Finite Element Analysis of a Wheelchair when Used with a Front-Attached Mobility Add-On

by

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The following individuals certify that they have read, and recommend to the Faculty of Graduate and Postdoctoral Studies for acceptance, the thesis entitled:

**Finite Element Analysis of a Wheelchair when Used with a Front-Attached Mobility Add-On**

submitted by Colleen Ogilvie in partial fulfillment of the requirements for the degree of Master of Applied Science in Biomedical Engineering.

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Abstract

Over the past two decades, a variety of mobility add-ons for manual wheelchairs have emerged in the assistive technology industry, including pushrim-actuated power-assist wheelchairs, motorized propulsion aids, manual and motorized front-end drive attachments, and passive attachable wheels. These technologies are typically used by long-term lightweight manual wheelchair users including those from spinal cord injury populations, and increase the mobility capabilities of the wheelchair, such as through the addition of all-terrain wheels or power-assistance.

Currently, little is known about how mobility add-ons affect the durability, strength, and lifespan of manual wheelchairs, and whether they increase the risk of component failures. In particular, very little research has assessed the likelihood of failures associated with front-attached mobility add-ons. Component failures can lead to wheelchair rider injuries or leave users stranded. Additionally, repairing or replacing damaged frames can incur significant costs.

Finite element analysis (FEA) is a technique frequently used in structural analysis. The goal of this thesis is to develop a finite element model of a wheelchair when used with a passive, front-attached mobility add-on that attaches at the footplate. The FEA model was physically validated using strain gauges under static loading scenarios. The validated model was then used to assess stresses and displacements under static loading considering several different design variables and dynamic loads based on experimental use cases, and considers how these factors impact number of cycles to fatigue failure in the system and therefore the overall lifespan of the wheelchair.

Results found that the use of a footplate-mounted mobility add-on increased stresses in the horizontal portion near the tube intersections of a D-shaped footplate. The thickness of the tubing in the footplate and the location of the rear axle created high stresses in the footplate under particular customizations. Furthermore, it was found that user mass and increased frontal impacts greatly reduced the hours of use to failure for the chair. Through identifying the location and magnitudes of points of failure, design guidelines
such as changes to attachment location or recommendations for reinforcement in manual frames can be provided to minimize risks of component failures.
Lay Summary

A recent trend in the assistive technology industry includes the increase in “mobility add-ons”, defined as relatively lightweight attachments for manual wheelchairs that usually contain one or more wheels. These devices increase the mobility capabilities of wheelchairs, such as through the addition of power-assistance or all-terrain wheels, and can be removed when not in use. These devices are typically used by long-term lightweight manual wheelchair users who are active in the community. It is important to understand how these devices change the direction and magnitude of the loads occurring on the wheelchair during use and if their use can lead to premature wheelchair frame failures. This information can be used to develop new manufacturing standards and improve wheelchair design. This study used the finite element method to provide a structural analysis of the change in stresses and deformations when wheelchairs are used with mobility add-ons in static and dynamic cases, including exploring the effect of common wheelchair configurations on stresses in the frame.
Preface

This research was performed under the supervision of Dr. H.F. Machiel Van der Loos with the Department of Mechanical Engineering and the School of Biomedical Engineering at the University of British Columbia, and Dr. Jaimie Borisoff with the British Columbia Institute of Technology. The author was responsible for performing background literature review, developing a physical and software-based experimental set-up, conducting experimental analysis and writing the manuscript. The supervisors provided guidance during the experimental analysis and writing.

A paper reporting preliminary results was presented at the RESNA (Rehabilitation Engineering Society of North America) 2018 conference in Arlington, VA, U.S.A. as a poster presentation.
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Glossary

ANSI- American National Standards Institute
CEN- Comité Européen de Normalisation
FEA- Finite Element Analysis
ISO- International Standards Organization
LRE- Low-Resource Environment
PAPAW- Pushrim-Activated Power-Assisted Wheelchair
RESNA- Rehabilitation Engineering Society of North America
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Chapter 1:

Introduction

Regaining one’s occupation, independence, and community participation are integral goals of rehabilitation for wheelchair user populations. Increasing and addressing barriers to an individual’s mobility are important steps in achieving these goals.

Wheelchairs are one of the most important technologies for regaining mobility during rehabilitation. Traditionally, wheelchair users have had two options for a personal wheelchair: power wheelchairs and manual wheelchairs, each having strengths and limitations for navigating terrain, performing activities, accessing buildings and other environments, and transportation. Since the early 2000s, a variety of mobility add-ons for manual lightweight wheelchairs have emerged in the assistive technology industry, including pushrim-actuated power-assist wheelchairs (PAPAWs), motorized propulsion aids, manual and motorized front-end drive attachments, and passive attachable wheels. These devices increase the capabilities of the manual wheelchair, such as through power assistance or enabling the wheelchair to access rougher terrain, while maintaining the benefits of a lightweight chair. Such devices can therefore combine the benefits of manual and powered wheelchairs or give the manual wheelchair other new capabilities.

The addition of a mobility add-on changes the loading conditions on the wheelchair frame. This can increase the risk of component failures and affect the overall lifespan of manual wheelchairs. Currently, little literature and no manufacturing standards exist that address the impact of mobility add-ons on the lifespan of the wheelchair. The purpose of this thesis is to provide a structural analysis of a wheelchair frame when used with a mobility add-on in static and dynamic scenarios using the finite element method to assess deformations and stresses. Through identifying likely areas of failure, design guidelines, such as recommendations for reinforcement in manual frames, can be provided. The results of this research have the potential to provide insight for designers and
manufacturers of manual wheelchairs, as well as for working standards groups that contribute to the development of manufacturing standards.

1.1 Overview of Wheelchair Mobility Add-ons

This thesis examines a set of devices that have emerged over the past two decades, described here as “mobility add-ons”. The following sections describe this term, the included scope of devices, and user demographics.

1.1.1 Definition and Examples of Mobility Add-Ons

This thesis defines “mobility add-ons” as accessories for manual wheelchairs that increase the wheelchair’s mobility capabilities and can be removed when not in use. Benefits of these devices include an improved ability to navigate rough terrain, a change in means of wheel propulsion, and power-assistance, which can increase the distances that can be travelled and compensate for reduced upper body function. Some examples of mobility add-ons currently on the market can be seen in Figure 1.

![Figure 1 Examples of Types of Mobility Add-Ons: The FreeWheel, the RIO Dragonfly, the BATEC Electric, the Permobil SmartDrive, and the Alber e-motion](image-url)
Mobility add-ons began to appear on the market in the early 2000s. An early device was the JWII (Yamaha Motor Co, Iwata, Japan), a removable wheel that provides power-assistance. These devices began the trend of removable power-assist wheels and removable front attachments, which include passive, manual and power-assist devices. Rear-attached power-assist devices, such as Permobil’s SmartDrive, appeared on the market in the early 2010s [1]. Notable companies who currently produce mobility add-ons in North America include Permobil (Lebanon, TN, USA), Alber (Oakdale, PA), Rio Mobility (Berkeley, California, USA) and BATEC (Concord, ON, Canada).

With the continued development of battery technology and increasingly smaller and lighter motors, the trend of increased capabilities of manual wheelchairs through attachable devices can be expected to continue.

1.1.2 Demographics of Manual Wheelchair and Mobility Add-On Users

As of 2017, the World Health Organization estimates that there are 75 million wheelchair users worldwide, making up approximately 1% of the total population [2]. According to Statistics Canada, an estimated 288,800 community-dwelling Canadians aged 15 or older were using wheeled mobility devices in 2012. Of these, an estimated 197,950 used a manual wheelchair [3].

Manual wheelchair user populations are characterized by having a mobility impairment. Major causes of mobility impairments requiring the use of a manual wheelchair include congenital conditions, disease and illness, both work- and non-work-related injuries, and aging [1]. A 2016 study by Smith et al. found that disease and illness are the largest known causes of activity limitation requiring a wheelchair in Canada [1]. Some specific populations of manual wheelchair users include people with spinal cord injury, lower-limb amputations, multiple sclerosis, Parkinson’s disease, and aging-related issues, among others.

There is currently limited research on the demographics of users who use manual wheelchairs with mobility add-ons. Power-assistance can be useful for people who have difficulty propelling a manual wheelchair because of pain, low cardiopulmonary reserves, insufficient arm strength, or the inability to maintain a posture effective for propulsion, as well as for those who are at risk for fatigue when using a manual wheelchair [4].
Tetraplegic populations have also been identified as benefitting from power-assist devices [5]. However, this list is not exhaustive; it can be assumed that many people who benefit from a power wheelchair would also benefit from a power-assist device, such as a PAPAW, as an alternative.

Often mobility add-ons are marketed to younger and more active users. However, elderly populations have also responded positively towards power-assist devices, with users in one study reporting that a power-assist chair required less effort to propel and was less tiring and that the PAPAW increased handling and maneuverability compared to a regular manual wheelchair [6]. Furthermore, elderly users stated a PAPAW would enable them to have greater activity and that they would travel more extensively in the community if they owned a PAPAW [6].

1.2 Need for Understanding of Failure Modes

1.2.1 Risks Associated with Mobility Add-Ons

Mobility add-ons have the potential to increase risks to the user. These include increased risk of component failures, increased user confidence in rough terrain and at higher speeds, risks associated with being stranded, and changes to stability.

The addition of a mobility add-on can result in unconventional loading scenarios for a wheelchair frame. For example, changes in torque and bending moments can occur if the wheelchair is pushed or pulled by a power-assist device rather than by manual pushrim propulsion. Changes in loadings for which the chair was not designed may lead to premature failures. For example, anecdotal evidence exists of plastic deformation as well as fracture near the welded regions of D-shaped footplates when used with footplate-mounted mobility add-ons such as the FreeWheel. Such wheelchair component failures can incur significant costs on the user. Wheelchair repairs are expensive; even simply replacing a wheelchair footplate can cost upwards of several hundred dollars, and component failures often require time to be taken off work if the user does not have a back-up chair available [7].
Mobility add-ons increase the capabilities of the wheelchair, in some cases allowing the user greater access to a variety of terrains and the ability to cover longer distances each day. This can also allow the user to access steeper slopes and travel at higher speeds, which has the potential to lead to increased impact forces on the wheelchair, particularly when the chair hits an obstacle such as a curb or rock. This can also lead to a decrease in stability and therefore increased risk of falls. Mobility add-ons with large wheels and pneumatic tires, such as the FreeWheel or the BATEC Manual, make it easier for wheelchair users to access hiking trails and cobbled roads by allowing them to easily roll over larger rocks and bumps. The FreeWheel has also been shown to decrease travel time, reduce required propulsive movements and reduce the amount of force to propel the wheelchair on snow-covered inclines [8].

Being stranded as a result of a fall is more critical for wheelchair users than for able-bodied users of similar mobility technologies, such as bicycles. Being stranded has been shown to be associated with pain, depressive symptoms, greater odds of rehospitalization and injury for wheelchair users [9]. As manual wheelchair users depend on their wheelchair for mobility, a loss of function of the chair will often require assistance from another person. In locations where help is not readily available, the severity of this risk can increase.

Mobility add-ons can also affect the stability of the chair. The addition of accessories has been identified as a major factor affecting stability in wheelchairs, as they can cause the system’s center of mass to move outside of the point of contact of the wheels in unexpected situations, particularly on slopes [10]. The consideration of the impact of mobility add-ons on stability is outside of the scope of this thesis but represents a direction for future research.

In summary, the risks of using a mobility add-on include increased risk of injuries from falls and component failures that result from traversing rougher terrain and with increased speed and confidence, therefore subjecting the wheelchair to higher loads than it may have been designed for. These risks can also lead to users being stranded and with high costs to replace components. Furthermore, mobility add-ons can impact the stability of the wheelchair.
1.2.2 Trend Towards Ultralight Wheelchairs

In manual wheelchair design, it is advantageous for the chair to be as lightweight as possible to make the chair easier to maneuver, more energy efficient for the user and easier to lift into a motor vehicle. Decreasing the weight of the manual wheelchair has been an industry trend over the past century, from heavier 20-30 kg steel folding chairs, developed at the beginning of the 20th century [11], to “ultralight” aluminum, titanium or carbon fibre chairs, which today can be found as light as 4 kg [12]. To achieve a lighter weight, wheelchairs are made with thinner and smaller tubes and are often reinforced only in necessary locations to withstand normal loads over an expected lifespan. For such chairs, it is plausible that even minor changes in loading, such as those introduced by add-ons, may become significant over time, contributing to either fatigue or impact failures.

1.3 Problem Statement and Thesis Objectives

Mobility add-ons are an emerging trend in the wheelchair industry. Currently, little literature and no manufacturing standards exist to guide how and where mobility add-ons attach to manual wheelchair frames and what loads they should place on the frame. Our long-term research goal is to contribute to the knowledge of how mobility add-ons change wheelchair frame deflections and stresses, which may change the type and severity of risks experienced by the user. By contributing to a body of knowledge that reports wheelchair failures for various types of wheelchair configurations and wheelchairs under various types of loading, wheelchair designs can be improved for use with add-ons. This information can provide guidance to manufacturers and designers to ensure that wheelchairs are safe and able to withstand reasonable forces throughout their expected lifespan when used with mobility add-ons. This will be achieved through both physical testing and computer simulations based on the finite element method.

This type of research serves an important purpose because wheelchairs are an important technology for improving mobility and greatly impact the quality of life of 75 million users worldwide. However, even with these numbers, the wheelchair industry has a relatively small market size and wheelchair users often represent an under-funded population. For this reason, there are less resources driving improvements in technology
and innovation in industry than in comparable industries such as the automotive or aerospace industries. Mobility add-on innovations often emerge from users who design custom devices to fit their own needs, rather than through industry-funded research and development, and therefore many designers of devices lack access to product testing.

The specific objectives of this thesis are:

- To develop a methodology based on the finite element method and strain gauge based analyses to assess stresses in manual wheelchairs when used with a simple mobility add-on,
- To explore common wheelchair configurations and their impact on stresses in the wheelchair frame when used with the mobility add-on, and
- To make an estimation of the likelihood of fatigue failures under five defined scenarios estimating realistic loads experienced by the wheelchair.

1.4 Thesis Outline

The thesis is laid out as follows:

Chapter 1 (Introduction): Information on the emerging trend of mobility add-ons for wheelchairs and the need for understanding the loading conditions on wheelchair frames when used with mobility add-ons are presented.

Chapter 2 (Background and Related Work): A review of relevant studies available in the literature, an overview of ISO manufacturing standards for wheelchairs, and relevant technical theory on the finite element method are presented.

Chapter 3 (Methods): An overview of the design and development of finite element analysis and physical validation model is detailed. Relevant finite element studies exploring the effect of various design parameters on the wheelchair are outlined.

Chapter 4 (Results): The results of testing are presented.

Chapter 5 (Discussion): A discussion of the implications of these results is included.

Chapter 6 (Conclusions and Future Direction): Thesis contributions, directions of this research, and limitations in the developed system are summarized.
Chapter 2:

Background and Theory

In this chapter, relevant background on mobility add-ons, including categorization of mobility add-ons, manufacturing methods, existing ISO standards, and standard loading conditions used to assess wheelchair durability, is provided. Relevant background theory on finite element analysis and strain gauge-based testing and a review of related literature are also included.

2.1 Categorization of Mobility Add-Ons

A system was created by the UBC CARIS Lab (Vancouver, Canada) to identify and categorize existing mobility add-ons. Based on this work, mobility add-ons can be classified into three main categories: power-assist wheels, front attachments, and rear attachments. These groups can be further subclassified into passive devices, manually-powered devices, and power-assist devices, described in further detail below. This categorization allows for consideration of groups of devices that place similar magnitudes and directions of force on the wheelchair frame. These categories are illustrated in Figure 2.

![Figure 2 Categories of Mobility Add-Ons as Defined by the Loading Conditions on the Wheelchair](image-url)
Power-assist wheels include devices that attach to or replace the wheelchair’s back wheels and provide supplementary power to the wheels, such as the eMotion. A power-assist wheel is often termed a “PAPAW”, or pushrim-activated power-assist wheelchair, which describes the control mechanism of the wheel; PAPAW systems sense the torque that the user applies to the pushrim and add power-assistance to help the user propel the chair. PAPAW controls are popular as they allow for an intuitive user interface, similar to the manual propulsion of a wheelchair. PAPAW, and other power-assist systems, are beneficial because they can reduce metabolic demands on the user [5], decrease injuries [13], and increase the distances users can travel [14].

2.1.1 Front Attachments

Front attachments attach to the front of the wheelchair’s frame or under the seat of the user. Commonly, these devices lift the wheelchair’s front caster wheels and replace the front ground contact surface with a large, centred front wheel, which is used to improve the chair’s ability to traverse soft or uneven surfaces. This design can be seen in passive devices such as the FreeWheel, which has been shown to improve the ability to navigate irregular surfaces including pavement and grassy areas [15]. It also includes manually-powered front attachments including the BATEC Manual [16] and the Rio Dragonfly [17], and power-assist front attachments such as the BATEC Electric and the Rio eDragonfly, as well as hybrid systems including the BATEC Hybrid. Manually-powered front attachments primarily include handcycle-like attachments, whose main purpose is to change the method of propulsion of the wheel.

2.1.2 Rear Attachments

Rear attachments push or stabilize the chair from behind and include devices such as Permobil’s SmartDrive [1]. These devices tend to be smaller and therefore more portable than front attachments, and can have a variety of control mechanisms, such as a Bluetooth wristband [1]. As they do not lift the caster wheels, they have limited abilities on soft rough terrain where the casters normally dig in or are too small to navigate larger rocks and bumps.
2.1.3 Power-assist, Manually-powered and Passive Add-Ons

Mobility add-ons can also be categorized by their method of propulsion. Categorization of mobility add-ons, or “propulsion aids”, based on method of propulsion has been proposed in a recent scoping paper by Choukou et al (2019), identifying “human-powered” devices, such as one-arm drive and lever-activated devices, and “power-assisted” devices including PAPAW devices and detachable motor drives. In this thesis, power-assist, manually-powered and passive add-ons serve as sub-categories to the aforementioned categories based on location relative to the wheelchair. Most power-assist devices, including front, rear and wheel attachments, are powered by a lithium-ion battery and a motor that provide power to the drive train of the wheels. Manually-powered attachments such as handcycle attachments are based on gears and levers that change the method or augment the power provided by the user. Passive devices primarily describe wheel attachments. Novel wheelchair attachments such as attachable skis are outside the scope of this thesis.

2.1.4 Examples of Mobility Add-Ons Currently on the Market in North America

Examples of products currently on the market, categorized by their locations relative to the chair, can be seen in Table 1. A few examples of products are described in further detail in Section 1.2.

2.2 Loading Conditions Associated with Front Attachments

While limited published literature or statistics exist on the subject, anecdotal evidence collected by this lab has found that failures are occurring in manual wheelchair frames when used with some types of front attachments. There is currently a gap in the literature around the structural impact of front-attached mobility add-ons on manual wheelchair frames, described further in Section 2.8.
Due to the added complexity of analyzing manually-powered or power-assist devices, the FreeWheel, which is a passive front attachment currently commercially available in North America, was selected as a basis for this analysis. Future directions of this work could include analysis of more complex types of front attachments for comparison purposes. Section 2.2.1 describes this device in more detail.

### 2.3 The FreeWheel

The FreeWheel is a simple front attachment that adds a large, swiveling wheel to the chair while lifting the wheelchair’s front wheels off the ground. The main benefit of this device is to improve rolling resistance over soft and uneven terrain. This includes terrain that would otherwise be hazardous, difficult or impossible for manual wheelchairs. It was selected for this analysis as it is one of the most simple add-ons and causes visible changes to the loading conditions on the wheelchair.

<table>
<thead>
<tr>
<th>Device Classification</th>
<th>Method of Power</th>
<th>Examples of Products Currently on the Market</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Front Attachments</strong></td>
<td>Passive</td>
<td>FreeWheel (FreeWheel)</td>
</tr>
<tr>
<td></td>
<td>Manually-Powered</td>
<td>Dragonfly (Rio Mobility)</td>
</tr>
<tr>
<td></td>
<td>Power-Assist</td>
<td>Firefly (Rio Mobility), eDragonfly (Rio Mobility), Electric (BATEC Mobility), Rapid (BATEC Mobility), Urban (BATEC Mobility), Hybrid (BATEC Mobility), Raptor (Progeo), Triride (Triride Italia)</td>
</tr>
<tr>
<td><strong>Rear Attachments</strong></td>
<td>Power-Assist</td>
<td>SmartDrive MX 2 (Max Mobility Inc.), Benoit Light Drive (Speedy Snail Mobility), ZX-1 (Spinergy)</td>
</tr>
<tr>
<td><strong>Power-Assist Wheels</strong></td>
<td>Power-Assist</td>
<td>eMotion (Alber), e-Fix (Alber), Twion wheel (Alber), z50 (Ottobock), e-Support (Ottobock), Xtender (Quickie), WheelDrive (Quickie), Servo (AAT), Solo (AAT), JWX-2 (Yamaha), Duo and Nomad (Autonomad Mobility)</td>
</tr>
</tbody>
</table>
As can be seen in Figure 3, the FreeWheel extends out in front of the wheelchair. The wheel has an approximately 12” pneumatic tire which serves to lift the front casters off the ground when it is being pushed over soft or uneven surfaces. This is beneficial as the casters are known to be the primary cause of poor manual wheeling performance on soft terrains [15]. The FreeWheel attaches either to the wheelchair’s footrest or to a support bar through a hand-tightened clamping plate.

![Figure 3 The FreeWheel, an Example of a Passive Front Attachment](image)

2.4 Materials and Manufacturing Methods of Manual Wheelchairs

Modern, manual ultralight wheelchairs typically used by active users are usually made from one of three materials: aluminum, titanium or carbon-fibre [18]. Aluminum alloy is one of the most common material types of manual wheelchairs as it has low weight and cost and high specific strength. Common aluminum alloys found in wheelchairs are 6000 and 7000 series alloys. Titanium also has a low weight and high specific strength, but can be more expensive and difficult to machine and weld. Carbon fiber-reinforced polymers have also become common material choices for very light wheelchairs over the past decade, but are higher-cost [19]. Steel alloys can also often be found in wheelchairs, however, they typically result in heavier wheelchairs and are subject to corrosion. Steel has traditionally been used for folding-style chairs that are primarily intended for temporary use such as in hospitals or nursing homes [18].
Common main steps of manufacturing an aluminum wheelchair frame include hydroforming and bending extruded tubing of aluminum to a particular profile and shape, miter cutting the tubes to the correct length, TIG welding, and finally heat treating to compensate for the reduction in material strength due to internal stresses induced during welding. The process of welding can have significant impacts on the material strength and other properties of aluminum, such as through the creation of a heat affected zone.

2.5 Standards for Wheelchairs and Mobility Add-Ons

Industry standards for wheelchairs exist to ensure quality, safety, and standardized products [20]. Existing standards include the International Standards Organization (ISO) standards and national standards such as RESNA/ANSI, CEN, and Health Canada. Wheelchairs are usually subject to compliance to industry standards at both the international and national levels.

2.5.1 ISO Standards for Wheelchairs

The ISO maintains the ISO 7176 standard, which guides the design and manufacture of wheelchairs internationally. This standard contains 37 sections that cover different parts of the wheelchair. ISO 7176-8 outlines static, fatigue and impact strength testing for wheelchairs, which intends to minimize frame and component failures in the chair over its expected lifespan.

During static strength tests, the wheelchair is positioned in a horizontal test plane and loads are slowly applied to various parts of the chair. Upon completion of the test, a visual inspection is used to assess any deformed, fractured or cracked components, detached adjustable components, nuts, bolts or screws, and displaced handgrips. These tests include static downward force tests on the armrest, footrests, tipping levers and handgrips, and upward force tests on the armrest, footrests and push handles.

Under the impact strength tests, a weighted pendulum is used to strike various part of the wheelchair, which simulates the impact applied by users falling against the wheelchair backrest and parts of the wheelchair colliding with obstructions. The parts of the wheelchair assessed for impact include the backrest, the handrim, the casters, the footrests, and the front structure.
There are two tests that assess wheelchair fatigue: the two-drum test and the curb drop test. The two-drum test involves positioning the chair’s wheels on two spinning drums with small obstructions for 200,000 cycles, specified to run at 1.0 m/s. A curb-drop test machine is used to drop the chair freely 50 mm ± 5 mm onto a rigid horizontal plane until 6666 cycles have been completed, which is used to simulate the effect of dropping off curbs. Schematics of these two tests can be seen in Figure 4.

![Figure 4 ISO Curb Drop (left) and Double Drum (right) Fatigue Tests](image)

During both static and fatigue tests, a test dummy is secured to the seat of the chair to simulate the user’s weight. The same failure criteria as the static strength tests are used to assess fatigue and impact strength requirements.

### 2.5.2 Maintenance of Standards

Standards are maintained with the assistance of working standards groups, such as RESNA, which gather experts to discuss and assess issues related to wheelchair standards and provide recommendations for standard maintenance and development [20].

### 2.5.3 Existing Gap and Possible Need for New Standards

Currently, no standards exist that directly pertain to the use of manual wheelchairs with mobility add-ons. In practice, companies may modify the double drum test and curb drop
tests to assess the fatigue life of manual frames when used with their mobility add-ons. However, these practices are not regulated by the ISO.

2.6 Finite Element Analysis Background and Justification for Use

Finite element analysis is a structural analysis technique that can be used to predict values of variables such as stresses or displacements in a body, given particular loading and boundary conditions. The premise of finite element analysis is that large, complex bodies can be broken down into small, simple pieces called “elements”, whose individual mechanical behavior under applied forces is known.

Newton’s 2\textsuperscript{nd} law describes the relationship between applied force and displacement within a structure for each element:

\[ F = m\ddot{u} + c\dot{u} + ku \]

In the static case, the load is assumed to be applied slowly, so the equation reduces to:

\[ F = ku \]

Here, the stiffness coefficient \( k \) is defined based on the known material properties:

\[ k = \frac{EA}{L} \]

where \( E \) is Young’s Modulus,
\( A \) is the cross-sectional area, and
\( L \) is the length of the material.

This concept is illustrated in Figure 5.

![Figure 5 Conceptual Model of Finite Element Analysis under Static Loading](image-url)
In dynamic finite element analysis, effects due to damping and inertia are not ignored.

\[ F = m\ddot{u} + c\dot{u} + ku \]

**Figure 6 Conceptual Model of Finite Element Analysis under Dynamic Loading**

Finite element analysis has three main steps: pre-processing, processing, and post-processing. The pre-processing step includes defining the geometry, loading and boundary conditions, and meshing the model into smaller elements. In the processing step, the set of algebraic equations defining all element displacement and force relationships is solved to obtain displacements at the nodes of the elements. Stresses can be calculated and areas of likely failure can be determined from these displacements during post-processing.

For complex structures, finite element analysis is most often achieved through specialized software such as ANSYS. These applications can produce detailed results that include graphs, reports, and color-scaled animations indicating major stresses, strains and displacements.

An additional and important step of finite element analysis is model validation. This is often achieved through the development of a physical model that reflects the geometry and applied load and the use of experimental methods to assess displacements and stresses.

Using finite element analysis for the structural analysis of manual wheelchairs has several benefits over physical testing. Physical testing equipment can be expensive, and access to double drum, curb drop and pendulum test equipment may be limited for designers. Testing wheelchairs to failure often requires the wheelchair to be replaced after each design iteration, adding to the overall cost. There is also a significant time component to designing – the part must be conceptually designed, prototyped, and tested before
iterations can be made. Finite element analysis allows for rapid design iterations and predictions around stress concentrations and failures, without the cost of physical testing.

2.7 Physical Validation using Strain Gauges: Background

Strain gauges are commonly used for experimental stress analysis of complex structures as well as for validating numerical methods such as finite element analysis, as they are accurate, affordable, small compared to many overall structures, and easy to acquire and set-up. For this reason, experimental analysis using strain gauges was selected to validate the finite element analysis for this project.

Strain gauges consist of flexible wire that changes resistance when strain occurs during an applied load, as shown in Figure 7. Experimentally, strain gauges are most often used to find the axial strain. Because the change in strain can be very small, strain gauges are often mounted in a Wheatstone bridge circuit, which allows for very small changes in resistance to be measured more accurately. There are three commonly used configurations of the Wheatstone bridge for structural analysis: the quarter bridge configuration, the half bridge configuration, and the full bridge configuration [21], as shown in Figure 8. The selection and configuration of strain gauges selected for this project are described in Chapter 3. An example of how strain gauges are used for structural analysis are shown in Figure 9.

![Figure 7 Example of a Strain Gauge](image)
2.8 Related Literature

This thesis has identified an emerging trend of mobility add-ons for wheelchairs. There is currently a lack of knowledge in the literature surrounding how front-attached mobility add-ons impact the lifespan of manual wheelchairs. We propose using finite element analysis and strain gauge-based physical validation to help address this gap in the literature. This section reviews related literature assessing wheelchair durability, mobility add-ons, wheelchair failures, and related finite element studies.
2.8.1 Assessment of Mobility Add-Ons

Of the mobility add-ons identified in this thesis, PAPAWs have received the most attention in the literature, and have been assessed for their impact on the wheelchair’s durability, the effect on the user’s oxygen consumption, and the users’ abilities in various tasks and skills.

Several studies have assessed PAPAWs for durability under the ISO tests. In 2001, Cooper et al. assessed the JWII for compliance with wheelchair standards, energy demand placed on the user during propulsion, and ergonomics [22]. In a similar study in 2008, Karmarkar et al. assessed three PAPAW wheels – the iGlide, the eMotion and the Xtender – under ISO standards, including static and dynamic stability, brake effectiveness, speed, energy consumption, and impact, static and fatigue strength [23]. Both studies found good compliance of PAPAW systems under ISO testing.

Other means of assessing PAPAWS include characterizing their impact on users’ VO2 consumption, and on the users’ ability to perform activities of daily living. A systematic review by Kloosterman et al. in 2013 found that power-assisted propulsion reduced the strain on the arms and cardiovascular system compared to hand-rim wheelchair propulsion, and tasks that require more torque were made easier with a power-assisted wheelchair [24]. However, precision tasks were found to be easier with a conventional hand-rim manual wheelchair. They also found that social participation was not affected significantly by the use of a hand-rim, powered or power-assisted wheelchair. Best et al. (2005) hypothesized that PAPAW users could accomplish a greater range of wheelchair skills than non-PAPAW manual wheelchair users, but concluded that PAPAW performance was not superior to that when using a manual wheelchair [25]. Guillon et al. (2015) compared 3 PAPAWs, including the eMotion and the Servomatic, and conventional manual wheelchairs used by users to assess users’ oxygen consumption (VO2) and heart rate, ability on indoor and outdoor courses, and the ability of the subjects to transfer themselves and the wheelchairs into and out of their car without help [4] They found that PAPAWs significantly decreased heart rate compared to manual wheelchairs, while car transfer ability was reduced.
An initial evaluation of the FreeWheel exists in the literature, including a qualitative description of one person’s experience, reporting that the device performed well in urban environments and grassy areas, when turning in tight areas, turning on hills, and traversing over drops and grooves in sidewalks [15]. The user also felt more confidence to travel at faster speeds on sidewalks.

Mobility add-ons have been shown to have positive influence on users’ autonomy. For example, Khalili et al. (2019) found that wheeled mobility assistive devices have positive benefits on user quality of life and life satisfaction.

2.8.2 Physical Testing of Wheelchairs

2.8.2.1 ISO Testing of Wheelchairs

Standard ISO testing results have been reported in the literature to assess and report on particular types, brands and designs of wheelchairs. Electric power wheelchairs were tested by Rentschler (1995) [26] as well as Fass (2004) [27], and low-cost nonprogrammable electric powered wheelchairs were tested by Pearlman et al. in 2005 [28]. Fatigue testing for manual chairs was reported by Fitzgerald et al. in 1999 [29], and of selected suspension manual wheelchairs using ANSI/RESNA standards by Kwarciaik in 2005 [30]. Kwarciaik et al. performed fatigue testing on suspension wheelchairs to compare with lightweight and ultralight chairs, finding no improvement in durability [30].

2.8.2.2 Other Means of Physical Testing of Wheelchairs

Other means for physical testing include the use of strain gauges and accelerometers. Strain gauges have been used in combination with the ISO double drum roller test to assess fatigue in wheelchairs [31]. VanSickle et al. used accelerometers to assess loads for box frame wheelchairs and a cantilever frame wheelchair using the ISO curb drop test to provide input loads for FEA models [32].

2.8.2.3 Wheelchair Durability in Outdoor Conditions

Because front-attachments increase the user’s abilities to navigate outdoor terrain, literature reporting wheelchair durability in outdoor conditions was also reviewed. A systematic review by Mahtre et al., gathered field evidence of premature failures of
wheelchairs when used in the rugged conditions of low-resource environments (LREs), and discussed the discrepancies between common wheelchair failures reported in ISO testing and field failure evidence from the rugged terrain LREs [33]. Common field failures included bent frames, non-functional brakes, worn out bearing, rusting and loosening of several parts, and caster failures, among others. Mahtre et al. also suggested ultraviolet light, high temperatures, dirt, dust, humidity and water exposure as possible causes in addition to different types of loads normally tested by the ISO standard. In response to these findings, caster testing equipment was proposed by Mahtre in 2017 [34].

2.8.2.4 Finite Element Analysis for Wheelchair Durability Assessment

Finite element analysis has many applications in both the biomedical and transportation fields. In the literature, FEA has been used in wheelchair design to optimize composite wheels [35] and composite frames [36], and to optimize specialized chairs for particular activities, such as racing [37], standing [38], and basketball [39]. FEA has been reported in the literature for analyzing bicycle frames, which can be considered a similar type of mobility device in terms of material and tubular fabrication. Some examples include optimizing a sandwich composite bicycle frame [40], an aluminum bike frame [41], and a bamboo bicycle frame [42].

2.8.3 Gap in Literature

While manual wheelchair durability assessment has been described in the literature based on a variety of methods, relatively few studies have assessed fatigue strength, impact strength, or risks to users of manual wheelchairs when with a mobility add-on attached. To the best knowledge of the author of this thesis, no studies have yet assessed failures associated with front-attached mobility add-ons, either through physical testing or computer-aided analyses such as finite element analysis. The author of this thesis believes that front attachments create a unique loading scenario on manual wheelchair frames that requires further examination.
Chapter 3:

Methods

3.1 Overview

The methodology of this thesis included the development of a physically validated finite element analysis (FEA) model of a lightweight manual wheelchair and an attached mobility add-on. The FEA model was validated using strain gauges mounted on a commercially acquired lightweight manual wheelchair mounted with a custom-built front-attached mobility add-on. Following validation, the model was used to examine a set of design variables and customizations that may affect the likelihood of failure in the footplate under static loading.

Dynamic loading during typical add-on use was examined through an experimental study in which typical impact loads were estimated over ten minutes of mobility add-on use for five defined scenarios. Consideration was given to the impact of mobility add-ons on the likelihood of premature fatigue failures in manual lightweight wheelchairs using an estimation of fatigue.

3.2 Experimental Design

3.2.1 Static Analysis

A static analysis was performed to assess downward loads on a manual lightweight wheelchair when fitted with a mobility add-on that attached at the footplate. The analysis contained both experimental stress analysis based on strain gauge measurements and stress analysis using FEA software.

The experimental set-up of the static analysis can be seen in Figure 10. The purpose of the analysis was to apply simple, realistic loads on the wheelchair frame. In the physical model, the setup included a Quickie (Sunrise Medical, Fresno, USA) Q7 wheelchair frame, a D-shaped footplate, two rear mounts that supported the wheelchair at the axle, and a simple front attachment, described in Section 3.1.1.
Figure 10 Experimental Design

In the physical model, shown in Figure 11, the setup included two triangle-shaped steel mounts that suspended the rear wheelchair axles, allowing them to rotate freely. The rear mounts were constructed previously by the BCIT Rehabilitation Engineering Design Lab (RED Lab, Burnaby, Canada). The footplate of the wheelchair was suspended by the front attachment. Strain gauges were mounted on the vertical portion of the footplate and were connected to a Wheatstone bridge circuit and digital multimeter, described in Section 3.4.

Figure 11 Photo of the Experimental Set-up

This experimental set-up was then modelled in SolidWorks and imported into ANSYS to determine output stresses in the structure given the loads and constraints defined by the experimental set-up.
3.2.2 Front Attachment Design

A front attachment was designed to replicate similar forces on the wheelchair to those caused by the addition of the FreeWheel wheelchair attachment, without needing to model the complex mechanisms of the FreeWheel in the FEA software.

Figure 12 shows the primary static forces that act on a wheelchair when used with a passive front attachment such as the FreeWheel. The FreeWheel attaches to the wheelchair frame at the footplate and lifts the casters so they are suspended above the ground, which causes loads that are normally directed through the caster wheels to the ground to be transferred instead through the footplate to the FreeWheel wheel and tire. This load creates compression and bending on the vertical portion of the footplate and shear, bending and torsional loads in the horizontal portion of the footplate. The location of the front attachment was the primary design requirement of the analysis add-on. A free body diagram of the wheelchair frame and footplate can be seen in Figure 14. The FreeWheel allows primarily for translation in the X-Y plane, as defined in Figure 12, through rolling, as well as rotation about the Z-axis from the rotation of the front wheel about its axle. These boundary conditions were also important design features.

![Free Body Diagram of a Wheelchair and FreeWheel](image)

**Figure 12 Free Body Diagram of a Wheelchair and FreeWheel**
The CAD design and physical model of the designed front attachment can be seen in Figures 15 and 16, respectively. This design fulfilled with the design requirements noted above and allowed the front of the wheelchair to be suspended at a height of 6 cm from the bottom of the footplate off the ground.
3.2.3 Dynamic Loads Study

The location, direction and magnitude of loads were estimated to represent four scenarios: hitting a crack in a sidewalk, hitting a small rock in a grassy area, hitting a curb at a slow speed, and hitting a curb at a faster speed. Schematics of these scenarios are shown in Figure 17. Figure 18 shows the approximate point of impact of a curb impact when using a FreeWheel which was used to determine the approximate location and direction of the load in ANSYS. At this location on the FEA model, a time history load was input into ANSYS.
Figure 17 Schematic showing three activities assessed during typical use of the FreeWheel: Curb Descent and Ascent while Crossing a Road, Traversing Grass, and Traversing Sidewalk

Figure 18 Curb Impact
To estimate the time history load, a block impulse was used. An example of a block impulse is shown in Figure 19. The block impulse is a simple model for an impact load in structural dynamics. It is typically used where damping is of less importance because the maximum response is reached in a very short amount of time, such as during an impact.

![Figure 19 Example of a Block Impulse used to estimate dynamic loads in ANSYS](image)

For each of the four scenarios noted above, a 10 msec duration of impact was assumed. Impacts associated with curb bumps encountered by bikes have been found to be on the order of 20-30 msec [43]. Because the impulse was assumed to be higher for mountain bikes than wheelchairs, a 10 msec impulse was used for this study. These numbers represent rough estimates that were assumed to represent typical loads. It was assumed that the user would hit sidewalk cracks at a rate of 1 Hz and small rocks at 0.2 Hz.

### 3.3 Finite Element Analysis Model Development and Validation

#### 3.3.1 Overview

A simple 2D theoretical calculation was used to provide preliminary insight into the expected stresses found in the vertical portion of the footplate under static loading. Next, FEA software was used to find the von Mises stress at the same location under several loading conditions. Finally, strain gauges mounted in the footplate were used to make a prediction of bending stress. These results were compared and used to assess the validity of the FEA model.
3.3.2 2D Theoretical Analysis

A simple 2D theoretical calculation was used to provide preliminary insight into stresses that could be expected at the strain gauge location in the footplate. Considering the structure as a solid beam, a free body diagram of the system described can be seen in Figure 20.

![Free Body Diagram of a Wheelchair and Mobility Add-on in 2D](image)

Figure 20 Free Body Diagram of a Wheelchair and Mobility Add-on in 2D

The stress component associated with each internal loading can be computed using simple beam equations.

**Normal Force**

\[ \sigma = \frac{N}{A} \]

**Shear Force**

\[ \tau = \frac{V}{A} \]

**Bending Moment**

\[ \sigma = \frac{My}{l} \]
where $I$ is the moment of inertia for a hollow tube:

$$I = \pi (R_o^4 - R_i^4)/2$$

**Torsion**

$$\tau = \frac{Tc}{J}$$

where $J$ is the polar moment of inertia for a hollow tube:

$$J = \pi (R_o^4 - R_i^4)/2$$

The reaction forces can be found using a balance of forces and moments:

$$\sum F = R_a + R_b - F = 0$$

$$\sum M_B = F \cdot \text{dist } a - R_a \cdot \text{dist } b = 0$$

$$R_a = \frac{F \cdot \text{dist } a}{\text{dist } b}$$

$$R_B = F - R_A$$

Given these forces and the known dimensions of the aluminum tubing in the footplate, the stress at the location of the strain gauge can be estimated. Further details on the calculations used for the preliminary analysis are found in Appendix A.

### 3.3.3 Finite Element Analysis Methodology

#### 3.3.3.1 Summary of FEA Methodology

The following steps summarize the FEA methodology:

*Pre-Processing:* A passive front attachment and manual wheelchair were modelled in SolidWorks and imported into ANSYS. Material properties, appropriate mesh parameters, boundary conditions indicating where the displacements would be fixed, and loading conditions indicating the location and magnitude of the loads were specified for each study.
**Processing:** An analysis was run using the industry standard ANSYS FEA software package.

**Post-processing:** The maximum von Mises stress in the wheelchair and the von Mises stress at the strain gauge location on the footplate were recorded. In ANSYS, displacements and resulting stresses were displayed graphically, and the locations of stress concentrations were identified.

**Validation:** During the physical validation studies, a physical wheelchair and mobility add-on system closely matching the FEA model was instrumented with strain gauges to assess the strains, which were used to calculate stresses, caused by the loading conditions modelled in ANSYS. This step was used to confirm the values of the stress at the strain gauge location predicted by the FEA.

### 3.3.3.2 Pre-Processing

**Geometry Characterization**

The Quickie Q7 chair is a rigid aluminum wheelchair, selected due to the simplicity of its geometry and as a common example of a long-term daily-use chair for active users. A photo of the Quickie Q7 can be seen in Figure 21.

![Figure 21 The Quickie Q7 Wheelchair](image)
The geometry of the frame was measured and modelled in SolidWorks and imported into an assembly based on the experimental set-up described previously. This assembly was imported into ANSYS. Several geometric simplifications were made in the frame. All holes and screws were ignored for the model, as they increase the number of parts and therefore greatly increase the runtime of the analysis. Effects due to pneumatic tires were ignored for this study, and as such, wheels and tires were not modelled. Filler material added during the welding process and any change in material properties due to the heat-affected zone were not modelled.

**Material Characterization**

The Quickie Q7 chair is made from aluminum series 7005. The footplate, a separate part fitted into the frame tubing, was made of 6061 T6 aluminum, a common material for footplates. Relevant material properties are listed in Table 2.

**Table 2 Material Properties of 7005 T6 Aluminum. Source: MatWeb Metal Material Data Sheets [44]**

<table>
<thead>
<tr>
<th></th>
<th>7005 T6 Aluminum</th>
<th>6061 T6 Aluminum</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Ultimate Tensile Strength</strong></td>
<td>350 MPa</td>
<td>300 MPa</td>
</tr>
<tr>
<td><strong>Young’s Modulus</strong></td>
<td>72.0 GPa</td>
<td>69 GPa</td>
</tr>
<tr>
<td><strong>Yield Strength</strong></td>
<td>290 MPa</td>
<td>300 MPa</td>
</tr>
</tbody>
</table>

**Loading and Boundary Conditions**

In the static studies, the rear boundary condition was defined using a cylindrical support to allow rotation about the Z-axis. This reflected the physical frame, which was able to rotate about the rear axle but not translate in any direction.

For the front boundary condition, it was determined that using a simple frictionless support along the base of the add-on would have been insufficient to represent realistic deformations that result from a wheel and axle because the frictionless support did not
allow for rotations. As such, the axle and wheels were modelled, and a revolute joint was used between the main body of the add-on and the wheel axle of the add-on, which can be seen in Figure 22. A frictionless support was used at the base of the add-on wheels. This allowed for more realistic rotation and forward translation at the front of the model. The boundary conditions input into ANSYS are shown in Figure 23.
The magnitude and direction of the loading applied to the frame was dependent on the study being run. These studies are described in further detail in Section 3.4.

3.3.3.3 Processing and Post-Processing in ANSYS

For the static study, the ANSYS static structural module was used. The selected metric for assessment was the von Mises stress.

3.3.4 Physical Validation Methods

As can be seen in Figure 24, two strain gauges were bonded to the vertical tubes of the footplate in a half-bridge configuration. A Wheatstone bridge circuit was set up and connected to a digital multimeter to display changes in voltage due to bending loads.

Figure 24 Strain Gauges Mounted on the Wheelchair Footplate
3.3.4.1 Wheatstone Bridge Configuration

For this project, two EA-13-060WR-120 strain gauges (Micro-Measurements, Wendell, USA) were selected. The reasoning for selecting these strain gauges can be found in Appendix B. The strain gauges were wired in a half-bridge Wheatstone bridge configuration, a standard configuration for assessing bending stress. A diagram of this configuration is shown in Figure 26.

Figure 26 Half Bridge Configuration of the Wheatstone Bridge. A voltage is applied across A and C, and the potential difference V is measured across B and D.
3.3.4.2 Strain Gauge Calibration

Bending was determined to be the primary stress expected at the location of the strain gauge. The system was calibrated for bending stress using the setup shown in Figures 27 and 28. This was achieved by clamping one end of the footplate, and systematically applying loading to the opposite end. This load was used to calculate bending stress at the location of the strain gauge. For each load, the output voltage was recorded. A more detailed description of the calibration calculations is given in Appendix C.

Figure 27 Strain Gauge Calibration

Figure 28 Schematic of the Strain Gauge Calibration
3.3.4.3 Method of Applied Loading

For the static loading, sandbags were used to apply the required load. Each bag was weighed before being applied. A 2”x4” plank of wood was used to more precisely control the location of the load on the frame. This setup is shown in Figure 29.

![Applied Load using Sandbags](image)

Figure 29 Applied Load using Sandbags
Chapter 4:

Results

In this chapter, the results from the development and physical validation of the static FEA model, the subsequent static studies exploring variable customizations based on this model, and the dynamic loading study are presented.

4.1 Description of Studies Performed

This section describes the loading and boundary conditions for studies that were run, as well as justification for each type of study. The studies can be grouped into two main types: physical validation studies and assessment of design variable studies.

4.1.1 Physical Validation Studies

The purpose of the initial set of experiments performed on both the FEA and the physical model was to validate the behaviour of the FEA model using the physical model. Two main studies based on a simple downward static load were used. The first study consisted of a downward static load, shifted horizontally along the top tube to five locations. The strain gauge readings were recorded for each location, and the FEA model was probed at the corresponding location. The second study assessed a centred static load with an increasing load. These studies are described in further detail in Section 3.5.1. These studies were used to validate the FEA model because they were simple with predictable outcomes.

4.1.1.1 Increasing Distance Study

The purpose of this study was to validate the behavior of the FEA model through systematically moving a static load away from the rear wheel axle and towards the footplate. The study set-up is shown in Figure 30. The distance $d$ represents the distance the load was moved from the rear axle.
The distances from the rear axle that were assessed are shown in Table 3.

**Table 3 Distances used in First Physical Validation Study**

<table>
<thead>
<tr>
<th>Distance from Axle (mm) $d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
</tr>
<tr>
<td>150</td>
</tr>
<tr>
<td>200</td>
</tr>
<tr>
<td>250</td>
</tr>
<tr>
<td>300</td>
</tr>
</tbody>
</table>

### 4.1.1.2 Increasing Load Study

The purpose of this study was to successively increase the weight of the static downward load. The loads were approximately based on the ISO 7076-8 standard for user weight classes: 25 kg, 50 kg, 75 kg and 100 kg [45]. The load was applied 200 mm from the wheel axle. The schematic can be seen in Figure 31.
The loads assessed are shown in Table 4.

Table 4 Loads used in Second Physical Validation Study

<table>
<thead>
<tr>
<th>Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>245</td>
</tr>
<tr>
<td>489</td>
</tr>
<tr>
<td>733</td>
</tr>
<tr>
<td>978</td>
</tr>
</tbody>
</table>

4.1.2 Variable Analyses

The next set of analyses included static studies exploring several common customizations and configurations of manual wheelchairs, to assess if particular variables had a more significant impact on stresses in the frame when used with the FreeWheel. For example, when purchasing a wheelchair, users can decide what angle they want between the horizontal portion of the frame and the vertical portion of the frame; this is often referred to as the “frame bend” and includes 70, 80 and 90 degree bends as common angles. The design iterations considered in this thesis included frame bend angle, footplate tube
thicknes, footplate tube diameter, rear axle distance, and user weight. Schematics of these design iterations can be seen in Figure 32. These design variables were explored in the FEA model only and are summarized in Table 5.

During the design variable analyses, a static, downward load was used. The distribution of the loading across the top of the frame and footplate was based on ISO test dummies. The ISO test dummies are intended “to provide an appropriate total load mass, to approximate the mass distribution of a human occupant, to avoid damage to the wheelchair, to be durable and to be inexpensive to manufacture”[45]. Modelling the loads in this way was used to represent more realistic loads on the frame.

ISO 7176-11 specifies the location of the overall center of mass of test dummies for given dummy masses ranging from 25 to 100 kg and the masses of the three body segments: a torso segment, thigh segment, and lower leg segment. The segments have associated loading pads and may also have rigid spacers between the segments and loading pads, which were ignored for this analysis.
For a 100 kg dummy, the following locations for the center of mass and the magnitude of the masses are specified:

Table 5 Location of Centre of mass of 100 kg test dummy segments relative to the rear backrest of the wheelchair.

<table>
<thead>
<tr>
<th>Location of CoM</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>x (mm)</td>
<td>y (mm)</td>
</tr>
<tr>
<td>m_torso</td>
<td>145</td>
</tr>
<tr>
<td>m_thigh</td>
<td>346</td>
</tr>
<tr>
<td>m_lowerleg</td>
<td>493</td>
</tr>
<tr>
<td>Total</td>
<td>235</td>
</tr>
</tbody>
</table>

In this table, the y-axis corresponds to the back-support reference plane, and the x-axis corresponds to the seat reference plane. The origin is referred to as the “seat reference point”.

The x-dimensions were used to define the centre of the three loads defined for the three segments of the ISO test dummies. The y dimension was ignored for the downward static load.

4.1.3 Dynamic Loads Study

The purpose of this study was to characterize typical use and the resulting loading conditions on the wheelchair and FreeWheel using accelerometers, considering both a curb impact study and a study examining ten minutes of use.
4.1.3.1 Scenario Definitions

10 minutes of “typical” use was modelled for five user cases. In scenario A, the user spends 8 minutes on the sidewalk and 2 on the grass, and hits two curbs. In scenario B, the user spends 5 minutes on the sidewalk and 5 on the grass and hits two curbs. In scenarios C, D and E, the user spends 2 minutes on the sidewalk, 8 minutes on the grass, and hits 4, 10 and 20 curbs, respectively. These scenarios are described in Table 6.

Table 6 Defined Scenarios for Estimated Cycles to Failure

<table>
<thead>
<tr>
<th>Cycles per 10 Minutes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sidewalk Crack</td>
</tr>
<tr>
<td>-------------</td>
</tr>
<tr>
<td>A</td>
</tr>
<tr>
<td>B</td>
</tr>
<tr>
<td>C</td>
</tr>
<tr>
<td>D</td>
</tr>
<tr>
<td>E</td>
</tr>
</tbody>
</table>

4.1.4 Summary of Studies

A summary of the studies described above is outlined in Table 7.
### Table 7 Summary of Studies Performed

<table>
<thead>
<tr>
<th>Physical Validation Studies</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>COG of Load Study</strong></td>
<td>A 1000 N load is applied at an increasing horizontal distance from the rear wheel axle.</td>
</tr>
<tr>
<td><strong>Increasing Load Study</strong></td>
<td>An increasing downward load is applied to the top portion of the wheelchair frame.</td>
</tr>
<tr>
<td><strong>Variable Analysis</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Geometric</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Footplate- Tube Diameter</strong></td>
<td>Several tube diameters are explored in the footplate.</td>
</tr>
<tr>
<td><strong>Footplate- Tube Thickness</strong></td>
<td>Several tube thicknesses are explored in the footplate.</td>
</tr>
<tr>
<td><strong>Frame Bend Angle</strong></td>
<td>The bend angle between the vertical and the horizontal portions of the frame are assessed.</td>
</tr>
<tr>
<td><strong>Rear Axle Position</strong></td>
<td>The rear axle is adjusted in reference to the back of the frame.</td>
</tr>
<tr>
<td><strong>Loading Conditions</strong></td>
<td></td>
</tr>
<tr>
<td><strong>User Mass</strong></td>
<td>The mass of the user is increased.</td>
</tr>
<tr>
<td><strong>Preliminary Fatigue Analysis Study</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Fatigue</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Frontal Impact</strong></td>
<td>A cycle is defined as the wheelchair hitting a small bump and is used to estimate cycles to failure for typical use.</td>
</tr>
</tbody>
</table>
4.2 Physical Validation Results

4.2.1 Increasing Distance Study

The purpose of this study was to systematically move the load away from the rear axle to evaluate the stress at the strain gauge in the FEA model compared to the physical model. In both the increasing distance and increasing load studies, the rear axle was set up at 120 mm from the back of the chair. In the FEA model, only the normal stress was considered as the shear stress was considered insignificant. The results of this study, as can be seen in Table 8 and Figure 33, compared the calculated bending stress based on the output reading, the FEA results and the results from hand calculations described in Appendix A. Confidence bands on the physical testing are shown indicating a 95% confidence interval on the calculated regression line. As can be seen, as the load was moved away from the rear axle, the stress in the footplate at the strain gauge location increased proportionally.

The normal stress was used in the validation of the FEA model. At this point, it was found that the shear stresses were much smaller than the normal stresses. Further assessment of normal and shear stresses found in the footplate can be found in Appendix D and represents a direction for future work.

Table 8 Increasing Distance Physical Validation Results

<table>
<thead>
<tr>
<th>Distance from Rear Axle (mm)</th>
<th>100</th>
<th>150</th>
<th>200</th>
<th>250</th>
<th>300</th>
</tr>
</thead>
<tbody>
<tr>
<td>FEA Normal Stress (MPa)</td>
<td>26.4</td>
<td>40.7</td>
<td>68.7</td>
<td>90.4</td>
<td>112.0</td>
</tr>
<tr>
<td>Physical Testing (MPa)</td>
<td>21.3</td>
<td>55.0</td>
<td>60.2</td>
<td>92.4</td>
<td>114.4</td>
</tr>
<tr>
<td>Hand Calculations (MPa)</td>
<td>31.9</td>
<td>47.8</td>
<td>63.8</td>
<td>79.7</td>
<td>95.7</td>
</tr>
</tbody>
</table>
Increasing Distance Physical Validation Results

A linear regression on the physical data found a slope of \(0.447\) and y-intercept of \(-20.78\). The standard error for these values was found to be \(0.0021\) and \(94.31\), respectively. These results give 95% confidence interval values of <0.3, 0.6> for the slope and <\(-51.69, 10.13\)> for the intercept.

A linear regression on the FEA results found a slope of \(0.447\) and a y-intercept of \(-20.78\). Given that the slope falls within the confidence interval of the physical testing results, it was determined that this study supported the physical validation of the FEA model.

4.2.2 Increasing Load Study

As described in further detail in Chapter 3, the purpose of this study was to systematically increase the load applied to a fixed point on the wheelchair to evaluate the stress in the strain gauge as the load was increased. The stresses found in the FEA model, the physical model, and the hand calculations for the increasing load study are summarized in Table 9 and displayed graphically in Figure 34. These results show that as the load is increased,
the stress at the strain gauge is increased proportionally in both the FEA and the experimental models.

Table 9 Increasing Load Physical Validation Results

<table>
<thead>
<tr>
<th>Load (N)</th>
<th>245</th>
<th>489</th>
<th>733</th>
<th>978</th>
</tr>
</thead>
<tbody>
<tr>
<td>FEA Normal Stress (MPa)</td>
<td>17.1</td>
<td>34.0</td>
<td>52.1</td>
<td>66.6</td>
</tr>
<tr>
<td>Physical Testing (MPa)</td>
<td>13.0</td>
<td>28.0</td>
<td>44.4</td>
<td>57.9</td>
</tr>
<tr>
<td>Hand Calculations (MPa)</td>
<td>16.0</td>
<td>31.9</td>
<td>47.8</td>
<td>63.8</td>
</tr>
</tbody>
</table>

A linear regression on the experimental data in the increasing load study found a slope of 0.0618 and a y-intercept of -1.97. The standard error for these values was found to be 0.0001 and 1.17, respectively. These results give 95% confidence interval values of <-0.0549 – 0.0688> for the slope and <-6.64, 2.68> for the intercept. A linear regression on the FEA results gave a slope of 0.0648 and y-intercept of 0.6751. While the intercept fell outside of the confidence range, the slope was within it. It was therefore determined that these experimental values also supported the validation of the FEA study.
4.3 Variable Studies

As described in Chapter 3, the motivation behind these studies was to compare the stresses found in the structure for five different design variables that represent common customizations: rear axle distance, footplate tube thickness, footplate tube diameter, frame bend angle, and user mass.

4.3.1 Summary

The location of the rear axle and the footplate tube thickness had the most significant effects on the output von Mises stress in the structure. These results indicate that changing these two variables caused the greatest fluctuation in stress at the footplate. Of the variables explored, a decreased footplate tube thickness also created the highest stresses noted in this study- when the footplate thickness is decreased to 1.5 mm, the output von Mises stresses reached 184 MPa.

It should be noted that in determining which design variables are most important to minimize stress, other factors must be considered. For example, there are other factors that OTs use to determine the optimal rear axle distance in wheelchair set-up for a particular user. Adjusting the rear axle distance, and therefore the location of the center of gravity, impacts stability, ease of propulsion, posture, and the ability to do a wheelie to traverse curbs. In contrast, changing the tube thickness in the footplate has little impact on the stability and comfort for the user, and therefore represents a relatively simple change to make. Cost is also a consideration. In Canada, the cost of the average footplate ranges on the order of several hundred dollars. Compared to a wheelchair, which can cost upwards of several thousand dollars, it is a relatively easy design modification to a frame model as often different footplate styles and sizes are interchangeable in the same frame. Even though the rear axle distance is one of the most sensitive design variables, the dimensions of the footplate may be an easier adjustment to make. The importance and significance of the sensitivity of each design variable is discussed in further detail in Chapter 5.
4.3.2 Rear Axle Distance Study

The rear axle distance relative to the back of the chair is commonly adjusted to change the load distribution between the front and rear wheels, which affects stability and comfort for the user. In this study, the distance between the rear axle and the backrest was adjusted from 40 mm to 120 mm by a change of 20 mm each iteration. In the model used in this research, an 80 mm rear axle distance corresponds to an “80-20” configuration: 80 percent of the load is carried by the rear wheel, and 20 percent is carried by the front. Figure 35 shows the maximum von Mises stress as a function of the rear axle distance and Table 10 provides the maximum von Mises stress compared to the rear axle distance as well as the equivalent load distribution. The calculations for the relationship between the load distribution and the rear axle distance can be found in Appendix E. These results indicate that a more forward rear axle distance decreases the von Mises stress.

![Figure 35 Output von Mises Stress vs. Rear Axle Distance](image)

\[ y = -0.9589x + 139.6 \]

\[ R^2 = 0.9933 \]
Table 10 Output von Mises Stress vs. Rear Axle Distance

<table>
<thead>
<tr>
<th>Rear Axle Distance (mm)</th>
<th>Equivalent Load Distribution</th>
<th>Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>76.4 - 23.6</td>
<td>99.52</td>
</tr>
<tr>
<td>60</td>
<td>78.1-21.9</td>
<td>86.07</td>
</tr>
<tr>
<td>80</td>
<td>80-20</td>
<td>60.57</td>
</tr>
<tr>
<td>100</td>
<td>81.6-18.4</td>
<td>43.25</td>
</tr>
<tr>
<td>120</td>
<td>83.4-16.6</td>
<td>25.04</td>
</tr>
</tbody>
</table>

The percent change for the input variable was 20 mm, or 25% of the initial value of 80 mm. The resulting change in output von Mises stress was approximately 19.2 MPa, a 77% change from the initial output stress of 60.6 MPa. The resulting ratio of output change to input change is 3.06. These results are summarized in Table 11.

Table 11 Percent Change Output to Input for Rear Axle Distance

<table>
<thead>
<tr>
<th>Input</th>
<th>Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rear Axle Distance</td>
<td>Stress</td>
</tr>
<tr>
<td>mm</td>
<td>MPa</td>
</tr>
<tr>
<td>Change</td>
<td>20</td>
</tr>
<tr>
<td>Initial Value</td>
<td>80</td>
</tr>
<tr>
<td>Percent Change</td>
<td>0.25</td>
</tr>
<tr>
<td><strong>Ratio</strong></td>
<td><strong>3.06</strong></td>
</tr>
</tbody>
</table>

50
The location of the maximum output von Mises stress was found in the footplate, approximately at the intersection of the D-shape portion of the footplate and the main footplate tubing. This location can be seen in Figure 36.

![Figure 36 Location of Maximum Stress in Rear Axle Study](image)

The FEA software showed the deformations, scaled by a factor of 48, and areas of high von Mises stress in Figure 37. Areas of relatively low von Mises stress appear in dark blue, and higher stress appear in light blue and green. The footplate and rear portion of the main frame had notably higher levels of stress.

![Figure 37 Von Mises stress for 80 mm Rear Axle Distance. Deformation is magnified by a factor of 48.](image)
4.3.3 Frame Bend Angle Study

The motivation behind this study was to assess the angle between the horizontal and vertical portions of a cantilever frame, which is an important customization a user or OT will choose when purchasing a new wheelchair frame. Similarly to the rear axle distance, the frame bend angle affects stability and comfort for the user. In this study, the frame bend angle in the model was altered from 70 to 90 degrees by 5-degree increments. Figure 38 and Table 12 show the results of the maximum von Mises stress as a function of the frame bend angle. These results indicate that as the bend angle is brought closer to 90 degrees, which also indicates the front of the frame is in a more vertical position, the maximum stress in the structure decreases.

![Graph showing von Mises Stress vs. Frame Bend Angle](image)

**Figure 38 Output von Mises Stress vs. Frame Angle (degrees)**
### Table 12 Output von Mises Stress vs. Frame Angle (degrees)

<table>
<thead>
<tr>
<th>Frame Bend Angle</th>
<th>Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>90.3</td>
</tr>
<tr>
<td>75</td>
<td>85.3</td>
</tr>
<tr>
<td>80</td>
<td>80.2</td>
</tr>
<tr>
<td>85</td>
<td>77.3</td>
</tr>
<tr>
<td>90</td>
<td>74.5</td>
</tr>
</tbody>
</table>

The results of this study found that for each 5-degree change in the frame bend angle, the output stress changed by 3.96 MPa, which corresponds to a change of 5.3% output for an input of 5.6%, or a ratio of 0.95. These results can be seen in Table 13.

### Table 13 Percent Change Output to Input for Frame Bend Angle

<table>
<thead>
<tr>
<th>Input</th>
<th>Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frame Bend Angle</td>
<td>Stress</td>
</tr>
<tr>
<td>mm</td>
<td>MPa</td>
</tr>
<tr>
<td>Change</td>
<td>5</td>
</tr>
<tr>
<td>Initial Value</td>
<td>90</td>
</tr>
<tr>
<td>Percent Change</td>
<td>0.06</td>
</tr>
<tr>
<td>Ratio</td>
<td>0.96</td>
</tr>
</tbody>
</table>
The maximum stress for all angles was found to be at the footplate near the weld location, as can be seen in Figure 39. Deformations and maximum von Mises stress are shown in Figure 40.

![Figure 39 Location of Maximum Stress in Frame Bend Angle Study](image1)

![Figure 40 Von Mises Stress for 70 degree frame bend angles in the Frame Bend Angle Study. Deformation is scaled 28x](image2)

### 4.3.4 Footplate Tube Thickness Study

The motivation behind the footplate tube thickness study was to assess the impact of changing the wheelchair footplate with a footplate of the same diameter with a thicker wall. In this study, the footplate tube thickness was adjusted from 1.5 mm to 2.5 mm by .25 mm increments. The results of this study are presented in Figure 41 and Table 14.
These results show that increasing the tube thickness decreases the von Mises stress in the wheelchair.

![Von-Mises Stress vs. Tube Thickness (mm)](image)

**Figure 41 Output von Mises Stress vs. Footplate Tube Thickness (mm)**

<table>
<thead>
<tr>
<th>Tube Thickness (mm)</th>
<th>Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>184.37</td>
</tr>
<tr>
<td>1.75</td>
<td>143.57</td>
</tr>
<tr>
<td>2</td>
<td>60.57</td>
</tr>
<tr>
<td>2.25</td>
<td>46.72</td>
</tr>
<tr>
<td>2.5</td>
<td>60.55</td>
</tr>
</tbody>
</table>

**Table 14 Output von Mises Stress vs. Footplate Tube Thickness (mm)**

The maximum stress for all angles was found to be at the footplate near the weld location, as shown in Figure 42. The deformations, scaled to a factor of 48, and von Mises stress can be seen in Figure 4.11.
Figure 42 Location of Maximum Stress for Footplate Tube Thickness Study

Figure 43 Deformations for Footplate Tube Thickness Study with 1.5 mm Thickness, scaled to a factor of 48.

4.3.5 Footplate Tube Diameter Study

The motivation of this study was to assess changing the wheelchair footplate with a footplate of several diameters but with the same 2 mm thickness. In this study, the footplate tube diameter was increased from 15 mm to 25 mm by 2.5 mm increments. The results of this study are presented in Figure 44 and Table 15. These results show that increasing the diameter decreases the stress in the footplate. It is important to note that changing the tube diameter in the CAD geometry also requires changing the geometry in
the wheel insert in the main frame and in the front attachment, which has an impact on the results.

The maximum stress for all angles was found to be at the footplate near the weld location. Von Mises stress and deformations can be seen in Figure 45.
4.3.6 User Mass Study

The motivation of this study was to assess the effect in increasing user mass on the stresses in the footplate. In this study, the user mass was increased for four ISO dummy masses: 25 kg, 50 kg, 75 kg, 100 kg, and 125 kg corresponding to a 25% change from the initial 100 kg load. The results can be seen in Figure 46 and Table 16. These results show that increasing the mass increased the stress in the footplate linearly.
As can be seen in Table 17, the results of this study found that the maximum von Mises stress in the structure increased by an average of 15 MPa for each 25 kg change in user load. This corresponds to a 25% change in output for each 25% change in input. The ratio of output change with respect to input was 1.0.
Table 17 Percent Change Output to Input for User Mass

<table>
<thead>
<tr>
<th>Input</th>
<th>Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>User Mass</td>
<td>Stress</td>
</tr>
<tr>
<td>kg</td>
<td>MPa</td>
</tr>
<tr>
<td>Change</td>
<td>25.0</td>
</tr>
<tr>
<td>Initial Value</td>
<td>100.0</td>
</tr>
<tr>
<td>Percent Change</td>
<td>0.25</td>
</tr>
<tr>
<td><strong>Ratio</strong></td>
<td><strong>1.00</strong></td>
</tr>
</tbody>
</table>

The maximum stress for all angles was found to be at the footplate near the weld location, shown in Figure 48. Von Mises stress and deformations can be seen in Figure 49.

![Figure 48 Location of Maximum Stress for User Mass Study](image)
4.4 Dynamic Load and Fatigue Estimation

The motivation of this study was to estimate dynamic loads representing typical obstacles experienced by the front of the wheelchair during normal use, including a sidewalk crack, a small rock, and a curb. These were used to give a rough estimation of hours to failure.

4.4.1 Estimations of Cycles to Failure

The loads were input into a transient analysis and the maximum von Mises stresses were noted. Using an S-N curve for 6061 T6 aluminum alloy, as can be seen in Figure 50, these stresses were used to determine the number of cycles to failure.

In this thesis, the von Mises stress was used to make an estimation of the fatigue life of the material. The von Mises stress is a measure of strain energy and is used to estimate failure under multiaxial loading. Under the maximum distortion energy theory, the von Mises stress that will cause a material to fail under multi-axial loading will be equal to the strain energy at which the material will fail in uniaxial loading.

The type of loading- for example, uniaxial tension or biaxial bending- is specified for a given S-N curve. The S-N curve used in this research was for a uniaxial tensile stress state. This S-N curve was used because it was not practical to find an S-N curve that would accurately provide the number of cycles to failure due to the complexity of the loading and geometry in the footplate. For regions under multiaxial loading with both
normal and shear stress components, there can be high error in the results if loading data for the uniaxial case is used. This error can be increased further by complicated geometry, such as the joining of the tubes in the footplate, which prevents deformation in some directions. Furthermore, ignoring the heat affected zone due to welding can have a significant impact on the validity of these results. Considering these factors, it is recommended that further analysis be performed to make a more accurate assessment of the cycles to failure for the given loading conditions.

The FEA model was used to find stresses at a critical point in the model over time under the estimated frontal impact loads. A 10 msec time history of a load was used as the input into the model.
The overall damage was calculated based on Miner’s rule, a linear damage model. The linear damage rule can be expressed as:

$$D_i = \sum \frac{n_i}{N_{i,f}}$$

where $n_i$ is the number of cycles at a particular load determined from an S-N curve of the material, and $N_{i,f}$ is the number of cycles to failure for that load for a given material. Failure is predicted to occur where:

$$D_i = \sum \frac{n_i}{N_{i,f}} \geq 1$$

Calculations based on a linear damage model were then used to estimate the number of hours to failure. The estimated hours to failure are shown in Table 18 for each scenario for three different user masses: 500 N, 750 N and 1000 N. Using these methods, a heavier user and a rougher terrain resulted in greatly reduced hours to failure.

**Table 18 Hours to Failure for Three User Masses for Five Typical User Scenarios**

<table>
<thead>
<tr>
<th>Scenario</th>
<th>User Mass</th>
<th>500 N User</th>
<th>750 N User</th>
<th>1000 N User</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>32638</td>
<td>20641</td>
<td>3156</td>
<td></td>
</tr>
<tr>
<td>B</td>
<td>25976</td>
<td>14363</td>
<td>3079</td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>13936</td>
<td>7992</td>
<td>1591</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>6759</td>
<td>4384</td>
<td>1046</td>
<td></td>
</tr>
<tr>
<td>E</td>
<td>3637</td>
<td>2501</td>
<td>633</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Colour</th>
<th>Hours to Failure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Red</td>
<td>&lt;3000</td>
</tr>
<tr>
<td>Orange</td>
<td>3000 – 10 000</td>
</tr>
<tr>
<td>Green</td>
<td>&gt;10 000</td>
</tr>
</tbody>
</table>
Chapter 5:

Discussion

This chapter discusses the results of the analysis in the context of related literature and its potential implications and impact. Major findings of this thesis are discussed in Section 5.1 and 5.2. The impact of this research is discussed in Section 5.3, and study limitations are discussed in Section 5.4.

5.1 Major Findings of the Study

5.1.1 Summary

The use of a front-attached mobility add-on results in the footplate becoming a load-bearing component for a wheelchair and therefore experiencing stresses not otherwise present. Under downward static loading, this thesis found that the maximum stresses are highly dependent on the dimensions and configuration of the chair. The magnitude of maximum stress increases particularly as the footplate tube thickness decreases and as the rear wheel axle is moved toward the back of the chair.

In the FEA model used, the critical points in the structure were found in the footplate, very close to the welded joint between the main tube and the D-shaped footplate tube. These points require attention as they represent areas susceptible to failure by both yield and fatigue.

Dynamic studies characterized and estimated frontal impact loads associated with typical add-on use and made an estimation towards the hours to failure based on an Aluminum 6061 T6 S-N curve and Miner’s rule for accumulative damage. These results suggest that both user mass and terrain are important factors in determining the number of hours of use to failure of the wheelchair.

5.1.2 Model Validation

A validated FEA model of a wheelchair with a front-attached mobility add-on was developed. This model can be used to assess various design parameters and loading
conditions that a wheelchair is subjected to when used with an add-on. As with any FEA model, there are limitations to its validity. The FEA model was physically validated for downward static loads. The model assumes that the material behaviour is elastic and therefore the relationship between stress and strain is linear under Hooke’s law. Loads that would cause plastic deformation would not produce validated results using this model.

With consideration of its limitations, there are extensions and numerous further configurations and loads that can be explored using the FEA model that are not within the scope of this thesis. For example, by changing the material library, the maximum stress and factor of safety can be compared for several different materials with the same geometry. Different types of steel such as chromoly, a variety of aluminum alloys and plastic represent common material choices in the footplate and are valuable directions for future studies. Furthermore, in the case of aluminum, the type of alloy, whether the material is heat treated, and the impact of heat affected zones could also be input and compared.

5.1.3 Results of Static Studies

5.1.3.1 Areas of High Deformation

Viewing the wheelchair and add-on from the side, the structure sags inwards under the downward static loading applied by the static studies. In the static studies, the areas of highest deformation were found in the footplate. Significant deformations noted included bending in the vertical portion of the footplate and bending in the horizontal portion of the footplate. These deformations were also found in footplates that had plastically deformed following use with the FreeWheel. These deformations are shown in Figures 51-54.
Figure 51 Field Evidence of Bending Found in the Footplate

Figure 52 Deformations in the Vertical Portion of the Footplate.
5.1.3.2 Critical Stresses

The critical point, or location of maximum stress, was consistently found in the footplate near the weld, as reported in Chapter 4. The critical point represents the point location most likely to yield under high static stresses or fracture due to fatigue under cyclic loading. Under a simple 1000 N load, the highest stress found during this research was 184 MPa. This value is below the yield strength of 276 MPa for 6061 T6 aluminum. However, given that the loads caused by the user’s mass are dynamic and variable based on the user’s position and movements, it can be expected the stresses would often be
higher during normal use. Therefore, it is reasonable to expect that plastic deformation in the footplate is a possibility. This is again supported by field evidence of yield caused under static loading, as shown in Figures 51 - 54 above. Furthermore, one important factor not considered by this analysis is the effect of welding on the material properties in the footplate. Welding is a common manufacturing process during the manufacturing of D-shaped footplates. It is known that materials such as aluminum will lose up to 30% of their ultimate tensile strength during the welding process due to the formation of precipitates during the extreme temperature change. Some of this can be recovered through heat treating, however, often wheelchair footplates are not heat treated after welding.

Depending on the configuration of the design variables, the maximum von Mises stress in the footplate ranged from approximately 20-180 MPa under static loading. The high variability in stress in the structure suggests that design parameters have a significant impact on magnitude of maximum von Mises stress, and that there is potential to greatly minimize stresses in the footplate by adjusting the design parameters of the system.

To the best knowledge of the author of this thesis, critical stresses or failures in the footplate have not been noted by studies assessing wheelchair durability in the literature. Other wheelchair components are commonly cited for various types of chairs and loading conditions. Two examples of related literature that have looked at failures in manual lightweight wheelchairs include failures due to rugged terrain [33] and failures due to the addition of a suspension element [30]. In the rugged terrain of low resource environments, Mahtre et al. noted flat and cracked tires, wobbly rear wheels, bent frames, non-functional brakes, worn out bearings, torn seat covers, loose upholstery, collapsed cushions and rusting and loosening of several parts, caster failures, corrosion and resistance to wheelchair rolling [33]. KwarciaK et al. (2005) found that the modification of adding a suspension element reduced the number of cycles to failure and noted failures in the heat affected zone of welded locations in the Invacare A-6S 1 and A-6S 3 and Quickie XTR wheelchair models [30].

The knowledge of wheelchair component failures that are reported in academic literature is often used by manufacturers, designers and working ISO standards development
groups to improve wheelchair design. As there is currently little research on footplate failures, the importance of design parameters that impact stresses in the footplate when used with an add-on have the potential to be overlooked by designers, engineers and manufacturing standards working groups to make more durable wheelchairs.

5.1.4 Results of Design Variable Studies

The rear axle distance and tube thickness are important design parameters to consider, as they have high sensitivity as well as resulted in the highest loads under the applied loading given the type of range of the design parameters considered.

As the rear axle was moved posteriorly, it was found that the stress in the footplate increased. However, in determining the optimal rear axle distance for a user, there are several other implications to consider, including propulsion biomechanics, potential for injury, and user stability and comfort. These considerations are well-examined in the literature. For example, Boninger et al. [46] found that the rear axle distance was significantly correlated with the frequency of propulsion as well as the push angle at multiple speeds. A more anterior rear axle position has been found to be associated with reduced stability, as it increases the likelihood of backwards tips [47]. Therefore, while moving the rear axle towards the front of the chair contributes to minimizing stresses in the footplate and better overall propulsion performance, it may not be a practical adjustment for a user with concerns over stability and comfort.

A decreased tube thickness and a decreased tube diameter were both predictably found to be correlated with an increased maximum von Mises stress. Within the category of D-shaped footplates, which is one of the five main types of footplates identified by ISO 7176, the selection of tube thickness and diameter have little effect on the stability and comfort for the user. Rather, the choice of footplate dimensions is often selected to minimize cost and weight, so long as it fits with the desired frame and provides adequate material strength. The footplate tube thickness is therefore one of the easiest design variables to modify during wheelchair selection and configuration, as it is a separate component from the main frame, has lower cost than the rest of the chair, and has little effect on user biomechanics, comfort or stability. It is also therefore relatively to change the material of the footplate, such as to chromoly.
The tube diameter is also a parameter that can be changed within the dimension defined by the frame tube. However, changing the tube diameter may also require that the wheel insertion component be changed to match the diameter of the footplate tube, contributing to an increased cost.

The results of the FEA indicate that as the frame bend angle is decreased, or the vertical portion of the frame is brought more horizontal, the stress in the footplate increases. Therefore, using a wheelchair with a frame bend angle of 90° is one method of minimizing stresses in the footplate. As with the rear axle distance, the frame bend has several other implications on the user’s comfort and stability that also need to be considered when making this customization. Having a large frame angle allows wheelchair users to tuck in their lower legs, which increases maneuverability. On the other hand, having a smaller frame angle provides advantages to some users, such as those who have plantar-flexion contracture in the ankles and may need a lesser frame angle to prevent their feet sliding off the footplate [48].

As the user mass is increased, the stress in the footplate increases proportionally. The importance and effect of user weight on the lifespan of the chair is well-established; the strength tests under the ISO 7176-8 standards are defined for five different user weight classes to ensure the wheelchair has an approximate five-year lifespan for each category of user weight [45]. However, currently, the definition of the user weight classes under the ISO standards does not consider the potential use of a mobility add-on with the wheelchair. As such, it is worth further investigation if a wheelchair designed for a particular weight category is still adequate if it is intended to be used with a mobility add-on.

In summary, based on the static studies performed during this thesis, it can be recommended to designers to reduce stresses in the footplate by increasing tube thickness and by considering the rear axle placement, if appropriate for the user. However, it is also recommended that further design parameters and customizations be explored to optimize wheelchair durability.
5.1.5 Fatigue due to Frontal Impact Dynamic Loading

Using estimated time-history loads meant to represent frontal impacts, the dynamic studies assessed and characterized frontal impact loads associated with typical add-on use, including sidewalk cracks, small rocks and sidewalk curbs. An estimation of hours to fatigue failure for five user scenarios found that the user mass and terrain greatly affected the lifespan of the chair. However, accurately quantifying the hours to fatigue failure requires a more detailed investigation and physical validation.

5.1.5.1 Importance of Frontal Impact when using a Front-Attached Add-On

The loading during a frontal impact, such as hitting a rock or curb bump, is an important change in loading conditions associated with a front-attached add-on for two reasons. First, it introduces a horizontal force into the footplate when it hits a curb or wall in situations where non-add-on users would typically use a wheelie technique to lift the front of the chair over the obstacle [49]. Second, one of the intended uses of front-attached add-ons is to more easily allow users to roll over curbs and rocks, so it greatly increases the frequency of such events.

5.1.5.2 Fatigue Estimation

For five typical scenarios, the preliminary fatigue estimation found that both increased impacts and user mass greatly reduced the number of cycles to failure. Fatigue failures are important to consider as it is well-known that fatigue causes a high percentage of component failures in transportation structures, such as vehicle chassis, bicycle frames, and wheelchairs, as such structures are subject to heavy vibration and randomly varying loads. These types of loads allow stresses well below the yield strength of particular materials like aluminum to cause failure over time.

While fatigue failures are commonly reported in the literature through ISO-based physical testing, such as work by Baldwin [31], Kwarciaik [30], Fitzgerald [29] and Todd [50], this thesis attempted to estimate dynamic impact loads experienced by the user during common impact events. The question of how to estimate realistic loads experienced by a wheelchair is difficult and is an area being explored in the literature
through quantifying loads such as those done using accelerometers [32], strain gauges [51] and through field studies identifying and tracking typical behaviours [52]. These studies aim to quantify dynamic reaction force, moment and acceleration data of the wheelchair during normal use.

A complicating factor in determining how the mobility add-on affects the lifespan of the chair is the variability in user lifestyle and daily activities. To withstand a 5-year lifespan, it is important to consider what the typical daily activities, and therefore stresses in the chair frame, are for a particular user. Considerations include vibration stresses from rolling on a relatively smooth surface such as concrete, larger transient events encountered on rocky terrains or curbs, as well as accounting for the distances travelled in a typical day. Generally, stresses due to small bumps such as sidewalk cracks would constitute a much higher number of cycles to failure. However, they also occur at a high frequency, and therefore have the potential to contribute to the accumulated fatigue damage of the chair. Higher impact events, including traversing on and off curbs in urban settings, and rocks, logs, or ledges in off-road settings, would occur less frequently but cause greater accumulated damage per event. The frequency of these events greatly depends on the lifestyle and activities enjoyed by the user. Within this, an obvious factor is how often an add-on is used. More accurate estimation of the magnitude, duration and frequency of dynamic loads is therefore an important direction for future work to better quantify the hours to failure for use with a device.

5.2 Impact of Research

Wheelchair users often try to optimize their wheelchairs for weight, cost, and maneuverability, along with secondary goals such as aesthetics. Designers work to achieve this optimization by adjusting the geometric, material and structural aspects of their designs, often trusting the ISO strength standards to meet the lifespan and durability requirements that are expected. Quantified knowledge of what stresses are occurring in the wheelchair when used with mobility add-ons can contribute to improved standards development, design decisions, and risk assessments for wheelchairs intended to be used with mobility add-ons. In addition to helping designers making manual wheelchairs more
durable and longer lasting when used with add-ons, this knowledge can also allow designers and manufacturers to make more innovative add-ons with confidence.

A second potential impact of this research is that it can provide a framework for occupational therapists and manual wheelchair users to consider the impact of configurations on the stresses in the footplate when using add-ons with manual wheelchairs. This can help assess the appropriateness of the use of the FreeWheel or other add-on, and provide guidance on considering modifications or reinforcements to the chair that could compensate for the newly introduced stresses.

5.3 Study Limitations

The following limitations were noted by this study:

**Estimation of Impact Loading:** One limitation in this study is the estimation of time history loading to represent the frontal impact loads during a curb impact in ANSYS. The assumed impact time was 10 msec and the forces were estimated to match the experimental output accelerations. As discussed, estimating dynamic loads is difficult and future work can therefore more thoroughly assess the time of impact considering the shape and stiffness of the system.

**Estimation of Cycles to Failure using an S-N Curve:** S-N curves for welded materials are specific to the type of loading and welding type and can be difficult to access. In the case of the heat affected zone for 6061 aluminum, the material properties, including the ultimate tensile strength, are known to decrease significantly. The post-welding heat treatment process can also considerably affect material properties. In some cases, this step is not performed for footplates, which leaves the material weaker. This thesis did not use an S-N curve that considered the impact of welding. Understanding the impact of welding on the material properties in the footplate will lead to greater accuracy in the predicted cycles to failure.
Chapter 6:

Conclusion

In this thesis, we presented a structural analysis using the finite element method to explore the effect of the addition of a mobility add-on on the loading conditions found in a manual lightweight wheelchair. We achieved this through the design of an FEA-based study examining vertical loading in both simulated and physical strain gauge-based studies, exploring design parameters in the FEA model, making a preliminary estimation of dynamic loads introduced by the mobility add-on, and finally considering how these forces could impact the cycles to failure under simple loading. In this chapter a summary of the contributions, limitations, and directions for future work are provided.

6.1 Summary of Contributions

The results of this thesis provide three main contributions. First, the results of a validated static finite element analysis quantified stresses in a Quickie Q7 wheelchair frame and footplate when used with a mobility add-on. This information has not previously been published in the literature and can therefore aid designers and manufacturers in improving the strength and durability of manual wheelchairs. Second, the effect of five design and structural parameters were assessed using the FEA software. Finally, a preliminary analysis of dynamic effects during typical use was conducted, with consideration on impact of dynamic loads on the lifespan of the wheelchair through a fatigue assessment.

In the validated static analysis, the following contributions were made:

- Design of an experiment aimed at quantifying the stresses in the footplate. The physical set-up is based on strain gauges and the simulation was set-up using ANSYS software
- Assessment of the validity of the FEA model using the experimental results
• Quantitative and qualitative assessment of the deformations and stress concentrations in the footplate as the result of use with a footplate-mounted add-on

In the design variable studies, the following contributions were made:

• Identification of several common customizations that users will make to the base model of the wheelchair, and design of several studies to assess the impact of these variables on the output von Mises stress

• Quantitative assessment of the sensitivity of these input variables on the output von Mises stresses in the horizontal portion of the footplate and in the vertical portion at the strain gauge location

Finally, in the fatigue estimation study, the following contributions were made:

• Design and completion of a preliminary study to assess common frontal impact loads, beginning a research direction to assess dynamic loading in manual wheelchairs that result from using a mobility add-on

6.2 Directions for Future Work

The following represent possible directions for future work:

Consideration of Vibrations Using a Modal Analysis: Much like vehicle chassis, wheelchairs are subject to vibration loads under many normal use conditions. Understanding the mode shapes and the resonant frequencies of the system is important in minimizing failures due to vibrations.

Validated Fatigue Testing and Analysis: The development and use of physical testing to validate the fatigue analysis would provide greater confidence to the results of the fatigue estimation and could contribute to the development of manufacturing standards designed to minimize fatigue failures over time.

Experimental Time History of Bending Stresses Using Strain Gauge Set-up: Another possible direction is to quantify the stresses in the footplate during dynamic loading using a data acquisition system to directly record the change in strain in the footplate over time. Simple adjustments to the current experimental set-up could be used to achieve this.
**Structural Design Optimization:** The combined use of FEA and sensitivity analysis are commonly used in optimizing structural design. The results of this research could be extended to consider the impact of many more of the structural design parameters in wheelchairs on output variables including von Mises stress, overall weight and cost.

**Field Studies Assessing Typical Use with a Mobility Add-On:** Understanding the typical duration and location of use of the mobility add-on is important in extending the results of a fatigue analysis to better estimate the lifespan of the chair in months or years. This type of work could be sensor-based, for example using accelerometers or a GPS system to track a user’s daily activities or based on qualitative assessments such as questionnaires to determine how often and for what types of activities an add-on is adopted.

**Development of ISO Testing Standards to Ensure Adequate Strength for Use with Add-ons:** Using the values for stresses as a baseline, further work could develop physical manufacturing tests to assess other models of wheelchairs when used with mobility add-ons to ensure quality assurance in commercial products.

**Development of Guidelines for OT’s and Distributors:** With increased knowledge of the impact mobility add-ons have on manual wheelchairs, guidelines could be created to help occupational therapists determine whether or not a mobility add-on is appropriate for a given wheelchair and user, given their activity level and expected wheelchair lifespan.

### 6.3 Conclusion

In closing, mobility add-ons increase the mobility capabilities of manual wheelchairs and represent a promising direction for commercial assistive devices for manual wheelchair user populations in the coming decade. However, as this research has shown, the addition of a mobility add-on changes the location and magnitude of stresses in the wheelchair frame. An understanding of the structural impact of these devices on manual wheelchairs is essential in optimizing the design of wheelchairs and developing manufacturing standards to ensure product quality is achieved.
References


Pushrim-Activated Power-Assisted Wheelchairs Using ANSI/RESNA Standards,”


Appendices

Appendix A: 2D Calculations

2D Calculations

The method of sections can be used to find the internal loading and stresses at two points—a point on the vertical portion of the footplate and a point on the horizontal portion of the footplate. These points are labelled G and W, respectively, on the figure below.

![Diagram of footplate with points G and W labeled]

Point G (Vertical Portion of the Footplate)

Calculation of Internal Loading

Support Reactions

The reaction forces from a two-dimensional drawing of the system. This system can be considered as a simply supported beam in two dimensions with a pin joint at the rear axle location and a roller at the front of the system.

The reaction forces can be calculated using the equations of equilibrium.
\[ \sum F = R_a + R_b - F = 0 \]
\[ \sum M_B = Fwc \times \text{dist}_a - R_a \times (\text{dist}_b) = 0 \]
\[ R_A = (Fwc \times \text{dist}_a)/(\text{dist}_b) \]
\[ R_B = F - R_A \]

**Free Body Diagram:**
Knowing the support reactions, an imaginary plane can be passed through point G, the location of the strain gauge. Using the method of sections, a free-body diagram of the portion of the system anterior to this point is shown.

In the X-Y plane:

![Free Body Diagram](image)

**Equations of Equilibrium:**
At this point, the bending (M), axial (P), shear (V) and torsion (T) internal loadings for points S and G can be calculated. Because there is no significant loading in the Z direction, at point S, torsion loads can be ignored at this step.

\[(\text{vertical}) \sum F = R_b - N\cos(\theta_2) - V\sin(\theta_2) = 0 \]
\[(\text{horizontal}) \sum F = N\sin(\theta_2) - V\cos(\theta_2) = 0 \]
\[ \sum M_G = M_G - R_B\cos(\theta_2) \times (d_\perp) = 0 \]
\[ R_b - 2V(\sin(\theta_2) + \cos(\theta_2)) = 0 \]

\[ V = \frac{R_b}{-2(\sin(\theta_2) + \cos(\theta_2))} \]

\[ N = -V\cos(\theta_2)/\sin(\theta_2) \]

\[ M_G = R_B\cos(\theta_2) \cdot (d_\perp) \]

*Calculation of State of Stress Combined Caused by Combined Loadings*

The stress component associated with each internal loading can be computed using simple beam equations.

**Normal Force**

\[ \sigma = \frac{N}{A} \]

**Shear Force**

\[ \tau = \frac{V}{A} \]

**Bending Moment**

\[ \sigma = \frac{M_y}{I} \]

where I is the moment of inertia for a hollow tube:

\[ I = \pi(R_o^4 - R_i^4)/2 \]

**Torsion**

\[ \tau = \frac{T_c}{J} \]

where J is the polar moment of inertia for a hollow tube:

\[ J = \pi(R_o^4 - R_i^4)/2 \]
The principle of superposition can then be used to determine the resultant normal and shear stress components.

**Point W (Horizontal Portion of the Footplate)**

*Calculation of Internal Loading*

*Free Body Diagram:*

At this point, bending and shear loads can be expected in the Z-Y plane and torsion can be expected in the X-Y plane.

In the X-Y plane:

In the Z-Y plane:
Equations of Equilibrium:

In the X-Y plane:

\[ \sum M_W = T_G - R_B \times (d_\perp) = 0 \]

In the Z-Y plane:

(\text{vertical}) \[ \sum F = V + \frac{R_A}{2} = 0 \]

\[ \sum M_W = M_W - R_A/2 \times (d_\perp) = 0 \]

\[ V = R_A/2 \]

\[ T_W = R_B \times (d_\perp) \]

\[ M_W = \frac{R_A}{2} \times (d_\perp) \]

The stress component associated with each internal loading can be computed using simple beam equations.
Normal Force

\[ \sigma = \frac{N}{A} \]

Shear Force

\[ \tau = \frac{V}{A} \]

Bending Moment

\[ \sigma = \frac{M_y}{I} \]

where I is the moment of inertia for a hollow tube:

\[ I = \pi (R_o^4 - R_i^4) / 2 \]

Torsion

\[ \tau = \frac{T_c}{J} \]

where J is the polar moment of inertia for a hollow tube:

\[ J = \pi (R_o^4 - R_i^4) / 2 \]

The principle of superposition can then be used to determine the resultant normal and shear stress components.
Appendix B: Strain Gauge Selection

Strain Gauge Selection

For this project, two EA-13-060WR-120 strain gauges were selected. The EA series of strain gauges is a standard series containing constantan foil with polyimide backing, primarily intended for general-purpose static and dynamic stress analysis. 13 is the self-temperature-compensation, or S-T-C, number, and is a standard selection for aluminum alloys. A 120 ohm resistance was selected, as it is the most commonly used strain gauges and there were no indications for specialized resistance. The gage factor of this strain gauge was 2.05 for aluminum
Appendix C: Strain Gauge Calibration

Strain Gauge Calibration

To calibrate the strain gauges, the base of the footplate was held by the clamp, and an increasing load was applied to the end for the plate. For each load, the output voltage was recorded. A schematic of this set-up is shown below.

To determine the stress at the strain gauge location for a given applied load, the bending equation was used:

\[ \sigma = \frac{My}{I} \]

where \( \sigma \) is the bending stress at the location in the beam, in MPa
\( y \) is the radius of the footplate tube
\( I \) is the polar moment of inertia
\( M \) is the bending moment calculated by:

\[ M = F \times d \]

In this case, \( F \) = the load in lb, converted to Newtons, and \( d = 0.104 \) m

The recorded output voltages for several increasing loads were obtained, and the stresses were calculated. This was used to establish a linear relationship between recorded voltage and corresponding theoretical stress.

The output for an example calibration is shown below.
In this example, the bending stress at the strain gauge would be equivalent to:

\[
\sigma = -0.0343V + 166.83
\]

where

V is the record voltage, in mV, and

\(\sigma\) is the computed bending stress in the footplate, in MPa
Appendix D: Normal and Shear Stress Probes in Footplate

To further analyze stresses in the footplate and support the validation of the FEA model, six points in the footplate were probed. The results found in this study are shown below. The loading used here was 1000 N using the ISO loading distribution.

The purpose of performing this study was to consider the proportions of the normal and shear stress components at points in the footplate. This work can be used to better assessing the validity of physical validation using strain gauges and the use of uniaxial S-N curves.

In the FEA model, the points shown in Figures X.1 and X.2 are shown. Probes were run for normal and shear stresses defined by the coordinate system shown in Figure X.3. Results are shown below.

Location of Probes in Footplate
Definition of Coordinate System
Normal, Stress and Shear Components at Points A, B, C, D, E and F in MPa

<table>
<thead>
<tr>
<th></th>
<th>Normal (Z) (MPa)</th>
<th>Normal Y (MPa)</th>
<th>Normal X (MPa)</th>
<th>Shear (XY) (MPa)</th>
<th>Shear (XZ) (MPa)</th>
<th>Shear (YZ) (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A- Center of Footplate</td>
<td>10.625</td>
<td>0.231</td>
<td>-1.374</td>
<td>-0.127</td>
<td>0.479</td>
<td>-0.073</td>
</tr>
<tr>
<td>B- D Shape Center</td>
<td>-0.803</td>
<td>0.968</td>
<td>2.239</td>
<td>0.134</td>
<td>-0.091</td>
<td>-0.398</td>
</tr>
<tr>
<td>C- Horizontal, Outside D Shape</td>
<td>-0.671</td>
<td>-0.085</td>
<td>0.456</td>
<td>-0.034</td>
<td>3.916</td>
<td>2.219</td>
</tr>
<tr>
<td>D- Vertical at Strain Gauge</td>
<td>-2.361</td>
<td>-26.685</td>
<td>-0.491</td>
<td>3.449</td>
<td>0.557</td>
<td>-0.542</td>
</tr>
<tr>
<td>E- At Weld</td>
<td>-8.368</td>
<td>-2.375</td>
<td>6.966</td>
<td>-1.390</td>
<td>15.490</td>
<td>0.644</td>
</tr>
<tr>
<td>F- At Weld</td>
<td>-0.194</td>
<td>3.340</td>
<td>-0.871</td>
<td>-0.667</td>
<td>0.339</td>
<td>15.887</td>
</tr>
</tbody>
</table>
Stress Components in Footplate

![Stress Components in Footplate](image-url)
Appendix E: Calculation of Relationship between the Rear Axle Distance and COG

The following results were calculated by SolidWorks:

Center of mass: (meters)

\[ X = -0.69 \]
\[ Y = 0.13 \]
\[ Z = -0.22 \]

The 100 kg ISO Dummy Mass Distribution. X location is defined from back of chair:

<table>
<thead>
<tr>
<th>Segment</th>
<th>Mass</th>
<th>X location</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torso</td>
<td>61</td>
<td>145</td>
</tr>
<tr>
<td>Thigh</td>
<td>31</td>
<td>346</td>
</tr>
<tr>
<td>Legs</td>
<td>8</td>
<td>493</td>
</tr>
</tbody>
</table>

The center of mass of the ISO test dummy is therefore:

\[ COM = \frac{(61 kg \times 0.145 m) + (31 kg \times 0.346 m) + (8 kg \times 0.493 m)}{100 kg} = 0.235 m \]

The combined center of gravity of the wheelchair and test dummy is:

\[ COM = \frac{(100 kg \times 0.235 m) + (5.87 kg \times 0.67 m)}{105.87 kg} = 0.259 m \]

Relationship between COG and Front-Rear Wheel Load Distribution
Given a load and a particular desired weight distribution, the reaction forces $R_a$ and $R_b$ can be determined.

For an 80-20 configuration,

\[ R_B = F \times 0.2 \]
\[ R_A = F \times 0.8 \]

Summing the moments around B, \( \text{dist}\_\text{rear axle} \) can be found:

\[
\sum M_B = F \times 0.777 \, m - R_a \times (1.306 - \text{dist}\_\text{rear axle} \times \cos(12^\circ)) = 0
\]

\[ \text{dist}\_\text{rear axle} = (1.306 - \frac{F \times 0.777 \, m}{R_a})/\cos(12^\circ) \]
The relationships found are:

<table>
<thead>
<tr>
<th>Load Distribution</th>
<th>Rear Axle Distance (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>0.26</td>
</tr>
<tr>
<td>95-5</td>
<td>0.23</td>
</tr>
<tr>
<td>90-10</td>
<td>0.18</td>
</tr>
<tr>
<td>85-15</td>
<td>0.14</td>
</tr>
<tr>
<td>80-20</td>
<td>0.08</td>
</tr>
<tr>
<td>75-25</td>
<td>0.02</td>
</tr>
<tr>
<td>70-30</td>
<td>-0.04</td>
</tr>
</tbody>
</table>