kinematic equations, $\theta_2$, $\theta_3$, and $\theta_4$ can be solved using $\theta_0$, and the verification process can be conducted by substituting them into Eqs. (1)–(4). The obtained error is within the range of $10^{-13}$ to $10^{-15}$. The dimensions of the seating place heave structure used in the simulation are $l_7 = 70mm$, $l_8 = 3mm$ and $l_9 = 70mm$. With regard to the inverse kinematic equations, $\theta_8$ and $\theta_9$ can be solved using $z$, and the verification can be conducted by substituting them into Eqs. (24)–(25). The result is that the obtained error is within the range of $10^{-13}$ to $10^{-15}$. For the kinematics of the chair’s back surge axis structure, the dimensions are $l_{10} = 388.5mm$, $l_{11} = 52.5mm$, $l_{12} = 192.2mm$, $l_{13} = 224mm$. The program is first executed the inverse kinematics equations in the subsection B part of section II part, and solved for $\theta_{11}$ and $\theta_{12}$ based on $\theta_{13}$. The verification could then be conducted by substituting them into Eqs. (33)–(34). The obtained error is within the range of $10^{-13}$ to $10^{-15}$. The sway axis structure of the back pad is the same as the heave axis. The dimensions used in the simulation are $l_7 = 75mm$, $l_8 = 15mm$ and $l_9 = 75mm$. The obtained error of the simulation verification is within the range of $10^{-13}$ to $10^{-15}$. From Eq. (43), we can observe that the kinematic model of the seat height is fairly simple. Hence, its verification is omitted here.

III. DESIGN AND ANALYSIS OF THE MOTION CHAIR

This section is divided into five regions. First, the design of the seat pan is mainly based on the roll-axis region and heave-axis region. Next, the design of the back pad is mainly based on the surge-axis region and sway-axis region. Finally, the seat height is designed so that the height of the entire seat is adjustable. In order to effectively verify the displacement, speed and acceleration of this designed mechanism, the SimWise 4D software is used to analyze the displacement, speed and acceleration of the mechanism. The CAD modeling of the work is constructed by the Solidworks software. Then, the solid modeling of the chair is inputted the SimWise 4D software to simulate the dynamic motion of the chair. The SimWise 4D is the mechanical software. SimWise 4D is for design and engineering professionals developing products involving assemblies of 3D parts [19].

A. SEAT PAN MECHANISM

1) MECHANISM DESIGN

According to Fig. 1, the roll region is the rotating chair base with a dual base plate. Fixed phase differences exist for the motion of the dual base plate. The design of the dual base plate is a design of 180° such that the single DoF on one side can be extended to the other side to execute the roll motion. The structure is designed to sustain the weight of a human body and the weight of the chair surface. Thus, a toggle joint structure module design is used. The size of the chair is 600 mm * 560 mm as shown in Fig. 8. Figures 9(a) and (b) respectively represent schematics of the platform surface and the supporting frame.

FIGURE 8. Chair dimensions.

The roll axis has to generate a rolling effect on the seat pan with a larger force required to be exerted larger body mass. Hence, a toggle structure is used for the design as shown in Figs. 10(a), (b), and (c). The structure length is 249.42 mm. As shown in Figs. 11(a) and (b), the design of the heave axis involves a combination of eccentricity and link.

2) ANALYSIS OF THE MECHANISM

The displacement, velocity, and acceleration of the seat’s base may be analyzed assuming that the roll axis motor rotates at a speed of 300 rpm. The results are shown in...
Figs. 12(a), (b), and (c), respectively. It is observed that the maximum displacement is $0.515 - 0.431 = 0.084$ m, the position is $\pm 0.042$ m. The speed is $1.53 - (-1.33) = 2.86$ m/s, and the speed is $\pm 1.43$ m/s. The peak to peak acceleration is $62.3 - (-38.2) = 100.5$ m/s$^2$, or the averaged acceleration is $\pm 50.25$ m/s$^2$.

Assuming that the heave axis motor rotates at a speed of 300 rpm but the roll axis motor is still, the displacement, velocity, and acceleration of the chair’s base are analyzed, the results are shown in Figs. 13(a), (b), and (c), respectively. Then the maximum displacement is $0.481 - 0.475 = 0.006$ m and the position is $\pm 0.003$ m. The speed is $0.0908 - (-0.0908) = 0.1816$ m/s, and the speed is $\pm 0.0908$ m/s. The peak to peak acceleration is $2.83 - (-3.09) = 5.92$ m/s$^2$, or it can be said that the averaged acceleration is $\pm 2.96$ m/s$^2$.

**B. BACK PAD MECHANISM**

1) MECHANISM DESIGN

The surge and sway regions are shown in Fig. 1. A design for surge is based on the design of rotary vanes. The main mechanism involves driving using a four-bar linkage such that the structure is actuated within a small space. On the other hand, the sway makes use of a four-bar linkage crank sliding structure and generates a movement rate of $\pm 15$ mm through the design of the crank shaft length. In addition, the chair’s back motion contains both surge and sway motion. Therefore, the mechanism design requires two motors to drive surge axis and sway axis. Figures 14 and 15 respectively show the dimensions of the supporting platform surface and the contact surface between the back pad and the chair’s back.
The design of the surge axis is based on rotary vanes such that the structure could be actuated within a small space. The main mechanism involves driving a four-bar linkage using a motor. The structure requires a small space and the required force exertion is not large. Figure 16 illustrates the eccentricity of the crank shaft. The sway axis motion mainly depends on the structure of the chair’s back. Therefore, the focus of the designed planning is on the crank–slider structure of a four-bar linkage. The dimensions of the crank and the link are shown in Figs. 17 (a) and (b).

2) ANALYSIS OF MECHANISM

At a rotating speed of 300 rpm, the surge and sway directions of the chair’s back are analyzed. The results are shown in Fig. 18 and Fig. 19, respectively. In Fig. 18(a), the maximum displacement is 0.032-0.0144 = 0.176m and the position is ±0.0088m. The speed in Fig. 18(b) is 0.277-(-0.294) = 0.571 m/s and the speed is ±0.285m/s. The acceleration in Fig. 18(c) is 9.24(-8.54) = 17.78m/s² and the acceleration is ±8.89m/s².

In Fig. 19(a), the maximum displacement is 0.000281-0.0297 = 0.029981m and the position is ±0.01488m. For Fig. 19(b), the speed is 0.476-(-0.476) = 0.952 m/s and the speed is ±0.476m/s. The acceleration shown in Fig. 19(c) is 17.7-(-11.8) = 29.5m/s² and the acceleration is ±14.75m/s².

C. SEAT HEIGHT MECHANISM

1) MECHANISM DESIGN

The design of adjustable height region is shown in Fig. 1. The height of the seat can be adjusted by the motor, slide rail and ball screw. The all-movable structure for the whole chair is composed of the whole-chair adjustable height structure.
Therefore, the seat will be pushed by a ball screw-driven platform. The design of the schematic diagram of the mechanism is shown in Fig. 20.

The modeling of the chair is constructed by the Solidworks software. The dimension of the modeling picture is verified dimension through the ANSYS design model. In order to understand the dimension and the FreeCAD, this redraws the solid modeling using the Solidware software. They are shown in Figs. 8-11 and Figs. 14-17. The analytical results of SimWise 4D simulation are shown in Figs. 12-13 and Figs. 18-19.

IV. IMPLEMENTATION

As implied in Fig. 21, the study has three main aspects, namely: (1) design, analysis, and production of motion chair structures, (2) construction and verification of the inverse kinematic models and (3) establishment of a control box for the motion control system. The motion control system completes the position control and velocity control of the multiaxial motor. Figure 22 shows the flowchart of the implementation. The system is implemented and shown in Figs. 23(a)-(b). The system includes four major parts: the control box, the seat pan, the back pan and the seat height.

The control box contains an industrial computer, five sets of server drivers, the main circuit power supply, and a 24V power. The desktop computer is connected to the six-axis motion control card via a peripheral component interconnect (PCI) express bus. The model of PCI express bus is PCI-1265-AE produced by Advantech. Table 1 shows the expected stroke, and acceleration values for the five movable structures. Table 2 shows the stroke, and acceleration data.
obtained from the structure analysis conducted in Section 2. The motor parameters are listed in Table 3.

V. EXPERIMENTAL VALIDATION

The measurement steps for these components and development of the four-axis chair motion system are next described.

For displacement measurement, the total maximum displacement is measured. The used tools are a height gauge, a vernier caliper, and a metal ruler. The measurement reference points of the moveable mechanism are shown in Figs. 24(a)-(f). The measured values for the four DoF and adjustable height conditions are as follows: 43 mm for total surge displacement, 30 mm for total sway displacement, 5.5 mm for total heave displacement, and 79 mm for total roll displacement. The
TABLE 3. The parameters of the motor.

| Item                        | Surge | Sway | Roll | Heave | Seat height |
|-----------------------------|-------|------|------|-------|-------------|
| Power dissipation (kW)      | 0.75  | 1.3  | 3.0  | 3.0   | 2.0         |
| Rated voltage               | 220 V | 220 V| 220 V| 220 V | 220 V       |
| Rotor inertia (Kg-m²)       | 0.00025| 0.00201| 0.00180| 0.00180| 0.001214    |
| Rated speed (rpm)           | 3000  | 1500 | 2000 | 2000  | 2000        |
| Maximum speed (rpm)         | 3800  | 3000 | 2500 | 2500  | 2500        |
| Rated torque (N-m)          | 2.39  | 8.34 | 14.33| 14.33 | 9.55        |
| Maximum torque (N-m)        | 7.16  | 23.3 | 42.69| 42.69 | 28.65       |
| Weight (kg)                 | 3.05  | 8.9  | 13.87| 13.87 | 10.16       |

FIGURE 24. The measurement reference point, (a) adjustable height, (b) surge axis, (c) sway axis, (d) heave axis, (e) left side of roll axis and (f) right side of roll axis.

The model of ball screw used is FSIW-R25-5T4-172-265, and the lead is 5 mm. The total displacement of the adjustable height structure is 55 mm. The pn354 parameter of five drivers is set to 2000, which means the pulse per unit (PPU) is 2000. Starting from zero, the five servo drivers increase the pulse rate by 200, and carry it out 10 times before completing a rotation around the servo motors. The relationships between the motion path of each axis and the pulse rate are shown in Figs. 25(a)-(e). Figures 25(a) and 25(b) show the measured sway-axis and heave-axis errors as the ±0.8mm and ±0.5mm, respectively. Since the structure of the back pad is a design with a four-link mechanism and a slider crank mechanism, the slight errors occur in the assembly tolerances and links processing. Figure 25(c) shows the measured heave-axis error around ±0.2 mm. The measured result...
of the roll axis is shown in Fig. 25(d). The seat plate is the toggle joint structure. Observed from the figure and the seat plate assembly tolerance, it can be seen that the error caused by the installation of the links is about ±0.6mm. Figure 25(e) demonstrates the measured adjustable height structure of the whole chair error around ±0.05mm. The measured displacement of each DoF is close to the expected displacement. Figure 26(a) displays the acceleration measurements of the roll axis, heave axis, and adjustable height using an accelerometer. Figure 26(b) shows the acceleration measurement of the surge axis. The accelerometer is attached to the chair’s back for this measurement. Figure 26(c) shows the acceleration measurement of the sway axis motion path. The accelerometer is attached to the support frame on the chair’s back for this measurement. Time domain analysis conducted using the accelerometer measurements reveals the changes in acceleration under a fixed rotation speed shown in Figs. 27(a)-(e). Figure 27(a) shows a time domain diagram of the surge axis at the rotational speed of 100 rpm. The maximum acceleration 50.1538 m/s^2 could be achieved under this rotation speed. Figure 27(b) shows a time domain diagram obtained as the sway axis is driven at a rotational speed of 100 rpm. The figure indicates that the maximum acceleration could be 3.88169 m/s^2 at this speed. The heave axis is driven at a rotation speed of 300 rpm, resulting in
the time domain diagram in Fig. 27(c), which reveals that the maximum acceleration $24.1206 \text{ m/s}^2$ could be reached. Figure 27(d) is a time domain diagram generated as the servo motor rotates at a speed of 100 rpm. The maximum acceleration could be reached $24.9898 \text{ m/s}^2$ under this condition. Figure 27(e) is a time domain diagram of the adjustable height structure under a rotation speed of 300 rpm. Counting from the starting point of the diagram, the maximum acceleration can be reached $3.49275 \text{ m/s}^2$. Figures 28(a)-(b) and Figures 29(a)-(b) show the responses of the roll axis in torque and rotation speed without and with a load. Under the influence of structure changes and parameter settings, the results indicate that the roll axis is the axis that directly sustains human body weight. Therefore, it is more sensitive to driver parameters. Figures 30(a)–(e) show the T-curve velocity response when the rotation speed is 300 rpm. Figures 31(a)–(e) present the velocity response at 300 rpm after the loading of a 90 kg individual. The figures indicate that that tracking of initiation and stopping on the T-curve is not ideal. The set parameters can be saved to the specified path through the file writing function. Figures 28–31 show the chair velocity responses, the measurement uses the motion control card to store the relative speed of the servo motor. In the load test, a person weighing 90 kg sits on the seat. Therefore, Figure 29 and Figure 31 mainly measure
the influence of seat startup, transient and steady state on load disturbance. Without modifying the control parameters of the control loop. The reason behind this is that the same controller parameters were used in all the testing sessions. The purpose of using the same parameters is to understand whether the same set of parameters could be used to obtain the expected outcomes under different conditions. Table 4 shows the measurement results.

VI. CONCLUSION
In this paper, a four DoF and adjustable height chair has been discussed. With the aim of achieving the preplanned motion paths and effects, the design, analysis, mathematical model building, simulation, and servo driver establishment for the chair structure are conducted. The target of the study is to develop a structure platform with four DoFs, adjustable height, relevant structure design, mathematical derivation, and software/hardware development. Analysis of different DoFs is performed to gain understanding of the assembly of the structure and further complete its production. Experimental results indicate that the developed chair can approximately sustain a weight of 90 kg. Moreover, the dynamic motion of the chair is simulated and the analytical results include the displacement, velocity and acceleration of the chair. However, the structure strength of the chair is not this main topic of the study that will be considered to evaluate the design in the future work.

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M.-Y. Wei et al.: Design, Analysis, and Implementation of Four-DoF Chair Motion Mechanism

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