Development of a fully flexure-based prosthetic hand

L. Garcia Rodriguez
DEVELOPMENT OF A FULLY FLEXURE-BASED PROSTHETIC HAND

PDEng Thesis

to obtain the degree of Professional Doctorate in Engineering (PDEng) at the University of Twente, on the authority of the rector magnificus, prof. dr. T.T.M. Palstra, on account of the decision of the graduation committee, to be defended on Friday the 22 of June 2018 at 09.30 hours

by

Luis Alberto Garcia Rodriguez

born on the 21 August 1986 in Caracas, Venezuela
This PDEng Thesis has been approved by:

PDEng-program director: Dr.ir. T.H.J. Vaneker

Thesis Supervisor: Prof.dr.ir. D.M. Brouwer PDEng

Member(s): Ir. M. Naves

Ir. E.E.G. Hekman

Dr. ir. R.G.K.M. Aarts
Summary

Most people that require assistive devices such as prosthetic hands are located in developing countries. Two basic requirements for prosthetic hands have been identified: affordability; and, acceptance of the device. Additive manufacturing offers the opportunity for local, low-cost manufacturing with the possibility to achieve acquisition cost below 3,000 USD. Moreover, the acceptance of the device is related to its weight, its cosmetic appearance and its capability to perform a power medium wrap, which is the most common grasp during activities of daily living.

Current flexure-based hands lack support stiffness throughout large range of deflections and power grasping capacity while performing power grasps, particularly at the metacarpophalangeal joint.

The focus of this study is to design a fully flexure-based hand which can perform successfully a power medium wrap. Special attention is given to the metacarpophalangeal joint since it presents the biggest challenge.

The concept of the fully flexure-based hand is studied in detail. Needs of the users are identified and are translated to requirements. The technical performance metrics are associated with: acquisition costs, power grasping force, and capability to grasp the most common objects. These metrics are addressed at the component level.

The presented flexure mechanisms follow a bottom-up approach from the component level to the fully flexure-based hand system. A monolithic, 3D-printed, fully flexure-based hand has potential to scale-up production of custom devices by minimizing assembly, which ultimately benefits the total acquisition costs of the assistive device.

At this component level, a methodology is developed to determine the optimal flexure layout and the corresponding design parameters for an anthropomorphic metacarpophalangeal joint. This methodology uses an implementation of the Nelder-Mead algorithm to optimize the hinges towards maximum grasping force. In total, five flexure layouts are investigated: the Leafspring, the Solid-Flexure Cross Hinge, the Three-Flexure Cross Hinge, the Hole Cross Hinge, and the Angled Three Flexure Cross Hinge.

In addition, an overload-protection mechanism is proposed to mitigate the effects of low support stiffness in flexure-based fingers at larger ranges of motion.
Acknowledgments

I would like to thank my supervisors from the University of Twente, Prof.dr.ir. D.M. Brouwer PDEng and Ir. M. Naves who guided me through the design process and for giving me invaluable input. Special thanks to Dannis Brouwer for having faith in me and changing my life for the better.

I also had the great support from my colleagues–M. Tjapkes, D. Zimmerman van Woesik, Z.A.J Lok, and L. Timmersma–who always helped me at my every need.

To Martin, Marijn, Mieke, Jan, and Jaap from the Precision Engineering Chair, for their patience, knowledge, input and criticism which pushed me to perform better. To Wessel, Rick, Roland, Jeroen and Bas for their contribution with their Bachelor’s degree and Master’s degrees assignments.

To my classmates and colleagues Anoek, Abhijeet, Martijn, Matthias and Roberto, for sharing the challenging courses and becoming my friends.

To my ultimate frisbee team, the Disc Devils Twente, for helping me integrate in The Netherlands. For putting in the effort to become a great team and for supporting each other. To the sport that allowed me to grow personally in so many ways.

To the de Bruijn family–Jens, Marleen, Aniek, Annette and Coco–who welcomed me in their family. They made me feel right at home in this beautiful country. For sharing their culture and for pushing me to learn the Dutch language. To the amazing afternoons chatting with Coco that filled me with perspective about life.

To Britt for her unconditional support, for listening to my work related stories, my ups and downs, for proof reading my texts and correcting my English grammar. For being my rock and for her love that allowed me to call a small piece of Enschede: “home”.

To my mom, dad, brothers and sister, who have supported me through my adventures, my study and my travels. Their advice, guidance, and love that made smooth the bumpy road.
The problem with SPACAR is always the USER

Anonymous
## Contents

Summary iii

Acknowledgments v

List of Figures x

1 Introduction 1
   1.1 Background ................................................. 1
   1.2 Motivation ............................................... 1
   1.3 Company ................................................ 2
   1.4 Outline of the PDEng report ......................... 2

2 Needs and Objectives 3
   2.1 Needs identification ...................................... 4
      2.1.1 Affordability ........................................ 4
      2.1.2 Acceptance of devices .............................. 5
      2.1.3 Compliant-based prosthetic hands - literature-review 6
      2.1.4 Function analysis system technique ............... 7
   2.2 Description of the design challenge .................. 8
      2.2.1 Operational requirements ............................ 8
      2.2.2 Technical performance metrics .................... 9
   2.3 Objectives of the design project .................... 10
   2.4 Evaluate technical performance metrics ............. 11
      2.4.1 Design for affordability ........................... 12

3 Design Methodology and Development 15
   3.1 Abstract ................................................ 15
   3.2 Contact based model for optimization of flexure-based fingers during a power grasp ......................... 16

4 Conclusions and Recommendations 25
   4.1 Conclusions .............................................. 25
   4.2 Recommendations and future work ..................... 26

Appendices 27

A Additional Requirements 29
   A.1 Risk management ......................................... 29
      A.1.1 Avoid ............................................... 30
# List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Controversy of system design [7]</td>
<td>3</td>
</tr>
<tr>
<td>2.2</td>
<td>Conceptual design [8]</td>
<td>4</td>
</tr>
<tr>
<td>2.3</td>
<td>Physical model of a prosthetic hand</td>
<td>4</td>
</tr>
<tr>
<td>2.4</td>
<td>On the left, prosthetic hook. On the right, prosthetic hand [9]</td>
<td>6</td>
</tr>
<tr>
<td>2.5</td>
<td>Functional analysis system technique (FAST) diagram</td>
<td>7</td>
</tr>
<tr>
<td>2.6</td>
<td>Physical V-model</td>
<td>9</td>
</tr>
<tr>
<td>2.7</td>
<td>Natural frequencies of the first unwanted vibration mode [10]</td>
<td>10</td>
</tr>
<tr>
<td>2.8</td>
<td>Proximal arm of torsion [11]</td>
<td>11</td>
</tr>
<tr>
<td>2.9</td>
<td>V-model for validation</td>
<td>11</td>
</tr>
<tr>
<td>2.10</td>
<td>Model for indicators</td>
<td>12</td>
</tr>
<tr>
<td>A.1</td>
<td>Stakeholders diagram</td>
<td>31</td>
</tr>
<tr>
<td>A.2</td>
<td>Mug grasp or median power wrap [12]</td>
<td>34</td>
</tr>
<tr>
<td>B.1</td>
<td>Percentage of prosthetic hands with compliant mechanisms</td>
<td>35</td>
</tr>
<tr>
<td>B.3</td>
<td>Monolithic 3D printed [15][16]</td>
<td>37</td>
</tr>
<tr>
<td>B.4</td>
<td>Straight, convex and concave leavesprings [17]</td>
<td>38</td>
</tr>
<tr>
<td>B.5</td>
<td>SDM hand [18]</td>
<td>38</td>
</tr>
<tr>
<td>B.7</td>
<td>Flexure-based finger. Top, non-symmetrical elliptic notch hinges; bottom, conceptual notch designs [1]</td>
<td>39</td>
</tr>
<tr>
<td>B.8</td>
<td>Leavespring design and PRB model [2]</td>
<td>40</td>
</tr>
<tr>
<td>B.9</td>
<td>From left to right: Cross-Leafspring, Solid Cross Flexure and Leavespring flexure hinges [3]</td>
<td>40</td>
</tr>
<tr>
<td>B.10</td>
<td>ISR Hand, thin element embedded on a rubber joint [19]</td>
<td>41</td>
</tr>
<tr>
<td>B.11</td>
<td>UC Softhand mold, wire flexure structure [20]</td>
<td>42</td>
</tr>
<tr>
<td>B.12</td>
<td>UC Softhand flexure joints for endoskeleton structure [21]</td>
<td>42</td>
</tr>
<tr>
<td>B.13</td>
<td>U. of Illinois, compliant finger hand [22]</td>
<td>43</td>
</tr>
<tr>
<td>B.14</td>
<td>WPI, contact-aided compliant mechanism [4]</td>
<td>43</td>
</tr>
<tr>
<td>B.16</td>
<td>Medium wrap [5]</td>
<td>44</td>
</tr>
<tr>
<td>C.1</td>
<td>Test rig for measurement of sideways stiffness, W. Pot design Section C.1.5</td>
<td>50</td>
</tr>
<tr>
<td>C.2</td>
<td>Experimental results, FEM from original design and FEM with modified dimensions</td>
<td>51</td>
</tr>
<tr>
<td>C.3</td>
<td>Improved base clamp concept</td>
<td>52</td>
</tr>
</tbody>
</table>
C.4 Rigid finger loaded with 6 N of sideways force. Base clamp material: a) PLA; b) Aluminum.
Chapter 1

Introduction

1.1 Background

The fully flexure-based prosthetic hand of the University of Twente is at the first stage of the system design: the conceptual phase. Flexure mechanisms could be attractive to prosthetic hands because they can help reduce to number of parts and, can reduce the weight and costs of prosthetic hands.

A literature research into the state-of-the-art in compliant prosthetic devices (Appendix B) reveal a 20% presence of compliant mechanisms in prosthetic hands (Fig. B.1). However, it seems that the precision and bio-mechanical fields are divorced. Biomechanical designers are missing the non-linear behavior of flexure mechanisms in a large range of motion while precision designers neglect the needs of the users.

The Precision Engineering Chair has developed flexure mechanisms that allow a large range of motion while maintaining guiding stiffness within limits. This combination of features can lead to stable grasps. A home-bred, efficient, non-linear computer-modeling tool named SPACAR and a state-of-the-art generic method of flexure synthesis can be used for the treatment design of the innovative product [10,23,24].

With a system-design approach and the knowledge and tools of the Precision Engineering Chair at the University of Twente, a synergy of fields can be achieved.

1.2 Motivation

The reason to design the fully flexure-based prosthetic hand is intrinsic to the motto of the University of Twente: "high tech, human touch" drives us to present a solution for an interdisciplinary product which meets societal needs. As Victor van der Chijs, President of the Executive Board of the University of Twente, has said, "The most relevant innovations are found at the interface between different disciplines" [25].

A combination of R&D technology from the Precision Engineering Research Chair and assistance from the Biomechanical Department can lead to an innovative solution. This product will have a societal impact in developing countries, where the majority of the users are located.

This professional doctorate in engineering (PDEng) intends to create a foundation which can further design phases. At the time of this writing, three bachelor's degrees in mechanical assignments have been completed and two master's degrees in mechanical assignments are in progress.
1.3 Company

The project was funded by the Science Based Engineering program of the Engineering Technology Faculty of the University of Twente.

1.4 Outline of the PDEng report

This PDEng report is organized as follows: Chapter 2 describes the status of the project and the system architecture decomposition. A detailed description of the system at the component level is presented in Chapter 3. This includes a condensed report of the design developed during this PDEng project. It also identifies distinguishing characteristics of this study, such as the identification of technical performance metrics and the development of a methodology which can be used to exploit an optimization method for these metrics. The treatment design was accomplished for five different flexure layouts. Each of these layouts carry at least four design-dependent parameters. Treatment validation was performed with a commercial FEM software and prototype fingers. A novel overload-protection mechanism for flexure-based fingers is presented.

The appendices correspond to intermediate steps that led to the condensed information presented in Chapter 3. Appendix A presents additional requirements that are defined at the concept-design stage but were not of primary concern during this PDEng project. Appendix B shows existing compliant hand manipulators. A thorough analysis of the existing devices helped to determine the needs for this project.

Appendix C provides a detailed description of the experimental setup presented in Chapter 3. It includes design targets and identifies improvements made to the setup to improve the accuracy of the test rig.

Appendix D consists of a treatment design named force-based flexure-hinge optimization. Further iteration of these results led to the new cost function from Chapter 3, which surpasses the grasping forces.
Chapter 2

Needs and Objectives

The controversy of system design is related to the infinite number of choices which are available at the start of a system design: the conceptual phase. Uncertainty can be reduced only as time advances and decisions are made, see Figure 2.1 [7].

![Figure 2.1: Controversy of system design](image)

However, the right decisions must be made, as ease of change reduces with time. The first step of the conceptual phase (Fig. 2.2) is to identify the needs of users with prosthetic hands by conducting an exploration of the available literature; the second step is to apply a research technology known as a flexure mechanism to the prosthetic device to meet these needs.

A literature review provided an understanding of the possible grasp types performed by users with prosthetic hands. Therefore, the main challenges to achieve a stable grasp have been identified and can be translated to the flexure mechanism, for instance, large range of motion and support stiffness.

A physical model, presented in Figure 2.3, provides the architecture of the system. The prosthetic hand represents the system. Actuators, fingers and palm represent the subsystems, and, the phalanges and joints represent the components. The R&D technology is located at the component level.

The fully flexure-base prosthetic hand should be body-powered and voluntary closing, implications are described in Section 2.1. These decisions at the early design-concept phase were constrained by the stakeholders from the University of Twente,


2.1 Needs identification

The needs of prosthetic hand users can be divided into two categories. The first relates to the affordability of the assistive devices; the second relates to the rate at which the devices are abandoned [26–28].

2.1.1 Affordability

Globally, 650 million people have a physical disability. 80% of them are living in low-income countries, and only 1-2% have access to rehabilitative services. More precisely, 0.5% of the population in developing countries requires a prosthetic/orthotic device, according to the 2011 report of the WHO [26, 27, 29]. According to population numbers of less-developed countries provided by the Population Reference Bureau, this represents roughly 304 million people for 2015 [30].

Figure 2.3: Physical model of a prosthetic hand.
The prices of current leading prosthetic hands range between 25,000 USD to 100,000 USD [31]. Although no range in acquisition costs for affordability in low-income countries was found, new initiatives with low-cost, 3D-printed prosthetic devices aim to bring production costs below the 3,000 USD range [32].

According to the 2011 report of the WHO, half of all assistive devices are purchased directly by the users or their families. In countries where insurance schemes cover acquisition costs fully or partially, it was found that maintenance and repairs were not considered and the users were left with defective equipment. In some cases, aging devices were not replaced until they were broken [27]. Therefore, a low-cost, low-maintenance device is of interest.

2.1.2 Acceptance of devices

Prosthetic hands have a fairly high rate of abandonment/rejection. Rejection rates of body-powered devices are as high as 65% to 80% for prosthetic hands and 32% to 51% for prosthetic hooks [33,34]. See Figure 2.4.

Evidence comparing rejection rates of body-powered devices and myoelectric devices have not been conclusive [35]. However in 2011, the World Health Organization (WHO) reported that the ongoing use of assistive devices was affected by a lack of access to batteries [27]. In addition, body-powered devices fulfill one of the most common needs of myoelectric devices, which is force feedback [34].

Furthermore, basic requirements for upper-limb prosthetic devices have been categorized as follows: cosmetics, comfort, and control [36]. Particularly in body-powered devices, rejection has been attributed to comfort and control: poor grasp force, weight of the device, slow movement, and high actuation forces [33,35]. Hooks usually provide comfort due to their low weight and high control due to good visibility of the object being held (visual feedback) [34,35]. For example, Hosmer Hooks weigh below 400 grams and allow enough control to accomplish activities of daily living (ADLs) [37]. Nevertheless, these devices lack an anthropomorphic shape, which makes them aesthetically unpleasant.

The weight of commercial prosthetic hands is \( \approx 550 \) grams (without battery and glove) [33,37], which has been reported as being too heavy by 73% of adult-users and all children in an Internet survey [28]. The weight of body-powered devices (\( \approx 350 \) grams) has been reported to be lower than that of myoelectric devices [33]. The weight of the devices has been reported to be the main contributor for interface discomfort and user fatigue [37].

Prosthetic hands offer a more appealing cosmetic look. The challenge lies in making them as comfortable and controllable as prosthetic hooks.

Other factors are also important for the acceptance of assistive devices. They must suit the environment, the manufacturing and fitting of the device in the country must attend to local needs, and they must be suitable for the user. A match with the needs of the user can be addressed by engaging the user in the assessment and selection of the device. Follow-up is also necessary to ensure safe and efficient use. This involves access to local maintenance [27]. This PDEng promotes local manufacturing by designing a 3D-printable device. It will allow a closer connection between the manufacturers and the users.
Functionality

Moreover, for ADLs, an amputee generally uses the prosthesis for secondary grasping; for example, this may involve holding an object while performing precision tasks with the dominant hand [36]. For ADLs, the power medium wrap is the most common grasp; this is the type of power grasp used to hold an object. See Figure A.2 [38–40]. The required force for a power grasp is 68 N [41]. Under this premise, load-carrying capacity becomes one of the most important challenges which must be overcome for prosthetic hands.

Voluntary-closing hands make it possible to manage the grasping force exerted in an object in a more intuitive way. More force in the input (for example, shoulder harness) is equivalent to more force in the grasp. This becomes important when handling compliant objects, for example, plastic cups. However, holding objects for longer periods could be tiring, as the force needs to be applied throughout the whole period [36].

2.1.3 Compliant-based prosthetic hands - literature-review

One proposed solution to meet the needs is to apply compliant mechanisms to prosthetic hands, as such mechanisms can reduce the number of parts, lower the weight, and decrease logistical and assembly costs. In addition, the availability will thereby be strongly improved in low-income countries, as flexure-based prosthetic hands can be designed to be manufactured by 3D printing.

The purpose of the systematic literature review was to study the state-of-the-art of compliant mechanisms applied in functional hand prostheses, referring to anthropomorphic yet artificial substitutes of upper limbs which have the objective of allowing ADLs. It was found that compliance analyses are still subject to the development of flexure mechanisms in prosthetic hands:

- Recent studies have shown an interest in compliance in undesired directions, but the studies have been only evaluated in the undeflected state.

- Others have evaluated compliances in the deflected state. However, their optimizing parameter was compliance in the actuation direction. If the compliance was too high in the unwanted direction, the decision was made to return to the pin joint.

Figure 2.4: On the left, prosthetic hook. On the right, prosthetic hand [9].
Optimization of the compliances in the unwanted directions on the deflected state (real position of the fingers in grasping) has not yet been studied.

To summarize, researchers who have applied flexure joints to prosthetic hands have overlooked the importance of studying power grasp capacity and the load-carrying capacity (support stiffnesses) in the full range of motion.

2.1.4 Function analysis system technique

Function analysis system technique (FAST) is a tool used to identify what adds value to a project. The value is defined as a ratio of the functions and the resources. The FAST diagram presented in Figure 2.5 was used to map the development of the project. It was constructed based on literature research (Appendix B) and discussions with technical experts (Appendix A.2).

![Figure 2.5: Functional analysis system technique (FAST) diagram](image-url)
The FAST diagram of Figure 2.5 is a 2-dimensional diagram which considers only the how and the why. When read from left to right, the question "How?" is addressed. When read from right to left, the question "Why?" is addressed. The functional analysis provides a clear image, in the top section of the diagram, of how unsatisfied needs are to be fulfill in this PDEng project. It is recommended, when the project advances and more parties are involved to perform a value workshop to identify functional requirements. Alternate ways to add value can be found when different perspectives are involved.

2.2 Description of the design challenge

Understanding the behavior of the flexures will allow for the 3D-printed manufacturing of a functional monolithic prosthetic hand without assembly. The number of parts is reduced, as is the probable weight of the system. Further studies of different flexure topologies that allow one rotational degree of freedom could lead to improved support stiffnesses.

Technological feasibility:

Two main questions need to be addressed if we are to understand the technological feasibility of a fully flexure-based prosthetic hand. These indicate the possibilities of using the technology in an upper-limb prosthetic device.

- What is the limit in grasping forces of flexure hinges for anthropomorphic prosthetic hands?
- Can the sideways deflection be kept to an acceptable level?

2.2.1 Operational requirements

**Mission definition:** Assistive device provide independence to the users during activities of daily living. This is accomplish by attending to the main grasps performed during the day. The prosthetic should be a custom design to match environment and special needs of the user.

**Performance and physical parameters:** A power medium wrap requires a force of 68 N [41] for performing ADLs. Objects below 500 grams are common for this type of grasp, and there is a direct relation between the weight of the object and the required force. Most of the grasps require hand openings of 50 mm or less [40].

The weight of a prosthetic hand should be below 400 grams [37]. Acquisition costs under 500 USD have been showed to be competitive against similar products [32].

**Utilization requirements:** Power medium wraps are used extensively to pick up and release objects. Approximately 154 objects per hour can be picked up [38]. It is the most common grasp in ADLs [38–40]. It accounts for 23% in duration of the grasps in the 7.45 hours study [38]. The mean duration per grasp is 12 seconds [38].
2.2.2 Technical performance metrics

Based on needs and operational requirements, the technical performance metrics for the PDEng project can be summarized as follows:

- Acquisition costs for the prosthetic hand should be under 500 USD.
- It must perform a power medium wrap with a grasping force of 68 N.
- It must be possible to grasp an object of at least 500 grams with a diameter of 50 mm.

This PDEng project focuses on pushing R&D technology (flexure mechanisms) to the prosthetic hand application. To meet the acquisition-costs metric—which is fairly important given the need—additive manufacturing techniques that have been successful for other designers are considered [32]. There is freedom in the technique if the acquisition costs and availability in the country where the prosthetic is going to be manufactured are within reasonable ranges. Selective Laser Sintering (SLS) and Fused Filament Fabrication (FFF) are both options for manufacturing the device.

Furthermore, the technical performance metrics—i.e., the grasping force and characteristics for the object to be held—are decomposed through a V-model (Fig. 2.6) and brought to the component level, the flexure hinge. The intention is to assess the feasibility of this R&D technology for this application.

The design challenges encountered in using flexures and strategies to measure the technical performance metrics are presented below.

Technical design challenge - Flexure hinges

In principle, flexures or compliant mechanisms can be seen to have high compliance in desired directions when compared to the support directions. The latter guide the motion of the compliant directions. When a flexure joint is deflected, the stiffness in the support directions decreases [10,42–44]. See Figure 2.7.

The support stiffnesses add in series, which means that the weakest stiffness determines the behavior of the hinge. As can be observed in Figure 2.7, the drop in the first parasitic frequency (directly related to the support stiffness) can be quite dramatic.
Nonetheless, it is possible to find certain configurations of parameters and layouts with a relatively flat behavior over the range of motion. For example, see Iteration 5 of Figure 2.7.

Current flexure-based hands present deficiencies in support stiffness in the large range of motion (Appendix B), and the reported grasping force goes up to 21.5 N in a three-finger, flexure-based robotic hand [11]. One of the issues reported by Odhner is the increase of the arm with respect to the metacarpophalangeal joint (MCP). See Figure 2.8 [11].

As observed in Figure 2.8, the distance $d_1$ increases as the finger is flexed to grasp an object. This distance is referred to by Odhner as the "proximal arm of torsion", and it can indeed produce torsion over the proximal flexure joint [11].

When a mug is grasped and carried in the air, the weight of the object produces a sideways force at the contact point. This force, translated to the proximal / metacarpophalangeal joint, acts as torsion due to the arm $d_1$ and acts in-plane bending in the flexure mechanism. In conclusion, at large deflections, when it is necessary to grasp and carry the weight of an object, the flexure joint is at its weakest point.

2.3 Objectives of the design project

The actual concept proposes the basis for the design of a flexure-based prosthetic hand and the allocation of requirements for flexure hinges.

Knowledge has been exploited regarding a large range of motion for flexure mechanisms from the Precision Engineering Chair of the University of Twente. The final design aims to reduce assembly and the need for qualified personnel for that assembly to a minimum in low-income countries. Therefore, the main objectives of the design project are as follows:

- Document unsatisfied need for prosthetic hand devices.
- Identify technical performance metrics that meet the main needs.
- Decompose the requirements of the system to apply flexure hinge technology.
- Create a methodology to analyze the technical performance metrics of flexure hinges for prosthetic fingers.

![Figure 2.7: Natural frequencies of the first unwanted vibration mode](image-url)
Besides a unique, optimum, prosthetic hand, it is more important to present a methodology for the development of the most critical hinge in flexure-based prosthetic hands. Two main reasons can be mentioned: the first concerns integration with other components and sub-systems. For example, the type of actuation and position of the forces and dynamic requirements such as fatigue or grasping speeds will affect the results. The design-dependent parameters are not linear with the technical performance metrics, and the optimized solution will be different in each case. The second main reason is the rapid growth of the 3D-printing field, including procedures and materials which are available at low costs.

2.4 Evaluate technical performance metrics

At the component level, the technical performance metrics were analyzed at three different levels. See Figure [2.9].

Conceptual flexure-hinge designs were analyzed with a home-bred, efficient, non-linear, multi-body computer-modeling method called SPACAR [45]. The efficiency of the method makes it possible to test different sets of design-dependent parameters in a relatively short time.
The validation was realized in two steps, and iterations occurred at different levels. See Figure 2.10. First, the set of optimized parameters were used to build a CAD model that was tested in a commercial FEM software package. Furthermore, a prototype was manufactured for validating the results experimentally. Details related to the design and improvements of the experimental setup are presented in Appendix C.

Several iterations were performed: after SPACAR optimization, after improvements in the test rig, and also after results were validated. Such iterations are expected to be continued as integration with other components and subsystems occurs. See Figure 2.3.

### 2.4.1 Design for affordability

As mentioned, affordability represents one of the main needs for prosthetic hands. Life-cycle costs must be considered and studied at different stages of the system. Main factors which affect life-cycle costs originate, according to Blanchard and Fabrycky [8], from the following: engineering changes which occur during design and development, changes in suppliers during the procurement of system components, system production or construction changes, and from unforeseen problems. For these reasons, flexibility in the methodology for the design and development is of interest.

A design-to-cost technical performance metric must be set in a proactive basis at the concept design stage. At this stage, design to unit acquisition cost is defined below 500 USD per hand.

The acquisition costs of the fully flexure-based prosthetic hand, manufactured in nylon, are 88.3 USD (75 euros) by FFF and 568.7 USD (483 euros) by SLS print. Both

![Figure 2.10: Model for indicators.](image-url)
quotes were obtained online in 3D Hubs, which is an initiative used to connect users with local manufacturers [46]. Acquisition and assembly costs related to the tendons and their routing must be added for the current prosthetic hand (3.5 USD for a Kevlar cord).

By outsourcing manufacturing to a local manufacturer from 3D Hubs, $C_{PM}$ costs are already included in the acquisition costs. For example, costs cover tooling equipment, fabrication, material, packing, and, shipping and manufacturing rework. Other possible costs ($C_{PC}$ and $C_{PL}$) for the local manufacturer are also included: manufacturing facilities, inventory warehouses, personnel and training, and training and equipment.

The process of producing a prosthetic hand starts with an assessment of functionality and measurements of the residual limb. These are used to select a device and customize the product dimensions for the user. This step is followed by manufacturing the device, post-processing, and assembly. Later, the fitting of the device must occur, which involves creating an interface between the device and the user through products like sockets and/or harnesses. The last step is prosthetic training and follow up.

During the whole process, there are costs that are not considered at this stage and can be difficult to quantify. For instance, these include costs involved with taking personal measurements before manufacturing or prosthetic training costs. Volunteer initiatives try to bring these costs down to only the acquisition costs of the devices. See the e-Nable Device Sizing web page [47].

For the nature of the fully flexure-based prosthetic hand, maintenance costs are either too low or nonexistent. Flexure mechanisms has no friction between moving parts, which usually leads to wear. However, the actual tendons—a kevlar cord—produce friction in each grasp. This will have a negative influence in the life of the prosthetic hand.

The current acquisition costs, of the fully flexure-based prosthetic hand, when printed by FFF are 91.8 USD (assembly costs are not considered). This meet the technical performance metric, acquisition cost below 500 USD. The acquisition costs when printed by SLS, 572.2 USD (excluding assembly costs), exceeds the metric. As the 3D-printing technology continues to grow these costs are expected to be reduced.
Chapter 3
Design Methodology and Development

3.1 Abstract

Flexure-based finger joints for prosthetic hands have been studied, but until now they lack stiffness and load-bearing capacity. In this paper, we present a design which combines a large range of motion, stiffness, and load-bearing capacity with an overload protection mechanism. Several planar and non-planar hinge topologies are studied to determine load capacity over the range of motion. Optimized topologies are compared in a 30-degree deflected state in terms of stresses by deflection and grasping forces. In addition, support stiffnesses were computed for all hinges in 45 degrees of range of motion. The Hole Cross Hinge presents the best performance over the range of motion in sideways stiffness and a grasping force up to 36 N when deflected 30°. A new concept, the Angle Three-Flexure Cross Hinge, provides outstanding performance in grasping forces up to 48 N when fully deflected; while loaded with a sideways force of 5 N in deflected position, a 32% reduction of the maximum grasping force and a deflection of 1 mm was observed. Experimental verification of the support stiffness over the range of motion shows some additional compliances, but the stiffness trend of the printed hinge is in line with the model. The presented joint’s power-grasping capability outperforms that of current flexure-base hands and is comparable to that of commercial, non-flexure-based prosthetic hands. In the event of excessive loads, an overload-protection mechanism is in place to protect the flexure-hinges.

This chapter is awaiting publication as a journal paper publication.
Abstract— Flexure-based finger joints for prosthetic hands have been studied, but until now they lack stiffness and load-bearing capacity. In this paper, we present a design which combines a large range of motion, stiffness, and load-bearing capacity with an overload protection mechanism. Several planar and non-planar hinge topologies are studied to determine load capacity over the range of motion. Optimized topologies are compared in a 30-degree deflected state in terms of stresses by deflection and grasping forces. In addition, support stiffnesses were computed for all hinges in 45 degrees of range of motion. The Hole Cross Hinge presents the best performance over the range of motion in sideways stiffness and a grasping force up to 36 N when deflected 30°. A new concept, the Angle Three-Flexure Cross Hinge, provides outstanding performance in grasping forces up to 48 N when fully deflected; while loaded with a sideways force of 5 N in deflected position, a 32% reduction of the maximum grasping force and a deflection of 1 mm was observed. Experimental verification of the support stiffness over the range of motion shows some additional compliances, but the stiffness trend of the printed hinge is in line with the model. The power-grasping capability of the presented joints outperform that of current flexure-base hands and is comparable to that of commercial, non-flexure-based prosthetic hands. In the event of excessive loads, an overload-protection mechanism is in place to protect the flexure-hinges.

Index Terms— Compliant joints, flexures, robotic hand, prosthetic hand, anthropomorphic, additive manufacturing.

NOMENCLATURE

MCP Metacarpophalangeal.
ROM Range of motion.

E Young’s modulus.
G Shear modulus.
SLS Selective laser sintering

I. INTRODUCTION

Flexure joints applied in prosthetic and robotic hands have been of interest in recent years [1]–[4]. Some of the advantages of an integrated flexure design include more stable grasps and a reduced number of parts [3]–[5]. Furthermore, when 3D-printing technology is used to manufacture a prosthetic hand as a single, monolithic structure, absence of assembly can be achieved, thereby reducing overall costs.

A major challenge for flexure joints in large range-of-motion applications is the strong decrease of support stiffness in load-carrying directions when deflected [6]–[8]. This loss of support stiffness for large ranges of motion has led to the reconsideration of flexures in the MCP joint. The accompanying poor load-carrying capacity currently prevents widespread applicability in robotic and prosthetic hands [3]. Therefore, it is of interest to study the mechanical behavior of monolithic, integrated flexure-joint designs over the whole range of motion. The decrement of the stiffness in the support directions also leads to a loss of the load-bearing capacity of the hand. Especially when including tendon actuation and high grasping forces, elastic instability of the joint (buckling) can result in reduced load-carrying capacity.

Researchers of the UB Hand compared several flexure topologies for robotic hands by analyzing compliance matrices in undeflected position
Additionally, Tavakoli et al. present new topologies and have analyzed the flexure stresses and deflections for the undeflected state [1]. Although analyzing the stiffness properties of flexure topologies in an undeflected state allows the use of simple linear beam equations, it gives no lead to the stiffness properties at larger deflection angles due to the strong non-linear behavior. Furthermore, as critical stiffness and load typically occurs at a maximum deflection angle, stiffness at a maximum deflection angle rather than at the undeflected state is of primary interest.

Kalpathy used a pseudo-rigid-body model with an approximation of Timoshenko beam theory to model soft leafsprings in a large range of motion [2]. Although pseudo rigid-body modeling allows for larger deflections, it is limited to simulation of its kinematic behavior and stiffness in the free-motion direction. Therefore, evaluation of the support stiffness at large deflection angles is still unavailable.

Odhner has presented the “Smooth Curvature model” to calculate compliance matrices in large deflections of planar leafspring designs, as this can be associated with stable grasps [10]. This method allows for evaluation of support stiffness at large deflections. However, it describes the compliance matrix for only the two-dimensional case. For typical loading-conditions, out-of-plane stiffness and load-carrying capacity are important also. Furthermore, it only allows for the evaluation of planar hinge designs.

In addition, the Medium Power Wrap was identified as the most common grasp used in Activities of Daily Living (ADLs) [11], [12]. It is widely used to pick up and release objects. Most objects weigh under 500 grams and require an up to 50 mm hand opening [13]. Therefore, this mode of grasping is the main focus of this research.

In this paper, we exploit a flexible, multibody method to calculate and optimize several flexure hinge topologies, including non-planar topologies, during a cylindrical medium power wrap (Fig. 2). First, we develop an optimization strategy to maximize grasping force for each topology in a deflected state. Second, several joints are presented and the optimized topologies are compared. The comparison is based on stresses due to grasping force and sideways loads. Furthermore, a comparison is made of the sideways support stiffness over the whole range of motion for the different topologies. Third, an overload-protection mechanism for the sideways force is presented. A FEM analysis is used to obtain the stiffness of the hinge, which is subsequently corroborated with measurements.

II. DESIGN METHODOLOGY

A. Optimization loadcase

A finger is designed to be in a rest position that allows 15° of passive extension (ROM\textsubscript{pass}) and −30° of active flexion (ROM\textsubscript{act}). See Fig. 1. This range of motion allows one to grasp objects in the medium wrap range.

Since the fingers have high compliance for rotations around the z-axis (Fig. 1), the passive extension is achieved by contact with an object. The contact will open the hand to allow larger objects to be grasped. The extension is actuated by a tendon force \( F_{\text{act}} \) which deflects the flexure up to −30° around the z-axis.

The metacarpophalangeal joint (MCP) has been identified as the critical joint [3]. When holding an object, the contact force and weight of the object result in a combination of in- and out-of-plane bending loads of the flexure elements. See Fig. 2.

Since it is of interest to study the functionality of hands while power grasping, a contact point common to all hinge topologies is defined (Fig. 2). Thus, the loads affect the hinges similarly and the anthropomorphic dimensions of the finger are independent of the size of the hinge.

For the optimization, the tendon is actuated to position the finger at −30° of rotation. At this
position, a contact profile is modeled and the reaction force $F_{\text{grasp}}$ is measured [14]. Friction between the object and the finger is not considered.

The actuation force $F_{\text{act}}$ in the tendon is increased until failure. The maximum grasping force $F_{\text{grasp}}$ is recorded when the allowable stress $\sigma_{\text{max}}$ is reached.

In an additional simulation, a $F_z = 0.5$ N sideways force in the z-direction plane is loaded at the contact point when fully deflected.

**B. Workspace**

The workspace is defined based on the dimensions of a human hand [15]. For the proximal joint (MCP), a workspace of 60 mm long, 18 mm wide, and 17 mm thick is used. Width and thickness represent an average of the proximal-joint dimensions of all fingers for both males and females, except the thumb.

The length of the hinge is designed so that half of it is inside of the palm. See Fig. 2. Thus, the center of rotation of the flexure hinge is at the end of the palm and the beginning of the finger, which is equivalent to the location in a human hand. The proximal phalange acts as a housing for the other half of the joint.

**C. Hinge Topologies**

A series of hinge topologies are defined in advance. See Fig. 3. Their performance during power grasp is compared.

- Leafspring (LS)
- Solid-Flexure Cross Hinge (SFCH)
- Three-Flexure Cross Hinge (TFCH)
- Hole Cross Hinge (HCH)
- Angled Three-Flexure Cross Hinge (ATFCH)

The initial topologies are designed such that, in the un-deflected position, there is one rotational degree of freedom for flexion and extension of the fingers, and the stiffnesses in support directions are high. For comparison, a flexure hinge consisting of only a single leafspring is also evaluated, which provides support stiffnesses only in three degrees of freedom. This topology is used as a reference, as it is often used for prosthetic and robotic hands [2]–[4]. An initially curved design is added to generate high support stiffness at large deflections while sacrificing stiffness at smaller deflections. See Fig. 3e. Several of these hinges were defined previously by [7], including their design parameters $p$.

The Hole Cross Hinge combines the constant bending moment of a Three-Flexure Cross Hinge with the full width of a Solid-Flexure Cross Hinge except at the crossing where reinforced parts are used.

The concept of the Angled Three-Flexure Cross Hinge is introduced in this paper, with a topology similar to that of the Three-Flexure Cross Hinge. The hinge is defined so as to obtain straight elements when a specific angle is achieved. See Fig. 4b.

The length of the leafsprings are equal, like the diagonals of an isosceles trapezoid, to allow for an even stress distribution during deflection around the z-axis. This hinge is parametrized by the parameter vector $p$:

$$p = \{L_{\text{flex}}, B_{\text{phal}}, W_{\text{in}}, t\}$$  \hspace{1cm} (1)

$L_{\text{flex}}$ is the length of elements, $B_{\text{phal}}$ is the distance of the base (short side of the isosceles trapezoid), $W_{\text{in}}$ is the width of the inner element and $t$ is the thickness of the elements. See Fig. 4.

**D. Optimization**

The flexible multibody software, SPACAR, is used to evaluate the performance of the intrinsic geometric nonlinearities of the hinges [16]. By using nonlinear 3D beam elements, it is possible to efficiently compute the performance of a series of design parameters in large displacement motions.
and small elastic deformations. As a result, a relatively small number of elements produce accurate results at low computational cost.

A shape optimization based on the Nelder Mead method is used. The objective is to find the set of design parameters $p$ that maximize the performance within the specified constraints $C(p)$ [17]:

$$
p_{\text{opt}} = \arg \min_p F(p), \quad \text{subject to: } C(p) \leq 0
$$

(2)

The method minimizes a cost function $F(p)$, which is defined to achieve the highest grasping force when in contact with an object at $-30^\circ$ of flexion:

$$
F(p) = (1 + \lambda)^5 \frac{1}{F_{\text{grasp}}}
$$

(3)

$\lambda$ is a performance “penalty” for unfeasible solutions and $F_{\text{grasp}}$ is the grasping force at the allowable stress $\sigma_{\text{max}}$:

$$
\lambda = \max_{\theta} \left( \frac{d_z(p, \theta) - d_{z,\text{max}}}{d_{z,\text{max}}} \right) \quad \text{if: } d_z(p, \theta) > d_{z,\text{max}}
$$

(4)

The “penalty” factor shown in eq. 4 corresponds to the deflections in the z-direction $d_z$ due to sideways force $F_z$. Where $d_z(p, \theta)$ is the deflection for the current set of parameters $p$ and $d_{z,\text{max}}$ is the maximum allowable deflection.

A similar performance “penalty” is also applied when dimensions exceed the defined workspace and/or when flexures collide. This ensures collision free designs [18]. By applying these penalties to the cost function, soft constraints are added to the unconstrained Nelder Mead algorithm [17].

For each iteration in the Nelder Mead algorithm, $N+1$ cost functions values ($N$ equal to the number of design parameters) are compared and sorted according to $F(p_1) \leq F(p_2) \leq \ldots \leq F(p_{N+1})$, being $F(p_1)$ the solution with lowest cost (highest performance). Based on these results, a new parameter set $p$ is determined. If the performance of the latter is better than $F(p_{N+1})$, this value is replaced in the set of solutions. This process continues until a certain convergence criterion is satisfied, which is defined by the following:

$$
\frac{F(p_1)}{F(p_{N+1})} > 0.995
$$

(5)

which corresponds to a 0.5% deviation in performance in the current set of solutions [17].

Sixteen shape optimizations were conducted per hinge topology, each one with a different initial parameter set. In each optimization, a global or local optimum can be obtained. By conducting
several optimizations, the probability of finding a solution within 5% of the global optimum is greatly increased. For example, the probability to find a result within 5% of the global optimum for the Three-Flexure Cross Hinge is 62%. When conducting sixteen optimizations, the probability of finding a solution close to the global optimum is approximately 99% [17].

E. Experimental Setup

To validate the numerical model, a setup for measuring stiffness was used. See Fig. 5. A parallel guidance, 1-DOF in the gravity direction, is actuated when weights are added to the end effector. The vertical displacement is measured through a linear-variable differential-transformer (LVDT) sensor. The actuation stiffness of the parallel guidance has been taken into account.

To attach the finger to the parallel guidance, a wire flexure is used in the DOF of the parallel guidance. Since the wire flexure constrains only 1-DOF, torsion and in-plane bending can be measured from the tip of the finger.

A finger with only the MCP joint was printed using the SLS process. In the finger, the proximal and median phalanges are hollow with a shell of 1.5 mm, and the distal phalange is printed with a 100% infill.

III. RESULTS

A series of optimized hinges were found by evaluating the cost function $F(p)$. The hinges were deflected until $-30^\circ$ where contact was modeled. The reaction/grasping force at contact was calculated, and it is plotted in Fig. 6.

When an object is about to be grasped, a tendon force $F_{act}$ is first required to close the hand. This produces an initial stress $\sigma_{flex}$ in the flexure hinge. This stress can be observed in Fig. 6 at 0 N of grasping force.

The ratio between the maximum allowable stress of the material ($\sigma_{max}$ in Table I) and the stress due to deflection $\sigma_{max}/\sigma_{flex}$ is lower than 1.25 for the Solid Flexure and Hole Cross Hinge. For the other hinges, the ratio is lower than 1.6. In general, a higher ratio is desired for flexure mechanisms that are going to be cyclically loaded.

After the object makes contact, the tendon force is increased and the grasping force is produced.

In most of the hinges, with the exception of the Angle Three-Flexure Cross Hinge, a change in the stress-grasping force slope can be identified. In these inflection points, the tendon actuation force creates instability in the flexure hinges.

The Solid Flexure Cross Hinge and the Hole Cross Hinge present slight increase of stress with increasing grasping force, as the grasping force increases until instability shows. The Leafspring and the Three-Flexure Cross Hinge exhibit a steep slope between the stresses and grasping force.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\theta_{min}/\theta_{max}$</td>
<td>$-15^\circ/30^\circ$</td>
<td></td>
</tr>
<tr>
<td>$t_{min}/t_{max}$</td>
<td>mm</td>
<td>0.5/2.5</td>
</tr>
<tr>
<td>E</td>
<td>GPa</td>
<td>1.7</td>
</tr>
<tr>
<td>G</td>
<td>GPa</td>
<td>1.5</td>
</tr>
<tr>
<td>$\sigma_{max}$</td>
<td>MPa</td>
<td>50.0</td>
</tr>
<tr>
<td>$\sigma_{max}/E$</td>
<td></td>
<td>29.4</td>
</tr>
</tbody>
</table>

Fig. 6. Comparison of optimized hinge topologies deflected at $-30^\circ$. 

Fig. 5. Experimental setup for stiffness measurement.
The Angled Three-Flexure Cross Hinge presents a behavior that is close to linear without inflection points.

Furthermore, it is of interest to analyze the behavior of the hinges under the weight of the grasped object. The sideways stiffness $K_{sw}$ is the inverse ratio of a measured displacement $d_z$ at the contact point and an applied load $F_z$. See Fig. 2. $K_{sw}$ is affected by the translational compliance in $z$ and rotational compliances:

$$K_{sw} = \frac{F_z}{d_z} \quad (6)$$

The behavior of $K_{sw}$ over the range of motion is presented in Fig. 7. A tendon force was applied to deflect the flexure joint to a specific angle. At that deflection, a load $F_z = -2$ N was applied, and a $K_{sw}$ was calculated as described in equation (6). In the optimization, while the grasping force was part of the cost function, the sideways stiffness was treated as a soft constraint.

The Hole Cross Hinge outperformed over most of the range of motion, with a drop of support stiffness of only 28.7% (Fig. 7). This behavior matches with the high $\sigma_{max}/\sigma_{flex}$ ratio from Fig. 6. It suggests a hinge with high stiffness in all directions.

The Leafspring performed the second best for deflections beyond $-5^\circ$ with a peak at $-15^\circ$. This behavior is related to the off-centric load produced by the weight of the object being held. By analyzing the dashed line from Fig. 8 it can be observed that, at $-15^\circ$, that the torsional component is almost not present. The same behavior was observed for the other hinges at different angles. The stiffness drop in the full range of motion for the leafspring is 46.7%.

The Angled Three-Flexure Cross Hinge exhibited almost symmetric behavior with a peak around $-5^\circ$. The stiffness drop was the second lowest. In general, it presented a decent performance with a deflection around 1 mm for 5 N (500 ml bottle of water) of sideways force at the lowest sideways stiffness ($-30^\circ$). See Table II.

The Hole Cross Hinge and the Angled Three-Flexure Cross Hinge have resulted in hinge topologies with the best performance. The Hole Cross Hinge outperformed in sideways stiffness over the range of motion. The almost linear behavior of the Angled Three-Flexure Cross Hinge in the stress-grasping force is of interest.

Furthermore, the influence of the sideways forces over the stress was analyzed. The hinges were loaded up to contact, where grasping force was produced and a sideways constant load was applied.

Figure 9 shows that the stress for the Hole Cross Hinge surpasses the allowable stress limit $\sigma_{max}$ at

<table>
<thead>
<tr>
<th>TABLE II</th>
<th>LOWEST SIDEWAYS STIFFNESS IN THE RANGE OF MOTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hinge</td>
<td>Stiffness (N/mm)</td>
</tr>
<tr>
<td>Three-Flexure Cross Hinge</td>
<td>3.8 @ -30 deg</td>
</tr>
<tr>
<td>Hole Cross Hinge</td>
<td>7.6 @ 15 deg</td>
</tr>
<tr>
<td>Leafspring</td>
<td>5.0 @ 15 deg</td>
</tr>
<tr>
<td>Solid Cross Hinge</td>
<td>5.3 @ -30 deg</td>
</tr>
<tr>
<td>Angle Three-Flexure Cross Hinge</td>
<td>5.4 @ -30 deg</td>
</tr>
</tbody>
</table>
$F_{\text{grasp}} = 36.25 \text{ N}$ and $F_z = 0 \text{ N}$. The sideways force limits the grasping force to $F_{\text{grasp}} = 25 \text{ N}$ at only $F_z = -2 \text{ N}$, which represents a 30% reduction of the performance.

The Angled Three-Flexure Cross Hinge at $F_z = 0 \text{ N}$ presents a linear and steady increase of the stresses until $F_{\text{grasp}} = 48 \text{ N}$. The influence of the sideways stiffness was moderate compared to that of the Hole Cross Hinge. A 32% reduction of grasping force was observed when loaded with a sideways force of $F_z = -5 \text{ N}$.

No clear relation between the sideways stiffness and the influence of the sideways force over the stress was found. Depending on the resulting hinge and the loads applied, the higher von Mises stress can be found in different points in the hinge. This means that, for a specific hinge, the sideways force can have a higher influence over the stress than for others, depending where the concentration of stresses occurs during grasping.

Odhner reports grasping forces as high as $21.5 \text{ N}$ for a three-finger robotic hand with flexure hinges only in the proximal joint position [3]. The latter measurement was accomplished in a grasping position that avoided sideways forces. Belter reports holding forces at the tip for commercial non-flexure-based prosthetic fingers in a range between $3–16 \text{ N}$ [20]. The presented performance of the Hole Cross Hinge and the Angled Three-Flexure Cross Hinge represent considerable improvements to current flexure-based hands and can be compared to current commercial non-flexure-based prosthetic hands [20].
the FEM makes it attractive for efficient flexure hinge optimizations.

B. Overload-protection mechanism

A mechanism located in both sides of the finger that prevents failure of the flexures when loads are over the limits is proposed. See Fig. 11. This concept prevents excessive displacements in torsion and in mostly all support directions with the exception of loading in the positive y-direction. Undesired translations in the z-direction are also constrained by a wall that is not shown in figures for convenience.

In Fig. 11b, contact is produced by excessive torsion on the finger. Also, when overloading due to lateral (x-direction) or compression forces (negative y-direction), contact is expected. Rolling contact is still possible between the palm and the phalange.

Figure 12 shows the kinematic analysis used to determine the shape of the mechanism. An exploration of the trajectory of the points of the phalange was done through the whole range of motion. See the red lines of Fig. 12. Where the points barely move during deflection suggests the approximate location of the center of rotation. Furthermore, a combination of points were selected based on their trajectories and connected with the blue lines from Fig. 12 to create a shape that constrains overloading in the support directions. Particular attention was paid to overloading in torsion.

The center of rotation of a Hole Cross Hinge is approximately at 2 mm to the right of the crossing of the flexures. See Fig. 12. The translation of the center of rotation is attributed to the tendon force which creates the deflection of the finger. At $-30^\circ$, contact is produced to constraint deflections beyond this point.

IV. CONCLUSIONS

In this paper, five flexure-based finger joints topologies are presented, optimized and compared. The joints were kept within stress limits of 50 MPa and MCP joint human dimensions while a combination of $45^\circ$ large range of motion and grasping force of at least 25 N was carried out. The topologies have been designed to withstand relatively high tendon actuation forces.

The Hole Cross Hinge presented the best sideways stiffness over the range of motion; however, high stress at deflection and a high influence of the sideways force over the stresses could limit its use. The Angled Three-Flexure Cross Hinge exhibited noteworthy behavior in terms of high grasping force and, almost linear stress behavior while grasping. It also exhibited a moderate influence of sideways force over the stresses.

Experimental verification of the support stiffness over the range of motion reveals some additional compliances, but the stiffness trend of the printed hinge is in line with the model. The power grasping capability of the joints outperforms that of current, ‘state-of-the-art’, flexure-base hands and is comparable to that of commercial, non-flexure-based prosthetic hands. In the event of excessive loads, an overload-protection mechanism is in place to protect the flexure-hinges.

ACKNOWLEDGMENT

We thank W. Pot for his contribution to the experiments.
REFERENCES


Chapter 4

Conclusions and Recommendations

4.1 Conclusions

A way was found to meet an important societal need for around 304 million people who require prosthetic / orthotic devices in developing countries.

The main needs are low cost, low weight, and ability to perform activities of daily living. In terms of functionality, a power medium wrap is essential if a prosthetic hand is to perform successfully and consistently.

The technical performance metrics related to the needs are as follows:

- Acquisition costs of the prosthetic hand must be under 500 USD.
- It must be possible to perform a power medium wrap with a grasping force of 68 N.
- It must be possible to grasp an object of at least 500 grams with a diameter of 50 mm.

Two models were developed during the PDEng program to optimize flexure hinges applied to prosthetic hands. First, the force-based flexure hinge model, presented in Appendix D resulted in grasping forces as high as 21 N with the Angled Three-Flexure Cross Hinge. Second, the contact-based flexure model, discussed in Chapter 3, produced grasping forces as high as 48 N. Again, this was accomplished with the Angled Three-Flexure Cross Hinge.

The contact-based flexure model allows for a more computationally efficient model. There were fewer runs per configuration than for the force-based flexure hinge model. Additionally, the cost function from the contact-based model directly maximizes the technical performance metric: the grasping force. Hence, the highest grasping forces were obtained for the contact-based model. However, the stresses for deflection were higher for the latter model, which potentially has negative effects on the mean time between failure of the system. However, it was decided to proceed with the contact-based model due the its efficiency.

The Angled Three-Flexure Cross Hinge found through the contact-based flexure model is able to withstand 500 grams of sideways forces with 1 mm deflection at the contact point while still achieving a maximum grasping force of 32 N. In addition, a novel overload-protection mechanism was designed using a kinematic analysis. It could be used for sideways forces above the 500 grams.
Acquisition costs are achieved under 500 USD via FFF 3D-print manufacturing. See Section [2.4.1]. Although different 3D-printing process can be considered, it is important to take the resulting anisotropies of the material into account to be included in the model.

A flexure-based prosthetic hand that meets the acquisition costs and capability of grasping common objects was achieved. The required grasping force was not concluded. However, the contact-based model showed promising results. Overall, the design could provide a solution for developing countries.

### 4.2 Recommendations and future work

Technical performance metric tradeoffs could influence further decisions. For example, the power-grasping force vs mean time between failure vs acquisition costs. Fatigue of the flexure joints will have an effect on the mean time between failure. Acquisition costs will have an influence in the manufacturing process and as consequence in the material properties.

Challenges remain in the interfaces and how these are integrated. Subsystems as the actuation mechanisms play a vital role that needs to be study and integrate it to the prosthetic hand. Furthermore, validation of actuation forces and transmission from the user to the prosthetic hinges. High actuation forces have been reported to be an issue for the fatigue of the user.

At this stage, the only part that requires maintenance is the tendons. By integrating the actuation in the 3D-printed design, a fully flexure-based prosthetic hand may be realized. Such a design will reduce the assembly time required for the hand and reduce maintenance.

It is recommended that a grasping force test with the full hand prototype be developed. During power grasping with an under-actuated hand and multiple hinges, several points of contact are expected, and the contribution of each digit may not be the same. Understanding force distribution could a goal of future studies.

Expand the optimization to multiple hinges where more contact points are considered, at the finger level. This should probably be done in conjunction with an analysis of the force distribution during the power grasping. An exploration in the workspace is suggested, including consideration of dimensions for children.

Societal embedding can be accomplished by open-source design for a nonprofit organization.
Appendices
Appendix A

Additional Requirements

A.1 Risk management

A strengths, weakness, opportunities, and threats (SWOT) analysis was performed at the system level. By evaluating weaknesses and threats, it is possible to identify the risks and propose potential risk treatments.

<table>
<thead>
<tr>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td>Assembly and wear-free mechanisms</td>
<td>Material anisotropies from manufacturing process</td>
</tr>
<tr>
<td>Knowledge/tools for efficient flexure mechanisms design</td>
<td>Difficult follow-up for rejection rates</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Opportunities</th>
<th>Threats</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manufacturing process allows freedom to consider complex mechanisms</td>
<td>Freeze of development of the prosthetic hand</td>
</tr>
<tr>
<td>Freedom to develop and explore technical opportunities</td>
<td>Interface between development projects</td>
</tr>
<tr>
<td>Growing low-cost prosthetic industry</td>
<td>Requirements are not met</td>
</tr>
<tr>
<td>Pool of volunteers open for development and deployment</td>
<td>Difficult to manage trade-offs</td>
</tr>
<tr>
<td>Scale-up production of custom prosthetic hands</td>
<td>User acceptance</td>
</tr>
</tbody>
</table>

Direct action can be taken over technical risks related to the flexure mechanisms. Other risks are involved in the environment, which are related to stakeholders. The risks can be categorized under the terms avoid, control, accept, and transfer (A-CAT):
A.1.1 Avoid

Material anisotropies from 3D-printing technologies, like Fused Filament Fabrication, are currently outside of the scope of the system. Rapid growth of the additive manufacturing field can lead to low-cost additive manufacturing that can produce isotropic products. Moreover, the accuracy of the prototyping process with regard to dimensions and/or warping effects can lead to poor quality in the flexure mechanisms. These risks are avoided by leaving them outside of the scope of the project.

A.1.2 Control

To mitigate the risk of being unable to achieve the required support stiffness, two actions are suggested: mechanical stops can add extra stiffness under undesired deflection, and the relax-starting position of the fingers could be at a certain angle instead of at 0 degrees. In the latter case for grasping a single object the required deflection angle of the joints will be reduced.

In case the required grasping force is not met, it is possible to use the same methodology for a redesign of the system. Different flexure mechanisms or actuation strategies can be evaluated.

User acceptance can be affected by factors like aesthetics or robustness. A value workshop would help to mitigate this risk by involving prosthetic technicians, industrial designers, marketers and users.

A.1.3 Accept

Follow-up for rejection rates can be complicated on a large scale. Although it is important for the success of a product to receive a feedback from the user to close the life-cycle, the location of the majority of the users in developing countries adds another degree of complexity.

A.1.4 Transfer

Freeze development and interface between projects are a consequence of the fact that the PDEng project has been concluded. These risks are accordingly transferred to the University of Twente. Funding would be required to develop the system to a complete the life-cycle: i.e. bring it to the market and to provide support, feedback, and disposability.

Furthermore, the interface between the projects is partially controlled by promoting reports which facilitate the transfer of knowledge and which are supported by the personnel involved with this PDEng project who will continue as employees of the University of Twente.

A.2 Stakeholders

The PDEng project is a cluster of research groups from the University of Twente (UT). It was sponsored by the UT via the Science Based Engineering (SBE) program in accord
with the university approach “high tech, human touch”. A stakeholder’s diagram is presented in Figure A.1.

The main stakeholders can be found in the University of Twente block. These persons are aware of, committed to, and have direct influence on the project. The approbation of the project is based on the satisfaction of their goals.

When thinking broader about the prosthetic hand and the environment, other stakeholders enter into the cycle. Their role in this project is secondary, but they are still considered in the decision-making process.

The end users are the people who will use the prosthetic hands. Although accurate numbers for the demand have not been found, some estimates were presented in Chapter 2. The importance of the involvement of the users in the selection process of the devices is also considered there [27]. Their participation was considered through the findings of researchers who are in direct contact with the end users. See Section 2.1.

The prosthetic technicians play an important role in the product life cycle. They recommend the use of a specific device to the end user and guide him/her during the adaptation process. The rejection rate for prosthetic hands is highly dependent on the adaptation process and the timing from amputation to prosthesis [48].

National regulatory authorities exist in 64% of the WHO members states, according to the 2017 Global Atlas of Medical Devices report [49]. This number accounts for at least some regulation. For example, classification of the medical devices or import controls. Most of the countries with no regulations or with no data are low-income and lower-middle-income countries. In Europe, most countries have a regulative agent.

According to the definition of medical devices of the Medical Device Directive of the European Commission, prosthetic hands are medical devices. Thus, they will require a CE marking that is accredited by the same directive, thereby allowing a medical device to enter the European market.

The industry is a part of the external entities which are formed by manufacturers of commercial prosthetic hands (Ottobock, Touch Bionics, Vincent Systems, TRS, etc.). The offer is quite limited, and their aim is usually towards high-performance devices. The costs start at 9,000 euros per device.

Due to the high cost of prosthetic devices, the high demand in developing countries, and the mass expansion of 3D printers, new opportunities have arisen for those in need. Non-profit volunteer initiatives (such as e-NABLING, The Open Hand Project, etc.) have emerged to provide affordable prosthetic solutions to those in need.
etc.) offer open-source collaboration. Designs can be freely downloaded and printed next to the user, thereby providing agile manufacturing.

Insurance companies or funding organizations play a determinant role in the field as external entities. The costs can be elevated for a person in need of such device. Actually, it is projected that a typical amputee will go through USD$1.4 million worth of treatment, including surgeries, prosthetic, as well as therapies [50]. Together with insurance companies, the users cover the cost of prosthetic devices.

Governments play a role with the regulative agents and facilitators to develop the life-cycle of prosthetic devices. According to the 2011 World Report on Disability, many people acquire their assistive devices in the open market. An economy of scale can reduce the purchasing and production costs. Some governments offer tax exemptions or loans to companies that produce assistive devices. In support, governments could promote 3D-printing farms and support centralized, large-scale purchasing of raw materials (3D printing filaments).

Governments can benefit from making prosthetic hands accessible to users. For instance, the Ontario Workers’ Compensation Board reports that 93% (473 out 508 people) of upper-limb industrial amputees return to the workforce. The report also found that users who wore the prosthetic hand more often were more likely to be employed [51].

The competition includes researchers from other universities, institutes or companies which are interested in developing similar mechanisms. Yale University and various universities (Coimbra, Bologna and Modena) have in recent years presented similar designs to that being developed by the University of Twente.

### A.3 Social embedding

The University of Twente, represents the main stakeholders, who will decide the path by which this technology becomes socially embedded, thereby possibly creating more research positions, offering the technology to the industry, or open-sourcing the device. Still, a path for social embedding has not yet been determined.

Currently, the maturity of the product is still far from social embedding. The actual PDEng project is a proof-of-concept of the technology which has yielded promising results. Though the product is still young, some impact will be created, and ideas for socially embedding can be studied.

As a proof of concept, our product will first have an impact on the research world. A methodology for comparing the performance of the flexure (sub-artifact) was developed, and it can be adopted by other researchers. It is possible that collaborations with other universities will be focused on other areas relevant to the prosthetic hands, like actuators or feedback devices.

External stakeholders represent the largest actuator group and can be clients of the product. To embed the flexure mechanism in the market, it is possible to sell the presented technology to already-established manufacturing companies.

Others have taken the technology to start-ups that can offer an artificial hand with flexure mechanism to a specific niche: i.e., robotic hands (https://www.righthandrobotics.com/). Right Hand Robotics is the initiative of a post-doctoral researcher, from Yale University, with a similar research project.

Non-profit volunteer initiatives could be the largest opportunity for this technology
to become societally embedded. Several social groups have already been linked, including designers, end users, manufacturers, and prosthetic technicians. This gives stability to the technology. A flexure-based prosthetic hand is already offered by one of these initiatives [47].

Table A.2: Social embedding

<table>
<thead>
<tr>
<th>Aim for Society-embedded Design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Present the behavior, advantages and disadvantages of the flexural hinges through publications.</td>
</tr>
<tr>
<td>Present the flexural hinges actuated by different mechanisms, focusing on actuators of commercial prosthetic hands.</td>
</tr>
<tr>
<td>Prove the reliability of the mechanisms.</td>
</tr>
<tr>
<td>Open-source the technology through volunteer initiatives, thereby creating the niche for technology transition.</td>
</tr>
</tbody>
</table>

A.4 Recyclability/Disposability

The 3D-printed techniques and materials considered for the manufacturing process of the flexure-based hand are thermoplastics. Nylon, acrylonitrile butadiene styrene (ABS) and polylactic acid (PLA) are recyclable materials, and there is already an infrastructure for producing new filaments from parts. Projects like the Perpetual Plastic Project aim to recycle thermoplastics and produce filaments. The cycle is comprised of cleaning and dying the plastic, shredding, extruding the filament, and 3D-printing a new product [52].

Nylon, ABS, and PLA are water-insoluble. Detrimental effect on plants, animals or microorganisms are not expected, according to the manufacturer. It was not possible to define a waste code number as per the European list of wastes [53].

The United States of America classifies these materials as significant materials under the Clean Water Act (40CFR 122.26); consequently, their disposal is regulated [54].

A.5 Operational Thinking

The thinking tracks make it possible to analyze the interaction of the system with the environment to determine how the system interacts with it and affects each other. The power medium wrap was analyzed with the help of one of the thinking tracks: the operational thinking track.

- The user identifies a mug resting on the table.
- The user approaches the mug with the prosthetic hand.
The user actuates the prosthetic hand to close the hand.

The mug is engaged by the hand and build force is produced.

The user uses elbow and shoulder to raise mug to mouth.

The wrist is used to incline the mug towards the mouth.

Wrist, elbow, and shoulder are used to return the object to the table.

The actuation of the prosthetic hand is released.

The prosthetic hand is moved away from the mug.

Based on the operational thinking, the forces involved in the process of holding an object can be identified. The two main forces are the grasping force and the sideways force. The latter corresponds to the weight of the object. In the case of holding a 500 ml bottle of water, a sideways force of 5 N is exerted at the contact point of the fingers. See Figure 2, Chapter 3.
Appendix B

Compliant Mechanisms in Hand Prostheses

The purpose of this systematic review is to study the ‘state-of-the-art’ of compliant mechanisms applied in functional hand prostheses, referring to anthropomorphic artificial substitutes of upper limbs with the objective of allowing ADLs.

B.1 Method

This literature search, updated to March 2018, intends to make an inventory of prosthetic hands developed in the last 20 years [1,4,18,20,22,31,33,37,39,55–69]; a list is presented in Table B.1. A small number of robotic hands were also registered because of their compliant nature [2,3,5,17,19,70].

Publications that focus on control, actuators or feedback mechanism, and those that did not present a novel mechanical design, were excluded from this review. A categorization of prosthetic hands into compliant (20%) and non-compliant (80%) is presented in Fig. B.1.

Subsequently, an analysis of the presented mechanical designs of the hand devices with compliant mechanisms is done in detail. The objective is to understand the ‘state-of-the-art’ and to identify possible improvements in the field.

As shown in Fig. B.1 and Table B.1, several researchers have attempted to make use of flexure-based mechanisms in hand manipulators.

Figure B.1: Percentage of prosthetic hands with compliant mechanisms.
Different compliant approaches for anthropomorphic hand manipulators have been used: rigid bodies connected by a flexure joint \([1–3, 14, 17, 18, 71]\), see Figs. B.2, B.3, B.4, B.5, B.7, B.8, and B.9; a flexure-based endoskeleton structure embedded on silicon \([19, 20, 22]\), see Figs. B.10, B.11, and B.13; and other compliant mechanisms \([4, 5, 65]\), see Figs. B.14, B.15, and B.17.

**B.1.1 Keywords**

Prosthetic hand, robotic hand, compliant finger, underactuated hands, ottobock, Michelangelo hand, bebionic, i-limb, anthropomorphic fingers, compliant grippers, prosthesis, e-NABLE, smarthand, ih2 azurra, 3d printed, disability, low-income.

**B.2 Rigid Bodies Connected by Flexures**

**B.2.1 University of Bologna (UB)**

The University of Bologna has been a pioneer in the use of flexure joints for fingers. Since 2002, a curved leafspring was proposed for robotic hands \([13]\), see top Fig. B.2. In 2004, they introduced the UB Hand 3, which included springs as a flexure mechanism \([14, 70]\), see bottom Fig. B.2.

![Figure B.2: UB Hand. Top, first flexural design \([13]\). Bottom, spring joints \([14]\).](image)

In their first versions, a monolithic finger was developed. The researchers used a Pseudo-Rigid body model with constant spring stiffness for kinetostatic analyses. The study involved only flexion and extension of the finger. Stress analysis or load-carrying capacity were not presented. Different actuation concepts were proposed and the 3DOF fully-actuated design was studied.

Later, the hinges were replaced by steel spiral coils which were used for routing the tendons. The hand is fully actuated in 4DOF (3DOFs flexion-extension and 1DOF adduction-abduction per finger).

The tendons were routed concentrically to the spiral coils until the respective actuated phalange. It was claim a simplified assembly and avoidance of parasitic motions when hinges were actuated independently.
On the functional side, the mechanical analysis of the steel spiral coils hinges was not found. 6.8 N of precision grasping force is reported in [37]. Neither the power grasping force nor the weight of the hand were reported.

B.2.2 Universities of Bologna and Modena

In recent years, different flexure layouts have been studied in a collaboration between the University of Bologna and the University of Modena to obtain monolithic 3D printed structures [15,16], see Fig. B.3.

The monolithic 3D printed fingers reduce the number of parts required. Berselli and Vassura integrated joints and actuation to obtain a monolithic, fully-actuated finger.

The studies identified that the compliance should be higher in one direction compared to the other directions. It focused on comparing the performance of flexure joints based on the compliance matrix in the undeflected position. Analytical equations of the elements of the compliance matrix are presented for initially curved leafspring for the undeflected state.

Closed-form equations to determine allowable rotation of different hinges are presented based on undeflected compliance terms. Rigid body transformations are used to find the center of compliance.

The compliance term in the flexion-extension direction is usually constant in the range of motion. The closed-form equations are valid to understand and compare the functionality of different compliance hinges. However, they do not consider the behavior in the other directions, as these compliances change in the large range of motion, and consequently the load-carrying capacity of the hand.

Furthermore, rigid body transformations are valid for small ranges of motion [16] and mostly for planar elements.

B.2.3 Huazhong U. and Georgia Tech

An equivalent pin model (EPM) approach to understand the non-linear behavior of flexure hinges, under pure moment, for robotic fingers was proposed by Guo and Lee [17], Fig. B.4.
The EPM is similar to a Pseudo-Rigid Body Model (PRBM), with the difference that the location of the joint is close to the center of rotation. The models consider the axis drift [72] existing in the compliance mechanism as the center of rotation changes for large displacements. The model is based on a multibody dynamics approach.

The intention was to develop a rigid body model less sensitive than the pseudo-rigid body model to the type of load applied. The errors in the orientation of the phalange and the position of the end effector were found to have a non-linear behavior. The latter is less sensitive to the type of load applied.

Guo and Lee studied three type of hinges: straight, convex and concave leafsprings [17] (Fig. B.4). According to the researchers, the center of rotation of the EPM plays an important role in the dynamic influence compared to a typical PRB model.

Guo and Lee’s publication focused on the model, and the flexure-based hand is a case study. Trajectories of the center of rotation and tip of the fingers are presented.

Functional requirements in terms of load-carrying capacity or weight of the device were not considered.

B.2.4 Yale University

Yale University introduced the use of leafsprings made of urethane for prosthetic hands [18], see Fig. B.5

The lower young modulus of the rubber joint offers high compliance in the actuation direction. However, it also offers undesired compliance in the other directions. The undesired compliance increase dramatically at large deflections, see Fig. B.6.

As shown in Fig. B.6, the distance \( d_1 \) increases as a large deflection of the joints occur. The increment of the arm \( (d_1) \) considerably increases the torsion on the proximal joint (MCP). For this reason, the elastic joint at the MCP was later replaced by a pin-joint [11].

The study of the flexure joints led to the development of the smooth curvature model [73–75]. Changes in the compliances at large deflections are considered in this model for planar structures (leafsprings).
The flexure joints are optimized in the actuation direction with interest in an ellipsis compliance in the contact point. However, the publications did not show special attention for the compliance in the unwanted direction and the influence in the load-carrying capacity.

Reported weight: 400 g. [76].

B.2.5 University of Wollongong [1]

Researchers presented a flexure-based finger aimed for prosthetic applications. It consisted of three notch joints.

Three conceptual notch designs were presented: a leafspring with round corners; a circular notch; and, an elliptical notch (Fig. B.7). The thickness (3 mm) and length of all elements was constant for all hinges.

The study is based on the compliance in the actuation direction. A Finite Element Analysis and validation were done with 5% relative error in the displacements of the end effector.
However, analysis on the stresses were not presented. The selected material was an elastomer, which makes sense for that thickness and the large range of motion.

Hysteresis was shown in flexion-extension of the finger, and it was attributed to material hysteresis and friction in the channels of the tendon.

### B.2.6 Ohio State University [2]

For soft joints, the compliance in the elongation direction becomes important. An adapted PRBM with an additional constant spring to model the elongation deformation was included.

![Figure B.8: Leafspring design and PRB model [2].](image)

The modified PRBM allows a better track of the tip of the finger than conventional PRBM or even the 3R PRBM [77], which is suited for large deformations.

However, this model only takes into consideration flexion and extension of the finger. The PRBMs consider the joints to be constraint in the other directions, and these loads can be important when considering compliant grasping hands.

### B.2.7 Tennessee Technical University [3]

Tennessee Technical University presented three type of flexure-based fingers for robotic applications. A monolithic 3D printed finger design was studied, and comparisons between Fused Deposition Modelling (FDM) materials were done.

![Figure B.9: From left to right: Cross-Leafspring, Solid Cross Flexure and Leafspring flexure hinges [3].](image)
An empirical flexion extension testing was conducted. The joints were deflected up to 184 degrees, for which they encountered plastic deformation. The Cross-Leafspring was selected because it presented less plastic deformations.

Furthermore, a comparison with an FEM package was done and differences in the reaction forces between testing and computational model were 8% for a deflection of 90 degrees. With $\sigma_y/E \approx 17$, it is expected that plastic deformation is present for that deflection. However, stresses were not presented in the publication.

Deflections in other directions and the weight of the hand were not reported as they were not part of the presented requirements.

B.3 Endoskeleton Structures

B.3.1 University of Coimbra - ISR Hand

The ISR Hand of the University of Coimbra used the urethane joints concept of the SDM Hand [19], see Fig. B.10. The importance of improving the compliance on the undesired directions was reported. For this reason a leafspring was integrated inside of the soft material.

![Figure B.10: ISR Hand, thin element embedded on a rubber joint [19]](image)

The hybrid approach led to increased stiffness in all directions compared to only the urethane joint. Calculations of the joint deflections of the rigid leafspring, in the actuation direction and in-plane bending, were made by traditional beam theory equations, which are only valid for small deflections. Reported weight: 530 g.

B.3.2 University of Coimbra - UC Hand

Later, the University of Coimbra presented a new prosthetic hand named the UC Hand [20]. Their design was an evolution of the previous ISR hand.

Fig. B.11 shows a 3D printed mold used to cast the silicon on a 3D printed endoskeleton. The objective was to offer a compliant finger with a soft contact able to deform into the shape of the graspable object.

The new wire flexure endoskeleton structure offers one degree of freedom more than the previous leafspring structure by releasing in-plane bending. This structure should be less beneficial for the support stiffness than the presented in the ISR Hand.

Recent studies of the University of Coimbra have focused on the understanding of the flexure-based joints [21], several designs are presented in Fig. B.12.

Comparisons between the original curved leafspring design used in the ISR Hand, Fig. B.12a, and several topologies were made. A non-linear Finite Element Analysis
The difference between the simulated and experimental results ranged from 45 - 83%; the experimental deflections were higher in all cases.

While power grasping an object, two forces are produced: a normal contact force and a sideways force product of the weight of the object. Their study [21] focused on the sideways force by comparing deflections: in the actuation direction, in torsion, and in sideways translation.

Although their approach [21] is quite interesting, it only focuses in the undeflected state. The downside is that compliances increase considerably at large deflections, as shown by [78]. This is the reason why Odhner and Dollar from Yale University changed to a pin joint in the MCP [11].

Reported weight: 280 g (incl. motors) [20].

B.3.3 University of Illinois

The University of Illinois presented an open-source, low cost myoelectric prosthetic hand [22], see Fig. B.13.

In their mechanical design, an endoskeleton urethane structure is embedded in a silicon matrix like the one presented by the University of Coimbra [19][20].

Compared to the University of Coimbra design, the material used for the endoskeleton structure is urethane instead of a material with a higher Youngs modulus. Even though a compliance and performance analysis of the fingers/joints was missing, it
can be expected that the fingers have higher compliance in all directions.

**B.4 Other Compliant Mechanisms**


A design methodology to translate from a crossed four-bar mechanism, used in prosthetic hands [37] to a flexible mechanism was presented by the Worcester Polytechnic Institute, see Fig. B.14.

A similar mechanism was conceptually presented by Lotti and Vassura (University of Bologna) [13], where two rigid bodies were kept in contact by a ligament. In this case, two cross-curved leafsprings act as the ligaments.

The trajectory of the mechanism was determined and used to create a contact-aided mechanism. The design presents advantages as reduced compliance in the unwanted directions. However, it lessens some of the advantages of flexure mechanism, like frictionless and backlash after wear.

The rolling contact concept requires a compression force to keep the mechanism in contact. The sideways stiffness is improved by a width mechanism but this is limited for anthropomorphic hands. This mechanism has been proposed for knee joints in which the compression is present when actuated [79].
B.4.2 Technical University of Berlin [5]

The Technical University of Berlin presented a compliant robotic hand [5].

The design consisted of a fully compliant finger actuated by a pneumatic chamber in each of the fingers and also in the palm, see Fig. B.15. It was achieved with a low-weight device (178 g.) capable of reproducing 31 out of 33 grasps from the Feix taxonomy.

The soft hand is very dexterous and was not intended for power grasps. The power grasps is actually listed in the limitations. The authors suggested that if it was necessary to increase the grasp stiffness, it could be done by increasing the stiffness of the actuator.

For a medium-wrap grasp, the load-carrying capacity was not presented. However, it is mentioned that the grasp was stable until 6-8 N of axial force was applied on the cylinder.

B.4.3 Delft University of Technology [6]

An underactuated compliant finger was presented by the Delft University of Technology [6], see Fig. B.17. The objective was to provide an underactuated design adaptable to the shape of the object. Requirements for out-of-plane loads were not found.

It was reported that the joints are able to achieve up to 30° of flexion with distributed contact force, especially in the intermediate and distal phalanges. The ratio between the actuation force and the grasping force (up to 2 N) is linear when each joint is deflected 30°.
The publication focused on the actuation and distribution of the grasping force over the object. A PRBM was used for conceptual to detail design, validation with an FEM commercial package and experiments to measure contact forces. Final weight per finger was 15.1 g.

B.5 Conclusion

A literature review on prosthetic hands and the introduction of compliant mechanisms over the last 20 years is presented. Due to the similitude of the requirements of compliant fingers, some robotic hands were included in the analysis.

- A total of 30 prosthetic hands were registered during the last 20 years.
- Of the 30 prosthetic hands, 20% included some type of compliant mechanism.
- Pseudo-Rigid Body models are commonly used for calculating behavior of flexure hinges. However, 3D analyses are missing.

Compliance analyses are still subject to development for flexure mechanisms in prosthetic hands:

- Recent studies have shown interest in compliance in undesired directions, but they have only been evaluated in the undeflected state.
- Others, have evaluated the compliances in the deflected state, although their optimizing parameter was the compliance in the actuation direction. If the compliance was too high in the unwanted direction, the decision has been to return to a pin joint.
- An optimization of the compliances in the unwanted directions on the deflected state (the real position of the fingers in grasping) has yet not been studied.

B.6 Discussion

Based on the literature findings, it is of interest to study a monolithic flexure-based prosthetic hand that could be 3D printed without assembly, reducing the number of parts and possibly weight of the system.

Researchers who have applied flexure joints on prosthetic hands have overlooked the importance of studying the load-carrying capacity (support stiffnesses) in the full...
range of motion. Understanding the behavior of the flexures will allow manufacturing of a functional monolithic prosthetic hand.

Further studies of different flexure topologies that allow one rotational degree of freedom could lead to improved support stiffnesses.
Table B.1: List of prosthetic and robotic hands (CP, commercial prosthetic; RP, research prosthetic; RH, robot hand), type of joints (NC, non-compliant; C, compliant)

<table>
<thead>
<tr>
<th>Device</th>
<th>Developer</th>
<th>Joints</th>
<th>Hand</th>
<th>No. Joints</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>SDM Hand</td>
<td>Yale University</td>
<td>C</td>
<td>RP</td>
<td>2</td>
<td>Rigid phalanges with urethane leafsprings. 2D compliance matrix in large deflections were considered.</td>
</tr>
<tr>
<td>Worcester Polytechnic Institute</td>
<td></td>
<td>C</td>
<td>RP</td>
<td>1</td>
<td>Rolling contact connected by flexible elements.</td>
</tr>
<tr>
<td>UC Soft Hand</td>
<td>U. of Coimbra</td>
<td>C</td>
<td>RP</td>
<td>2</td>
<td>Tendon actuated hand (280 g) with wire - flexure structure for the joints.</td>
</tr>
<tr>
<td>U. of Illinois</td>
<td></td>
<td>C</td>
<td>RP</td>
<td>2</td>
<td>Urethane skeleton with silicone cover for compliant contact.</td>
</tr>
<tr>
<td>Delft U.T.</td>
<td></td>
<td>C</td>
<td>RP</td>
<td>3</td>
<td>Monolithic finger with compliant actuation and distributed contact force.</td>
</tr>
<tr>
<td>U. of Wollongong</td>
<td></td>
<td>C</td>
<td>RP</td>
<td>3</td>
<td>Notch hinges designs in thermoplastic elastomer material.</td>
</tr>
<tr>
<td>ISR Soft Hand</td>
<td>U. of Coimbra</td>
<td>C</td>
<td>RH</td>
<td>2</td>
<td>Beam equations in undeflected state, combination of leafspring embedded in soft material.</td>
</tr>
<tr>
<td>RBO Hand 2</td>
<td>TU Berlin</td>
<td>C</td>
<td>RH</td>
<td>-</td>
<td>Actuated pneumatic chambers allow finger deformations.</td>
</tr>
<tr>
<td>Huazhong U. and Georgia Tech</td>
<td></td>
<td>C</td>
<td>RH</td>
<td>3</td>
<td>Kinematic analysis of leafsprings.</td>
</tr>
<tr>
<td>UB Hand 3</td>
<td>U. of Bologna</td>
<td>C</td>
<td>RH</td>
<td>3</td>
<td>Analysis of compliance matrices in undeflected state.</td>
</tr>
<tr>
<td>Ohio State University</td>
<td></td>
<td>C</td>
<td>RH</td>
<td>2</td>
<td>Pseudo-Rigid Body Model including elongation to calculate behavior of soft leafsprings.</td>
</tr>
<tr>
<td>Tennessee Tech. U.</td>
<td></td>
<td>C</td>
<td>RH</td>
<td>2</td>
<td>Monolithic 3D-Printed of 2D and 3D flexure designs.</td>
</tr>
<tr>
<td>Vincent Hand</td>
<td>Vincent Systems</td>
<td>NC</td>
<td>CP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>iLimb Pulse</td>
<td>Touch Bionics</td>
<td>NC</td>
<td>CP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>-------------------</td>
<td>---------------------</td>
<td>----</td>
<td>----</td>
<td>-----</td>
<td></td>
</tr>
<tr>
<td>Bebionic</td>
<td>RSL Steeper</td>
<td>NC</td>
<td>CP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Bebionic v2</td>
<td>RSL Steeper</td>
<td>NC</td>
<td>CP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Michelangelo</td>
<td>Otto Bock</td>
<td>NC</td>
<td>CP</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>IH2-Azurra</td>
<td>Prensilia s.r.l.</td>
<td>NC</td>
<td>CP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>UT Hand I</td>
<td>U. of Twente</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>TBM Hand</td>
<td>U. of Toronto</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>MANUS Hand</td>
<td>Spain, Belgium and Israel</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>Smarthand</td>
<td>Scuola Superiore</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>UNB Hand</td>
<td>Uni. Of New Brunswick</td>
<td>NC</td>
<td>RP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Stanford U. School of Medicine</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Worcester Polytechnic Institute</td>
<td>NC</td>
<td>RP</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Delft Hand</td>
<td>Delft U.T.</td>
<td>NC</td>
<td>RP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>ACT Hand</td>
<td>Carnegie Mellon U.</td>
<td>NC</td>
<td>RP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Cyborg Beast</td>
<td>Creighton U.</td>
<td>NC</td>
<td>RP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>SMA Hand</td>
<td>Saarland U.</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>Tact Hand</td>
<td>U. of Illinois</td>
<td>NC</td>
<td>RP</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Remedi</td>
<td>University of Southampton</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>University of Colorado-Denver</td>
<td>NC</td>
<td>RP</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rehand</td>
<td></td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>RIT</td>
<td>Rochester Institute of Technology</td>
<td>NC</td>
<td>RP</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>The University of Electro-Communications</td>
<td>1</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Appendix C

Experimental Setup

C.1 Test rig for sideways stiffness of fingers

C.1.1 Introduction

From a system engineer perspective is necessary to measure the technical performance metrics through the decomposition of the system. At the component level one of the performance metric is the sideways stiffness. It is related to the capacity of a hand to carry weight while performing a power grasp, see Fig. 2 Appendix D. It has been reported in literature as one of the limitations of the flexure-based hands [11].

In principle, a sideways force will have influence in torsion and in-plane bending stiffnesses of the flexure hinge. Both are acting in series, meaning that the weakest will determine the deflection of the tip of the finger.

The Bachelor Assignment of W. Pot was the design of a test rig, section C.1.5. It was intended to measure independently both stiffnesses by constraining all other degrees of freedom. The measurements could give valuable information for the conceptual design of the hinges.

Additionally, the test rig was intended to accommodate fingers from commercial and research hands. Comparison between existing and own designs could show advantages of the PDEng prosthetic fingers. However, due to time constraints the test rig was not completed.

Further in the PDEng project, the requirement of the test rig was switched from an understanding tool to a validation tool. The experimental measurements were used to validate computational models. The initial requirement for Pot of accommodate different fingers without modifying them was removed and the clamping could be improved.

In the presented appendix, further analysis of the existing design and improvements are carried. The main focus is the sideways stiffness measurement.

C.1.2 Measured Stiffness

The stiffness measured in the test rig is a relation between the load applied and the displacement measured by the LVDT sensor. Furthermore the stiffness of the elements of the test rig must be considered to calculate the stiffness of the finger.

From Fig. C.1 it can be observed that the parallel guidance is in parallel with the finger. The stiffness measured by the test rig must be subtracted by the stiffness of the parallel guidance in order to obtain the stiffness of the finger.
\[ K_m = K_{pg} + K_l \]  \hspace{1cm} (C.1)

Where \( K_m \) is the measured stiffness, \( K_{pg} \) the stiffness of the parallel guidance and \( K_l \) is the stiffness of the components in series.

\[ K_l = \left( \frac{1}{K_{clamp}} + \frac{1}{K_{finger}} \right)^{-1} \]  \hspace{1cm} (C.2)

Where \( K_{clamp} \) is the stiffness of the clamping of the finger and \( K_{finger} \) the stiffness of the finger, including the phalange. Fig. [C.4] shows the components stack in series, as the base clamp is fixed to the ground and the finger is loaded at the tip.

**Parallel Guidance**

The stiffness of the parallel guidance was first determined by simulations in a finite element commercial package (FEM orig., Fig. [C.2]) and later measured experimentally (Exp., Fig. [C.2]). It was found, in both cases, linear relation between the load and the displacement. However important differences (87\%) were found in the stiffness results, being the experimental stiffer than the simulations.

The original FEM analysis, Fig. [C.2] considered a 100\% infill (material properties from supplier [80]) of the 3D printed parts while in reality the parts were printed with 25\% infill. Aside from the material properties, a convergence analysis was carried out as results can vary considerable (32\% difference from default mesh), large stress concentration in the notch flexures required a local fine mesh.

Furthermore, measurements on the dimensions of the notches resulted in significant differences from design to the prototype. The original design ratio \( h/D = 0.1 \) changed to \( h/D = 0.17 \), being the biggest difference in the thickness of the notch. The measured dimensions were modified in the simulations file and the results are presented in FEM mod., Fig. [C.2]. Additionally, material properties for 50\% infill were applied [81].
The relative error between the experimental and the simulations is relatively small, 6%. Although, the infill of the parts is 25%, as reported by Pot, in the notch hinges 60% of the thickness have full infill. The experimental stiffness was used in equation C.1 for determining the stiffness of the finger.

Effects such as creep and relaxation of the polymer are expected. However, the stiffness of the parallel guidance was measured with a difference of 3 months and a difference within 1% was found.

### C.1.3 Improvements

The mechanisms designed by Pot to attach the finger, in both ends, are presented in Fig. 7, Section C.1.5 The base clamp and the interface between the finger and the parallel guidance were suggested as improvement points at the end of his report. Failure under high loads and lack of stiffness were found.

The clamping in both ends of the finger relied in friction and clamping power to keep the finger in position. Furthermore, the alignment of the finger was not guaranteed and repetitive measurements were either. A redesign was done considering that additional interfaces could be print in the finger.

**Interface Finger - Parallel guidance**

The concept of the measurement was modified from a constraint stiffness to a sideways stiffness. The constraint stiffness allows only one degree of freedom on the flexure hinge, translation in the actuation direction of the parallel guidance. The sideways stiffness allows torsion and translation of the flexure hinge.

The interface of the tip of the finger with the parallel guidance was changed from a rigid connection (constraint stiffness measurement) to connections through wire flexures (sideways stiffness measurement). Wire flexures constraint one degree of freedom, the elongation stiffness of the wire is considerably higher than other directions.

Two wire flexures were used, one for applying the sideways force and another to deflect the finger in the actuation direction. The position of the tip of the finger is calculated with the computational model and connections are printed to deflect the finger to this position.
As the parallel guidance is loaded, the same force is translated to the finger in a combination of torsion and in-plane bending to the flexure hinge.

Results with the new concept and the original base clamp are presented in Fig. 9 of Appendix D. Differences between the computational models and the experimental results lead to the next improvement in the test rig, the base clamp.

Base clamp

The main consideration for redesign of the base clamp is the alignment of the finger. The largest differences between the models and the experimental results were found in rest position. It is understood that the stiffness of flexures are sensitive to small misalignment. A new concept was developed and it is presented in Fig. C.3.

By attaching the finger with bolts to the base clamp the alignment depends in the manufacturing processes. Besides the stiffness of the base clamp was studied to create a requirement for the test rig.

The stiffness of the base clamp should be higher than 338.95 N/mm, according to equation [C.2] to produce an error below between 1% the measurement and the finger stiffness. The 3D-print PLA considered in original Pot design was studied and compared to a machined aluminum design. The higher elastic modulus of the aluminum will increase the stiffness of the base clamp. By considering the finger to be rigid, it is possible to evaluate the stiffness of the base clamp. The results are presented in Fig. C.4.

A difference of 93% was found between the PLA base clamp and the aluminum version. The stiffness of the PLA base clamp is not enough and affect the measured stiffness by more than 1%. However, it can be subtracted from the measured stiffness. The aluminum version fulfills the required stiffness by a factor of 6.

C.1.4 Conclusions

The concept of the test rig was modified to measure the performance of the finger and validate computational models. The stiffness of the parallel guidance was determined and it is used to be subtracted from the measured stiffness. Modifications of original dimensions of the parallel guidance and consideration on material properties allowed to find a computational model within 6% of error.
The stiffness of all components in series with the finger has been determined to be higher than 338.95 N/mm. A redesign base clamp is presented which takes care of the aligment of the flexure-based hinge. The aluminum version satisfy the stiffness requirements for the test rig.

It is concluded that the biggest influence in the end results found in Fig. 9 of Appendix D are related to the misaligment of the flexure hinge. It can be observed as small variations in deflection can affect the stiffness of the flexure hinges.
ABSTRACT: Prosthetic fingers, fully consisting out flexure hinges are the newest innovation in the world of prosthetics. At the moment these newly developed fingers lack in rotational stiffness, which is needed to grasp relatively medium sized objects. Rotational stiffness depends on resistance for torsion and lateral bending. Since simulations are not able to point out which of the two properties causes this lack of stiffness, testing is needed. Therefore a 3D printed test rig is designed, able to accurately measure both properties. In order to ensure high accuracy, several key parts are explained and verified. Eventually, a test rig is constructed which has most capabilities to perform the different tests on a broad range of fingers.

Key words: Test rig, prosthetic fingers, flexure hinges, 3D printing manufacturing, lateral bending, torsion

1 INTRODUCTION

Prosthetic limbs date back to the Ancient Egypt and Roman times. It took until the sixteenth century for the first hinged prosthetic hand to be introduced by Amorise Par. In the 20th century prosthetic limbs started to become more advanced. Especially with the improvements in 3D printing technologies, the future in prosthetics seems very bright. [1]

The chair of Precision Engineering (PE) at the University of Twente (UT) is very innovative with many new developments in this field. The main goal of their research is to design prosthetic fingers with solely flexure hinges (see figure 1). A flexure hinge moves within the elastic limits of a material and does not consist of multiple different parts like conventional hinges. Reasons to design by using flexure hinges are: excellent repeatable motion (no friction, no backlash, low hysteresis), no maintenance and assembly needed due to their monolithic nature, and a strong reduction in the number of parts, mass and cost.

This paper will be supporting the developments in the design of prosthetic fingers, by enabling validation and testing of new concepts from the UT. The simulated and 3D printed fingers that already have been realised need a test rig to see if their behaviour is as designed. In literature is found that, rotational stiffness was found to be the overall weakest property in many commonly used hand grasps [2] and therefore chosen as property of interest. A confirming example was found in a research by Tavakoli [3]. In figure 2 the fingers rotate and are not able to hold the object in place due to the gravitational force.

Too little rotational stiffness can be due to a lack in resistance for torsion or lateral bending. Next to these properties, flexure hinges are multiple times weaker when they are bend. Since bending is needed to grasp, this is also taken into account. Therefore the function
of the test rig is to test properties for torsion and lateral bending of the fingers in different bend and straight situations. With this demand, it is of importance to have a test rig which can test a broad spread of fingers with different dimensions. The so called main question of this paper is:

How should the optimal test rig be designed in order to validate the 3D printed prosthetic fingers on their rotational properties?

2 METHOD

To test 3D printed prosthetic fingers (or similar behaving specimen) a test rig has to be designed. In order to do so, several sub questions have to be answered:

- What is the best method to test rotational stiffness on test specimen?
- How can different sized specimen be tested on the rig?
- What conditions are of importance to make sure testing is accurate and reproducible?

These questions were taken into account during design process. During this process one cannot foresee all features of a test rig on forehand. Extra sub questions might be added to fulfil the functions of the test rig properly. With the function of being able to test the rotational stiffness of fingers with different dimensions (based on findings in [4], [5], [6] and [7]) comes a List of Demands. This list is divided in four main headings, the tested specimen, the test rig, Verification and Organisational (see Appendix 1).

By studying the sub questions and taking the demands in mind, concepts can be developed. After development, in consultation with the end-user (the UT), the most optimal concept is picked for further development. This contains designing to specifications, tuning dimensions and optimisation to become producible within the facilities of the University’s workshops. The main focus is to have the test rig constructed by the use of the chair’s 3D printer (printer: BCN3D Sigma). This decision is made for its availability within the chair and since a 3D printer can produce small and precise parts, is relatively cheap and its prints are easily reproducible.

After realisation of the test rig, it should be verified on accuracy and be proven to work. Verification will be done by finding its inaccuracies in terms of friction, stiffness, strength and its degree of agreement to the List of Demands. During these validation the List of Demands is the guiding principle to determine if a validation has succeeded or not. Then, the test rig will be compared to a specimen, whereof the properties easily can be calculated (or are known). If the test rig is validated properly, the results of testing on this specimen will be compared to its properties. If the result corresponds to the expected values the main question is answered and an optimal test rig is delivered. This specimen will be used as a benchmark, to check the rig on accuracy after realisation. Reasons to check the test rig are long periods without being used, or damage. With the benchmark the user will be able to re-calibrate the test rig.

3 DESIGN AND CONSTRUCTION

Fingers can be approximated as multiple beam elements when testing their stiffness. When these fingers are bend, the beams create independent directions of deformation. Since the bending behaviour of the different phalanges is not collinear, the test rig has distinguish pure bending and pure rotation. In this way the quantities of stiffness due to torsion and lateral bending can be found. For that reason the test rig will be testing these properties in separate situations. Eventually when testing a finger on both testing principles, the user can find which of the two properties that are part of the rotational stiffness is weakest. In both cases, the applied force will be of vertical direction, since it is due to the gravitational force of a grasped mass. Two separate situations ask for two different designs of test rigs, these are explained in the next section.

3.1 Global design

Global design rules that apply on both test rigs are explained before zooming in to the separate testing principles. Besides these two principles the following parts are explained in separate sections: clamps, measurement tools and limitations in construction. In figure 3 the full test rig is seen, which gives an overview of all components.

As mentioned before, mostly all parts of the test rig will be 3D printed. The printed material is be polylactic acid (PLA). This material has been chosen since this plastic is the easiest and most precise to print.
with nowadays. The assumed properties of the PLA are found in the List of Demands (see Appendix 1).

However, since 3D printing might cause weaker properties than the base material, a safety factor of three is used in calculations for designing plastic parts. Most parts are printed in black, the parts that are important for testing are printed orange to make it more user-friendly.

A support frame is needed to construct both test principles on. A widely used aluminium profile called F-40*40 Blocan® is used. This material can be assumed to be rigid, compared to the material properties of the plastic components. Other advantages of the profile are its versatility, being connectible on all sides with fixing slots for m6 bolts.

All other connections of the test rig made with m5 threaded inserts, whose need a material thickness of 12.2 mm. This is mostly the overall thickness used in the components, unless no inserts were needed or Finite Element Method (FEM) asked for thicker parts due to weaknesses.

3.1.a Torsion case

For the torsion case it is important that the finger will only be able to resist against the applied force in the rotating direction around the longitudinal axis. There exists a wide variety in torsion testing machines [8], but mainly there are two reasons why these are not useful for this research. Firstly, the amount of torsion that these machines can apply is way above the regime that is of interest to 3D printed fingers. Secondly, these conventional machines are only able to test straight profiles and pipes. For this research it is important to test from straight to a range of different bended situations.

As mentioned, to design a test rig for torsion, it is of importance that specimen can only be tested in rotational direction. Therefore all other degrees of freedom should be constraint. Next to that the specimen should be perfectly aligned since it will be clamped at both ends. If not, the test rig will not purely test on torsion. At one end the specimen will be fixed, at the other end the specimen should be able to rotate, within every kind of bended situation. Therefore a disc is chosen to attach the finger to, where an moveable clamp should be used to created different angles of bending. The angle is created by moving the tip clamp in normal direction out of the middle of the disc over the two slots. This can be seen in figure 4.

Fig. 4: CAD model of the torsion case
To be able to test specimen with different lengths, the distance between the clamp and the disc is adjustable. To be able to test specimen with different diameters, the height of the disc is also adjustable, since the longitudinal axis of the specimen is depended on its thickness. In order to have as little friction as possible a ball-bearing is used for rotation. In general the friction coefficient (μ) of a ball bearings is between 0.001 and 0.0015, which is lower than a roller or needle bearing. Since the radial forces are very little, a ball bearing is chosen. The bearing goes in the bearing house (see figure 4 which is connected to the sup-
To apply a moment on the tip of the specimen, which generates the torsion, masses are hung on the holders that can be seen in the figure 4. There is a holder on both sides of the disc to be able to balance the test rig with contra weights.

3.1. b Lateral bending case

Fig. 5: CAD model of the lateral bending case

For the lateral bending case, the focus is to purely move one end of the specimen in vertical direction and to fix the other end. Also for this testing principle there are many machines on the market [9], but these machines also do not fulfil the demands due to the same arguments as for the torsion case. The notch flexure that is used is the optimal way to get this single direction movement with the least friction (see figure 5). Hereby should be noted that the deflection in vertical direction is small enough to neglect the shortening effect in horizontal direction (for further explanation about the shortening effect and why this can be neglected, see Herman Soemers (2011, para. 3.3.1) Design principles of precision mechanisms [10]. The force that is needed to make this displacement is calculated to be within the region of maximum applied force used during testing. This amount has to be validated afterwards to obtain a factor for the force that goes into shifting the flexure itself in vertical direction. Furthermore, the force will be applied in the center between the two notch flexure to keep the test rig balanced, a holder similar to the torsion case is placed on top of the flexure. Other options next to a flexure mechanism, like a slot with a guider or linear guidance were not chosen, respectively due to problems with friction and availability.

To compensate for the dead weight of the notch flexure and the clamp, a spring is attached between the weight holder of the flexure and the support frame. At this location the forces of the weights and the spring are working on the same axis (see figure 3 number eight and ten. The spring is chosen by calculating the spring constant using Hooke’s Law (Beuche, p. 95 [11]) since the mass, and therefore gravitational force of the dead weight is known. Equal to the torsion test rig, the height of the tip clamp is adjustable in vertical and horizontal direction by several slots as seen in figure 5.

Fig. 6: Parallel guiding notch flexure mechanism

One of the two notch flexure is seen in figure 5, used to enable the vertical displacement. According to the specifications the maximum needed deflection ($\Delta$) is 5 mm. As mentioned in section 3.1 Global design, a thickness ($t$) of 12.2 mm is needed for the tread inserts. For $D$ to $h$, a ratio of 0.1 is ideal [10]. Then, the applied force for a certain deflection is the least when dimension $p$ is taken the longest. Due to limitations with the 3D printing bed it is taken as 140 mm. With these input values, the method for calculating notch flexures in Herman Soemers (2011, section 3.7) Design principles of precision mechanisms [10] is followed.

By iterating the parameters with FEM calculation (see Appendix 2), optimal dimensions for $D$ and $h$ are found, 15 and 1.5 mm respectively. A maximum tension of 22.9 MPa was obtained. That is more then three times smaller then the Yield Strength, which agrees to the safety limit set in the List of Demands.

3.2 Clamps

The connecting part between the two test principles and the specimen are the clamps. Since there were no clamps available fitted perfectly within the total design, it was chosen to design and develop new clamps. The clamps are designed to fit on both test rigs and deal with many different sized specimen. After many different concepts the final designed clamp is able to properly clamp many 3D printed specimen that are mirror symmetrical in longitudinal direction (see fig-
As stated in the List of Demands, this is done without modifying to the specimen. This is done by two 120 degrees rectangular shaped clamp plates that are forces together. This ensures the specimen to be clamped in the exact middle, which is needed to apply a constraint force or moment as described in section 3.1. The top layer of the clamp is of natural rubber to avoid slip when the specimen are tested.

In the test rig, two versions of this type of clamps are used, a tip clamp and a base clamp for both sides of the specimen. The base clamp has got two holes for clamping the specimen, each hole is used for correct alignment with its own test case. This is done to keep the test rig as compact as possible and have the two test cases on a single support frame. FEM calculations have been done on the base clamp (see Appendix 2). A maximum displacement less then 0.042 mm and a maximum von Mises stress of 1.75 MPa satisfy the List of Demands.

The tip clamp is attached to one of the two test principles. It consist of to bars, the clamp itself and a plate. The plate ensures correct positioning of the specimen by locking the end in longitudinal direction. The plate is also used to enable the clamp to fit on both test cases. To enable testing with the specimen under different angles, many clamps come with the test rig. To ensure similar testing conditions all clamps are designed to have the same weight. This is easily done by designing within geometrical relations to keep the same volume when modifying angles. Also, by 3D printing it is relatively cheap to produce several clamps. This directly is a Poka-Yoke solution for inaccurate testing under bending angles.

### 3.3 Measurement tools

Several measurement tools have been used for verification and have to be used by the user of the test rig. This section briefly describes them.

#### 3.3.a Linear Variable Displacement Transformers

To be able to measure the displacements of the specimen during testing Linear Variable Displacement Transformers (lvdt’s) are chosen. Compared to other options, these tools give far more accurate measurements than analogue measuring tools. Other digital measurement tools were not chosen, due to accuracy, availability and cost. The lvdt’s measure up to 0.001 mm, have a range of 10 mm and can be clamped to make sure they are always in the same position. The measurement tools are attached to the support frame with clamps that are similarly designed as the clamps for the specimen. In between the clamps is a holder which has 10 slots for adjustment of the lvdt in height for accurate placement. The assembly of the lvdt clamps and holder can be seen in figure 8.

#### 3.3.b Lateral bending case alignment tool

To be able to put the lateral bending case in its neutral position (5 mm above its maximum deflection point) a steel tool is used. This tool cut from a cold deformed square steel bar. To align the lateral bending case the tool is put in the horizontal slot of both notch flexures.

#### 3.3.c Torsion case alignment tool

For adjusting the disc of the torsion case a tool is developed as seen in figure 9. This cone shaped tool has grooves which indicate the radial thickness of the cone by steps of 1mm. When the tool is attached to the bearing, the bearing house can be shifted vertically. When the base clamp (described in section 3.2) is put underneath, the correct height can be set by placing
the tool on the clamp at the indent that resembles the thickness of the specimen that is to be tested.

Fig. 9: CAD model of Torsion case alignment tool

3.3.d Digital slide gauge
For adjustments of all other components of the test rig a digital slide gauge will be used. This mostly comes with adjusting the test rig for different fingers or replacing a tip clamp. For ease of use, measuring tape is added to the support frame for rough adjustments.

3.3.e Test Weights
To be able to measure accurately on the test rig, weights of equal mass (100 gram) are designed. By taking the maximum thickness for laser cutting (3mm), small and identical discs can be produced very accurately. The discs have a slot which easily slides around the weight holder of the test rig.

3.4 Limitations in Construction
During the development and construction phase, some important borders formed which did not fully support the initial designs. Many of these problems have to do with 3D printing, in this section they mayor issues are explained.

Dependability of quality, the parts are very depended of correct settings of printing parameters. PLA is the material where of generally most parameters are precisely known. However, even when printing the same part twice with the same settings, the quality of the parts can differ (a lot). In the worst case, this means that part had to be reprinted.

Shrinkage, outer diameters and dimensions mostly are as modelled with CAD software. Inner diameters and dimensions mostly are smaller then modelled. This requires finishing treatments, which slows down the productivity at the cost of more man-hours. More importantly, this also reduces the precision of printed parts. In the design all screw holes are scaled up to be able to print the right diameters. Also, the notches of the lateral bending case are calculated on a bigger deflection than enabled by the design, to deal with the printing error.

Production time, next to more man-hours, printing time increases a lot as well when printing bigger parts. The lateral bending case consist out of multiple parts due to the fact that printing times would exceed the limit of 48 hours. For this reason it is under a lot of internal tension, since the 3D printed parts do not perfectly fit as modelled. Furthermore, most parts are only 20% filled to increase printing speed, with the negative effect of uncertainty about the FEM calculations. This is taken into account by designing with a safety factor of 3.

Other problems were limitations due to availability, money and time.

4 VERIFICATION OF TEST RIG

In order to verify if the test rig operates as designed for and to find its accuracy several test have been executed as described in the Method. To enable verification various parts of importance have to be measured to ensure a basis accuracy.

The designed weights are measured with 0.01 grams accuracy. The average weight is 98.7 grams, with a maximum deviation of 0.2 grams, therefore they can be assumed of similar weight in further tests. Other weights that have been used are certified measurement tools.

The designed torsion case alignment tool is remeasured with 0.01 mm accuracy with an external micrometer with v anvil. Due to 3D printing errors the tool generally became 0.25 mm thicker in diameter. With this error included, the maximum deviation between the designed thickness (plus 0.25 mm) and the measured thickness is 0.08 mm. Therefore the steps between the indents are 1 mm $\pm$ 0.08 mm, from a 7.25 mm to a 14.25 mm radius. The distance between the indents (longitudinal axis) are 2 mm with a maximum deviation of 0.1 mm.

4.1 Verification of springs
As mentioned in the design (section 3.1a), the lateral bending case is designed with a spring to compensate for its dead weight and the executed force of the lvdt in neutral position. This spring was calculated to have a spring rate of 0.2 Newton/mm. The spring rate was verified by doing a linear experiment
by step-wise adding 100 grams of mass. A spring rate of 0.1996 N/mm was found, by fitting a linear plot with a correlation coefficient of 0.9996 (very strong correlation). Also the spring rate of the lvdt had to be determined, in order to find the balance point where the notch flexure is in neutral position. This was done by measuring the amount of force the lvdt executed on a digital scale of 0.01 grams accuracy. In advance can be said that a scale is not ideal for point contact measurements. To make this test significant enough 24 samples were taken, which resulted in a spring rate of 0.05 N/mm, with a correlation coefficient of 0.941 (also a very strong correlation. These results are seen in figure 10.

![Fig. 10: Determination of the linear correlation for the spring rate of the lvdt](image1.png)

4.2 Verification of stiffness in Lateral bending case

With all springs verified, the test rig itself can be tested. To start testing, the notch flexure was put in neutral position by the use of the 5mm rod, described in section 3.3 Measurement tools. With this in place the spring and lvdt are adjusted to their calculated positions based on the verified spring rates. The vertical position of the notch flexure changed marginally when the rod is manually taken out of the test rig. This change in position could not be prevented. Due to very little internal friction the flexure does not get back to its calculated neutral position. This phenomenon is called hysteresis. Five tests have been carried out on the deflection of the lateral bending case, with the use of the disc weights. These weights have been applied on the test rig in specific order due to their small differences in weight (to make the initial errors of the different tests comparable). The difference in between the five tests is the method of positioning the weights. By using different methods the maximum inaccuracy due to hysteresis of the lvdt is verified. Test is done as following: After the notch flexure is put in neutral position, on of the methods of positioning is applied: 1) subtle placement, 2) placement and shifting the point of the lvdt above the contact, 3) placement and manually shifting the flexure to the lowest position, 4) placement and manually shifting the flexure to the highest position, 5) placement and manually shifting the flexure to the neutral position. After the positioning action, the rig was left loose to settle in its new balanced position. The average value over the five experiments is plotted in figure 11.

![Fig. 11: Deflection of the lateral bending case by adding weights, fit to a very strong linear correlation](image2.png)

The error bars indicate the error over all five samples at a certain point, where a standard deviation of 0.087 mm is found. After sample six the measurements are becoming inaccurate because the limit of the notch flexure is reached. The first 6 measurements have a strong linear correlation. By extrapolating this linear fit, the deflection of each weight is found in figure 11. With the deflections known, the relative deflection of the test rig can be calculated by subtracting the neutral position (at zero weights) from each value. This relative displacement is of importance when measuring specimen, since the weight of the set-up itself helps to deflect the specimen that are to be tested. The factors with each weight are found in table 1. Furthermore the rate of deflection in the notch flexure, including the spring and lvdt, can be calculated. The constant \( k \) at which the flexure flexes is found with Hooke’s law: \( F = -kX \) and is calculated at 0.75 N/mm. This is of use when other weights then the 100 gram disc designed for the test rig are used. One should pay attention to stay within the elastic deformation regime when applying weights bigger then...
Table 1: Relative deflection of the Lateral bending case by adding weights

<table>
<thead>
<tr>
<th>No. of weights</th>
<th>Total mass [g]</th>
<th>Deflection [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>98.7</td>
<td>0.7174</td>
</tr>
<tr>
<td>2</td>
<td>197.2</td>
<td>1.4846</td>
</tr>
<tr>
<td>3</td>
<td>295.5</td>
<td>2.2366</td>
</tr>
<tr>
<td>4</td>
<td>394.2</td>
<td>3.0112</td>
</tr>
<tr>
<td>5</td>
<td>492.8</td>
<td>3.7318</td>
</tr>
<tr>
<td>6</td>
<td>591.5</td>
<td>4.4862</td>
</tr>
<tr>
<td>7</td>
<td>690.1</td>
<td>5.2736</td>
</tr>
<tr>
<td>8</td>
<td>788.8</td>
<td>6.0236</td>
</tr>
<tr>
<td>9</td>
<td>887.5</td>
<td>6.7736</td>
</tr>
<tr>
<td>10</td>
<td>985.8</td>
<td>7.5236</td>
</tr>
</tbody>
</table>

tested. Also, the test rig has not been calculated on higher loads, therefore this is not recommended.

4.3 Verification with maximum load

To verify if the test rig is able to perform under the most extreme conditions due to the List of Demands, several test have been carried out. First the base clamp was tested on bending. This was test by the use of a rigid steel rod as a specimen of maximum specified length. A maximum load of one kilogram was used. In this case the lvdt was used to measure the deflection on the other side as the applied force. The steel rod was attached in the lower hole of the base clamp, since the higher hole is used for the torsion case. Therefore there will not be a vertical force acting on the specimen when it is place in the higher hole of the base clamp. The measured deflection is 0.135 mm, which is more than calculated and also more than allowable to ensure accurate testing. A note to this result is that the lvdt, as mentioned before, puts a force on the clamp itself. Since the maximum force exerted by the lvdt is in the regime of 2.5 N (250 grams as seen in figure 10), this excessive deflection cannot fully be caused by the lvdt’s force.

Secondly the base clamp was tested by applying a moment. The lvdt was shifted to the side of the base clamp to see the amount of bending in sideways direction. While performing more tests on the base it became clear that the steel test specimen began to slip. Therefore no accurate test results were found. After the specimen was taken out of the clamp the rubber looked damaged and to be moved as seen in picture 12. Due to limitations in time, no other test have been carried out.

4.4 Verification of stiffness in Torsion case

In the torsion case there are initially no springs to be verified. The bearing which enables the rotation has a static friction coefficient which is small enough to be neglected. But the lvdt presses on the disc while measuring and in its neutral position. The lvdt has already been verified in terms of excitation of force in the section 4.2. For a minimal excitation of the lvdt’s tip, a force of 2.16 Newton (220 gram) is applied. The excitation should be as little as possible, because the majority of the lvdt’s range is needed to measure the angle of rotation. To apply 2.16 Newton, the monitor of the lvdt should measure 9.0163 mm. With the use of a digital slide gauge the disc was kept in balance, to be able to install the lvdt on the calculated height. The disc stayed in balance with 220 grams of contra weights added to the opposite side of the lvdt. Thus the balance was found.

4.5 Verification of Deflection regime

Next to all mechanical verifications, the test rig has to be tested for different sized fingers. Within the design phase, the range of length and thickness has been taken into account. After realisation this was tested with different specimen. For clamping, thickness of 15 to 28 mm fit in the clamp which is stated in the List of Demands. In the total test rig is enough space for fingers between 60 and 120mm, from tip to base. There are issues found for dimensioning, which where not specified in the List of Demands. Namely, with small fingers the phalanges proportionally decrease to a size that does not fit under both braces of the tip clamp. Therefore the grip on the fingers will be decreased. This was not tested due to damaged clamps as described in section 4.4.
Looking back at the sub questions listed in the method a conclusion is drawn to this research. In the early phase of this research was found that the rotational stiffness is most accurately tested by making a distinction between lateral bending and torsion. For testing these two properties a broad variety of test rigs exist, whose working principles were used as inspiration for designing. In the designs of both cases is ensured that all DOF’s are constraint, except the DOF in the direction of testing, to find pure properties for torsion and lateral bending. This approach answers to the first sub question.

The designed clamps were able to constrain the broad spread of dimensions for the specimen without modifications. The two test cases are fully adaptable to align fingers of different dimensions under different bend or straight situations. Therefore, also the second sub question is fulfilled.

The third sub question is taken into account during the design phase, by setting limitations to maximum stresses and deflections. The different parts have been calculated with multiple FEM analyses and have been approved within the set demands. However, during verifications became clear that some parts deflected more than calculated, which make the accuracy of the test rig less then intended. The base clamp is the part that deviates the most with 0.135 mm (almost three times as much a demanded). Since the clamps failed to correctly clamp the specimen without slip, there could no further verification be done. This concludes that testing with the test rig, will be denoted as not reproducible within the demanded margins. Therefore, no satisfying answer is found to the main question. The test rig that is designed and constructed is not able to determine the rotational stiffness of different sized specimen in different situations. However, this first research is a big step into the design for an optimal test rig, since: The measurement tools are successfully verified and able to be used for testing. The spring constants of the various springs in the test rig are determined. The lateral bending case has been fully verified, the amount of error due to hysteresis is found to be smaller than demanded. The bending factors of deflections due to the dead weight of the notch flexure are calculated and a linear flexure rate is found.

5.1 future development

For future development of the test rig, the first look should go to the clamp and the natural rubber layer, since these should properly clamp the specimen. Although it could also be the steel specimen, which surface might be too smooth to be a good specimen for this verification. At this moment both possibilities should be investigated, which might lead to redesign of the clamps. When the problem is solved, the base clamp should be retested to find if this part is the source of the inaccurate testing conditions.

Currently, within the Chair of Precision Engineering, a research is carried out on the properties of 3D printed PLA. The results of this research might clarify the huge deformation measured in section 4.4 on the base clamp. In that case, all other important parts have to be recalculated with new material properties as well.

Overall, manufacturing by 3D printing has two sides. It comes with a lot of benefits, but also limitations were found (as described in section 3.4). After finding a strong decrease in strength for parts (as for example with the base clamp) another consideration for future development is to use a different production technique and therefore a different material (for some important parts). Since the primal demand of a test rig is to produce accurate results, a fully 3D printed test rig might not be able to fulfill such demand.

ACKNOWLEDGEMENTS

I would like to thank a couple of people, without them I would have never been able to construct this research. At first I want to express my gratitude to my supervisors prof. dr. ir. D. M. Brouwer and L.A. Garcia Rodriguez of the Chair Precision Engineering, University of Twente, who offered me this research. At this moment both possibilities have never been able to construct this research. At first I want to express my gratitude to my supervisors prof. dr. ir. D. M. Brouwer and L.A. Garcia Rodriguez of the Chair Precision Engineering, University of Twente, who offered me this research.

REFERENCES

3. Mahmoud Tavakoli, Andriy Sayuk, João Lourenço, and Pedro Neto. Anthropomorphic finger for grasping applica-
6 APPENDICES

Appendix 1: List of Demands

<table>
<thead>
<tr>
<th>Main Headings</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Test rig</strong></td>
<td></td>
</tr>
<tr>
<td>Material</td>
<td>Polylactic Acid (PLA)</td>
</tr>
<tr>
<td>Manufacturing</td>
<td>3D printing</td>
</tr>
<tr>
<td>Elastic modulus</td>
<td>3.5 GPa</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.36</td>
</tr>
<tr>
<td>Shear modulus</td>
<td>318.9 MPa</td>
</tr>
<tr>
<td>Mass Density</td>
<td>1251.5 kg/m³</td>
</tr>
<tr>
<td>Tensile strength</td>
<td>30 MPa</td>
</tr>
<tr>
<td>Yield strength</td>
<td>70 MPa</td>
</tr>
<tr>
<td>Safety factor</td>
<td>3</td>
</tr>
<tr>
<td>Dimensions</td>
<td>Printable within print margins: 200<em>200</em>300 mm (due to limitations in software, margins are reduced with 10mm) and within 48 hours</td>
</tr>
<tr>
<td>Applicable force</td>
<td>10 N total (steps of 1N)</td>
</tr>
<tr>
<td>Total weight</td>
<td>Maximum 23 kilogram</td>
</tr>
<tr>
<td>Support frame</td>
<td>F-40*40 Blocan® profile</td>
</tr>
<tr>
<td>Maintenance</td>
<td>little to none</td>
</tr>
</tbody>
</table>

**Specimen**

<table>
<thead>
<tr>
<th>Dimensions</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>60 to 120 mm</td>
</tr>
<tr>
<td>Thickness</td>
<td>15 to 28 mm</td>
</tr>
<tr>
<td>Bending angle’s</td>
<td>0, 5, 10, 15, 20, .30 ,45, 60, 90 degrees</td>
</tr>
<tr>
<td>Lateral bending</td>
<td></td>
</tr>
<tr>
<td>Deflection range</td>
<td>5 mm</td>
</tr>
<tr>
<td>DOFs</td>
<td>1 (translation in vertical direction)</td>
</tr>
</tbody>
</table>

**Torsion**

<table>
<thead>
<tr>
<th>Rotation range</th>
<th>5 degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>DOFs</td>
<td>1 (rotation around longitudinal axis)</td>
</tr>
</tbody>
</table>

**Base clamp**

<table>
<thead>
<tr>
<th>Modifications</th>
<th>None</th>
</tr>
</thead>
</table>

**Verification**

<table>
<thead>
<tr>
<th>Stresses</th>
<th>Maximum 23.3 MPa (von Mises)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deflection</td>
<td>Maximum 1% of Deflection range</td>
</tr>
</tbody>
</table>

**Organisational**

<table>
<thead>
<tr>
<th>Budget for parts</th>
<th>None to relative low cost for crucial parts. Printing material is provided</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time schedule</td>
<td>8 weeks</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Applied conditions</th>
<th>for static nodal FEM analysis on all parts:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Connections</td>
<td>Global contact between parts</td>
</tr>
<tr>
<td>Mesh</td>
<td>Fine auto-mesh</td>
</tr>
<tr>
<td>Material</td>
<td>PLA, as stated in the List of Demands</td>
</tr>
<tr>
<td>Stress</td>
<td>Max. 23.3 MPa (von Mises)</td>
</tr>
<tr>
<td>Displacement</td>
<td>Max. 0.05 mm</td>
</tr>
</tbody>
</table>

**Base clamp**

<table>
<thead>
<tr>
<th>Connections</th>
<th>Fixed at the four holes for bolts.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Applied forces</td>
<td>Vertical (gravitational) force at hole for weight holder of 10 N.</td>
</tr>
<tr>
<td>Mesh</td>
<td>Extra refined at notches.</td>
</tr>
<tr>
<td>Results</td>
<td>Max. von Mises stress in base clamp: 1.75 MPa.</td>
</tr>
<tr>
<td>Notes</td>
<td>Max. deflection in base clamp: 0.04 mm.</td>
</tr>
</tbody>
</table>

**Notch flexure**

<table>
<thead>
<tr>
<th>Connections</th>
<th>Fixed at the two slots at the bottom.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Applied force</td>
<td>Vertical (gravitational) force at hole for weight holder of 10 N.</td>
</tr>
<tr>
<td>Mesh</td>
<td>Extra refined at notches.</td>
</tr>
<tr>
<td>Results</td>
<td>Max. von Mises stress in flexure: 22.9 MPa.</td>
</tr>
<tr>
<td>Notes</td>
<td>Max. deflection in flexure: 6.14 mm.</td>
</tr>
</tbody>
</table>

**Torsion case**

<table>
<thead>
<tr>
<th>Connections</th>
<th>Fixed in end of the two slots, for the most extreme situation.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Applied forces</td>
<td>Vertical (gravitational) force at hole for weight holder of 10 N.</td>
</tr>
<tr>
<td>Results</td>
<td>Max. von Mises stress in disc: 0.66 MPa.</td>
</tr>
<tr>
<td>Notes</td>
<td>Max. deflection in disc: 0.005 mm.</td>
</tr>
</tbody>
</table>

Table 2: List of Demands

Appendix 2: FEM calculations

**General**

Applied conditions for static nodal FEM analysis on all parts:

<table>
<thead>
<tr>
<th>Connections</th>
<th>Fixed at the four holes for bolts.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Applied forces</td>
<td>Vertical (gravitational) force at hole for weight holder of 10 N.</td>
</tr>
<tr>
<td>Results</td>
<td>Max. von Mises stress in base clamp: 1.75 MPa.</td>
</tr>
<tr>
<td>Notes</td>
<td>Max. deflection in base clamp: 0.04 mm.</td>
</tr>
</tbody>
</table>

**Notch flexure**

<table>
<thead>
<tr>
<th>Connections</th>
<th>Fixed at the two slots at the bottom.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Applied force</td>
<td>Vertical (gravitational) force at hole for weight holder of 10 N.</td>
</tr>
<tr>
<td>Mesh</td>
<td>Extra refined at notches.</td>
</tr>
<tr>
<td>Results</td>
<td>Max. von Mises stress in flexure: 22.9 MPa.</td>
</tr>
<tr>
<td>Notes</td>
<td>Max. deflection in flexure: 6.14 mm.</td>
</tr>
</tbody>
</table>

**Torsion case**

<table>
<thead>
<tr>
<th>Connections</th>
<th>Fixed in end of the two slots, for the most extreme situation.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Applied forces</td>
<td>Vertical (gravitational) force at hole for weight holder of 10 N.</td>
</tr>
<tr>
<td>Results</td>
<td>Max. von Mises stress in disc: 0.66 MPa.</td>
</tr>
<tr>
<td>Notes</td>
<td>Max. deflection in disc: 0.005 mm.</td>
</tr>
</tbody>
</table>
Appendix D

Force-based flexure hinge optimization

D.1 Abstract

Flexure-based finger joints for prosthetic hands have been studied, but until now they lack stiffness and load bearing capacity. In this paper we present a design which combines large range of motion, stiffness and load bearing capacity, with an overload protection mechanism. Several planar and non-planar hinge topologies are studied to determine load capacity over the range of motion. Optimized topologies are compared, in 30 degrees deflected state, in terms of stresses by deflection and grasping forces. Additionally, support stiffnesses were computed for all hinges in the whole range of motion (45 degrees). The Hole Cross Hinge presented the best performance over the range of motion with a grasping force up to 15 N while deflected 30 degrees. A new concept, the Angle Three-Flexure Cross Hinge, provides outstanding performance for deflections from 17.5 up to 30 degrees with a 20 N maximum grasping force when fully deflected. Experimental verification of the support stiffness over the range of motion shows some additional compliances, but the stiffness trend of the printed hinge is in line with the model. The presented joints power grasping capability outperform current state flexure-base hands and are comparable to commercial non-flexure-based prosthetic hands. In the event of excessive loads, an overload protection mechanism is in place to protect the flexure-hinges.

This chapter has been submitted for possible publication as:

3D-printed flexure-based finger joints for anthropomorphic hands*

L. Garcia, M. Naves and D.M. Brouwer
Precision Engineering
Faculty of Engineering Technology
University of Twente
7500 AE Enschede, The Netherlands
d.m.brouwer@utwente.nl

Abstract—Flexure-based finger joints for prosthetic hands have been studied, but until now they lack stiffness and load bearing capacity. In this paper we present a design which combines large range of motion, stiffness and load bearing capacity, with an overload protection mechanism. Several planar and non-planar hinge topologies are studied to determine load capacity over the range of motion. Optimized topologies are compared, in 30 degrees deflected state, in terms of stresses by deflection and grasping forces. Additionally, support stiffnesses were computed for all hinges in the whole range of motion (45 degrees). The Hole Cross Hinge presented the best performance over the range of motion with a grasping force up to 15 N while deflected 30 degrees. A new concept, the Angle Three- Flexure Cross Hinge, provides outstanding performance for deflections from 17.5 up to 30 degrees with a 20 N maximum grasping force when fully deflected. Experimental verification of the support stiffness over the range of motion shows some additional compliances, but the stiffness trend of the printed hinge is in line with the model. The presented joints power grasping capability outperform current state flexure-base hands and are comparable to commercial non-flexure-based prosthetic hands. In the event of excessive loads, an overload protection mechanism is in place to protect the flexure-joints.

Index Terms—Compliant joints, flexures, robotic hand, prosthetic hand, anthropomorphic, additive manufacturing.

NOMENCLATURE

MCP Metacarpophalangeal.
ROM Range of motion.
E Young’s modulus.
G Shear modulus.
SLS Selective laser sintering

I. INTRODUCTION

Flexure joints applied in prosthetic and robotic hands have been of interest in recent years [1]–[4]. Some of the advantages of an integrated flexure design are more stable grasps and a reduced number of parts [3]–[5]. Furthermore, when 3D-printing technology is used to manufacture a prosthetic hand as a single monolithic structure, absence of assembly can be achieved reducing overall costs.

A major challenge for flexure joints in large range of motion applications is the strong decrease of support stiffness in load carrying directions when deflected [6]–[8]. This loss of support stiffness for large range of motions led to reconsideration of flexures in the MCP joint and the accompanying poor load carrying capacity currently prevents widespread applicability in robotic and prosthetic hands [3]. Therefore, it is of interest to study the mechanical behavior of monolithic integrated flexure joint designs over the whole range of motion. The decrement of the stiffness in the support direction also leads to loss of load bearing capacity of the hand. Especially when including tendon actuation and high grasping forces, elastic instability of the joint (buckling) can result in reduced load-carrying capacity.

Researchers from the UB Hand compared several flexure topologies for robotic hands by analyzing compliance matrices in undeflected position [4], [9]. Additionally, Tavakoli et al. presented new topologies and analyzed the flexure stresses and deflections for the undeflected state [1]. Although analyzing the stiffness properties of flexure topologies at undeflected state allows the use of simple linear beam equations, it gives no lead to the stiffness properties at larger deflection angles due to the strong non-linear behavior. Furthermore, as critical stiffness and load typically occurs at maximum deflection angle, stiffness at maximum deflection angle is of primary interest rather than at the undeflected state.

Kalpathy used a pseudo-rigid-body model with an approximation of Timoshenko beam theory to model leafsprings in a larger range of motion [2]. Although pseudo rigid-body modeling allows for larger deflections, it is limited to simulation of its kinematic behavior and stiffness in the free motion direction. Therefore, evaluation of the support stiffness at large deflection angles is still unavailable.

Odhner presented the “Smooth Curvature model” to calculate compliance matrices in large deflections of planar leafspring designs, as this can be associated with stable grasps [10]. This method allows for evaluation of support stiffness at larger deflections, however, it described the compliance matrix only for the 2-dimensional case. For typical loading-conditions, out-of-plane stiffness and load carrying capacity are important also. Furthermore, it only allows for the evaluation of non-planar hinge designs.

In this paper, we exploit a flexible multibody method to calculate and optimize several flexure hinge topologies, including non-planar topologies, during a cylindrical medium.
power wrap (Fig. 2). This power grasp is identified as one of the most common used grasps [11], [12] and therefore the main focus of this research. First, we developed an optimization strategy to minimize stresses and maximize grasping force for each topology in deflected state. Secondly, several joints are presented and the optimized topologies are compared. The comparison is based on stresses due to grasping force and sideways loads. Furthermore, a comparison of the support stiffnesses over the whole range of motion for the different topologies is done. Third, an overload protection mechanism for the sideways force is presented. An FEM analysis is used to obtain the stiffness of the entire finger, which is subsequently corroborated with measurements.

II. DESIGN METHODOLOGY

A. Optimization loadcase

A finger is designed to be in a rest position that allows 15° of passive extension (ROMpas) and −30° of active flexion (ROMact), Fig. 1. This range of motion allows to grasp objects in the medium wrap range.

Since the fingers have high compliance for rotations around the z-axis, the passive extension is achieved by contact with an object. The contact will open the hand to allow bigger objects to be grasped. The extension is actuated by a tendon force \( F_{act} \) which deflects the flexure up to −30° around the z-axis.

The MCP has been identified as the critical joint [3]. When holding an object the contact force and weight of an object result in a combination of in- and out-of-plane bending loads of the flexure elements, Fig. 2.

Since it is of interest to study the functionality of hands while power grasping, a contact point common to all hinge topologies is defined (Fig. 2). By doing so, there is a similar effect of the loads on the hinges and the shape of the finger is independent of the size of the hinge.

For the optimization, the tendon is actuated to position the finger at −30° of rotation. In the same plane a normal contact force, \( F_{grasp} = 5 \) N, is applied in the contact point. While the contact force \( F_{grasp} \) attempts to open the hand, the actuation force \( F_{act} \) in the tendon is increased such to maintain contact. Additionally, a \( F_z = 0.5 \) N sideways force in the z-direction plane is loaded in the contact point.

B. Workspace

The workspace is defined based on anthropomorphic dimensions of a human hand [13]. For the proximal joint (MCP) a workspace is used allowing for 60 x 18 x 17 mm, corresponding to the length, width and thickness respectively. Width and thickness represent an average of the proximal joint dimensions of all fingers, both for male and female, except the thumb.

The length of the hinge is designed so that half of it is inside of the palm, see Fig. 2. By doing so, the center of rotation of the flexure hinge is at the end of the palm and beginning of the finger, equivalent to the location in a human hand. The proximal phalange acts as a housing for the other half of the joint.

C. Hinge Topologies

A series of hinge topologies are defined in advance, see Fig. 4, and their performance during power grasp is compared.

- Leafspring (LS)
- Solid-Flexure Cross Hinge (SFCH)
- Three-Flexure Cross Hinge (TFCH)
- Hole Cross Hinge (HCH)
- Angled Three-Flexure Cross Hinge (ATFCH)

The initial topologies are designed such that in the undeflected position there is one rotational degree of freedom, for flexion and extension of the fingers, and the stiffnesses in support directions are high. For comparison, a flexure hinge consisting of only a single leafspring is evaluated too, which only provides support stiffnesses in three degrees of freedom. This topology is used as a reference as it is often used for prosthetic and robotic hands [2]–[4]. An initially curved design is added to generate high support stiffness at large deflections while sacrificing stiffness at smaller deflections. Several of these hinges were defined previously by [7] including their design parameters \( p \).
The HCH combines the constant bending moment of a TFCH, with the full width of SFCH, except at the crossing where reinforced parts are used.

The concept of the ATFCH is introduced in this paper, with a similar topology to the TFCH. The hinge is defined such to obtain straight elements when a specific angle is achieved, Fig. 3b.

The length of the leafsprings are equal, as the diagonals of an isosceles trapezoid, to have an even stress distribution during deflection around the z-axis. This hinge is parametrized by the parameter vector $p$.

$$ p = \{ L_{flex}, B_{phal}, W_{in}, t \} \quad (1) $$

Where $L_{flex}$ is the length of elements, $B_{phal}$ is the distance of the base (short side of the isosceles trapezoid), $W_{in}$ is the width of the inner element and $t$ is the thickness of the elements, see Fig. 3.

**D. Optimization**

Flexible multibody software SPACAR is used to evaluate the performance of the intrinsic geometric nonlinearities of the hinges [14]. By using nonlinear 3D beam elements, it is possible to efficiently compute the performance of a series of design parameters in large displacement motions and small elastic deformations. As a result a relatively small number of elements produce accurate results at low computational cost.

A shape optimization based on the Nelder Mead method is used. The objective is to find the set of design parameters $p$ that maximize the performance within the specified constraints $C(p)$ [15].

$$ p_{opt} = \arg \min_{p} F(p), \quad \text{subject to:} \quad C(p) \leq 0 \quad (2) $$

The method minimizes a cost function $F(p)$ which is defined to achieve the lowest ratio between stress and grasping force at $-30^\circ$ of flexion.

$$ F(p) = \lambda \frac{\sigma_p}{F_{grasp}} \quad (3) $$

Where $\lambda$ is a performance “penalty” to unfeasible solutions, $F_{grasp}$ is the grasping force and $\sigma_p$ is the stress value defined as follows,

$$ \sigma_p = \begin{cases} \sigma_{limit} & \text{if} \quad \max \sigma_p \geq \sigma_{limit} \\ \max \sigma_p & \text{if} \quad \max \sigma_p < \sigma_{limit} \end{cases} \quad (4) $$

$$ F_{grasp} = \begin{cases} F_{grasp@\sigma_{limit}} & \text{if} \quad \max \sigma_p \geq \sigma_{limit} \\ \max F_{grasp} & \text{if} \quad \max \sigma_p < \sigma_{limit} \end{cases} \quad (5) $$

$$ \lambda = \max \frac{d_z(p, \theta) - d_{z, max}}{d_{z, max}} \quad (6) $$

The stress $\sigma_{limit}$ is defined as 60% of the maximum allowable stress of the material, $\sigma_{max}$. The objective to introduce this stress limit is to filter out flexure hinges that show high stresses by only deflection. These hinges would result in low capacity for carrying grasping loads.

The “penalty” factor shown in 6 corresponds to the deflections in z-direction $d_z$ due to sideways force $F_z$. Where $d_z(p, \theta)$ is the deflection for the current set of parameters $p$ and $d_{z, max}$ is the maximum allowable deflection. A similar performance “penalty” is also applied for the allowable stress $\sigma_{max}$ (Table I) of the material and dimensions exceeding the defined workspace. An additional penalty is applied when leafsprings collide to ensure collision free designs [16]. By applying these penalties on the cost function, soft constraints are added to the unconstrained Nelder Mead algorithm [15].

For each iteration in the Nelder Mead algorithm $N + 1$ cost functions ($N$ equal to the number of design parameters) are compared and sorted according to $F(p_1) \leq F(p_2) \leq \ldots \leq F(p_{N+1})$, being $F(p_1)$ the solution with lowest cost (highest performance). Based on these results, a new parameter set $p$ is determined and added to the set of solutions. This process continues until a certain convergence criteria is satisfied, defined by:

$$ \frac{F(p_1)}{F(p_{N+1})} > 0.995 \quad (7) $$

which corresponds to 0.5% deviation in performance in the current set of solutions [15].

Eight shape optimizations per hinge topology were conducted, each one with a different initial parameter set. In each optimization a global or local optimum can be obtained. By conducting several optimizations the probability of finding a solution within 5% of the global optimum is greatly increased. For example, when conducting eight optimizations for the Three-Flexure Cross Hinge the probability of finding a solution within 5% of the global optimum is approximately 97% [15].

**E. Experimental Setup**

In order to validate the numerical model, a setup for measuring stiffness is used, Fig. 5. A parallel guidance, 1-DOF in the gravity direction, is actuated when weights are added to the end effector. The vertical displacement is measured through a linear variable differential transformer.
TABLE I

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>( \theta_{min} / \theta_{max} )</th>
<th>( t_{min} / t_{max} )</th>
<th>E</th>
<th>G</th>
<th>( \sigma_{limit} )</th>
<th>( \sigma_{max} / E )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \theta_{min} / \theta_{max} )</td>
<td>mm</td>
<td>-15°/30°</td>
<td>0.5/2.5</td>
<td>1.7</td>
<td>1.5</td>
<td>35.0</td>
<td>34.0</td>
</tr>
<tr>
<td>( t_{min} / t_{max} )</td>
<td>mm</td>
<td></td>
<td>0.5/2.5</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>E</td>
<td>GPa</td>
<td></td>
<td></td>
<td>1.7</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>G</td>
<td>GPa</td>
<td></td>
<td></td>
<td></td>
<td>1.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>( \sigma_{limit} )</td>
<td>MPa</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>35.0</td>
<td></td>
</tr>
<tr>
<td>( \sigma_{max} / E )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>34.0</td>
</tr>
</tbody>
</table>

III. RESULTS

A series of optimized hinges were found by evaluating the cost function \( F(p) \) at a contact force \( F_{\text{grasp}} = 5 \text{ N} \) and sideways force \( F_z = 0.5 \text{ N} \) while a deflection of \(-30^\circ\) was maintained.

After the optimizations were completed a sideways load \( F_z = -2 \text{ N} \) and an increasing grasping force on the contact point was modeled at a deflection of \( \theta_{max} = -30^\circ \), Fig. 6.

When an object is going to be grasped, first a tendon force \( F_{\text{act}} \), required to close the hand, is increased. This produces stress \( \sigma_{\text{flex}} \) in the flexure hinge. When the object is grasped and held, a combination of grasping force, increased tendon force and the sideways loads generate further stresses on the hinge.

The initial stress shown in Fig. 6 is only caused by deflection of the hinges, \( \sigma_{\text{flex}} \). The ratio between the maximum allowable stress of the material (\( \sigma_{\text{max}} \) in Table I) and the stress due to deflection, \( \sigma_{\text{max}} / \sigma_{\text{flex}} \), is lower than 2.5 for the Leafspring, the Hole Cross Hinge and Three-Flexure Cross Hinge. On the other extreme, the Solid-Flexure Cross Hinge accounts for a ratio lower than 1.6. In general, a higher ratio is desired for flexure mechanisms that are going to be cyclically loaded.

From \( F_{\text{grasp}} > 1 \text{ N} \) the object is being grasped and held in the air. At this point the initial stress \( \sigma_{\text{flex}} \) adds with the induced stress by the constant sideways force, the grasping force and the tendon force required to keep the finger in place. A steep increase of the stresses is observed for the hinges with the lowest \( \sigma_{\text{max}} / \sigma_{\text{flex}} \) ratio.

The Angled Three-Flexure Cross Hinge is a balance between a ratio \( \sigma_{\text{max}} / \sigma_{\text{flex}} = 2 \) and a low slope stress/contact ratio.
The Solid-Flexure Cross Hinge presents elastic instability, at 11 N of contact force, due to high tendon forces. Although it shows a steady stress behavior because it is stiffer in all directions compared to the other flexures.

The sideways stiffness $K_{sw}$ presented in Fig. 6 is the inverse ratio between a measured displacement $d_z$ at the contact point and the applied load $F_z = -2$ N at $-30^\circ$. $K_{sw}$ is affected by both translational compliance in $z$, and rotational compliances with the rotation axis in the $x/y$ plane.

$$K_{sw} = \frac{F_z}{d_z}$$

The Three-Flexure Cross Hinge displayed the lowest $K_{sw}$ stiffness while the Angled Three-Flexure Cross Hinge presented the best performance as expected, due to the straight flexures at the deflected position.

Further analysis of $K_{sw}$, Fig. 7, in the range of motion was studied. In this case a tendon force was applied to deflect the flexure joint to an specific angle. At that moment a load $F_z = -2$ N was applied and a $K_{sw}$ was calculated as described in 8.

The Hole Cross Hinge has the best performance over the range of motion, with a drop of support stiffness of 47.4%. The angled three-flexure cross hinge outperformed for deflection angles above 17.5°, but over the whole range of motion shows a drop of stiffness of 80%. The Leafspring, the Three-Flexure Cross Hinge and the Solid-Flexure Cross Hinge show differences above 50% in the support stiffness from undeflected to deflected position.

From Fig. 7 a non-symmetry between extension and flexion can be observed. In extension, the flexures are deflected but the influence of the torsion component is diminished. This is due to an alignment of the contact point with the center of rotation along the y-axis.

The Hole Cross Hinge and the Angled Three-Flexure Cross Hinge have resulted in hinge topologies with better performance. To compare the hinges in more detail, they were submitted to a deflection of $-30^\circ$ and load $F_{grasp}$ and $F_z$ are increased up to the maximum load carrying capacity to gain insight in failure behavior of the hinges.

Fig. 7. Comparison of optimized hinge topologies over the range of motion while loaded with a sideways force $F_z = -2$ N.

Fig. 8. Influence of sideways force ($F_z = [0; -1; -2]$ N) in optimized Angled Three-Flexure Cross Hinge (ATFCH) and Hole Cross Hinge (HCH).

Fig. 8 shows that the stress for the Hole Cross Hinge surpasses the allowable stress limit $\sigma_{max}$ at $F_{grasp} = 15$ N. The change of the stress behavior presented at $F_{grasp} = 5$ N is due to increasing tendon $F_{act}$ in an already deflected hinge. The Angled Three-Flexure Cross Hinge presented a linear and steady increase of the stresses until $F_{grasp} = 21$ N, where elastic instability appears.

Odhner reported grasping forces as high as 21.5 N for a three finger robotic hand with flexure hinges only in the proximal joint position [3]. The latter measurement was accomplished in a grasping position that avoided sideways forces. While Belter reported holding forces at the tip for commercial non-flexure-based prosthetic hands in a range between $3 - 16$ N [18]. The presented performance of the Hole Cross Hinge and the Angled Three-Flexure Cross Hinge are a considerable improvement to current flexure-based hands and can be compared to current commercial non-flexure-based prosthetic hands [18].

A. Experimental test

Before measuring the finger, the parallel guidance was characterized and the stiffness was measured when loaded up to displacements of 4.3 mm. The stiffness of the parallel guidance was linear in the whole range of motion. This was used later in order to subtract from the stiffness of the finger, as these are in parallel.

With the experimental setup, shown in Fig. 5, $[0^\circ; -15^\circ; -30^\circ]$ deflections angles were tested for a Hole Cross Hinge. These measurements are compared for validation with the flexible multibody and FEM model in Fig. 9.

Differences of 60% were found between the flexible multibody analysis and the experimental results. This model considers the attachment of the finger and the phalange as rigid. By comparison the FEM included the phalange as a deformable body. At $0^\circ$ the difference can be attributed to the clamp of the finger. As the deflection increases the loss of stiffness of the hinge becomes more important than the clamping. For this reason the differences between the FEM and the experimental at $-30^\circ$ are 25%. Small compliances
in any element contribute significantly as these elements are in series. Despite the differences, the efficiency of the flexible multibody analysis over the FEM makes it attractive for efficient flexure hinge optimizations.

B. Overload Protection Mechanism

A mechanism that prevents failure of the flexures when loads are over the limits is proposed. The concept prevents excessive displacements in torsion and in mostly all support directions with the exception of loading in positive y-direction. An initial kinematic analysis resulted in the geometry shown in Fig. 10.

In Fig. 10b contact is produced by excessive torsion on the finger. Also, when overloading due to lateral (x-direction) or compression forces (negative y-direction), a rolling contact is still possible between the palm and the phalange.

IV. CONCLUSIONS

In this paper five flexure-based finger joints topologies are presented, optimized and compared. The joints were kept within stress limits of 50 MPa and MCP joint human dimensions while a combination of 45° large range of motion, grasping force of 20 N and sideways load of 2 N was carried out. The topologies have been designed to withstand relatively high tendon actuation forces. The Hole Cross Hinge showed the best combination of high grasping force and low stress over the range of motion. The Angled Three-Flexure Cross Hinge however performs particularly good near the end of the range of motion at full flexion, and has the highest grasping force capacity. Experimental verification of the support stiffness over the range of motion shows some additional compliances, but the stiffness trend of the printed hinge is in line with the model. The presented joints power grasping capability outperform current state of the art flexure-base hands and are comparable to commercial non-flexure-based prosthetic hands. In the event of excessive loads, an overload protection mechanism is in place to protect the flexure-hinges.

ACKNOWLEDGMENT

We thank W. Pot for his contribution to the experiments.

REFERENCES


References


