Design of Customized Rehabilitation Devices and Bench Testing System

A Thesis Presented

By

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ABSTRACT

Off-the-shelf rehabilitation devices are currently prescribed to assist patients with stroke. Current fabrication processes of custom-made rehabilitation devices are time consuming and laborious. The process could be only performed by skilled therapists. In addition, quantitative assessment of mechanical properties is crucial in the design of customized rehabilitation devices. By the design and the real time implementation of a 3D printed hand exoskeleton and a biomimetic testbed for Ankle-Foot Orthoses (AFOs), the improved digitalized methodologies of design and bench testing systems for customized rehabilitation devices were presented in this study.

A customized 3D printed hand exoskeleton (the EXCELSIOR) was developed and prototyped to assist stroke patients for finger extension exercises. 3D printing was combined with 3D scanning to create a custom-fit clamp. Compliant finger elements were designed and optimized utilizing Finite Element Analysis. Embedded strain gauges were applied to measure angular positions of the finger joints.

In addition, a novel biomimetic testbed was designed to perform stiffness measurement and functional analysis for AFOs. A biomimetic footplate was designed to adjust pivot centers for the metatarsophalangeal (MTP) joint and the ankle joint according to the patient specific anatomy. Feedback control systems were developed and real time implemented to perform AFO stiffness measurement. An impedance control system was developed and real time implemented to simulate the kinematics of the human ankle for further functional analysis in gait.

Real time implementation of the hand exoskeleton and AFO testbed proved the concepts of the design and the testing for customized rehabilitation devices.

Keywords: Ankle Foot Orthoses, Exoskeleton, Rehabilitation, Rapid Prototyping, Bench Testing
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1 Introduction and Problem Description

1.1 Problem Description
Off-the-shelf rehabilitation devices, such as ankle-foot-orthosis (AFOs) and hand rehabilitation exoskeletons, are currently designed to assist the therapist, and improve patients’ motor skills. However, the standard rehabilitation devices do not provide individualized comfort or support the wide anthropometry and motion conditions of affected populations. Currently, custom-made medical devices typically provide improved comfort but the fabrication is a laborious and time-intensive manual process performed only by highly skilled orthotists [1].

The unique advantages of Addictive Manufacturing and 3D scanning have been adopted by health care industries because of their fabrication flexibility and complexity [2]. This also coincides with the trend of increasing quantitative human body sensing by design of smart wearable devices [3]. Therefore, an improved digitalized technique for designing and manufacturing customized medical devices could liberate therapists from the laborious tasks. In addition, embedding sensors in customized rehabilitation devices could provide quantitative feedback to the therapists and health care practitioner. Furthermore, a biomimetic robotic assessment system could help therapists measured mechanical properties for customized medical devices, and use the data to optimize the design.

1.2 Problem Statement
The goal of this research is to improve and develop current computerized methods to design and testing of customized rehabilitation devices. To develop a new testing method of customized orthotic devices, a biomimetic AFO testbed is developed to simulate the dynamics of patient specific ankle-foot complex in gait cycle and to evaluate mechanical properties of customized 3D printed AFOs. In addition, a customized 3D-printed hand exoskeleton with embedded strain gauges (EXCELSIOR) is developed utilizing the computerized design method of customized rehabilitation devices.

1.3 Significance
Approximately 795,000 people in the US suffer a stroke annually, making it the leading cause of long-term disability in the nation [4]. Of these stroke victims, 85% have arm impairment [5], 65% recover with some degree of impairment or gait abnormalities, and require rehabilitation to help regain motor functions.
Independently perform activities of daily living (ADL) is one of the main goals for post-stroke patients. Upper extremity rehabilitation aims to restore patients’ hand function such as eating, writing, grasping manipulation. For the lower extremity rehabilitation, the therapy aims to help patients to stabilize gait cycle[6, 7].

1.4 Overview
This thesis is organized in a project-based format. In chapter 2, the background of the AFO testbed and hand rehabilitation devices are introduced. Chapter 3 describes the mechanical design, control system design and physical implementation of the AFO testbed. Chapter 4 presents the mechanical design, sensors integration, and sample testing of the EXCELSIOR device. Finally, chapter 5 presents the conclusion and future works.
2 Background

2.1 Background on the Upper Extremity Rehabilitation Devices

2.1.1 Hand Anatomy

The hand is responsible for nearly 90% of upper limb function. A large number of joints and bones provide the motion capabilities of the hand[8]. The hand has 27 bones, and the fingers are labeled digits 1-5, with the thumb as digit 1, and the little finger as digit 5 (seen in Figure 1).

Every finger is connected to the palm with metacarpophalangeal (MCP) joints. The MCP joints allow flexion-extension as well as a slight degree of axial rotation. The MCP joint of the thumb allows approximately 75-80° of flexion, while the second through fifth MCP allows for about 90°. Active extension at these joints is 25-30°. Approximately 20° of abduction-adduction is allowed at the MCP. However, the abduction-adduction movements of the MCP joints are restricted in flexion [8].

The interphalangeal (IP) joint is a hinge joint provides flexion and extension movements of the finger. The thumb has one interphalangeal joint, and each of the second through the fifth digits has one proximal interphalangeal (PIP) joint and one distal interphalangeal (DIP) joint. About 110° of flexion are allowed at the PIP joints and 90° at the thumb IP joint. Extension reaches 0° at the PIP and 25° at the thumb IP joint. The motions available at the DIP joints consist of nearly 90° flexion and 25° extension.
Grasping is one of the main functions and movements of the hand. Grasps can be classified based on power and precision movements. People use different numbers of fingers to grasp an object depending on the size of the object. During the grasping, the thumb could be always separated from the other fingers to form a jaw [10, 11]. Joint compressive forces at the MCP, PIP, and the DIP joints of the index fingers during isometric hand functions are presented in Table 1.
Table 1 Joint Compressive Force (N) for Various Hand Functions [13]

<table>
<thead>
<tr>
<th>Hand Function</th>
<th>DIP Joint</th>
<th>PIP Joint</th>
<th>MCP Joint</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tip pinch</td>
<td>2.4-2.7</td>
<td>4.4-4.9</td>
<td>3.5-3.9</td>
</tr>
<tr>
<td>Key pinch</td>
<td>2.9-12.5</td>
<td>4.9-19.4</td>
<td>14.7-27.1</td>
</tr>
<tr>
<td>Pulp pinch</td>
<td>3.0-4.6</td>
<td>4.8-5.8</td>
<td>4.0-4.6</td>
</tr>
<tr>
<td>Grasp</td>
<td>2.8-3.4</td>
<td>4.5-5.3</td>
<td>3.2-3.7</td>
</tr>
<tr>
<td>Briefcase grip</td>
<td>0.0-0.0</td>
<td>1.7-1.9</td>
<td>1.0-1.3</td>
</tr>
<tr>
<td>Holding glass</td>
<td>2.5-2.9</td>
<td>4.3-4.4</td>
<td>4.0-4.1</td>
</tr>
<tr>
<td>Opening big jar</td>
<td>5.2-9.5</td>
<td>7.2-14.2</td>
<td>14.8-24.3</td>
</tr>
</tbody>
</table>

(Data from An et al., 1985)

2.1.2 Hand Rehabilitation Devices

Much work has been done in the area of hand rehabilitation devices. There are two main groups of upper limb robotic systems classified by mechanical characteristics: the exoskeleton and operational machines/end-effectors [14]. Several existing products will be presented in this study.

Interactive Motion Technologies, Inc (Watertown, MA) sells a hand robot mounted on a companion planar robot (MIT-Manus). The MIT-Manus allows 2 active DOFs at the shoulder and the elbow, while the InMotion Hand is optional module mounted senses grasping forces. A software platform is developed to provide exercise games for patients [15]. However, the hand module only allows 1 DOF motion. Therefore, it could not capture information from each finger.
The University of California, Irvine developed a hand-wrist assisting robotic device (HAVARD) to assist functional grasping and releasing movements for stroke patients. The device is actuated by pneumatic cylinders, and it provides 3 DOFs for thumb, fingers, and wrist. The range of motion (ROM) of the MCP joint is about 25° to 90°, and 20° extension to 15° flexion at the wrist [16]. However, pneumatic actuation system is complex and expensive.

Compared with end-effectors, the exoskeleton could carry with patients. Therefore it could help patients to perform rehabilitation exercises at home.

There are a lot of hand exoskeletons with various structures. Linkage mechanisms and cable-driven mechanisms are widely adopted in existing hand exoskeletons. Figure 5 is an application of hand
exoskeleton using four-bar linkage mechanisms. The device is developed by the Harbin Institute of Technology, China. Each finger elements are driven by a brushless motor, and the maximum output force is up to 8N. However, these types of exoskeleton are very complicated. The linkage mechanisms will increase the weight and the size. In addition, building this type of exoskeleton is very expensive.

Figure 5 Hand Exoskeleton using Four-bar Linkage Mechanisms[17]

Cable-driven mechanism is another widely used structure in existing hand exoskeleton products. Saebo, Inc sells a type of cable-driven hand exoskeleton (the SaeboFlex) for stroke rehabilitation (as seen in Figure 6) [18]. The advantage of cable-driven mechanism is that this mechanism is light-weight and low cost and easy to adjust to customized hand geometry. The disadvantage is that the cable works only in traction, there for a couple of cables should be implemented for both flexion and extension. Also, the resistive force in SaeboFlex is generated by one spring. Therefore, force distribution within DIP, PIP, and MCP joint is uncontrollable.
2.1.3 Rapid Prototyping Techniques

There are unique challenges in designing and manufacturing rehabilitation devices for patients. Variation in the biomechanical properties of the patients leads to various design requirements. In addition, customized rehabilitation devices are small batch products. Therefore, the traditional mass manufacturing process is difficult and expensive to fabricating personal rehabilitation devices.

Rapid prototyping techniques (also known as “Additive Manufacturing”, “3D Printing”, and “Layered Fabrication”) is a fast, accurate, and cost-effective way to fabricate complex parts. It is uniquely suited in biomedical fields. In the rapid prototyping process, parts are built in vertical layers. At each point on the z-axis, it deposits structural material at the desired location. In order to create complex shapes, support materials will be filled in the gaps where there is no part material. After the material is solidified then the machine can move to build the next layer.

The drawbacks of the current rapid prototyping techniques are the cost and the strength of the material. Since the parts are built layer by layer, the strength of the 3D printed parts are not as strong as ones fabricated by traditional manufacturing processes. Also, the time and the cost of the 3D printed parts are higher than traditional ways in mass fabrication tasks. A combination of rapid prototyping techniques and traditional manufacturing processes will make the rehabilitation devices much cheaper and personal.
2.2 Background on the Testing of Lower Extremity Medical Devices

2.2.1 Ankle-Foot Anatomy

The ankle-foot complex has two important functions in walking: stable the stance, and provide propulsion [19]. Therefore, an understanding of the anatomy of this area is essential for designing lower extremity medical devices.

The ankle joint is the articulation between the talus and the two bones (the distal tibia and the fibula) of the lower leg (seen in Figure 7). The ankle joint is a hinge joint allows flexion-extension movement as well as inversion-eversion (seen in Figure 8). The ankle allows approximately 25° dorsiflexion as well as 50° plantar flexion. For normal people, the inversion is about 5-10°, and the eversion is about 5°. The talus and the calcaneus provide inversion and eversion. During the passive motion, the articular surfaces and ligaments govern joint kinematics, with the articular surfaces sliding upon each other without appreciable tissue deformation [19-21].

![Figure 7 Ankle-Foot Anatomical Bone Structure](image-url)
Figure 8 Movements of the foot at the ankle and tarsal joints: (a) dorsiflexion and plantar flexion; (b) supination and pronation [19]

The ankle joint is actuated by several muscle groups. The prime dorsiflexors of the foot are tibialis anterior, extensor digitorum longus, while the major plantar flexors are gastrocnemius and the soleus[20].

Figure 9 Prime Dorsiflexors of the Ankle[20]
The foot is usually described as an elastic arch structure consists of 26 bones with numerous articulations. Together, the foot provides the support of the body. During the walking, the metatarsophalangeal (MTP) joint provides propulsion (seen in Figure 11). The MTP joint allows approximately 70° dorsiflexion [19-21, 23, 24].

Figure 10 the Major Plantar Flexors of the Ankle[20]

Figure 11 Dorsiflexion of the metatarsophalangeal Joint[25]
2.2.2 Human Gait

The human gait cycle includes two phases: the swing phase and the stance phase (seen in Figure 12). During the stance phase the foot is on the ground. The stance phase constitutes about 60% of the gait cycle and ends at the point of toe-off. The swing phase covers the remaining 40% [23].

Drop foot is a gait abnormality which leads to inability to dorsiflex the foot during the swing phase. Nearly 20% of stroke survivors suffer from drop foot. Ankle-foot Orthoses (AFOs) are frequently prescribed to help stabilize the ankle-foot complex with limited dorsiflexion [26].

![Figure 12 Phases of Human Gait Cycle](image)

2.2.3 Custom-made Ankle-Foot-Orthoses (AFOs)

The design of the AFO is essential as different designs are reported to affect outcomes in temporospatial parameters, ankle kinematics and knee kinematics. Parameters such as AFO ankle
angle, AFO stiffness is based on clinical measurement [28, 29]. Some evidence indicates that the AFOs require individual adjustment so as to obtain optimal function [30].

Currently, creating a custom-made orthosis is a time-consuming laborious work. The process showed in Figure 13 could take up to 4 hours by an experienced orthotist. The shape of the ankle-foot complex is captured by wrapping a shock and casting the leg. In this configuration key anatomical features are marked to help perform corrective modifications in a later fabrication process. The cast will be filled with plaster, and preheated thermoplastic is formed around the plaster. Once cool, the unwanted plastic is cut away, and the remaining part is ground down and smoothed to a wearable AFO [1].

Figures 13 Traditional fabrication process of a customized AFO[1]

Embraced with the rapid prototyping technique and 3D-scanning technique, several customized ankle-foot orthoses have been rapid prototyped to demonstrate the feasibility of the computerized fabrication process [31].

Northeastern University’s Biomedical Mechatronics Laboratory developed a patient specific AFO utilizing rapid prototyping technique (Seen in Figure 14). In this prototype, a kind of nylon-like
Selective Laser Sintering (SLS) material is used to fabricate AFO. Conductive piezoresistive material was injected into the SLS AFO so as to capture stress and stiffness data [32].

Figure 14 SLS AFO with C-SHIP Sensors [32]

University of Delaware also developed a passive-dynamic AFO fabricated by fused deposition modeling (FDM) 3D printing technique. In addition, the strut thickness was tuned, and correlated to the experimental stiffness data to develop a virtual bending stiffness determination process [33].

Figure 15 Passive Dynamic AFOs [33].
It is predictable that with the development of the advanced manufacturing process, orthotics will be further personalized to suit the patient.

2.2.4 Measurement of Lower Extremity Medical Devices

AFO stiffness plays an important role in determining the biomechanical function of AFO. Currently, AFO stiffness measurements could be divided into two main methods: bench testing and functional analysis. In the functional analysis, an AFO is embedded with sensors such as strain gauge, and is worn by subjects while walking. In the bench testing method, mechanical parameters are usually measured with an AFO attached to a measurement apparatus. [34, 35]

The advantage of functional analysis is that the properties of the AFO could be combined with the lower-limb joint kinematics. However, parameters generated from human factors such as pathology will influence the results, which make it difficult to apply these results to improve the design of new AFOs. Conversely, experimental parameters are easy to control in bench analysis. Although the ISO structural test standard for an AFO has not been established, various measurement devices have been developed to analysis AFO performance [34, 35].

The UV University Medical Center has developed a measurement device named BRUCE to capture AFO several characteristics. Figure 16 indicates the schematic overview of the BRUCE. The design of the BRUCE is based on human leg anatomy which is driven manually. Besides, the BRUCE allows the MTP flexion-extension motion. The stiffness and the neutral angle around the ankle as well as MTP joint could be obtained by the device, and the results have been described as reliable and clinically applicable [36]. However, since the actuation is driven manually, the device cannot be able to perform dynamic measurement.
Hong Kong Polytechnic University designed an automated device to measure the stiffness of articulated AFOs (seen in Figure 17). The device is actuated by a commercial hydraulic servo fatigue testing machine. A potentiometer is aligned with the rotary plate to capture ankle angle, while the force measurement is performed by a hand-held force gauge. The device has accuracy of 4% within repetitive 15° plantar flexion as well as 1% within dorsiflexion at the angular velocity of 10°/s [35]. However, the device does not have automated mechanisms to measure stiffness. Also, the device might not be able to simulate the complexity of actual loading and individual biomechanical environments.

Accurate measurements of orthotics are crucial and under-development in quantitative design of customized devices. The next generation bench testing systems for biomedical rehabilitation devices must be able to adjust to customized biomechanical anatomy parameters. The simulation of loadings should accurately reflect the actual human-device interface. Automated actuation and data acquisition is needed to simulate repeatable human motion, and obtained both static and dynamic mechanical characteristics. With the development of assessment systems for biomedical devices, quantitative
prescription and fabrication of patient-specific rehabilitation devices will be practicable in industry in the future.

Figure 17 An Automatic Articular AFO Stiffness Measurement Device [35]
Development of Customized 3D Printed Hand Exoskeleton with Embedded Sensors: EXCELSOR

3.1 Overview to Previous Work

The first prototype of Customized 3D printed hand exoskeleton with embedded sensors (The EXCELSIOR) was developed in November of 2011 by an undergraduate team at Northeastern University’s College of Engineering.

The previous EXCELSIOR (seen in Figure 18) is a customized hand cognitive and rehabilitation device for stroke patients to provide extension exercise. The exoskeleton consists of a 3D printed MCP clamp, 3D printed finger extenders, polypropylene finger links and LED finger thimble. All the exoskeleton parts are made separately, and assembled together after fabrication. Several customized target objects were designed and fabricated as hardware game interface. In each finger, the device used one film piezo-resistive bending sensor to capture extension and flexion motion [32].

However, there are some drawbacks of this prototype. The exoskeleton was made by multiple materials and assembled after fabrication. Therefore, it created the complexity of the device. Also, stiffness between MCP, PIP, and DIP joints were not adjustable. In addition, data from the bending sensors have large nonlinearity. Since the device only use one flex sensor for one finger, there is no measurement in MCP, DIP, and PIP joints respectively, and the relationship between the measurement and these joint angles are not verified.

![First Prototype of the EXCELSIOR](image)

Figure 18 First Prototype of the EXCELSIOR[32]
3.2 Development of EXCELSIOR

A new EXCELSIOR prototype was developed in this study to improve the previous work. The primary design goal of the new EXCELSIOR is to create a customized hand exoskeleton to assist stroke rehabilitation. Also, computerized methodology of design a customized rehabilitation devices will be studied during the design and prototyping process. With these objectives the design requirements of the new EXCELSIOR were listed below:

1. The device should fit the customized biomechanical properties. To provide comfort for the patients, the device should be able to fit the patient’s hand anatomy. The finger joints of the exoskeleton should fit the correlative human finger joints.

2. The device should be able to provide resistive forces and ranges of motion for rehabilitation exercises. To provide assistance to extension exercise, ranges of the motion for the EXCELSIOR must satisfy the function ranges of motion for the human fingers. Table 2 gives the functional range of motion of the hand:

<table>
<thead>
<tr>
<th>Joint Motion</th>
<th>Function Range of Motion (Degrees)</th>
</tr>
</thead>
<tbody>
<tr>
<td>MCP flexion</td>
<td>60</td>
</tr>
<tr>
<td>PIP flexion</td>
<td>60</td>
</tr>
<tr>
<td>DIP flexion</td>
<td>40</td>
</tr>
<tr>
<td>Thumb MCP flexion</td>
<td>20</td>
</tr>
</tbody>
</table>

And the compressive forces required in every joint are given in Table 1.

3. The device should contain embedded sensors to capture the angle and grasping force data.

3.2.1 Implementation of Customized Interface

In this project, a customized MCP clamp is a very important part of the device, since the anatomy of this area differs significantly between various people. The exact dimensions of the patient’s hand anatomy not only provide individual comfort, but also help to provide the correct neutral MCP joint position. Therefore, 3D-Scanning technique was adopted to capture geometry data from the patient’s hand. Two 3D-scanner systems, DAVID-laser scanner and XBOX Kinect Sensors are used to capture hand geometry respectively.
3D-Scanning by DAVID-Laser Scanner

DAVID Laser scanner is a very low-cost system for contact-free scanning of 3D objects. The only hardware requirements are a simple commercial hand-held laser and a standard camera. Figure 1 shows the DAVID laser scanner setup (Figure 19). With reasonable hardware, the scan accuracy is around 0.1% of the object size.

![Figure 19 DAVID Laser Scanner Setup](image)

Figure 20 shows the physical implementation of the 3D-scanning process by DAVID laser scanner. The basic components are: a webcam, a hand-held line laser, two plain broods in the background, and a Windows PC with the software DAVID. The basic setups for the scanning processes include:

1. Setup the webcam by adjusting the focus and the exposure/brightness according to the object.
2. Place the 90° calibration corner at the object’s location, and calibrate the camera.
3. Hold the laser line, and move slowly over the object.
4. With one additional click (possibly also adjustment of light conditions or camera settings) a texture is grabbed with each scan.
5. One by one, the collected scans are aligned to each other. The user defines the order (which scans are neighbors). As long as the scans overlap sufficiently, the correct alignment is found automatically (no markers necessary).

6. When all scans are aligned, the Fusion computes one closed triangle mesh without overlapping (possibly textured).

Figure 20 Implementation of 3D-scanning via DAVID Laser Scanner

After using the 3D scanner, raw 3D surface samples are imported to Meshlab. Figure 21 shows the 3D scanned hand model after reconstruction in Meshlab.
Using 3D scanning technologies, such as structured light or laser scan, detailed human models could be created. However, these devices are very expensive and often require expert knowledge of the operation. Moreover, it is difficult for people to stay rigid during the capturing process. Low cost 3D laser scanners such as the DAVID-laser scanner will produce much more missing holes and duplicated faces during the scanning, which increase the difficulties for image processing and decrease the resolution.

As a new kind of devices, depth cameras such as Microsoft Kinect have attracted much attention in the community recently. Compared with conventional 3D scanners, they are able to capture depth and image data at video rate and have little consideration of light and texture condition. Kinect is compact, low-price, and as easy to use as a video camera, which can be acquired by general users. Based on these advantages, a Kinect Sensor is selected to compare scanning results with the DAVID laser scanner.[38]
Kinect(Figure 22) use a RGB camera, a depth sensor and a multi-array microphone for full body 3D motion capture, facial recognition and voice recognition. The depth sensor consists of an infrared laser projector combined with a monochrome CMOS sensor. By projecting a known speckle pattern onto the scene, the depth could be inferred from the deformation of the pattern.

Kinect Fusion is a real time 3D model builder developed by Microsoft. Kinect Fusion takes the incoming depth data from the Kinect for Windows sensor and uses the sequence of frames to build a highly detailed 3-D map of objects or environments. It enables users holding and moving a standard Kinect camera to rapidly create detailed 3D reconstructions of an indoor scene. Only the depth data from Kinect is used to track the 3D pose of the sensor and reconstruct, geometrically precise, 3D models of the physical scene in real-time.

To capture 3D hand anatomy in this study, keep the subject’s hand in neutral position on the bench, then move the Kinect sensor around the hand (Seen in Figure 23). Once the raw 3D model is reconstructed by Kinect Fusion, Meshlab is used to perform post image processing for scanned samples.
Compared with DAVID laser scanner, the scanned sample from Kinect Sensor has much less missing holes and invalid faces, hence it is much easier for MeshLab to perform image processing automatically. Therefore, scanned samples from Kinect sensors are adopted to generate a customized 3D hand model. Finally, SolidWorks ScanTo3D toolbox is used to automatically transfer the meshed data from Meshlab into solid model (Shown in Figure 24). The flow chart of customized MCP clamp design processes is given in Figure 25. This digitalized process is easy to be implemented automatically.

Figure 23 Left: 3D Scanning Process via Kinect Fusion; Right: Post-processing in MeshLab

Figure 24 CAD Model of Customized Clamp
3.2.2 Design of Finger Elements

Finger element is the most important mechanical parts of the EXCELSIOR. The finger element should be allowed to rotate around all the finger joints. Also, the finger elements should be able to generate resistive extension force so as to keep patients’ hand in neutral position. Finally, considered that the MCP clamp is prototyped by additive manufacturing, therefore print-in-one designs are preferred to be implemented.

Based on the design requirements listed above, a series of concepts were generated and detailed below.

**Concept 1: Non-assembly Pivot Finger Joint**

One essential benefit of additive manufacturing is that it makes the fabrication of complex geometries applicable. This allows the one-step fabrication of multi-link systems as a whole without requiring assembly. Figure 26 shows the CAD drawing and the 3D-printed part of a non-assembly revolute joint. To design non-assembly joints, clearances should be implemented between joints in CAD drawings. Therefore, determination of clearance is very important.\[39\] Figure 27 illustrates the experimental clearance tuning method for the 3D-printer. A serial of shaft-hole assembly with various clearances was designed and imported into 3D-printer. After fabrication and cleaning, shaft-hole assemblies which have smaller clearances than 0.375mm could not be taken apart. Therefore, the optimum clearance for the Objet Eden 333 Machine is 0.375mm.
Figure 26 Non-Assembly Pivot Joints Utilizing Rapid Prototyping Method[39]

Figure 27 Clearances Tuning for Objet 3DPrinter. LEFT: CAD drawings of experimental samples. RIGHT: Actual parts fabricated by Objet Eden 333 3D Printer

The concept shows in Figure 28 uses non-assembly pivot joints as DIP, PIP finger joint elements. This design allows the links rotate along with the fingers by designing slots for the hinges. The main drawback of this concept is the difficulties to capture rotation angle measurement, and no resistant force could be generated during the bending process.
Concept 2: Compliant Finger Element

Another way to design non-assembly joints is compliant mechanisms. Compliant mechanisms are flexible mechanisms that transfer input forces or displacement by elastic energy. Figure 29 indicates the effects of the compliant structure in 3D-Printed objects. The structural material for this sample is Objet Vero Blue, which has similar mechanical properties as ABS. However, after designing zigzag cutout, the sample could achieve large deformation as that of rubber material.

Inspired by this sample, this concept of compliant finger elements used the zigzag cutouts to create compliance (seen in Figure 30). The zigzag cutouts are implemented as serial elastic elements. While
bended, they will provide large deformation. At the same time, the beam aligned with zigzag cutouts will function as stiff elastic elements, and provide mounted surface for embedded strain gauges.

**Concept 3: Compliant Finger Element Using Spiral Spring Structure**

This concept is inspired by spiral springs (seen in Figure 31). One benefit of this design is that the design formula for spiral spring is given by:

\[
K = \frac{Gd^4}{8nD^3}
\]  

(3-1)

Where, \( K \) is the stiffness of the spiral spring, \( G \) is the modulus of the elasticity in shear, \( d \) is the wire diameter, \( n \) is the number of active coils, \( D \) is the mean diameter of the spring. Once given the desired displacement and desired stiffness, a spiral spring element could be design to follow the desired trajectory, and the structural parameters could be tuned to fit the desired stiffness. However, since the materials for addictive manufacture usually have low shear strength, therefore to design a spiral spring using 3D-Printed materials, it will take more space than a metal spring. In addition, the spiral structure is difficult to clean after fabrications. Finally, it is also difficult to embedded strain gauge or other force sensors in this structure to capture the finger joint motion.
**Concept 4 Compliant Finger Element with Ball Joints**

This concept combines the ball joints with zigzag cutouts as an elastic element. When the finger is bended, the ball joints could perform pivot rotation, while the zigzag cutouts will provide elastic tension. Figure 32 gives the CAD drawing and the 3D-printed prototype of this design. In this design, potentiometer could be implemented with the ball joint to capture angular displacement.

*Figure 31 Finger Elements using Spiral Spring Structure*

*Figure 32 Compliant Finger Elements with Ball Joints. UP: CAD drawings. DOWN: Prototype fabricated by 3DSYSTEM Viper SLA 3D-Printer*

**Concept Selection**

Finally, Concept 2 is selected to be implemented into the final design. In Solidworks, the finger elements and the MCP clamp is assembled into one assembly files. After assembly, Solidworks could
save the complete assembled CAD into one single STL file. Imported the STL file into objet job manage software, the CAD then will be fabricated as one “non-assembly” part. Figure 33 shows the final CAD assembly and the prototypes fabricated by Objet Eden 333 3D-printer.

![Figure 33 Final Prototype of EXCELSIOR. LEFT: CAD drawing of final assembly. RIGHT: EXCELSIOR Prototype fabricated by Objet Eden 3D Printer](image)

**Analysis of Finger Joint with Zigzag Cutouts**

The zigzag-cutout finger joint consists of two parts: zigzag spring and a cantilever beam. All these two parts are serial aligned. The cantilever beam is mainly subjected to bending force from two zigzag springs, and the two zigzag springs are subjected to shear force and tension from fingers.

As shown in Figure 34, when given the bending force, the maximum stress will be given as:[40]

\[
\sigma_{\text{max}} = \frac{6FL}{bh^2} \]  \hspace{1cm} (3-2)

Where the deflection of the beam could be represented as:

\[
\delta_B = \frac{4FL^3}{Ebhl^3} \]  \hspace{1cm} (3-3)

When the beam is subjected to extension force, the deformation could be represented as:

\[
\delta_e = \frac{FL}{Ebhl^2} \]  \hspace{1cm} (3-4)

Compared the deformation from bending with that from extension when \(h \ll L\):
Therefore, increase deflection is the best way to increase elongation of the zigzag cutouts. Equation (3-2) and (3-3) also indicate that when the thickness $h$ increased, the cantilever beam could suffer larger bending stress, however it will decrease the deformation, which will reduce the signals from Strain Gauge. Hence, the thickness should be placed as small as possible under the safety condition.

**Analysis of Zigzag Cutouts**

The fundamental element of zigzag cutouts could be simplified as a $\pi$-shape mechanism consists of three beams (seen in Figure 36). Using Euler-Bernoulli beam theory to model the mechanisms, then the stiffness matrix of each beam could be represented into local coordinates as:

$$
K_{i,\text{local}} = 
\begin{bmatrix}
\frac{EA_i}{l_i} & 0 & 0 & -\frac{EA_i}{l_i} & 0 & 0 \\
0 & \frac{12EI_i}{l_i^3} & \frac{6EI_i}{l_i^2} & 0 & -\frac{12EI_i}{l_i^3} & \frac{6EI_i}{l_i^2} \\
0 & \frac{6EI_i}{l_i^2} & \frac{4EI_i}{l_i} & 0 & -\frac{6EI_i}{l_i^2} & \frac{2EI_i}{l_i} \\
-\frac{EA_i}{l_i} & 0 & 0 & \frac{EA_i}{l_i} & 0 & 0 \\
0 & -\frac{12EI_i}{l_i^3} & -\frac{6EI_i}{l_i^2} & 0 & \frac{12EI_i}{l_i^3} & -\frac{6EI_i}{l_i^2} \\
0 & \frac{6EI_i}{l_i^2} & \frac{2EI_i}{l_i} & 0 & -\frac{6EI_i}{l_i^2} & \frac{4EI_i}{l_i}
\end{bmatrix}
$$ (3-6)
Transfer into global coordinates, the stiffness matrix could be:

$$K_i = T_i^T K_{i,\text{local}} T_i$$  \hspace{1cm} (3-7)$$

Where, \( T \) is the linear transformation matrix:

$$T_i = \begin{bmatrix} \cos(\theta_i) & \sin(\theta_i) & 0 & 0 & 0 & 0 \\ -\sin(\theta_i) & \cos(\theta_i) & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & \cos(\theta_i) & \sin(\theta_i) & 0 \\ 0 & 0 & 0 & -\sin(\theta_i) & \cos(\theta_i) & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$  \hspace{1cm} (3-8)$$

Assemble every stiffness matrix, and then the stiffness matrix of \( \pi \)-shape mechanisms is:

$$K_\pi = \sum_{i=1}^{3} K_i$$  \hspace{1cm} (3-9)$$

The equation of the \( \pi \)-shape mechanisms is:

$$K_\pi [d] = \{F\}$$  \hspace{1cm} (3-10)$$

Where, \( \{d\} = \{u_1 v_1 \theta_1 u_2 v_2 \theta_2\}^T \) is the nodal displacement vector, \( F = \{F_{x_1} F_{y_1} M_1 F_{x_2} F_{y_2} M_2\}^T \) is the nodal force vector.
From the equation (3-5), \( l_2 \) should be small but larger than the deflection produced by the beam 1. \( l_1 \) should be large enough so as to increase elongation. Width \( b_1 \) helps control the stiffness, and \( b_2 \) should be large enough to avoid failure.

Figure 36 shows results from finite element analysis on one test sample. Under 10N bending forces, the maximum stress is 35.5 MPa, which is under the yield strength of the Objet VeroBlack material. The maximum deformation also satisfied the requirements.

![Figure 36 Finite Element Analysis of Zigzag Cutouts Finger Element](image)

### 3.2.3 Embedded Sensors Integration

In order to determine the amount of force that each finger or each joint is able to exert, sensors will be used to either directly measure the force on of the finger on each segment or measure properties that can be correlated to force based on the geometrical design.
While there are several methods of measuring strain, the most common selection is the strain gauge. In this study, the SGD-10/120-LY13 Omega strain gauge was chosen for use on the joints as it is maximized the use of expected space, and limited the possibility of fatigue wear.

To attach the strain gauge to the spring, a cyanide-akrylate based adhesive was used after prepping the surfaces by leveling, roughing, and degreasing. For protective purposes, the gauges were covered by kapton tape. More tape or epoxy should be used on the final device for strain relief at the solder joint.

In order to turn the change in resistance of the strain gauge to a change in voltage that a DAQ can read, a type-I quarter-bridge Wheatstone bridge was used. The general form of the circuit is shown in Figure 38.
3.2.4 Experimental Results

In order to verify the concept of design and calibrate the strain gauge, testing samples of EXCELSIOR were fabricated by Objet Eden 3D printer. As shown in Figure 39, the testing sample was mounted with strain gauges and connected to a Labview DAQ. The sample was placed on a hand that flexed to a certain angle. At each angle the strain was read by a Labview DAQ. The testing results and the linear fitting curve were shown in Figure 40. The results showed that the data from strain gauges had very good linearity, and the $R^2$ is 0.9923.

Experimental results indicated that the EXCELSIOR finger could achieve the desired angular displacement and extension resistance. The embedded strain gauge could capture finger joint angle motion. The data from embedded sensor has very high linearity.

Figure 38 Strain Gauges Integration. UP: the Schematic Drawings of quarter bridge Wheatstone Circuit. DOWN: Implementation of 8 Wheatstone bridges

The circuit takes in an excitation voltage uses two and three position terminals to connect to DAQ signal inputs and 3-wire strain gauges. Completion resistors are precision 120ohm resistors that can withstand 5 volts of excitation before heating up.
Figure 39 EXCELSIOR Testing Samples with Embedded Strain Gauge

Figure 40 Experimental Results of Testing Sample

\[ y = 0.1022x + 9.7259 \]

\[ R^2 = 0.9923 \]
4 Design, Development and Real-Time Implementation of AFO Testbed

4.1 Introduction

The purpose of the study is to design a new biomimetic stiffness measurement device. The device consists of a biomimetic footplate mechanism, a parallel 4-bar linkage mechanism, and a ball screw transmission mechanism. The following requirements for AFO test bed are established:

1. The device must be able to measure the stiffness and the neutral angle of the AFO around the ankle and MTP joints, over a functional range in the sagittal plane.
2. The device must be able to perform force control and position towards the AFO.
3. The device has to be able to accommodate a wide variety of AFOs designs.
4. The device has to be able to simulate the body weight.

4.2 Mechanical Design of the AFO Testbed

The two major mechanical subsystems of AFO test bed, as shown in Figure 41 are:

1. Biomimetic footplate, which is designed to locate accurate ankle joint center information for custom specification, and provide corrective gait cycle to the ankle-foot anatomy. The biomimetic footplate could provide two degrees of freedom: the ankle joint rotation and the MTP joint rotation.
2. Actuation system, which is consisted of a parallel four-bar linkage and a ball screw mechanism, is designed to provide corrective moments and displacements for testing patient-specific AFOs.
4.2.1 Design of Biomimetic Footplate with Ankle Position Sensor

In order to achieve AFO stiffness measurement, a biomimetic footplate is designed to accommodate for patient-specific joint centers. Simplified from ankle-foot anatomy (as shown on Figure 42), the footplate could be represented as a combination of a heel bone, a metatarsal bone, a metatarsophalangeal (MTP) joint and an ankle joint. An adjustable clamp is applied to provide desired body weight.
Figure 42 Schematic Drawing of the Biomimetic Footplate (a), Human Ankle-Foot Anatomy (b) and Simplification (c) [42]. Note that the Serial Elastic Mechanism is not depicted.

Figure 43 demonstrates the key components of the biomimetic footplate. 80/20 T-slotted Frames are used for adjustable biomimetic bones. A potentiometer is connected to the ankle joint axis by 3D-printed close fit potentiometer holders.
Figure 43 Components of Biomimetic Footplate. Both the metatarsal bone and the heel bone are made by 80/20 T-slotted framing (a), which provide length adjustability. MTP joint is made by 80/20 T-slotted pivot joint (b). Ankle angle measurement is performed by a potentiometer and 3D-printed sensor holders (c).

Hence, as shown on Figure 44, the coordinates of the ankle joint could be derived by the following equations:

\[
\cos \theta_M = \frac{l_{HM}^2 + L_{AM}^2 - l_{AH}^2}{2L_{HM}L_{AM}} \quad (4-1)
\]

\[
X_{Ankle} = -L_{AM} \sin \theta_M \quad (4-2)
\]

\[
X_{Ankle} = L_{AM} \cos \theta_M \quad (4-3)
\]

Figure 44 Coordinates Calculation for the Ankle Joint Center. Given the geometric dimension of the heel bone and the metatarsal bone, the coordinates of the ankle joint could be derived from the law of cosines.
The free body diagram of the footplate is shown on Figure 45. The relationship between real body weight and the equilibrium compression force is given by:

\[ GL_g = F_d L_c \]  \hspace{1cm} (4-4)

Where, \( G \) is the desired body weight, \( L_g \), \( L_c \) are the moment arm of the ankle and clamp respectively. Hence, the formula of compression force is given in (4-5):

\[ F_d = \frac{L_g}{L_c} G \]  \hspace{1cm} (4-5)

---

4.3 Actuation System and Sensor System

4.3.1 Actuation System Overview

Figure 46 shows the schematic drawing of the AFO actuation system. The test bed is actuated via the DC brush motor (3863024C) from FAULAHBER Inc. A ball screw mechanism is used to transform rotary motion into linear motion. Via two pivot joints and 80/20 frame, the ball screw mechanism is connected to a parallel four-bar linkage, which generates horizontal tension towards the AFO.

To implement torque measurement, both force and position sensing are needed. Therefore, the parallel linkage actuation system is outfitted with a tension/compression load cell, as well as a motor encoder. Therefore, the torque could be measured by the following:

\[ T_{AFO} = F_l L_{frame} \cos \theta_{frame} \]  \hspace{1cm} (4-6)
Where, $T_{AFO}$ is the torque around the ankle joint, $F_L$ is the Tension obtained by Load cell, $L_{frame}$ is the arm of the parallel four-bar linkage, and $\theta_{frame}$ is the pivot angle of the linkage.

To guarantee horizontal tension measurement, the load cell is mounted in series between the linear bearing and the four bar linkage, seen in Figure 47. A 3D-printed clamp is fabricated to provide close-fit a mounted surface, which stabilizes the load cell. In addition, a ball joint rod end is implemented between the load cell and the linear bearing to isolate tension force, allow angular adjustability between the AFO and the Four bar Linkage.

Figure 46 Actuation System of AFO Test Bed
4.3.2 Design of Actuation Systems
To design and analysis mechanical systems, mathematical modeling the actuation system is a must. Figure 48 presents the free body diagram of the AFO test bed. To establish a mathematical model for the whole system, separate it with two parts:

Part I, the biomimetic footplate and the parallel four-bar linkage.

Part II, the Ball screw Actuation System.

Geometry parameters of Parallel Four-Bar Linkage are shown in Table 3:

<table>
<thead>
<tr>
<th>Geometry Parameters</th>
<th>Symbols</th>
<th>Values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of linkage 2</td>
<td>$l_2$</td>
<td>0-17” (Variable)</td>
<td>Inch</td>
</tr>
<tr>
<td>Length of linkage 3</td>
<td>$l_3$</td>
<td>4.5”</td>
<td>Inch</td>
</tr>
<tr>
<td>Length of linkage 4</td>
<td>$l_4$</td>
<td>17”</td>
<td>Inch</td>
</tr>
<tr>
<td>Length of linkage 4a</td>
<td>$l_{4a}$</td>
<td>0-17” (Variable)</td>
<td>Inch</td>
</tr>
<tr>
<td>Length of linkage 4b</td>
<td>$l_{4b}$</td>
<td>0-17” (Variable)</td>
<td>Inch</td>
</tr>
<tr>
<td>Length of linkage 5</td>
<td>$l_5$</td>
<td>16”</td>
<td>Inch</td>
</tr>
<tr>
<td>Length of linkage 5a</td>
<td>$l_{5a}$</td>
<td>9.5”</td>
<td>Inch</td>
</tr>
<tr>
<td>Length of linkage 5b</td>
<td>$l_{5b}$</td>
<td>7.5”</td>
<td>Inch</td>
</tr>
<tr>
<td>Coordinates of Ankle</td>
<td>$(x_a, y_a)$</td>
<td>(-11”,0.125”)</td>
<td>(Inch, Inch)</td>
</tr>
<tr>
<td>X Coordinates of Ball Nuts</td>
<td>$x_1$</td>
<td>(Variable)</td>
<td>Inch</td>
</tr>
<tr>
<td>Y Coordinates of Ball Nuts</td>
<td>$y_1$</td>
<td>2.1875”</td>
<td>Inch</td>
</tr>
<tr>
<td>Lead of Ball Screw</td>
<td>$P$</td>
<td>13/64”</td>
<td>Inch/Turn</td>
</tr>
<tr>
<td>Pivot Angle of Parallel Linkage</td>
<td>$\theta_4$</td>
<td>$-\frac{\pi}{2} \sim \frac{\pi}{2}$ (Variable)</td>
<td>Rad</td>
</tr>
<tr>
<td>Pivot Angle of Ankle</td>
<td>$\theta_a$</td>
<td>$-\frac{\pi}{2} \sim \frac{\pi}{2}$ (Variable)</td>
<td>Rad</td>
</tr>
<tr>
<td>Pivot Angle of Linkage 2</td>
<td>$\theta_2$</td>
<td>$-\frac{\pi}{2} \sim 0$ (Variable)</td>
<td>Rad</td>
</tr>
</tbody>
</table>
**Design of Parallel Four-Bar Linkage**

Figure 49 illustrates the free body diagram of the parallel four-bar mechanism.

![Figure 49 Free Body Diagram of Parallel Four-Bar Linkage](image)

To derive the expression of $\theta_a$, we have:

$$\tan \theta_2 = \frac{x_{5a} - x_a}{y_{5a} - y_a} = \frac{l_4 \sin \theta_4 - l_{5a} - x_a}{l_4 \cos \theta_4 - y_a} \quad (4-7)$$

$$\theta_a = \tan^{-1} \left( \frac{l_4 \sin \theta_4 - l_{5a} - x_a}{l_4 \cos \theta_4 - y_a} \right) \quad (4-8)$$

Figure 50 illustrate both the original ankle angle curve and the linearized expression. The curve indicates that the expression has very high linearity. Hence, the equation of the ankle angle could be simplified as the following:

$$\theta_a = P_1 \theta_4 + P_2 \quad (4-9)$$

In fact, by calculating the root-mean-square-error (RMSE) and coefficient of determination in MATLAB, quantitative evaluation of the linearity is given in Table 4:
Figure 50 Comparison of Original Ankle Angle Curve and Linearized Expression

Table 4 Ankle Angle Linear Fitting Parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_1$</td>
<td>0.9569</td>
</tr>
<tr>
<td>$P_2$</td>
<td>0.1316</td>
</tr>
<tr>
<td>RMSE</td>
<td>0.2438</td>
</tr>
<tr>
<td>R-Squared</td>
<td>0.9245</td>
</tr>
</tbody>
</table>

**Design of Ball Screw Actuator**

As shown in Figure 51, the expression of $\theta_4$ and $\theta_2$ is given by the following equations:

\[
x_1 + l_2 \sin \theta_2 = l_{4a} \sin \theta_4 \tag{4-10}
\]
\[
y_1 + l_2 \cos \theta_2 = l_{4a} \cos \theta_4 \tag{4-11}
\]

Isolating $\theta_2$, $\theta_4$ respectively, then \((4-10)\) \((4-11)\) are transformed to:

\[
(x_1 + l_2 \sin \theta_2)^2 + (y_1 + l_2 \cos \theta_2)^2 = l_{4a}^2 \tag{4-12}
\]
\[
(x_1 - l_{4a} \sin \theta_4)^2 + (y_1 - l_{4a} \cos \theta_4)^2 = l_2^2 \tag{4-13}
\]

Therefore, the range of $l_2$ and $l_4$ could be given by the following:
\[ l_2 \geq l_{4a} \cos \theta_4 - y_1 \]  
\[ l_{4a} \geq l_2 \cos \theta_2 + y_1 \]  
To satisfy all the \( \theta_2 , \theta_4 \) in the range of motion, we have:

\[ l_2 \geq l_{4a} - y_1 \]  
\[ l_{4a} \geq l_2 \cos \theta_{2\min} + y_1 \]  
\[ x_{1\min} \leq x_1 \leq x_{1\max} \]  

Set \( l_{4a}, \theta_4 \) as variables, \( l_2 = 2.5 \) as a constant, the solution of \( x_1 \) is shown in Figure 52. The contour shows that as \( l_{4a} \) gets longer, the linkage could provide larger transmission ratio, shorten the stroke of ball screw nuts, however it will decrease the linearity of \( x_1 \). Similar conclusions could be performed in Figure 53.
Figure 52 Range of Motion of $X_1$ While $\theta_4$, $l_{4a}$ changed, and $l_2=2.5$ as a constant

$\theta_4$ to be a constant 2.5, tuning $l_2$, the results are represented in Figure 54. This indicates that $l_2$ has fewer influences in linearity and transmission ratio, but it constrains the range of motion of $X_1$ and $l_{4a}$. 
Finally, Select $l_2=2.5$, $l_{4a}=4$, Linearized the expression with the following expression:

$$x_1 = p_1 \theta_4 + p_2 \quad (4-19)$$

Figure 55 demonstrates the comparison of the linear fitting and the original curve. After optimizing the linkage parameters, the linearity of the actuation system is improved dramatically. In fact, the quantitative linearity evaluation is given in Table 5.
Table 5 Linear Fitting Parameters for Expression $x_1(\theta_4)$

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_1$</td>
<td>3.951</td>
</tr>
<tr>
<td>$P_2$</td>
<td>1.801</td>
</tr>
<tr>
<td>R-Square</td>
<td>0.9928</td>
</tr>
<tr>
<td>RMSE</td>
<td>0.06819</td>
</tr>
</tbody>
</table>

Implemented $l_2$, $l_4$ and the ROM of $x_1$ back to the equation. In MATLAB, used fzero() function to find the solution of $\theta_2$. applied cftool to obtain the cubic fitting expression as shown in Figure 56. The fitting polynomial of $\theta_2$ is:

$$\theta_2 = p_1 x_1^3 + p_2 x_1^2 + p_3 x_1 + p_4$$  \hspace{1cm} (4-20)$$

![Figure 56 Cubic Fitting of $\theta_2$](image)

Fitting parameters and fitness evaluation are given in Table 6.
### Table 6 Fitting Parameters of $\theta_2$

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_1$</td>
<td>0.01485</td>
</tr>
<tr>
<td>$P_2$</td>
<td>-0.1512</td>
</tr>
<tr>
<td>$P_3$</td>
<td>0.39</td>
</tr>
<tr>
<td>$P_4$</td>
<td>-1.059</td>
</tr>
<tr>
<td>R-Squared</td>
<td>0.9999</td>
</tr>
<tr>
<td>RMSE</td>
<td>0.0002983</td>
</tr>
</tbody>
</table>

### Motor Selection of the Actuator

#### Table 7 3863024C DC Bruch Motor Parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Symbols</th>
<th>Values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal Voltage</td>
<td>$U_n$</td>
<td>24</td>
<td>V</td>
</tr>
<tr>
<td>Terminal resistance</td>
<td>$R$</td>
<td>0.62</td>
<td>$\Omega$</td>
</tr>
<tr>
<td>Rotor Inductance</td>
<td>$L$</td>
<td>130</td>
<td>$\mu$H</td>
</tr>
<tr>
<td>Rotor Inertia</td>
<td>$J_m$</td>
<td>110</td>
<td>gcms²</td>
</tr>
<tr>
<td>Output power</td>
<td>$P_{2max}$</td>
<td>220</td>
<td>W</td>
</tr>
<tr>
<td>No-load speed</td>
<td>$n_0$</td>
<td>6700</td>
<td>rpm</td>
</tr>
<tr>
<td>No-load current</td>
<td>$I_o$</td>
<td>0.24</td>
<td>A</td>
</tr>
<tr>
<td>Stall torque</td>
<td>$M_{H}$</td>
<td>1.25</td>
<td>Nm</td>
</tr>
<tr>
<td>Friction torque</td>
<td>$M_g$</td>
<td>8</td>
<td>mNm</td>
</tr>
<tr>
<td>Back-EMF constant</td>
<td>$K_e$</td>
<td>3.49</td>
<td>mV/rpm</td>
</tr>
<tr>
<td>Torque constant</td>
<td>$K_t$</td>
<td>33.3</td>
<td>mNms/A</td>
</tr>
<tr>
<td>Torque Limit</td>
<td>$M_{max}$</td>
<td>110</td>
<td>mNm</td>
</tr>
<tr>
<td>Gear Ratio</td>
<td>$r_g$</td>
<td>134:1</td>
<td></td>
</tr>
</tbody>
</table>
The DC motor is driven by a Junus DC motor amplifier from Copley Motion Inc. (seen in Figure 57). Using an inner PI loop (seen in Figure 58), the amplifier allows the control of current to the motor by a PWM input signal.

Figure 57 Junus Motor Amplifier from Copley Controls Inc[43].

Figure 58 Outline of the inner current loop contained in the Junus[44]

**Finite Element Analysis of the Collar Holder**

To clamp the ball screw and the motor shaft, a 3D-printed collar holder is prototyped to mount two shaft collars at both sides. Among the whole mechanical system, the collar holder is the weakness part. Therefore, performing finite element analysis of the collar holder is required. In Solidworks, implement material properties given in Table 8, given the maximum torque from Table 7, run and analysis the Von Mises Strength. Results in Figure 59 indicate that the maximum strength generated in the collar holder is 6.5Mpa, much lower than the tensile strength of the material. Hence, the 3D-printed collar holder could satisfy desired requirements.

<p>| Table 8 Mechanical Properties of Objet Vero Black Material |
|---|---|
| <strong>Mechanical Properties</strong> | <strong>Values</strong> |</p>
<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile Strength</td>
<td>50.7 MPa</td>
</tr>
<tr>
<td>Elongation at Break</td>
<td>17.7 %</td>
</tr>
<tr>
<td>Tensile Modulus</td>
<td>2.192 GPa</td>
</tr>
<tr>
<td>Flexural Strength</td>
<td>79.6 MPa</td>
</tr>
<tr>
<td>Flexural Modulus</td>
<td>2.276 GPa</td>
</tr>
</tbody>
</table>

Figure 59 Finite Element Analysis of 3D-Printed Coupling Holder

4.3.3 Sensor Integration

*Signal Conditioning for Potentiometer*

A potentiometer is used to detect the actual ankle angle. To calibrate the potentiometer, a 500-line encoder is connected to the potentiometer by a coupling (seen in Figure 60). Rotate the coupling, and
used the LABVIEW DAQ to collect both signals from the potentiometer and the encoder. A low-pass filter is designed to filter noises.

![Diagram of Potentiometer, Coupling, and Encoder](image)

Figure 60 Calibration of Potentiometer. Connect the potentiometer with Encoder, and rotate the coupling. Use LabView DAQ to obtain both data from potentiometer and Encoder.

**Signal Conditioning of Load Cell**

To perform tension measurement, an omega LC703 load cell is selected into the test bed. To calibrate the load cell, a force gauge is used to drag the load cell. The signals captured from the amplifier were compared with the readings in force gauge. After linear fitting (seen in Figure 61), fitting parameters could be obtained via MATLAB. A 3rd order low pass filter was implemented in LABVIEW with the low-cut frequency 5Hz. Figure 62 shows the results after filters. As the low cut frequency becomes smaller and as the orders of the filter become higher, the filtered results will be smoother, and the delay will become much longer.
Figure 61 Signal Calibration of Load Cell

Figure 62 Digital Signals Processing of the Load Cell
4.4 Control System of AFO Test Bed

4.4.1 Dynamics Modeling of AFO Test Bed

To establish the kinematic equation of the system, recall Lagrange operator:

$$\mathcal{L} = \sum T_i - \sum V_i + W$$  \hspace{1cm} (4-21)

Where, $T_i$ represents the kinematic energy of each linkage, $V_i$ represents the potential energy of the system, $W$ represents the work generated by the DC motor.

For AFO and the footplate:

$$T_{AFO} = \frac{1}{2} I_{AFO} \dot{\theta}_a^2$$ \hspace{1cm} (4-22)

$$V_{AFO} = \int \tau_{AFO} \theta_4 \text{d} \theta_4 + \frac{1}{2} M_{AFO} g l_{AFO} \cos \theta_a$$ \hspace{1cm} (4-23)

For the parallel four-bar linkage (Link 3-5):

$$T_4 = \frac{1}{2} \sum_{i=3}^{5} J_i \dot{\theta}_i^2$$

$$= \frac{1}{2} \left[ \frac{2}{3} m_4 l_4^2 + m_3 l_4^2 + m_5 l_5^2 \right] \dot{\theta}_4^2$$ \hspace{1cm} (4-24)

$$V_4 = \left( m_3 l_{4a} + \frac{1}{2} m_4 l_4 + m_5 l_5 \right) g \cos \theta_4$$ \hspace{1cm} (4-25)

For the Linkage 2:

$$T_2 = \frac{1}{2} l_2 \dot{\theta}_2^2 + \frac{1}{2} m_2 \dot{x}_1^2$$ \hspace{1cm} (4-26)

$$V_4 = m_2 g (l_2 \cos \theta_2 + y_1)$$ \hspace{1cm} (4-27)

Linearized the fitting polynomial (4-20) at the initial position, then it could be simplified as:

$$\theta_2 = p_{1e} x_1 + p_{2e}$$ \hspace{1cm} (4-28)

For the ball screw actuator:
\[ T_m = \frac{1}{2} I_{bat} \dot{\theta}_m^2 + \frac{1}{2} I_{motor} \dot{\theta}_m^2 \] (4-29)

Where,

\[ \dot{\theta}_m = \frac{2\pi}{p} x_1 \] (4-30)

Recalled the Lagrange equation, perform variation at \( \theta_m \), we have:

\[ \Gamma_m = \frac{d}{dt} \sum \frac{\partial L}{\partial \dot{\theta}_i} \cdot \dot{\theta}_i - \sum \frac{\partial L}{\partial \theta_i} \cdot \dot{\theta}_m \] (4-31)

\[ \Gamma_m = K_i r_i i(t) - \sum B_i \frac{\partial \theta_i}{\partial \theta_m} \dot{\theta}_m \] (4-32)

From equation (4-9), (4-19), and (4-28), The linear expression of \( \theta_a, \theta_4 \) and \( \theta_2 \) could be represented as the following:

\[ \theta_a = K_{p1} x_1 + K_{p2} \] (4-33)

\[ \theta_4 = K_{p3} x_1 + K_{p4} \] (4-34)

\[ \theta_2 = K_{p5} x_1 + K_{p6} \] (4-35)

Linearized parameter could be seen in Table 9.

<table>
<thead>
<tr>
<th>Linearized Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>( K_{p1} )</td>
<td>0.2422</td>
</tr>
<tr>
<td>( K_{p2} )</td>
<td>-0.3046</td>
</tr>
<tr>
<td>( K_{p3} )</td>
<td>0.2531</td>
</tr>
<tr>
<td>( K_{p4} )</td>
<td>-0.456</td>
</tr>
<tr>
<td>( K_{p5} )</td>
<td>0.39</td>
</tr>
<tr>
<td>( K_{p6} )</td>
<td>-1.059</td>
</tr>
</tbody>
</table>

Therefore, the mechanical system of AFO test bed could be simplified as:

\[ T_m = J_{eq} \dot{\theta}_m^2 + B_{eq} \dot{\theta}_m + K_{eq} \] (4-36)
The dynamics model of the motor amplifier could be simplified as:

\[ T_m = K_r r_g i \]  

(4-37)

Where, \( K_r \) is the torque constant, and \( r_g \) is the gear ratio. Therefore, the transfer function of the plant could be represented as the following:

\[ G_p(s) = \frac{\theta_a(s)}{I(s)} = \frac{2\pi K_r r_g K_p}{J_{eq}s^2 + B_{eq}s + K_{eq}} \]  

(4-38)

The parameters of the characteristic equation are presented in Table 10. Based on the given values, the roots of the characteristic equations are:

\[ S = -0.2222 \pm 16.3284i \]  

(4-39)

Apparently, the plant is an under-damped system.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>( J_{eq} )</td>
<td>0.045</td>
</tr>
<tr>
<td>( B_{eq} )</td>
<td>0.02</td>
</tr>
<tr>
<td>( K_{eq} )</td>
<td>11.47</td>
</tr>
</tbody>
</table>

### 4.4.2 Feedback Control System Design

#### Position Control System

One objective of the AFO test bed is to obtain the mechanical characteristics of the AFOs, such as stiffness and damping ratio. To achieve stiffness and damping ratio measurement, an accurate position and force control system is required to be implemented into the system. Therefore, feedback control systems for both position control and torque control are designed respectively. Figure 63 shows the block diagram of the position feedback control system, where \( \theta_d \) is the desired angular input, \( \Gamma_d \) is the disturbance torque input due to the modeling error, and other unknown disturbance. The algorithm of the feedback control system is based on the compensation of the error, therefore it has good robustness, and it is easy to be implemented. Among all the control algorithms, P.I.D controller is the
most widely used algorithm. To design P.I.D controller, the feedback system are modeled and analysis in SIMULINK.

Since the AFO is prototyped by Nylon-like 3D-printed material, to avoid large plastic deformation, it is necessary to transfer the system towards an over-damped system. Therefore, P.D control algorithm is considered the best candidates to minimized overshoot.

Implemented the plant model with a PD controller into MATLAB, calculated the root locus of the system (Shown in Figure 64). Next, added the Saturation model in the Simulink, starting with the PD controller parameters P=5, D=6.25. Under the PD control law, simulation shows that without saturation, the system has settling time 1.4612e-5s and the overshoot are 0.0862. However, within saturation, the system will have a limit cycle. To decrease the magnitude of the limit cycle, setting the PD parameters by P=0.1, D=0.08. Results of the two PD controllers are shown in Figure 65. Simulations indicate that decrease PD parameters will decrease the limit cycle, however it will introduce the steady state error. Hence, additional Feed-forward compensation is needed to compensate the steady state error. After PD controller, the transfer function of the complete feedback system becomes:

\[ G_c(s) = \frac{\theta_a(s)}{\theta_d(s)} = \frac{4.482 \times 10^5 s + 5.602 \times 10^5}{0.045s^2 + 4.482 \times 10^5 s + 4.482 \times 10^5} \]
Figure 64 Root Locus of the Position PD control system

Figure 65 Simulation Result of Position Feedback Control with Current Saturation
**Torque Control System**

To achieve the fatigue testing, a torque control system is required to be implemented into the system. From the previous testing, the AFO is modeled as a simple linear spring. Similar to the position control system, the linearized torque feedback control system is built and simulated. Using the same PD controller, the simulation shows that without saturation, the ideal torque control system has $3.9274\times 10^{-7}$s settling time and zero overshoot. Within the saturation, system exists limit cycle, and is marginally stable. After tunneling PD controller, by setting $P=0.1$, $D=0.08$, results show that although the settling time is increased, limit cycle is dramatically reduced (shown in Figure 67).

![Figure 66 Block Diagram of the Test Bed Torque Control System](image)

Figure 66 Block Diagram of the Test Bed Torque Control System
Figure 67 Simulation Results of Torque Feedback Control Using Position PD controller

**Conclusion**

The simulation results show that both the position controller and the torque controller could achieve the accurate position and torque control. Both controllers are using PD control algorithm. By using PD controller, the test bed became an over-damped control system. Motor amplifier saturation could introduce limit cycle, which leads to vibration. By decreasing the PD parameter and introduce the feed-forward controller, the magnitude of the vibration is decreased, however it will increase the settling time.

4.4.3 **Impedance Control System Design**

With the widely applications of the AFOs, there are several studies about the influences of the AFO mechanical properties upon the patients’ gait cycle. In most cases, those evaluations are based on subject testing, where patients are asked to put on an AFO, and motion sensors are utilized to capture patients’ gait cycle. Subject testing could provide visual performance of the AFO. However, it is very difficult to ask one subject to test hundreds of AFO samples. Also, since the AFOs are patient-specific products, subjects would have large variations from each other. In addition, there are few safety mechanism to protect human subjects if the sample AFO failures. Hence, a biomimetic test bed is
required to replace human subject, and perform repetitive testing. The goal of the test bed is to fully simulate the dynamic properties of human subjects so as to evaluate the human-AFO interaction. Since in nature, most of the mechanical system could be simplified as a mass-spring-damper system, as long as the actuation system could have the same performance of the desired mass-damper mechanism, the test bed could fully reflect the actual subject-AFO interaction. Driven by this motivation, an Impedance control algorithm is designed to model the dynamics of human ankle-foot mechanism.

**Introduction of Impedance Control**

Generally speaking, *Mechanical impedance* (or simply impedance) is the dynamic generation of the desired stiffness and damping, and is defined as a complex function[45]:

\[
Z(\omega) = \frac{F(\omega)}{X(\omega)}
\]

(4.41)

Figure 68 shows the block diagram of an impedance controller. The outer position loop measure the position error between the ideal position command and the actual position, and calculate the desired force accordingly. Usually this is accomplished by a PD controller, where the proportional term represents virtual spring stiffness, and the derivative term perform as a virtual damper. The goal of the inner force loop is to eliminate the error between desired force and the actual force.

![Figure 68 the Block Diagram of an Impedance Controller](image)

**Modeling and Analysis of Impedance Controller**

The transfer function of the force loop is:
\[ G(s) = \frac{F_a(s)}{F_d(s)} = \frac{C(s)Kr_g}{C(s)Kr_g + 1} \]  

**Remarks:**

- When \( C(s) \) is a P controller, when \( CKr_g \gg 1 \), then the feedback controller will act as a deep negative feedback; if \( CKr_g < 1 \), then the feed-forward control law is needed to compensate the steady-state error.

- When \( C(s) \) is a PD controller, then the force loop yields to:

\[ G(s) = \frac{F_a(s)}{F_d(s)} = \frac{(K_ds + K_p)Kr_g}{K_dsKr_g + (K_pKr_g + 1)} \]  

If \( K_d \) is small, the force loop will add an zero, it will help to stabilize the system when the desired spring-damper is light.

### 4.4.4 Physical Implementation of AFO control system

**Implementation of P.I.D Control**

The control hardware of the AFO test bed comprises a personal computer, a LABVIEW BNC 2120 DAQ, a Junus DC motor amplifier. The computer runs LABVIEW with a graphic user interface, through which user could read and write signals to the DAQ. The DAQ is used to read analog inputs from the potentiometer and the load cell, read counts from the encoder, and sent the PWM signal to the motor amplifier.

Select the \( P=0.08, D=0.02 \), the step response of the position control system is shown in Figure 69. The actual system has settling time 1.76s, which is very close to the simulation results.

Increase the values of \( P, D \) decrease the settling time, however, due to the saturation and the mechanical backlash, larger \( P \) will introduce limit cycle, and the larger \( D \) will enlarge the noise signal.
Secondly, Select Load Cell signal as feedback, tuning the P.D controller to stabilize the test bed. After tuning, the test bed achieved the desired force signal. The results are shown in Figure 70, the controller parameters are:

\[ K_p = 0.08, K_d = 0.012 \] (4-44)

The vibration of the system is caused by signal noise. It could be eliminated by setting lower cut-off frequencies for the low-pass filter.
Implementation of Impedance Control

In LABVIEW, construct an impedance controller (seen in Figure 71). Start the testing by setting desired angle to be 5, then jumping to 10 after 10 seconds. Figure 72 shows the comparison of the actual angle output and the actual Force, while Figure 73 demonstrates the performance of the force loop. Given the desired angle, the test bed starts to bend the AFO. Since the AFO has elasticity, it generates resistant forces. The system is balanced after 1.2s, when the test bed remained at $\theta_a = 4.15^\circ$, and the external force remained at 8.45N. When the desired angle jumped to 10, the test bed began to track the angle input. As the desired angle inputs get larger, the AFO generated larger elastic forces. Hence, in the new balanced point, the actual angle input remained at 8.85, and this time the actual Force outputs became 11.45N. Results indicated that the impedance controller could achieve desired impedance. The system parameters are given in Table 11.
Table 11 Parameters for the Impedance Controller

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Virtual Spring (P)</td>
<td>10</td>
</tr>
<tr>
<td>Virtual Dampter</td>
<td>0.02</td>
</tr>
<tr>
<td>Force Loop $K_p$</td>
<td>0.08</td>
</tr>
<tr>
<td>Force Loop $K_d$</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Figure 71 Implementation of Impedance Control via LABVIEW
Figure 72 Experimental Results of Impedance Control: the Actual Angle Position and the Actual Force

Figure 73 Experimental Results of Impedance Control: Desired Force and the Actual Force
5 Conclusions and Future Works

5.1 Conclusions

5.1.1 Conclusions of the EXCELSIOR

The Design of a novel, low cost customized 3D printed Hand exoskeleton with embedded sensors (EXCELSIOR) was successfully implemented. The new prototype of EXCELSIOR was fabricated in non-assembly structure by an Object Eden polyjet 3D Printer. The embedded strain gauges could measure joint angle motion in every MCP, PIP, and DIP joints of the fingers.

The 3D scanned contours by a low cost depth sensor (the Kinect) could be post-processed easily by off-the-shelf open source image processing software (Meshlab). The 3D-printed clamp utilizing the scanned data could provide sufficient accuracy and fit the wearer’s personal geometry.

The 3D-printed compliant finger joint was designed to assist the finger extension exercise. The stiffness and range of motion could be adjusted by tuning the structural parameters of the finger joints. Finite Element Analysis on the finger joint prototype indicated that the design could satisfy desired locomotion and desired forces.

The mechanical structure of the EXCELSIOR was fabricated without assembly, and the processed could be easily implemented by therapists. Non-assembly structure not only simplified the design but also reduced the weight. The design process could be computerized by software.

The embedded low cost strain gauge could obtain joint angles in every finger joints. The results from sample tests showed that the strain gauge could obtain joint angles within high linearity.

5.1.2 Conclusions of the AFO Testbed

A biomimetic testbed system for AFO stiffness measurement was designed, developed and real-time implemented. The testbed could adjust to personal biomechanical properties, perform position control and force control to measure stiffness of AFO. Finally, the testbed could perform impedance control so as to simulate the interaction between human ankle-foot complex and the AFO in gait.

In this design, a biomimetic footplate was designed and fabricated to provide customized ankle joint location. Extension was allowed in MTP joint. Embedded Potentiometers could capture the ankle joint motion.
Actuation System consisted of a four-bar linkage mechanism and a ball screw transmission mechanism was designed and optimized to apply and measure torques. Mathematical model of the actuation system was built, and linkage parameters were optimized so as to improve linearity of the control system.

Feedback control system was designed and simulated so as to perform accurate torque and position control. In addition, an impedance control algorithm was developed so as to simulate the dynamics of human ankle in gait. Real time implementation was performed by LABVIEW, and control parameters were real time tuned to improve system performance.

The results show that the biomimetic AFO testbed could perform stiffness testing and functional analysis for the customized AFOs.

5.2 Future Works

5.2.1 Future Works on the EXCELSIOR

Future works of the design of EXCELSIOR will focus on optimization of the computerized design process. Improvement of image-processing for the 3D scanned data will be researched so as to simplify current processes. The mechanical structure of compliant finger joint will be further developed, and algorithms of parameter optimization will be developed. The thumb joint will be designed and tested. More materials and embedded sensors could be tested and optimized. Software interface will be developed.

5.2.2 Future Works on the AFO Testbed

Future works on the development of the AFO testbed will focus on the design of the clamp elements so as to provide quantitative body weight during the gait. Actual human gait cycle data and ankle stiffness could be implemented into the control system so as to verify the functional analysis of the AFO. Further study of the energy return in gait will be analysis in this AFO testbed. Control system will be developed in the real time machine so as to improve the performance.
REFERENCES


Owen Elaine, "Shank angle to floor measures" and tuning of "Ankle-foot orthosis footwear combinations" for children with cerebral palsy, spina bifida and other conditions," Master of Science, University of Strathclyde 2004.


APPENDICES A: LABVIEW Diagram

Figure 74 AFO Testbed LABVIEW Platform
APPENDICES B: MATLAB Codes

MATLAB Code for EXCELSIOR

clear;
%% system parameter initialization
syms E Ix A l_1 l_2 ;
Theta=[0 pi/2 0 -pi/2 0];
L=[l_2/2 l_1 l_2 l_1 l_2/2];
K=zeros(18,18);
K=sym(K);
M=@(x)[cos(x) sin(x) 0; -sin(x) cos(x) 0; 0 0 1];
T=@(x)[M(x) zeros(3); zeros(3) M(x)];
StiffMatrix=@(x) [E*A/x 0 0; -E*A/x 0 0; 0 12*E*Ix/x^2 6*E*Ix/x^2; 0 6*E*Ix/x^2 4*E*Ix/x; 0 6*E*Ix/x^2 2*E*Ix/x; -E*A/x 0 E*A/x; 0 -12*E*Ix/x^2 -6*E*Ix/x^2; 0 12*E*Ix/x^2 -6*E*Ix/x^2; 0 6*E*Ix/x^2 2*E*Ix/x; 0 -6*E*Ix/x^2 4*E*Ix/x];
%% obtain element stiffness matrix:
for elem=1:5
    SE=T(Theta(elem))'*StiffMatrix(L(elem))*T(Theta(elem));
    for i=1:6
        for j=1:6
            K(3*(elem-3)+i,3*(elem-3)+j)=K(3*(elem-3)+i,3*(elem-3)+j)+SE(i,j);
        end
    end
end
%% Obtain active Stiffness/Load matrix:
Kact=K(4:18,4:18);
Load=zeros(15,1);
Load=sym(Load);
syms M;
Load(15,1)=-M;
Dofact=Kact\Load;
MATLAB Code for AFO Testbed Simulation

den=[0.045 0.02 12];
num=0.033*134*((64*2*pi)/13)*0.2422*180/pi*47.22;
Gp=tf(num,den);
C=tf([0.02 10],1);
P=tf(1,[0.1 1]);
G=Gp*C;

PD=tf([5 6.25],1);
Go=PD*Gp;
Gf=feedback(Go,1);
step(Gf)