Department of Precision and Microsystems Engineering

Towards Topology Optimization for constructing Compliant Optical Mount Mechanisms by means of Additive Manufacturing

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Challenge the future

Towards Topology Optimization for constructing Compliant Optical Mount Mechanisms by means of Additive Manufacturing

by

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Abstract

Optical systems in high precision equipment become increasingly more complex due to the raising number of optical mounts used for mounting and positioning of optical components. Additional, the installation of electronics and inert gas purging systems to protect the optics against the abrasive optical beam extends the amount of components in the optical system, while keeping the surface area of the overall optical system as small as possible. Recent developments in optical systems lead to innovations of optical flexure mounts towards a more compact, accessible and user-specific system with integrated functionality at consistently high standards of optical stability.

Additive manufacturing, commonly named 3D printing, offers new possibilities of designing optical flexure mounts, since the layer-wise manufacturing approach allows the production of highly complex parts compared to traditional processes. Topology optimization is a mathematical design tool that can help to design these increasingly complex parts, such as the optical mount, by computing optimal material distributions for a given objective with a determined set of constraints and boundary conditions.

The thesis objective is bilateral: The first research objective investigates the design of compliant mechanisms for industrial optical mounts by means of additive manufacturing. The second research objective investigates the suitability of Topology Optimization for designing compliant mechanisms for optical mounts. This research is providing insights into the working principles of optical mounts and compliant mechanisms as well as topology optimization and additive manufacturing. The present case studies demonstrate and evaluate topology optimization design techniques for compliant structures and mechanisms. Further, the use of additive manufacturing for compliant optical mounts mechanisms designs is evaluated.

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Pieter Smorenberg
Delft, January 2017

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Chapter 1

Introduction

1.1 Optical mounts

Optical mounts are used as mounting and positioning device for various optical components [1] (i.e. lenses, gratings, filters,...). By adjusting the input screws of the optical mount, the angle or displacement of these optical components can be changed. Figure 1.1 represents such an optical mount that allows pitch and tilt of the attached optics. Optical mounts are often applied in high precision optical systems. Applications of these systems can be found in manufacturing equipment machines such as wafer inspection tools but also in communication satellites.

Figure 1.1: Newport stability top-adjustable kinematic optical mount for 1 inch optics. Retrieved from *https://www.newport.com/c/opto-mechanics*, Copyright Newport corp. (2015)

The overall performance of an optical system is critically affected by optical mounts. Particularly when selecting an optical mount for a specific application, ease of installation and maintenance as well as accuracy and adjustability of the optical mount in the system is essential for optimal functioning and reliable performance [2]. Optical misalignment from vibration, thermal drift or long-term mechanical creep leads to optical beam wandering during operation. Hysteresis in the mount mechanism crucially influences reproducibility and quality of the optical operation, mostly resulting in time consuming realignment procedures [1].

Optical systems in high precision equipment are increasingly complex due to the vast number of optical mounts and the possibility to install various electronics or additional systems for specific application purposes such as inert gas purging systems. Since the surface area of the overall optical systems remains as small as possible, recent developments in this field result in sophisticated systems, such as illustrated by the optical setup for quantum research of the Max-Planck-Institut fur Quantenoptik [3] (figure 1.2).

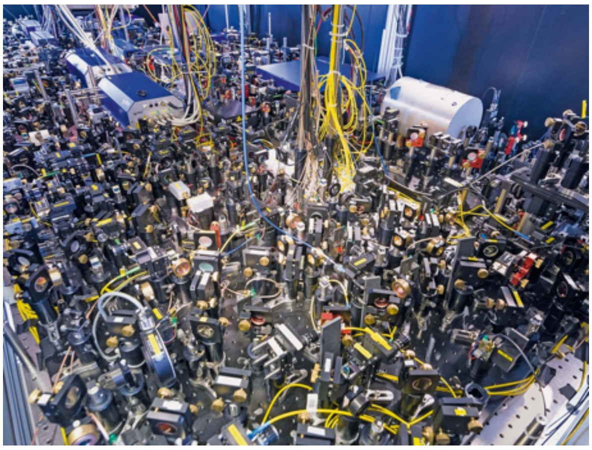

Figure 1.2: Optical table with experimental setup, at *'Max-Planck-Institut fur Quantenoptik'*. Retrieved from *https://www.mpg.de/9142418/magnetism-in-artificial-crystal*, Copyright MPinstitute (2010)

1.2 Novel design and manufacturing technology

The developments in optical systems and possible optical mount improvements emphasizes the necessity to design novel optical mount solutions with reduced surface areas, sufficient room for integrated functionality, improved accessibility and enhanced usability, whilst maintaining the high standards of optical stability.

Additive Manufacturing (AM), commonly named 3D printing [4] offers new possibilities to designing optical mounts. Compared to the traditional processes (substractive, forming), mechanical parts of significantly greater complexity can be produced due to the layer-wise manufacturing approach. AM provides significantly greater design freedom and enables manufacturing strategies that are closer to the optimum design than as it is the case in traditional processes [5]. This allows, for example, improvements of the basis frame structure of the optical mount as well as new forms of precision (compliant) mechanisms, actuating the optics of the optical mount. Nonetheless, these novel approaches require experienced designers with broad knowledge and specific expertise for 3D space and the constraints of AM [6].

The revolutionary engineered optical beam alignment module [7], specifically designed for AM at VDL-ETG during my internship, is a representative for the contribution of AM advantages to new innovative designs for optical flexure mounts. This optical module is compact, compared to existing solutions and allows two mirrors to be adjusted in a coordinated manner, offering separate adjustment for angular and linear displacements of an incoming optical beam. The key advantages compared to existing solutions are;

1. Angular and linear displacement are uncoupled, which saves adjustment time during installation or service.

2. Well accessible from one side, creating more space for other elements in the optical system.
3. The monolithic compliant structure (introducing flexures) preventing stick-slip and play in the adjustment mechanism, making optical movements better to predict.
4. Manufacturing tolerances in the mechanism between parts are non-excistent, saving part alignment problems.

Most of these advantages could not have been achieved by conventional manufacturing and not without additional investments in the module manufacturing process.

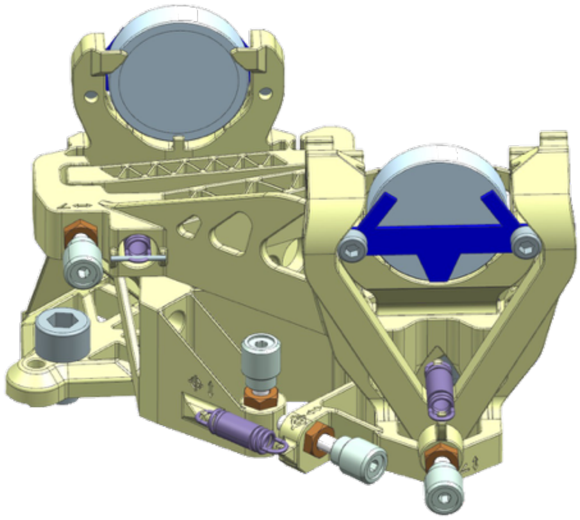

Figure 1.3: Optical beam alignment module designed by VDL-ETG. Reprinted from *"Entry points for metal additive manufacturing in precision"* by T.Peijnenburg, C. Wijnstok and G. Oosterhuis, ASPE, 2016

However, not every design for an optical mount is printable. As printing techniques evolve, process dependent constraints still apply, such as minimum feature size and limited overhang [8]. Besides, wrapping of the part can occur during the manufacturing process, which is introduced by local thermal stresses resulting from false designs and print strategies [9].

Topology Optimization (TO) can help to design an increasingly complex system, such as integrated optical mounts. TO is a mathematical design tool which computes optimal material distributions for a given objective with a determined set of constraints and boundary conditions [10]. TO is occasionally referred to as a 'free-from design' technique based on the natural-looking shapes generated with a minimal input from a designer [10]. Consequently, the TO designs exhibit complex geometries, which are supported by AM production capabilities [11].

This beneficial combination makes TO an interesting design tool for optical mounts produced by AM. Recent research has focused on the advantage to integrate AM constraints into TO, which results in algorithms strategies coping with constraints such as overhang [12]. Although these strategies are not sufficiently developed yet, TO is presently regarded as one of the most promising design tools for AM [13].

Considering the collaborative advantages of TO, AM and innovations in optical mounts, a new research area has emerged.

1.3 Research focus and objective

This thesis is focusing on internal compliant mechanisms of optical flexure mounts, which are responsible for the flexibility and, transmission of the users work into the actuation of the optics. Other components and functions are excluded from this research. This thesis investigates the possibility of using Topology Optimization (TO) for designing of these compliant mechanisms. Further, the use of Additive Manufacturing (AM) for compliant optical mounts mechanisms designs is evaluated.

The thesis objective is bilateral:

The first research objective investigates the design of compliant mechanisms for optical mounts by means of additive manufacturing.

The second research objective investigates the suitability of Topology Optimization for designing compliant mechanisms for optical mounts.

1.4 Report outline

This report is structured into three main parts. Part I presents the background and a critical review of the state of the art concerning compliant optical mount adjustment mechanisms. Further, Additive Manufacturing (AM) constraints for compliant optical mounts mechanisms are evaluated. Part II Introduces Topology Optimization (TO) and outlines TO of structures with tailored compliance and compliant mechanisms. Finally, conclusions and recommendations are given in Part III.

Part I: Optical mount mechanisms

- Chapter 2 Compliant optical mount mechanisms
- Chapter 3 Additive Manufacturing for compliant optical mount mechanisms

Part II: Compliant topology optimization

- Chapter 4 Topology Optimization
- Chapter 5 Topology Optimization of structures with tailored compliance
- Chapter 6 Topology Optimization of compliant mechanisms

Part III: Conclusion and Recommendations & Outlook

- Chapter 7 Conclusions
- Chapter 8 Recommendations for future work and outlook

Appendix A: Conference poster October 2015

Part I

Optical mount mechanisms

Chapter 2

Compliant optical mount mechanisms

2.1 Optical mounts in a nutshell

An optical mount is a device used in the field of optical systems that securely holds optics in place while permitting precision adjustment in both translational or rotational directions [1]. In order to allow more degrees of freedom depending on the application, many variations of optical mounts are available. In the introduction a two Degrees Of Freedom (DOF) tip-tilt optical mount is introduced (figure 1.1). The mounts can be adjusted by hand with a micrometer head or adjustment screw and can further be motorized for automation by linear actuators for dynamic control of the optics [14].

2.2 Adjustment mechanisms

Optical mounts are differentiated by the type of adjustment mechanism, which transfers input motion of the adjustment screw into displacement of the optics. Optical mount adjustment mechanisms can be divided into three categories: Flexure (compliant), kinematic and gimbal mounts mechanisms (figure 2.1). Each mounting system has different features and thus bear specific advantages and disadvantages. A summarized qualitative comparison is presented on the next page (table 2.1).

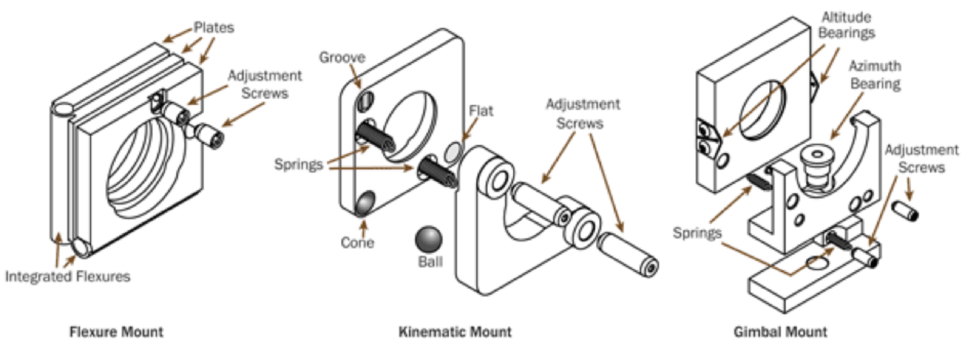

Figure 2.1: Three types of optical mount adjustment mechanisms. Retrieved from Article *https://www.photonics.com/Article.aspx?AID=56964*, Copyright J. Wingerd (2010)

Flexure Mounts
Flexure mounts obtain mobility from the flexibility of their parts rather than from the bearings, hinges and slides. This flexibility permits structures to function as mechanisms, earning their mobility from elastic body deformation [15]. These mechanisms are known as compliant mechanisms. Flexure mounts are compact and have a good repeatability of motion. For this reason, flexure mounts are popular for high precision measurements. The flexure mount is more sensitive than other adjustment mechanisms for alignment drifts from temperature fluctuations since the most critical element in the mount, the flexure, has a relatively small thermal mass, which is disadvantageous for conducting heat to the mounting surface [14].

Kinematic Mounts
Kinematic mounts typically make use of a ball bearing, which pivots in a hole of the fixed frame in order to allow the movable frame, holding the optics, to pivot on top of it. Kinematic mounts are widely used due to excellent mechanical stability at a low product price. However, technical drawbacks, such as cross-coupled adjustments, lead to unexpected angular beam translation. Besides, hysteresis occurs due to the friction in components when adjusting the optics, which makes the installation of optics more time consuming [1].

Gimbal Mounts
A gimbal mechanism has fixed orthogonal axes of rotation in space, where the intersection creates a uncoupled rotation point, preferably close to the surface of the optics [16]. The uncoupled rotation point eliminates beam translation (crosstalk of the mechanism) during optical adjustments. Gimbal mounts have optical travel ranges that are significantly larger than kinematic and flexure mounts. They are more expensive and not suitable for industrial applications due to poor mechanical stability of the rotational joints (bearings support rotation) [17].

	Flexure	Kinematic	Gimbal
Range of motion optics	-	0	+
Resolution	+	0	0
Repeatability adjustments	+	-	-
Orthogonality (crosstalk)	0	0	-
Mechanical stability (vibration)	+	+	-
Thermal drift	+	-	0
Size of mount	-	0	+
UV/vacuum compatibility	+	0	-
Purchase costs	+	-	-

Table 2.1: Qualitative comparison of different mount types mechanisms; + high contribution to the subject, 0 moderate contribution to the subject, - low contribution to the subject.

2.3 Application of optical mounts and design requirements for adjustment mechanisms

The research conducted in this thesis will focus on the application of optical mounts in the field of high precision optical measurement applied in the semi-conductor industry. The mounts operate in a vacuum environment and are climate controlled in order to ensure constant humidity and temperature. The optics attached to the mount are used to guide and split the deep ultra violet optical beams (with a typical wavelength of 266nm or larger). Due to the fact that research with optics is sensitive, stable conditions must be maintained, which is achieved by mounting the optical component to an optical table for assuring a high level of vibration isolation.

Structural requirements:

- The adjustment mechanism is free of friction - Friction leads to hysteresis leading to unpredictable effects during the adjustment of the optics.
- The adjustment mechanism is free of play - Play will cause difficulties during the adjustment and requires to be diminished in order to lock the position of the optical element.
- The adjustment mechanism is mechanically stable in lifetime performance - Maintaining the position of the optical element throughout its assigned lifetime as well as from external vibration is essential for a consistent operation of the optical system.

Functional requirements:

- Top adjustment of the optical mount - Adjustment screws are often located on the top of the mount assembly in space-constrained optical-based instruments, rather than the back, in order to make the mount better accessible in confined spaces.
- Converging adjustment of optics - Optical systems undergoing subsequent set of adjustments do not or only minimally influence previous adjustments.
- Sensitivity correlation - A prescribed sensitivity between input (screw adjustment) and output (rotation or displacement of optics) of the mechanism is required.

Requirements related to environmental conditions:

- Prevention of outgassing in a vacuum environment - Outgassing is the release of absorbed and adsorbed gases as well as the evaporation of material itself. This can be problematic in systems based on ultraviolet lasers, since UV optics are particularly sensitive to any material deposited on their surfaces. Any outgassed material can lower optics transmission or mirror reflectivity.
- Reduced adjustment mechanism dimensions - The size of the mechanisms has to be compact while providing enough space for easy use, maintenance and connectivity to additional instruments.

2.4 Flexure mounts for a high precision application

Flexure mounts are suitable optical mount types for industrial application and meet the according design requirements stated in the previous section. Due to their monolithic structure, these flexure (compliant) mounts have no motion among pieces and thus no overlapping pieces, which is a major advantage over kinematic and gimbal mechanisms. The absence of relative motion implies the absence of sliding friction and therefore eliminates wear, vibration and the need for lubrication [18]. Consequently, less maintenance is required and outgassing of lubrication is prevented. Furthermore, backlash is cancelled out, which leads to reduced positioning errors and therefore increased precision [15].

The lack of overlapping pieces allow the mechanism to contain fewer parts, thus reducing assembly efforts and furthermore enhances compactness and miniaturization characteristics. Another significant advantage of monolithic construction is the vacuum compatibility. In contrast to monolithic compliant mechanisms, kinematic and gimbal mechanisms constructed from numerous diverse parts trap gas or pollution in various areas of the assembly, requiring escape channels to let the gas exit from 'blind holes' [1]. Further, the thermal drift of the optics is minimized due to the controlled atmosphere, keeping the compliant mechanism at a constant temperature.

The main disadvantage of advanced designs with flexures, is increased cost due to the fabrication of monolithic components, especially compared to lower priced kinematic mounts. However, due to technical advantages, usability increases (i.e. labor savings in optical adjustment time and maintenance reductions) and the overall cost of ownership can be reduced.

2.5 Design approaches for compliant mechanisms

Generally, there are two approaches for the design approaches for compliant mechanisms:

The kinematic synthesis approach
The kinematic synthesis approach is based on traditional rigid-body kinematics [19] [20] [21]. The first step is to synthesize a rigid-body mechanism, with this information flexibility is introduced to obtain a pseudo-rigid-body mechanism and finally a fully compliant mechanism is analyzed. Due to potential energy storage in the compliant segments, the input-output relationship is affected, in particular, energy efficiency is challenged. As consequence, synthesis and analysis cannot be done by separating kinematics and dynamics, which makes the next approach an interesting alternative for designing compliant mechanisms. This research will not further elaborate on the kinematic synthesis approach of compliant mechanisms.

The continuum synthesis approach
The continuum synthesis approach is based on the (Topology) Optimization method of continuum structures [22] [23] [24] [25]. Chapter 4 will give more insight into Topology Optimization.

2.6 State of the art of flexure mounts

Flexure mounts in all variations are commercially available as well as custom made for usage in precision tools. The Siskiyou™ mount [26] is selected as benchmark to perform function and usability tests. The Siskiyou™ mount (figure 2.2) is a top adjustable tip-tilt mount and well known for compactness and userfriendliness.

In this research, practical tests performed with the Siskiyou™ mount, demonstrated the occurrence of hysteresis, when optical adjustments were made. Introducing over- and undershooting, which makes the positioning of the optics challenging when a high accuracy of optical beam pointing is required. Analysis of the mount has proven that hysteresis is caused by two factors:

- A kinematic mechanism used as transmission to actuate the 'compliant mechanism' introduces friction.
- The angle of displacement of the adjustment screw is not in line with the movement of the transmission, creating friction in the point of contact causing slip-and-stick.

Figure 2.2: Siskiyou™ top adjustable tip-tilt monolithic flexure mount for 1 inch optics. Retrieved from Article *https://www.photonics.com/Article.aspx?AID=56964*, Copyright J. Wingerd (2010)

These insights in 'off-the-shelf' flexure mounts give input for technical improvements of flexure mounts. First, not only the 'core flexibility' of the mount should be made out of compliant material, also the transmission should be turned into a compliant mechanism to prevent hysteresis. Further, in order to decrease the relative motion, thus friction, between the adjustment screw and the actuated mechanism by the adjustment screw, the contact point should move in one line when actuated.

Chapter 3

Additive manufacturing for compliant optical mount mechanisms

Additive Manufacturing uses digital design data for constructing optical parts. The manufacturing process begins with a digital CAD model, where digital data precisely describes the geometric shape of the part and each of its layers. For each layer individual design data is provided, generating a tool path that forwards motion coordinates to the 3D printer. The printer reads the digital data to form successive layers of material in order to build up the part. Post processing is often needed after printing in order to trim off excessive material and to fulfil other requirements such as surface roughness and cleanliness [11].

There are numerous types of additive manufacturing processes on the market, previously extensively reviewed in [4]. In this project, powder bed fusion as manufacturing process is chosen, since it is currently the most developed and commercially used process on the market. The powder bed fusion process produces parts with different types of metals alloys suitable for compliant mechanisms and includes five main printing techniques: Direct Metal Laser Sintering (DMLS) [27], Electron Beam Melting (EBM) [28], Selective Heat Sintering (SHS) [29], Selective Laser Melting (SLM) [30] and Selective Laser Sintering (SLS) [31].

3.1 Powder bed fusion process

Powder bed fusion of metal takes place in a controlled inert atmosphere or partial vacuum in order to shield off the molten metal and to control the melting process [32]. Material is selectively melted by a laser beam, describing a cross section of a part obtained from the CAD model during the manufacturing process. The layer thickness is controlled by step-wise vertical displacement of the built platform, which is filled with powder material from another chamber and moved into place with a sharp blade. The result is a powder bed cake, which fills the entire chamber space and includes the manufactured part. The schematic overview of the powder bed fusion machine is shown in figure 3.1.

The powder bed fusion processes inherently requires support structures to avoid sacking of molten material in case of overhanging surfaces, heat dissipation in critical parts and distortions. These support structures can be removed by mechanical treatment during post processing [33]. After support removal, the part may undergo post processing treatments such as shot peening, machining and heat treatment depending on the requirements for the part. Critical components go through hot isostatic pressing (HIP) to ensure adequate part density.

Power of laser source, writing speed, distance between laser paths and thickness of powdered layers are the main process parameters of power bed fusion. A typical layer thickness of $20 - 100\ \mu m$ is considered standard for this process [34]. Power bed fusion can produce fully dense metallic parts from a wide range of metal alloys such as titanium alloys, inconel alloys, cobalt chrome, aluminium alloys and stainless steels.

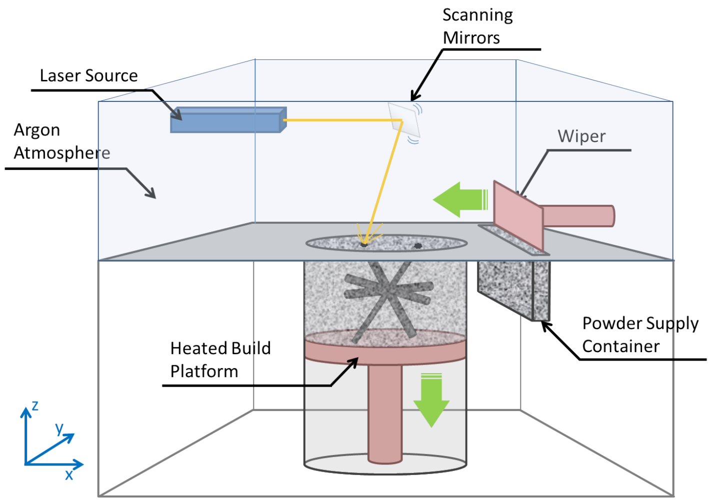

Figure 3.1: Schematic representation of a powder bed fusion machine. Retrieved from *http://blogs.nottingham.ac.uk/innovate/2015/08/10/can-we-3d-print-an-electric-motor/*, Copyright Michele Garibaldi (2015)

3.2 AM design considerations for compliant mount mechanisms

For this research, knowledge is gained on designing flexure mounts for AM. Contributing to the design of the laser alignment module (figure 3.2), introduced in the chapter 1, gave practical insight in designing optical mounts for AM as well as for the AM process in general. This section explicitly elaborates on design stepping stones for compliant mechanisms by means of AM.

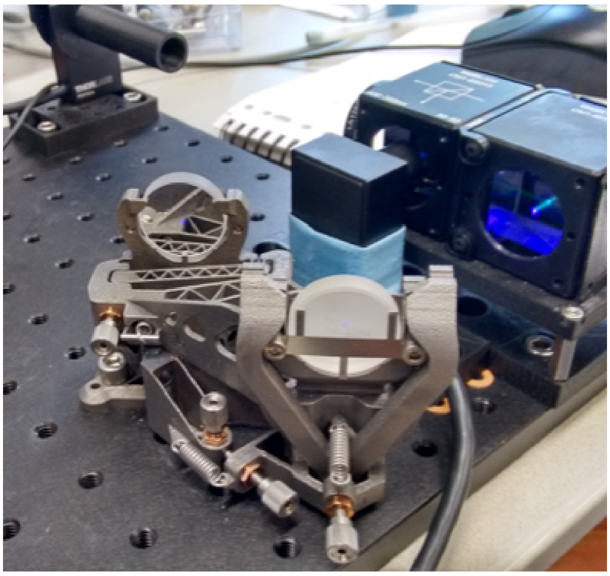

Figure 3.2: Test set-up of the optical beam alignment module. Retrieved from *"Entry points for metal additive manufacturing in precision"*, Copyright by G. Oosterhuis, 2015

General AM constraints
In general, AM constraints, such as minimal thickness, overlap and overhang, need to be respected during the design process. These parameters are dependent on the material and the type of powder bed fusion technique used [4].

Material choice
The material choice for compliant mechanisms is based on the (application) environment, mechanical properties and availability. Polymers and rubbers are optimal materials for compliant mechanisms due to their low Young's Modulus. The chemical composition of polymers and rubbers induce outgassing, which is the release of absorbed and adsorbed gases as well as evaporation of the material itself. Contaminating the ultra-clean systems, these materials are not suitable for these environments. Titanium alloys (Ti6Al4V) is a more suitable material for compliant structures in ultra clean environments, since it does not outgas and since it has higher yield strength as well as lower Young's modulus compared to steel alloys [35]. Powder bed fusion is, at its current state, the best process to manufacture titanium parts [4].

Compliant structures
Designing compliant mechanisms for AM is complex [36]. Since there are numerous ways to design flexures, an overview of considerations regarding designs of flexures for AM of compliant mechanisms stated below:

- Powder bed fusion technologies do not allow titanium parts to be designed with an angle larger than 45 degrees (overhang constraint) with respect to the building direction, reducing the configuration possibilities of flexures [37]. Violating the overhang constraint results in deformation, sacking and quality issues of materials (voids), changing the properties and function of the mechanism. Overhang is the most limiting constraint for designers, reducing the possibilities of orientation to implement elements such as flexures.

- Large thin leaf (slender) flexures are not recommended for AM. Due to the thermal stress encountered during the building process, thin structures have the tendency to warp.

- Printing inaccuracies (surface roughness and size tolerances) can lead to higher stresses levels in the flexures, which can cause fractures. Research at VDL-ETG has proven, depending on the layer-height and the surface area of the laser, that a minimal tolerable thickness of $0.7mm$ is sufficient for solid flexure properties in their AM titanium process. This thickness feature is process, material and orientation dependent.

- The type of process and intensity of surface treatment during post processing is relevant for the volume and shape of flexures, influencing the properties of the mechanism.

- Surface roughness will not lead to fatigue since the use of optical mounts has only a limited amount of quasi-static loading cycles.

- Inaccuracies are introduced by converting the initial CAD designs to the STL file format used by 3D printers (figure 3.3). By removing construction data and modelling history, a series of triangular facets that represent the surface of the design are created. The minimal size of these triangular planes can be set in the CAD software. The basic rule of thumb is the minimal triangle offset to be smaller than the resolution of the AM machine. Critical regions, such as thin compliant structures, will be affected, because offsets in manufacturing have influence on flexure behaviour. The format conversion into STL files is automatic within most CAD software tools, but there is a possibility of errors occurring during this phase. These errors, for example voids and small-scale design errors, require rectification by means of other software tools.

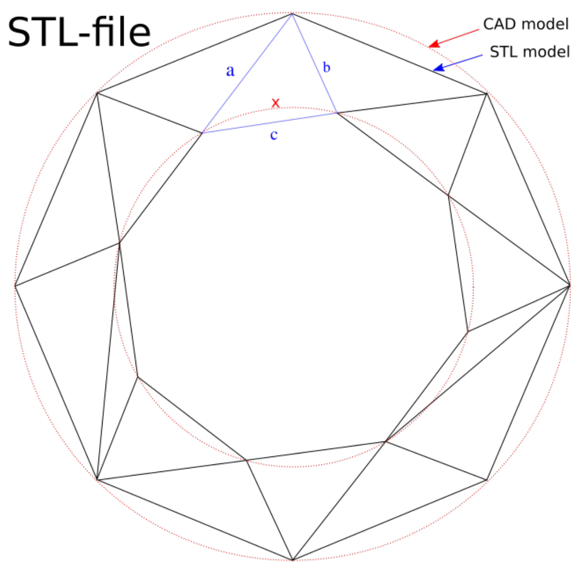

Figure 3.3: Example of STL vs CAD format. Retrieved from *http://www.fablabplus.be/3d-printen/aan-de-slag-met-3d/stap-2-stl-bestand-maken/*, Copyright M. van Lieshout (2011)

Post processing, machining
Post processing is needed to remove excessive support structures, drill holes, 'tap' holes or flat surfaces to make the optical mount usable. Due to compliant properties, the optical mount has some internal flexible structural regions, which are hard to post process if not retained.

Tools will exert a force and momentum on the mount, which results in a displacement of the work area, consequently leading to inaccurate shaping of the optical mount during the process.

Surplus support material that stiffens the flexible structures for post processing prevents these undesired post processing effects. After post processing, the extra material can later be removed by Wire Electrical Discharge Machining (WEDM). WEDM does not exert force on the material, allowing a good surface finish and a straight cut. The accurate reference face (cut) of WEDM can be used as overall reference to align further post processing and the optical mount. Besides, the surplus support material for compliant mechanisms can also be utilized to support material with overhang during manufacturing, making the excess support material valuable for manufacturing and post processing.

Surface finish
In order to meet the design cleanliness requirements, surface treatment of the part is essential. Tests have proven that small particles (partly attached or clammed) remain on the surface of a part after AM (figure 3.4). These particles can possibly fall off over time due to frequencies in the system, flow, installation or transportation, and can be harmful to the operation of the optical system. Also, unprocessed rough parts can scratch gloves, suits or other objects. Surface treatment can be achieved by: Beat Blasting/Shot Peening ($\pm 3-5\ \mu m$), Silk Cleaning (mechanical/chemical treatment) or Corrosive Treatment [4].

Due to safety concerns, optical mounts must not reflect light in industrial applications. Kinematic aluminum mounts are anodized since no top coating can be added due to outgassing. An exception to this is titanium, which cannot be anodized and which must be shielded by another material.

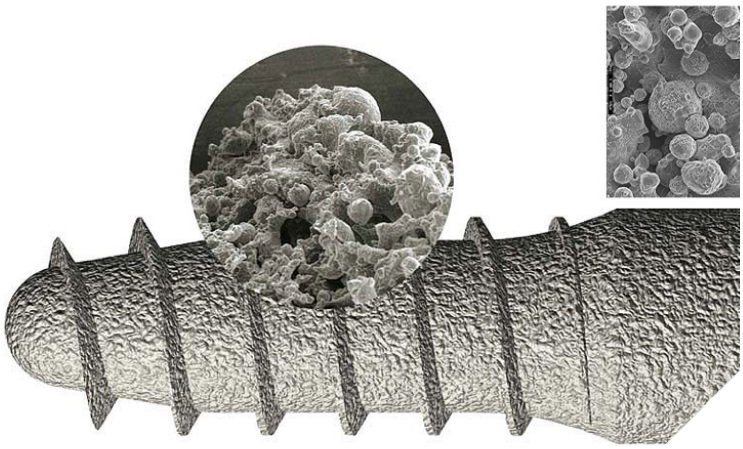

Figure 3.4: Surface roughness of a dental implant screw manufactured with DMLS (200x magnified). Reprinted from *"Dental Implant Surfaces - Physicochemical Properties, Biological Performance, and Trends"* by Ahmed M. Ballo et. al., 2014, ASPE, Aspen

Transportation
According to the guidelines of VDL-ETG, transportation can possibly exert a force up to $7G$ on the optical mount making the flexible structure vulnerable for exceeding the yield point in the flexures. Since it is impossible to design compliant structures with such robustness, stops can be added in the system that prevent optics to be actuated further than the yielding point of the material in the flexures. Not only for transportation but also during operation it is useful to limit the actuation well below the material yield point. More complex systems with undefined flexures, such as distributed compliant mechanisms, temporarily need support structures that can be removed after transportation to protect the optical mount mechanism.

Clean room convenience
Apart from the performance of the optical mount, the user friendliness is important. The clean room suit, gloves and mask disturbs the user and limits the sensitivity and work-space. Screws with a high thread counts (100 TPI+) are not recommended and stops need to be added to limit flexure deflection for accidental over-actuation.

Part II

Compliant Topology Optimization

Chapter 4

Topology Optimization

4.1 Topology Optimization methods

In the field of structural design, optimization techniques offers advantages [10] to find an optimum solution for predefined design objectives such as minimizing mass, stress or defining a displacement of a mechanism, for example. Structural optimization can be divided in three different techniques: Sizing Optimization, Shape Optimization and Topology Optimization [10]. The goal of Size Optimization is to find the optimal dimensional distribution of a predefined plate or truss structure. Shape Optimization optimizes the shape of a predefined layout through additional degrees of freedoms. Topology Optimization (TO) is not limited to size and shape changes and allows the determination of topology features, such as shape of holes and the connectivity of the design domain [10].

"TO is a numerical approach to define the best distribution of individual densities in a finite element model within design space under specific boundary conditions and loads, that meets a prescribed set of performance targets."

4.2 Gradient and non-gradient methods

TO can be categorized into gradient based and non-gradient based methods. In non-gradient based approaches, the design variables are discrete values and the methods are based on repeated function evaluations using a stochastic or population-based algorithm. The iterative solution for a TO problem is the distribution of either solid or void finite elements of the structure. Various methods have been developed, such as the Evolutionary Structural Optimization (ESO) method [38] and the Level Set-based method [39].

In gradient-based optimization the design variables ρ are defined as continuous variables ($0 < \rho_{min} \leq \rho \leq 1.0$), facilitating the evaluation of the first or possibly second-order derivatives of response functions with respect to design variables and the implementation of mathematical programming techniques for the solution of the optimization problem.

In this research, TO is based on a Solid Isotropic Material with Penalization (SIMP) gradient method [40] (figure 4.1), also known as the penalized proportional stiffness model. The fundamental TO algorithm implemented is written in C++ by M. Langelaar (2013) based on the Method of Moving Asymptotes (MMA) [41].

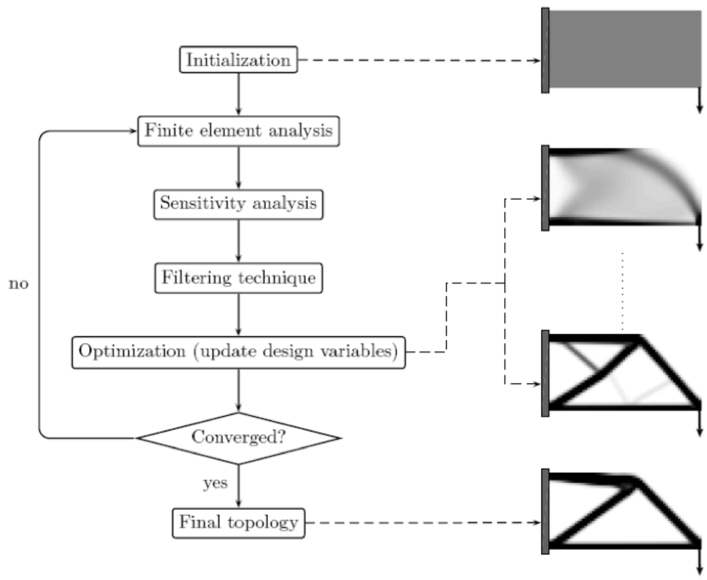

Figure 4.1: Flow chart of the SIMP method. Reprinted from *"Krylov subspace methods for topology optimization on adaptive meshes"* by S. Wang, University of Illinois at Urbana-Champaign, 2007

4.3 Solid Isotropic Microstructure with Penalization

The SIMP method affects the elasticity tensor E_{ijkl} of the Finite Elements (FE) model, and thereby the stiffness of the element e.

$$E_{ijkl}(x) \;=\; \rho(x)^p E^0_{ijkl};\; p > 1 \tag{4.1}$$

The volume of a structure made of the material are given by

$$\int_\Omega \rho(x) d\Omega \leq V \;; \quad 0 \leq \rho(x) \leq 1 \;; \quad x \in \Omega, \tag{4.2}$$

where $\rho(x)$ is a density function of the material and E_0 the linear elasticity tensor of a given solid isotropic reference material. The presence of the penalization power parameter 'p' is needed to favorably diminish the intermediate density (grey values) elements contributing to the structural stiffness of the design and encourage the development of elements which are either void ($\rho_e = 0$) or solid ($\rho_e = 1$). Choosing the penalization power value $p > 1$ makes intermediate densities unfavourable because the stiffness to volume ratio of the element is decreased (figure 4.2).

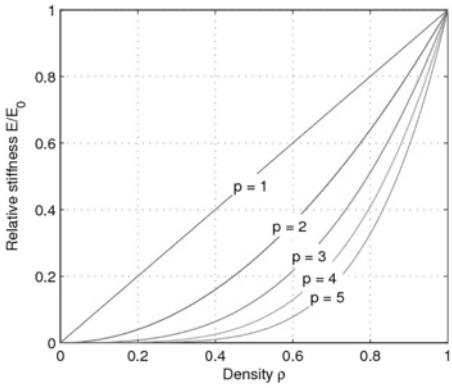

Figure 4.2: E/E_0 vs. volume density for different values of the penalization power p. Reprinted from *"Topology Optimization of Continuum Structures: A review"* by H. A. Eschenauer and N. Olhoff, Applied Mechanics Reviews, 2001

Optimizing a structure starts with defining the design space (Ω) into a grid of N finite elements (isotropic solid micro-structures), each finite element e having a predefined initial fractional material density ρ_e. The boundary conditions and body forces are applied on the finite element model. Further, the classical TO problem of minimizing the compliance while constraining the mass can with the density method, assuming linear elasticity, can be formulated as:

$$\begin{cases} min\ C(\rho) = F^T U(\rho) \\ st. \begin{cases} V \leq V^* \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{4.3}$$

The first step in each iteration is a finite element analysis, considering current material distribution. The results are used to evaluate the sensitivity of each element (the impact the variation of its density has on the objective function), expressed as the derivative of the objective function with respect to its density. Sensitivity can be calculated by the direct and adjoint method. The choice of sensitivity analysis depends on the number of design variables, type of model (cheap, expensive, linear, non linear) and implementation effort.

The third step, filtering, is necessary to prevent checkerboarding. Checkerboarding refers to the checkerboard pattern that is formed with alternating elements with density of 1 and 0. These patterns appear since they are locally stiffer than any other distribution of the two constituent materials [42], making this configuration for stiff structures more favourable. Topologies with checkerboard patterns are not well manufacturable since the elements are only connected at the corner nodes, and for this reason a linear density filter is implemented in this thesis.

The linear density filter [43] redefines the weighted average density of each element $\tilde{\rho_e}$ linearly in a neighbourhood of the element, and thereby impose stiffness to be a function of the densities in a specified neighbourhood of an element N_e, within a given filter radius r_{min} of the centre of element e.

$$E_e = E_e(\rho) = E_e(\tilde{\rho_e}) = E_{min} + \tilde{\rho_e}(E_0 - E_{min}), \tag{4.4}$$

The filtered density measure is:

$$\tilde{\rho_e} = \frac{\sum w(x_i)v_i}{\sum w(x_i)v_i\rho_i}, \tag{4.5}$$

where v_i denotes the volume fraction of element i. The weighting function $w(x_i)$ is given by the linearly decaying function

$$w(x_i) = r_{min} - ||x_i - x_e||. \tag{4.6}$$

The linear density filter limits the space of admissible solutions to the design, by imposing material with a minimum physical size, which can be used as a parameter to define the minimal thickness of flexures in mechanisms.

The Method of Moving Asymptotes (MMA) [41] solves the TO problem using a convex approximation method. In each iteration of the solver, the current iteration point $(X(k), Y(k), Z(k))$ is given. Then an approximating (k) sub problem, in which the functions $f_i(x)$ representing the objective and constraints are replaced by certain convex functions $\tilde{f}_i^{(k)}(x)$, are generated. The approximating functions are based on gradient information at the current iteration point as well as on the Moving Asymptotes (MA) $u(k)$ and $l(k)$. The MA are updated in each iteration based on information from previous iteration points. With this information the sub problem is solved, and the new optimal solution becomes the next iteration point $(X(k+1), Y(k+1), Z(k+1))$.

The last step in the process is to match the results with the stopping criteria. If either the variation of objective function value does not improve or the maximum set of iterations is reached, the iteration loop will terminate.

Chapter 5

Topology Optimization of structures with tailored compliance

This chapter will give insight into various design strategies for structures with a tailored compliance using TO. Such structures, which allow motion and flexibility from flexures, are crucial for structures belonging to the current state of optical mounts. Case studies in a 2D space illustrate the behaviour of structures with a tailored compliance for different TO strategies, where the optimization models describe only in-plane motions assuming linear displacement.

5.1 Example 1: Deflection of a Beam

The first example represents a one-sided clamped structure, allowing a high compliance in lateral and low compliance in axial direction, creating a structure with a tailored compliance. For this example two load cases are formulated. First, the compliance of the beam is calculated based on the axial point force of $F_z = 1\ N$. The second load case calculates the compliance based on a lateral point force $F_x = 1\ N$. The design space Ω has a length of $l = 150\ mm$ and a height of $h = 30\ mm$. One finite element is $1x1x1\ mm$ in size. The solid element E-modulus is set to $2.1x10^5\ N/mm^2$ and $\nu = 0.33$ representing steel. A minimum length scale of $r_{min} = 1.5\ mm$ is ensuring a minimal thickness of 3 elements throughout the system. The penalizing power $p = 3$ is chosen.

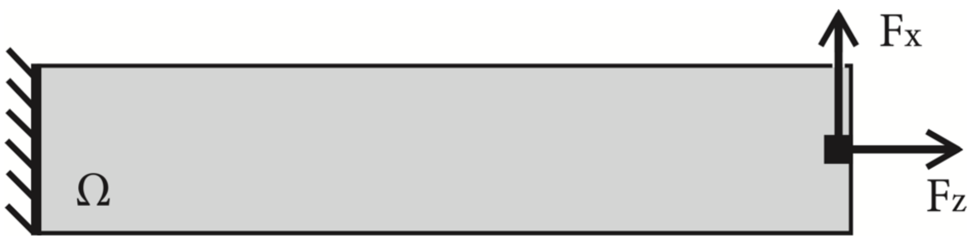

Figure 5.1: Schematic representation of the design domain for a beam

5.1.1 Objective: Maximizing compliance in lateral direction

The compliance in lateral direction will be maximized while the axial compliance is upper bounded constrained to ensure axial stiffness. The volume constraint fraction V^* of the design space is set to 0.3 with $C_{z\ ref} = 10^{-6}\ J$ as upper bound constraint.

$$\begin{cases} min \ \ -C_x(\rho) \\ st. \begin{cases} C_z \leq C_{z\ ref} \\ V \leq V^* \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{5.1}$$

Figure 5.2: Design of beam with objective: maximizing compliance in lateral direction for $C_{z\ ref} = 10^{-6}\ J$

Discussion of design:
The strategy with compliance in lateral direction (C_x) as objective results in designs with intermediate densities between the fixed end and point of actuation (figure 5.2). The reason of this behaviour is that the MMA optimizer has no incentive to drive the density of the elements to either 0 or 1, since enough intermediate density material is distributed to satisfy the axial compliance and the volume constraint, making this design feasible (figure 5.4 and 5.3).

The point force becomes a free-moving body when the axial compliance constraint is not taking part in the strategy. This leads to designs with no stiffness in the structure, which means it results in an unfeasible design and the optimizer aborts the algorithm.

The outcome of the strategy to maximize compliance in lateral direction is sensitive for the maximal required compliance in axial direction. The design changes to a feasible discrete structure when the allowable compliance in axial direction is further reduced to $C_{z\ ref} = 10^{-7}\ J$ (figure 5.5). This results in a compliance of $C_x = 10^{-9}\ J$ which is actually lower than the axial compliance. The combination of $C_{z\ ref}$ and the maximal volume constraint are important to make the design feasible. Sufficient volume needs to be allowed in the algorithm in order to assure axial stiffness. Further, lateral compliance cannot be controlled. The MMA optimizer satisfies the constraint before maximizing the lateral compliance.

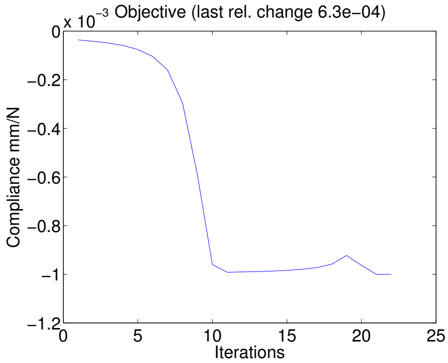

Figure 5.3: Objective: maximizing compliance in lateral direction for $C_{z\ ref} = 10^{-6}\ J$

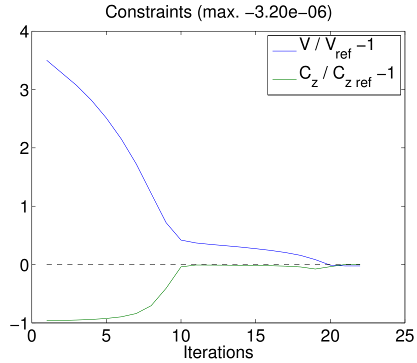

Figure 5.4: Constraints: maximizing compliance in lateral direction for $C_{z\ ref} = 10^{-6}\ J$

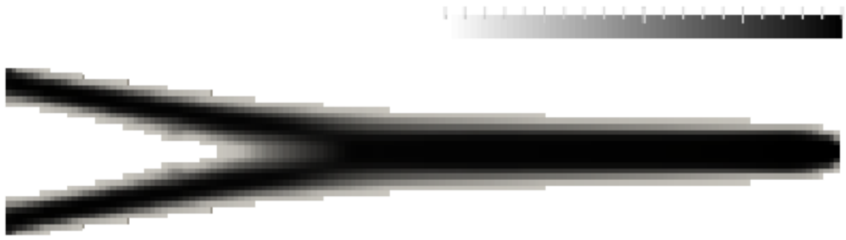

Figure 5.5: Design of beam with objective: maximizing compliance in lateral direction for $C_{z\ ref} = 10^{-7}\ J$

5.1.2 Objective: Minimizing volume

The volume of the design is minimized while the compliance in axial direction is upper bounded $C_{z\ ref} = 10^{-6}\ J$ and the lateral compliance is lower bounded $C_{x\ ref} = 10^{-3}\ J$.

$$\begin{cases} min\ V \\ st. \begin{cases} C_x \geq C_{x\ ref} \\ C_z \leq C_{z\ ref} \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{5.2}$$

Discussion of design:
The strategy of minimizing volume results in defined discrete structures. Since every finite element density has to contribute efficiently to the stiffness of the design, the optimizer forces the design into a structure with elements of 0 - 1 densities. In comparison to maximizing the compliance in lateral direction, this strategy bounds the required lateral compliance, controlling the lateral compliance (figure 5.8). Figure 5.6 represents the outcome of the above mentioned optimization.

The same strategy with an increased allowable axial compliance $C_{z\ ref} = 10^{-2} J$ results in

Figure 5.6: Design of beam with objective: Minimizing volume, $C_{z\ ref} = 10^{-6}\ J$, $C_{x\ ref} = 10^{-3}\ J$

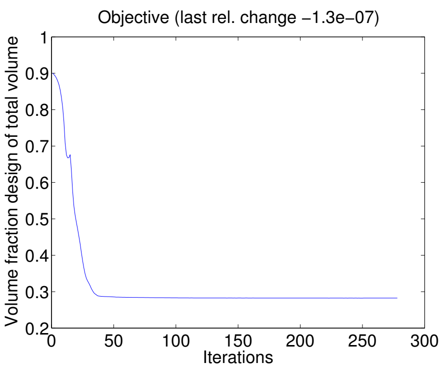

Figure 5.7: Objective: Minimizing volume $C_{z\ ref} = 10^{-6}\ J$ and $C_{x\ ref} = 10^{-3}\ J$

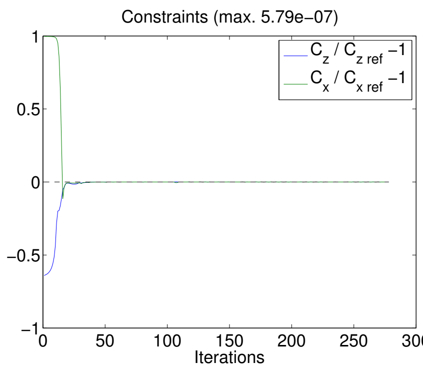

Figure 5.8: Constraints: Minimizing volume $C_{z\ ref} = 10^{-6}\ J$ and $C_{x\ ref} = 10^{-3}\ J$

an infeasible design. The optimizer is not able to find a structure that is more compliant in the axial direction.

The filter radius (r_{min}) has an effect on the design. The minimal size of the filter radius determines the size of the notch flexure. When the radius is increased in size, it results in a thicker beam structure up to the point that intermediate densities in the notch are in favor of satisfying the lower bounded lateral reference compliance constraint $C_{x\ ref}$.

5.1.3 Objective: Minimizing compliance in axial direction and maximizing compliance in lateral direction

The last example of optimization of a beam structure minimizes the displacement in axial displacement and maximizes the lateral displacement. To assure a maximal axial compliance, the axial direction is upper bounded $C_{z\ ref} = 10^{-6}\ J$ and the lateral compliance is lower bounded $C_{x\ ref} = 10^{-3}\ J$ to assure a minimal displacement of the beam.

$$\begin{cases} min\ C_z - C_x(\rho) \\ st. \begin{cases} C_z \leq C_{z\ ref} \\ C_x \geq C_{x\ ref} \\ V \leq V^* \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{5.3}$$

Figure 5.9: Design of beam with objective: Minimizing compliance in axial direction and maximizing compliance in lateral direction, $C_{z\ ref} = 10^{-6}\ J$, $C_{x\ ref} = 10^{-3}\ J$

Discussion of design:
Figure 5.9 represents the result of this optimization with an increased allowable axial compliance and $C_{z\ ref} = 10^{-6}\ J$, showing a far more distributed beam structure in comparison with the results of the volume minimization. Due to the minimization of the lateral displacement and the reduction of compliance in axial direction, the design is more distributed. Notch flexures, such as seen in the volume minimization, do not appear since the optimizer minimizes the compliance in axial direction, making locally thin structures less favorable (see C_z constraint in figure 5.11). Intermediate density elements are formed because there is no incentive for the optimizer to reduce volume further.
With this approach it is not possible to define the exact compliance in every direction since it does not bound the minimization of the axial compliance (see figure 5.11).

5.2 Example 2: Rigid body in space

The second TO example represents a rigid body in a 2D space structure with a tailored compliance in space. This example consist out of three load cases. In the first load case a force of $F_z = 10\ N$ is applied in the z-direction. The second load case consist out of a $F_x = 10\ N$ load on the rigid body. The last load case uses a clockwise momentum of $M = 10\ Nmm$ on the rigid body. The rigid body has a length of $l_{rb} = 30\ mm$ and a height of $h_{rb} = 30\ mm$. The design space Ω has a length of $l = 150\ mm$ and a height of $h = 150\ mm$. One finite element is $1x1x1\ mm$ in size. The E modulus of a solid element is chosen as $2.1E5\ N/mm^2$ with $\nu = 0.33$, representing steel. A minimum length scale of $r_{min} = 1.5\ mm$ is ensuring a minimal thickness of 3 elements in the system.

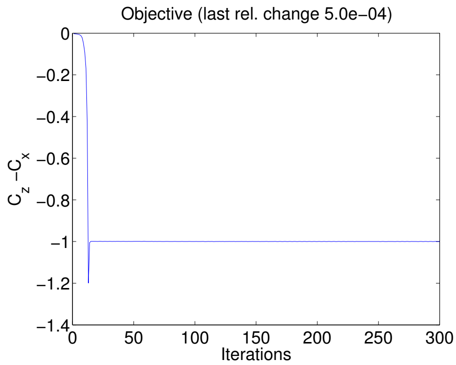

Figure 5.10: Objective: Minimizing compliance in axial direction and maximizing compliance in lateral direction, $C_{z\ ref} = 10^{-6}\ J$, $C_{x\ ref} = 10^{-3}\ J$

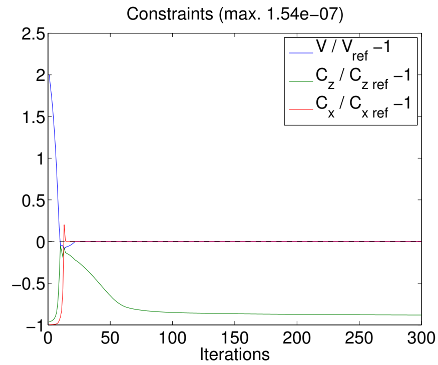

Figure 5.11: Constraints: Minimizing compliance in axial direction and maximizing compliance in lateral direction, $C_{z\ ref} = 10^{-6}\ J$, $C_{x\ ref} = 10^{-3}\ J$

5.2.1 Objective: Minimizing volume for a high compliance in rotational direction and a high translational compliance

In this optimization the goal is to create a structure for a rigid body that allows a low rotational stiffness and a low compliance in both in-plane directions x and z.

$$\begin{cases} min\ V \\ st. \begin{cases} C_z \leq C_{z\ ref} \\ C_x \leq C_{x\ ref} \\ C_M \geq C_{M\ ref} \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{5.4}$$

Discussion of design:
The outcome of this optimization problem results in a single beam connected to the rigid body (figure 5.13). The rotation compliance C_M is higher than the constraint bounds, which means that rotation stiffness is lower than expected. The compliance in lateral direction is lower than bounded. Noticeable is that this design does not control the displacement of the rigid body, it does only define the compliance. When a moment is applied on the rigid body it will not turn around the centre point.

5.3 Controlling the Degrees Of Freedom of a rigid body

A rigid body in a 2D space has three Degrees of Freedom (DOF): Two translational and one rotational Degree of Freedom. Constraining one or more of these DOF determines a specific freedom of motion in space. For optical mounts, a rigid body (holder of optics) is supported by a structure with a finite stiffness in the constrained as well as in the free moving direction determining DOF.

An exception are the free-floating rigid bodies that can be represented by zero-stiffness mechanisms. The geometry, stiffness and pre-stress enables zero stiffness in structures to change

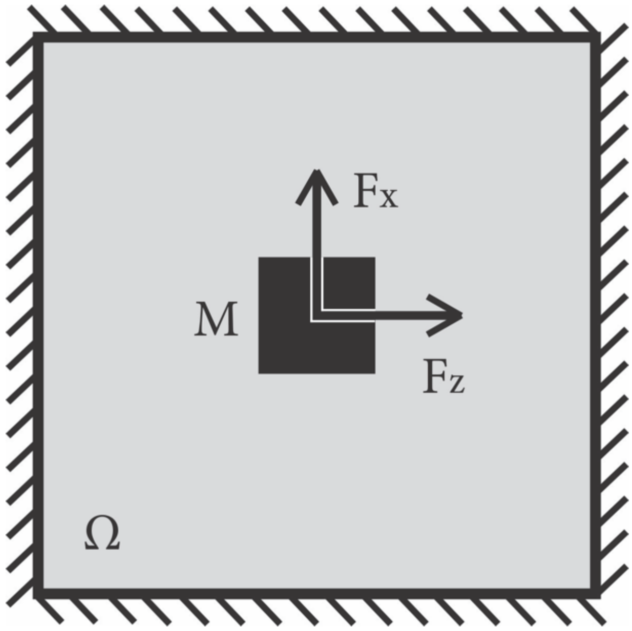

Figure 5.12: Schematic representation of a design domain for rigid body in space, with fixed domain edge

shape and maintain their deformed configuration without any external force, creating a free-floating mechanism. A geometrically non-linear analysis is necessary to distinguish between infinitesimal and large-displacement zero stiffness modes, which will not be further discussed in this research.

5.4 Constraining undesired displacement

Constraining a rigid body in space with TO can be enforced by two methods. The first method constrains a displacement due to a force acting on the structure. The second method uses the symmetry of the initial design of the structure to remove DOF.

5.4.1 Example 3: Constraining displacement by a bounded formulation

a displacement constraint is formulated to control the displacement due an external force on a rigid body. By restricting the displacement because of the moment on the rigid body ($u_x \leq u_{M_x\ ref}$ and $u_x \geq u_{M_x\ ref\ +\ allowed\ space}$) it possible to constrain unwanted translations. Since this is a numerically problem, extra space is needed to give the optimizer freedom to constrain the displacement. The number of constraints can be reduced by taking the square over the unwanted displacement. For comparison with the previous example, the same design geometries and values are used as in Example 2 (figure 5.12).

Figure 5.13: Design of rigid body in space with objective: Minimizing volume, $C_{z\ ref} = 10^{-5}\ J$, $C_{x\ ref} = 10^{-5}\ J$

$$\begin{cases} min\ V \\ st. \begin{cases} C_z \leq C_{z\ ref} \\ C_x \leq C_{x\ ref} \\ C_M \geq C_{M\ ref} \\ (u_{M_x})^2 \leq u_{M_x\ ref} \\ (u_{M_z})^2 \leq u_{M_z\ ref} \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{5.5}$$

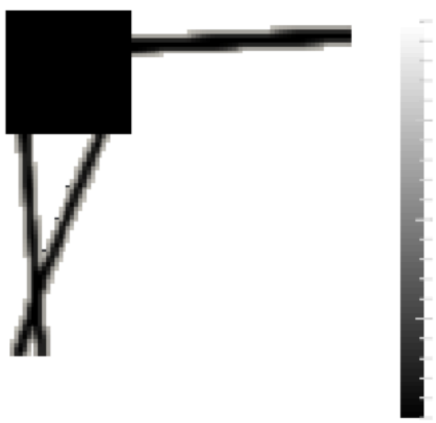

Figure 5.14: Design of rigid body in space with objective: Minimizing volume, constraining displacement due to moment on rigid body, $C_{z\ ref} = 10^{-5}\ J$, $C_{x\ ref} = 10^{-5}\ J$ and a displacement error of $u_{M_x\ ref} = 0.1\ mm$

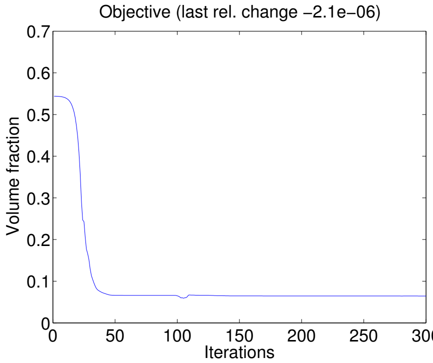

Figure 5.15: Objective: Design of rigid body in space with minimizing volume, $C_{z\ ref} = 10^{-5}\ J$, $C_{x\ ref} = 10^{-5}\ J$ and a displacement error of $u_{x,z_M\ ref} = 0.1\ mm$

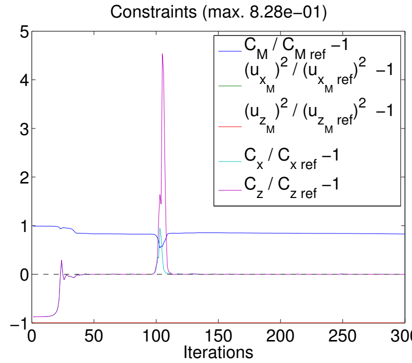

Figure 5.16: Constraints: Design of rigid body in space with minimizing volume, $C_{z\ ref} = 10^{-5}\ J$, $C_{x\ ref} = 1e^{-5}\ J$ and a displacement error of $u_{x,z_M\ ref} = 0.1\ mm$

Discussion of design:
The design changes into a three beam structure (figure 5.14) compared to the single beam design of Example 2 (figure 5.13). This design is not feasible since the minimal rotation compliance constraint is not satisfied (figure 5.16). Constraining the displacements due to a moment acting on the rigid body leads to less compliant structures in the x as well as in the z direction than is constrained. Adding a displacement constraint leads to more stiffness in the structure which reduces the compliance in favorable directions. More test have been performed with displacement constraints due to a translation of a rigid body in space, which resulted in more stiff structures and reduction of translation.

5.4.2 Example 4: Constraining displacement by symmetry of the design domain

When a rigid body is placed on a symmetry plane, reaction forces in the model constrain 2 DOF: The displacement u_x (perpendicular to the symmetry plane) and the in-plane rotation (figure 5.17). The DOF parallel to the symmetry plane (in x direction) is free, creating an 'exactly constrained' rigid body. For this example half the design domain of previous Example 3 is used as well as the force $F_x = 5\ N$. TSymmetry reduces the computational effort since half of the finite element analysis must be executed in every iteration.

$$\begin{cases} min\ V \\ st. \begin{cases} C_x \leq C_{x\ ref} \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{5.6}$$

Discussion of design:
Since it is only possible to guide the rigid body parallel to the symmetry plane, the amount of options is limited to use the symmetry geometry as constraint for DOF. Figure 5.18 represents the over the symmetry axis reflected structure (half of the full design domain) of the guided rigid body. The optimizer searches for an optimal way to reduce the volume of the design while limiting the compliance of the rigid body in the x-direction. The result is a beam in the direction of constrained compliance. Since no external force in the direction perpendicular

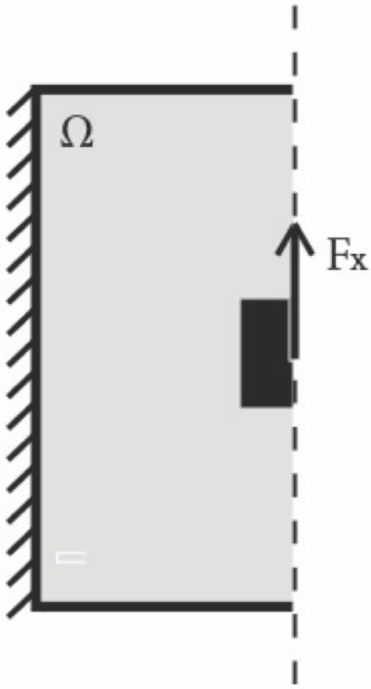

Figure 5.17: Schematic representation of half a design domain for rigid body in space divided by a symmetry plane with one fixed domain edge

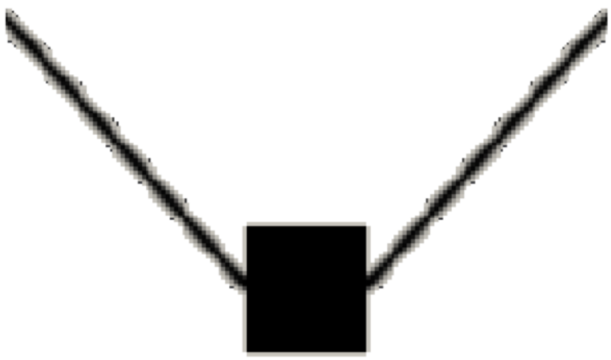

Figure 5.18: Design of rigid body in space with objective: Minimizing volume, constraining displacement by making use of symmetry, $C_{x\ ref} = 10^{-5}\ J$

to the symmetry axis can be exerted, the compliance in this direction cannot be determined nor constrained.
This strategy is sensitive for the maximal allowed compliance. A stiff structure (Figure 5.18, 5.19,5.20) is well defined while more compliant structures ($C_{x\ ref} = 10^{-3}\ J$) lack structure and are infeasible (figure 5.21, 5.22). The optimizer is not able to design feasible compliant structures smaller than a thin beam of low density material. Constraints with a higher compliance result in cloud structures, which are less compliant than constrained.

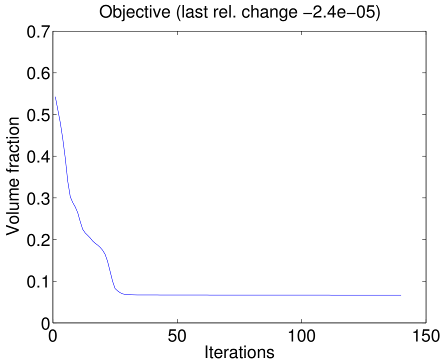

Figure 5.19: Objective: Design of rigid body in space with objective: Minimizing volume, constraining displacement by making use of symmetry, $C_{x\ ref} = 10^{-5}\ J$

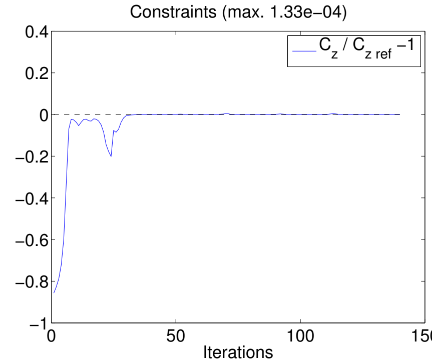

Figure 5.20: Constraints: Design of rigid body in space with objective: Minimizing volume, constraining displacement by making use of symmetry, $C_{x\ ref} = 10^{-5}\ J$

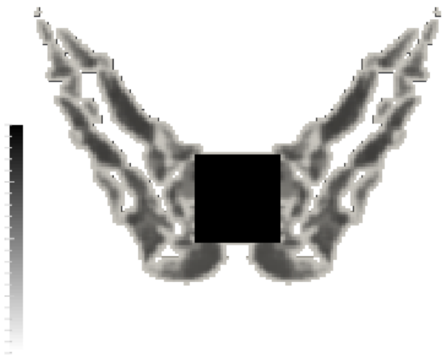

Figure 5.21: Design at iteration 150: Design of rigid body in space with objective: Minimizing volume, constraining displacement by making use of symmetry, $C_{x\ ref} = 10^{-3}\ J$

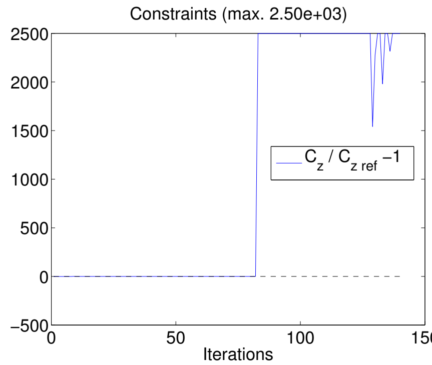

Figure 5.22: Constraints: Design of rigid body in space with objective: Minimizing volume, constraining displacement by making use of symmetry, $C_{x\ ref} = 10^{-3}\ J$

5.5 Intermediate densities at the boundary of the design

The linear density filter has an effect on the boundaries of the structure per definition. Density material in the structure is spread out over the boundaries to void elements, creating low density material around the edges. The influence of the filter radius r_{min} can be limited by forcing the constraint to distribute the densities over three elements in a radius. This is not in favor of the thickness of the thinnest part since the minimal feature size will scale with the radius. The linear density radius is independent of the mesh resolution and will therefore not be affected by resolution changes.

In order to prevent elements with intermediate densities on the boundary of the design, projection methods are developed by B.S. Lazarov [44]. This projection method based on erosion, intermediate and dilation projections by using the Heaviside function assures discreteness of the design.

5.6 Conclusion

In general, the problem formulations are non-convex and highly dependent on the initial conditions such as required compliance and volume. The user needs to define realistic constraints before using TO.

Minimization of volume gives the most discrete designs, while the compliance in every DOF is defined according to the constraint. Minimization of the unwanted compliance versus the maximization wanted compliance gives the most distributed designs, but they suffer from low density material around the edges.

Constraining DOF in TO is most effective by introducing symmetry in the design. This limits TO designs to translations parallel to the symmetry plane for a single input. Displacement constraints due to a translation or rotation reduces the compliance because of the applied force, resulting in either stiff or infeasible designs.

Volume minimization is the best strategy for the design of optical mounts. A lower bound compliance constraint for compliance in the wanted displacement assures minimal compliance while an upper bounded compliance in the unwanted displacement assures minimal stiffness in the other DOFs.

For TO every DOF of the optical mount should be separated in a design space which can be optimized for a tailored compliance. These tailored compliant structures combined (build on top of each other) form the optical mount.

Any structures with high compliance in multiple DOF are not favorable in practice. The operator (KLA-Tencor™) prefers independent control of DOF to prevent uncontrollable crosstalk between DOF.

Chapter 6

Topology Optimization of compliant mechanisms

Mobility of a compliant mechanisms distinguishes a compliant mechanism from a structure. In previous chapter a structure was designed to support external loads with an allowable compliance, while in this chapter a compliant mechanism is designed to deliver output displacements via deformation of the material. Which means that flexibility is a significant factor for the mobility of compliant mechanisms. Besides, compliant mechanisms need to be stiff to resist external loads. The optimal design of compliant mechanisms has therefore two conflicting design criteria. First, the structure needs to be flexible enough to give the output displacement, and secondly be stiff enough to support external loads.

Formulations with this combination of criteria is called stiffness-flexibility multi-criteria formulations. Many multi-criteria formulations have been developed, such as the ratio form of mutual strain energy (MSE) and strain energy (SE) [45], Mechanical Advantage (MA), Geometric Advantage (GA), and Mechanical Efficiency (ME) [46].

Essentially, all the formulations are of a form with the ratio of mutual strain energy and strain energy. Strain energy (compliance) is a global measure of the displacements is the structure under the prescribed boundary conditions. The lower the strain energy the higher the stiffness of the structure. The mutual strain energy represents the displacement of the output. By maximizing the ratio of mutual strain energy and strain energy, the output displacement and the stiffness can be maximized.

6.1 Single input, single output mechanism

To create a compliant mechanism two design criteria must be combined in TO. By introducing two load cases, one for flexibility and one for stiffness of the mechanism, a compliant mechanism can be designed. The first load case (figure 6.1) introduces a input force on the design domain, while noticing the deflection at the output, assuring flexibility of the mechanism. The second load case introduces a pseudo force on the output while the input is fixed, defining the stiffness of the mechanism (figure 6.2).

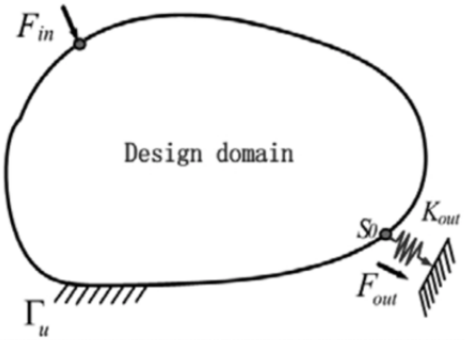

Figure 6.1: Load case 1: Input force. Reprinted form *Topology Optimization for Large-displacement Compliant Mechanisms Using Element Free Galerkin Method* by Y. Du and L. Chen (2008)

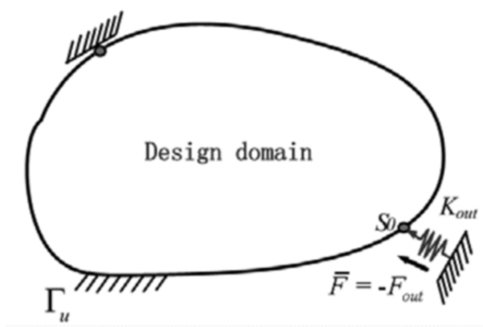

Figure 6.2: Load case 2: Pseudo load. Reprinted form *Topology Optimization for Large-displacement Compliant Mechanisms Using Element Free Galerkin Method* by Y. Du and L. Chen (2008)

6.2 Example 1: The compliant force inverter

The force inverter [47] is a compliant mechanism which is often used in TO as benchmark. Because of symmetry half the structure (finite elements) needs to be calculated and only one DOF for the input and output remains free. There are several ways of implementing objectives and constraints. The optimization problem for the force inverter example in this research is written as:

$$\begin{cases} min\ u_{out}(\rho) + C_{out}(\rho) \\ st. \begin{cases} C_{out} \leq C_{out\ ref} \\ u_{out} \geq u_{out\ ref} \\ V \leq V^* \\ \rho_{min} \leq \rho_e \leq \rho_{max} \quad e = 1, ..., n \end{cases} \end{cases} \tag{6.1}$$

The design domain (half of the full domain) is discretized with 120 by 75 elements of $1x1x1$, the filter radius is R = 1.5, the input force is $F_{in} = 1N$, the pseudo force $F_{out} = 1N$ and the input and output spring stiffnesses are $k_{in_z} = 1$ and $k_{out_z} = 0.01$, respectively. The walls in the design are 5 elements high (figure 6.7). The solid $\rho = 1$ E modulus of $E = 2.1E5N/mm^2$ and $nu = 0.33$ represents steel as material.

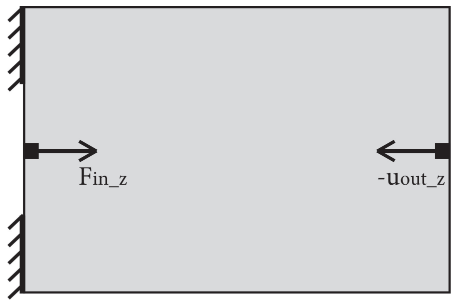

Figure 6.3: Schematic representation of the force inverter

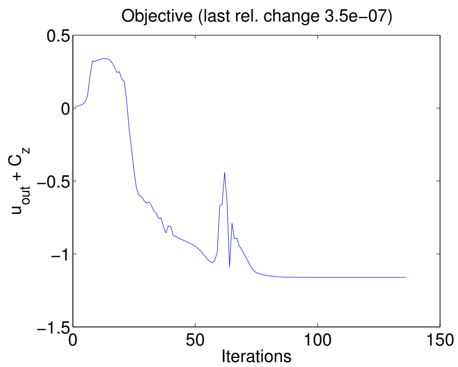

Figure 6.4: Objective: Force inverter

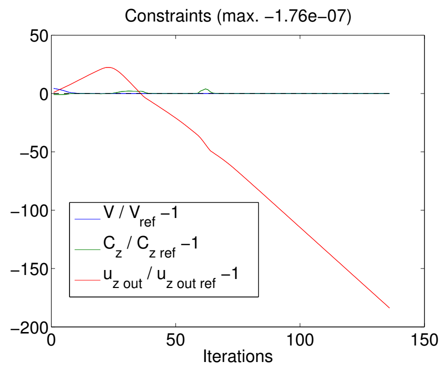

Figure 6.5: Constraints: Force inverter

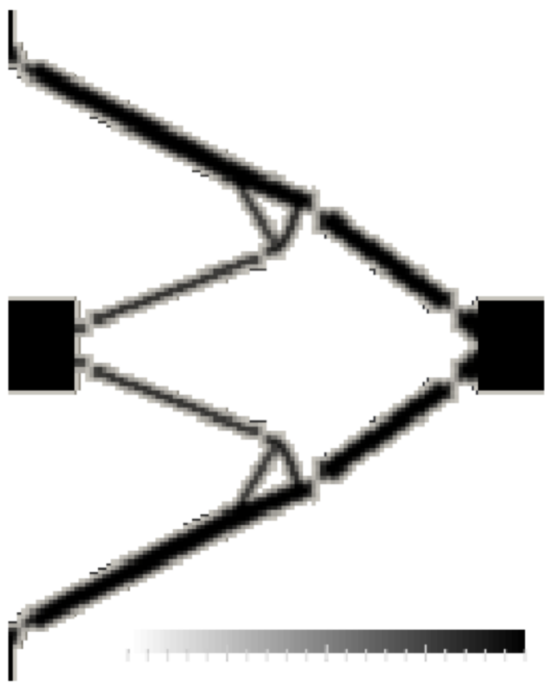

Figure 6.6: Design of the force inverter

Discussion on design:
The design of the force inverter resulted in solid structures with lower density material 3 elements wide lumped flexures. The node connections in this design have the advantage that they are good in transmitting forces but deliver no resistance to bending. For higher stiffness of the output spring, the hinge-like connection becomes more solid (distributed compliant) on the cost of smaller output displacement.

The artificial spring attached to the input simulates the character of the input by its blocking force F_{in_z} and its free displacement. The spring at the output port simulates the resistance from a work piece which is necessary for creating stiffness in the structure ('If you cannot feel resistance, you also do not know what you push away').

6.3 Example 2: Rotation of a rigid body mechanism

A rigid body (20 x 20) is rotated by an input (15 x 5 elements) in a design domain of 150 x 100 elements of $1x1x1$ mm. This example exists out of 4 load cases: In the first load case a force of $F_{in} = 20$ N actuate the input while the rotation and displacements are noticed. In the second load case the input is fixed while the output is actuated by a force of $F_x = 100$ N. In the third load case the output is actuated by a force of $F_z = 100$ N while the input is fixed. In the fourth load case, the output is actuated by a moment of $M = 400$ Nmm while the input is fixed. To constrain the undesired displacements of the output and input, distance constraints are used in the TO algorithm. $u_{z,x\ out\ ref}$ and $u_{z\ in\ ref}$ are 0.1 mm. The $C_{x\ out\ ref}$, $C_{z\ out\ ref}$ and $C_{M\ out\ ref} = 10^{-6}J$. The translational and rotational stiffness of this example having the stiffness requirements of an optical mount mechanism designed by KLA-Tencor (first Eigen-frequency 100Hz +).

$$\begin{cases} max\ u_{M\ out}(\rho)(counterclockwise) \\ st. \begin{cases} C_{z\ out} \leq C_{z\ out\ ref} \\ C_{x\ out} \leq C_{x\ out\ ref} \\ C_{M\ out} \leq C_{M\ out\ ref} \\ -u_{z\ M\ in} \geq u_{z\ M\ in\ ref} \\ u_{z\ M\ in} \leq u_{z\ M\ in\ ref} \\ -u_{z\ M\ out} \geq u_{z\ M\ out\ ref} \\ u_{z\ M\ out} \leq u_{z\ M\ out\ ref} \\ -u_{x\ M\ out} \geq u_{x\ M\ out\ ref} \\ u_{x\ M\ out} \leq u_{x\ M\ out\ ref} \\ V \leq V^* \\ \rho_{min} \leq \rho_e \leq \rho_{max} \qquad\qquad e = 1, ..., n \end{cases} \end{cases} \tag{6.2}$$

Discussion on design:
A stiff structure is designed due to the low compliance constraints (figure 6.10). Optical mounts need to stiff in every DOF to meet the requirements of KLA-Tencor for optical mounts. The displacement constraints give allowance for 0.1 mm offset of displacement when the rigid body is rotated by the input (figure 6.11). The rotation objective is chosen to let the mechanism naturally bend when the input is actuated. There is no convergence of the problem when the direction of rotation for the objective is chosen clockwise.

6.4 Multiple input, single output mechanism

This section will discuss a multiple input, single output mechanism. Optical mounts have often multiple adjustments screws which adjust a single mirror. By reducing the influence of the DOF from one screw to the other DOF, crosstalk between the adjustments can be limited. In this section two inputs act on a single output. In the literature, rigid body in space are actuated by 3 inputs to fully constrain 3 DOFs with TO. With optical mounts the

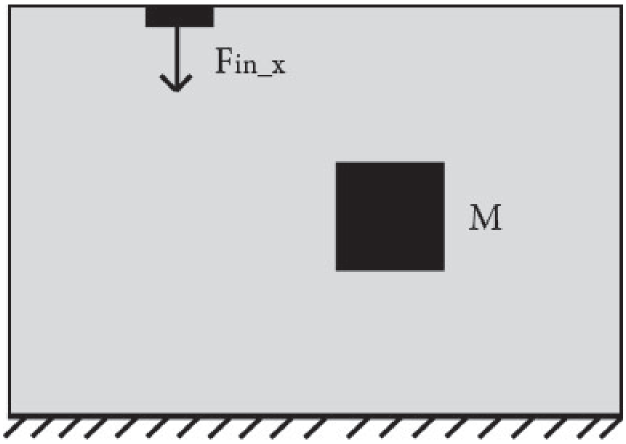

Figure 6.7: Schematic representation of the rigid body in the design space

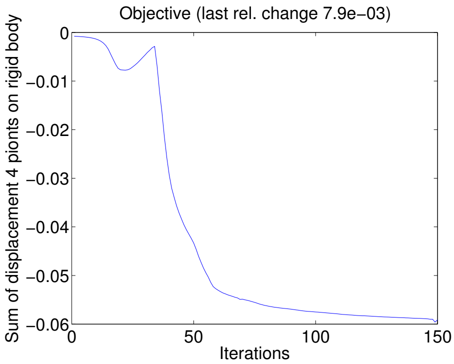

Figure 6.8: Objective: Rotation of a rigid body mechanism

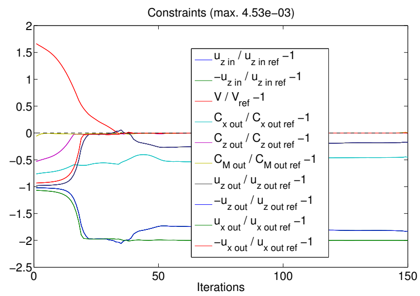

Figure 6.9: Constraints: Rotation of a rigid body mechanism

operator is only able to control 2 input screws at the time.

For this research a minimal bounded formulation is used to define multiple output directions in one structure. While one direction of the output due to the actuation is maximized, the other output direction due to the other input force needs to be at least as minimal actuated as the other. Also, when both inputs are actuated the output needs to move in both directions at the same time.

6.5 Example 3: Translations of a rigid body with two inputs

The last example uses two inputs, Fin_x and Fin_z to control the displacements of one rigid body (figure 6.12). The design domain is $150x100$ elements of $1x1x1mm$ and the inputs have a size of $15x5$ and the output is centered and has a size of $10x10$ elements. Fin_x and Fin_z are 30 N. Five load cases are used: First, Fin_x is fixed while Fin_z actuates the body in z direction. Second, Fin_z is fixed while Fin_x actuates the body in x direction. Third, Fin_z and Fin_x actuates the body in x and z direction. Fourth, Fin_z and Fin_x are fixed while the compliance of the rigid body is measured in x direction. Fifth, Fin_z and Fin_x are fixed

Figure 6.10: design of the rigid body rotation mechanism

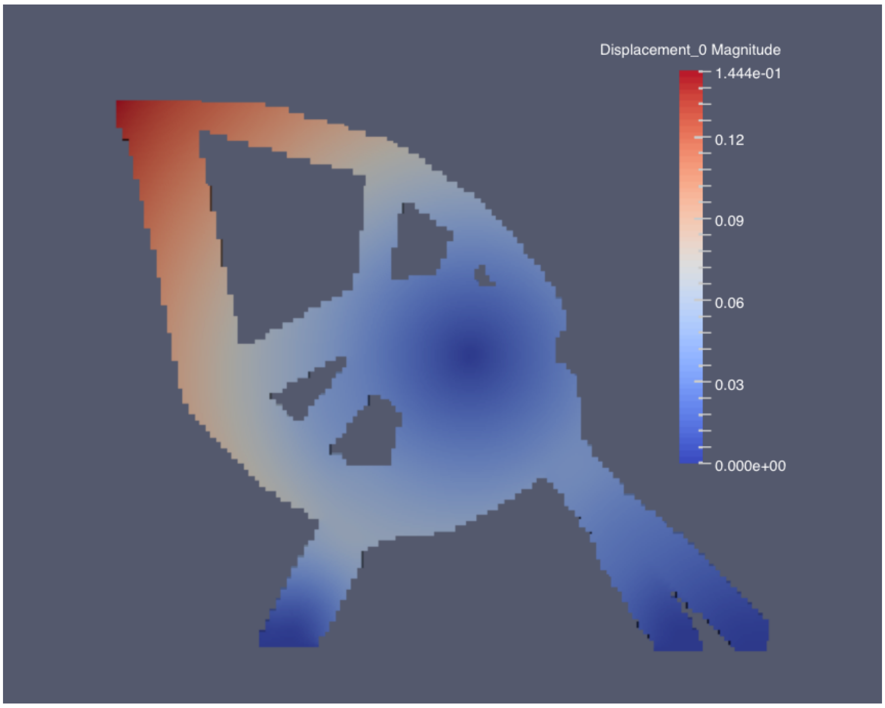

Figure 6.11: Rotation of rigid body due to input force $F_{in} = 20\ N$

while the compliance of the rigid body is measured in z direction. The following objective and constraints are used:

$$\begin{cases} max\ u_{x\ out}(\rho) = A \\ st. \begin{cases} C_{z\ out} \leq C_{z\ out\ ref} \\ C_{x\ out} \leq C_{x\ out\ ref} \\ C_{z\ in} \leq C_{z\ in\ ref} \\ C_{x\ in} \leq C_{x\ in\ ref} \\ -u_{z\ out} \leq -u_{z\ A} \\ u_{x\ out} \leq u_{x\ A} \\ -u_{x\ out} \leq -u_{x\ A} \\ u_{xz\ out} \geq u_{xz\ A} \\ -u_{xz\ out} \geq -u_{xz\ A} \\ u_{z\ out} \geq u_{z\ A} \\ C_{M\ out} \leq u_{M\ out\ ref} \\ V \leq V^* \\ \rho_{min} \leq \rho_e \leq \rho_{max} \qquad e = 1, ..., n \end{cases} \end{cases} \tag{6.3}$$

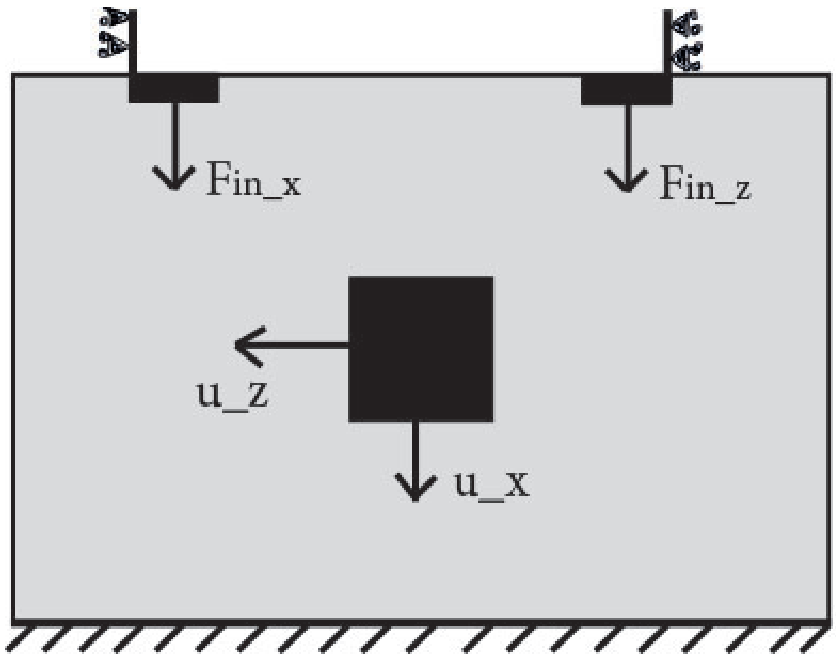

Figure 6.12: schematic representation of Two translations in x and z direction of a rigid body

Discussion on design:
This optimization problem ends with an undefined cloud of material. The minimal distance constraints (A) cannot be feasible. The optimizer has a lot of troubles to satisfy all constraints. Every time when the displacement in the x direction of the output moves further, constraints have to be satisfied again since they need to move at least the same distance as the rigid body has in the x direction.

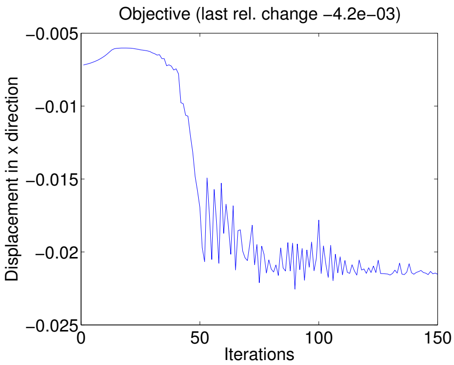

Figure 6.13: Objective: actuation of rigid body by two inputs

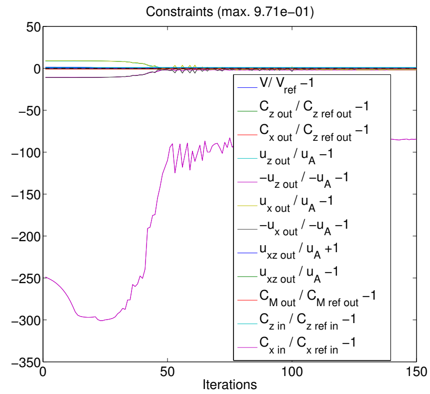

Figure 6.14: Constraints: actuation of rigid body by two inputs

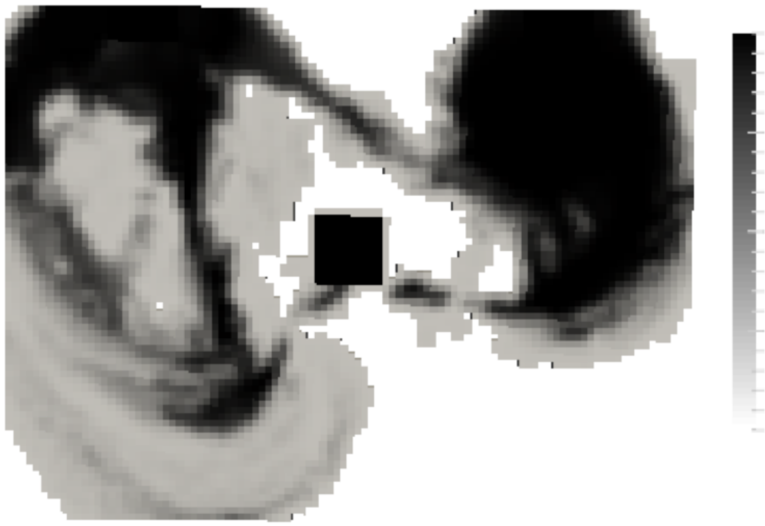

Figure 6.15: Two translations in x and z direction of a rigid body

6.6 Topology optimization for compliant optical mounts mechanisms

In this chapter several cases of compliant mechanisms are discussed. The first example, force inverter, is simple TO problem due to the symmetry only one DOF has to be controlled in the structure what brings almost discrete mechanisms. In this mechanism the input output relationship was not constrained. This relationship is necessary for optical mounts. Sensitivity of the mechanisms determines the accuracy of adjustment.

A realistic second case study is performed where the structure had to satisfy the mechanical stability constraints of the design team of KLA-Tencor. This design is far more stiff than other mechanisms seen in the literature. Kinematic mounts have screws to fasten the mechanism when it is calibrated, reducing the stiffness constraints of the mechanism for actual use. Since compliant mechanisms deform and translate through their entire structure their stiffness must come from their structure and cannot be constrained.

The last example was an attempt to design a mechanism with limited crosstalk. Many attempts are made to find the right volume to stiffness ratio but this was unsuccessful.

During my thesis TO of a real optical mount has been performed (tip-tilt mount) (figure 6.16) on basis of figure 1.1. This mount is able to tip-tilt 1 deg. in each direction with less than 1/100 crosstalk using the same bounded formulation as in example 3. Low density material around the structure made the design unfeasible to manufacture.

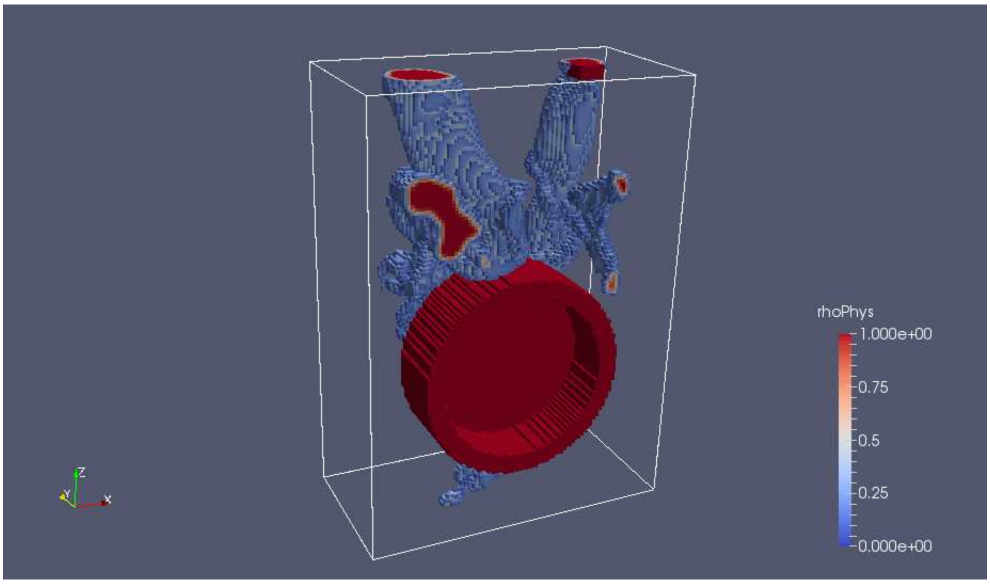

Figure 6.16: Tip-tilt mirror mount

Part III

Conclusions, Recommendations and Outlook

Chapter 7

Conclusions

Based on the functional and environmental industrial requirements in this research, flexure mounts are the most suitable type of optical mounts, based on internal compliant mechanisms.

Technical analysis and practical tests conducted with a representative commercially available flexure mount (Siskiyou tip-tilt mount), showed the occurrence of hysteresis the process of adjusting optics. Siskiyou mounts have both kinematic mechanisms and flexible components, and their motion is a combination of the motion permitted by the kinematic pairs and the deformation of the flexible components. Improvements in usability can be made to replace the kinematic 'transmission' into a fully compliant mechanism, preventing hysteresis and play in optics displacement.

This research is a step forward of design improvements of optical mounts by considering Additive Manufacturing (AM) as production method to procedure monolithic compliant mounts mechanisms without hysteresis and play. Topology Optimization (TO) is considered as potential structural design technique for designing compliant structures and mechanisms. Since the full implementation of AM into TO is not fully matured yet this thesis has a bilateral objective.

Based on this research the following conclusions are conducted:

The first research objective investigates the design of compliant mechanisms for optical mounts by means of additive manufacturing:

- The following conclusions of general design strategies are based on the design process of the optical alignment module(figure 3.2).
- Titanium (Ti6Al4V) is the most suitable material for AM of compliant structures in ultra clean environments, due to the lack of outgassing and optimal material properties for compliant structures.
- General AM constraints, such as minimal feature size, overlap and overhang, need to be respected during the design process.
- Design of well orientated flexures is required, satisfying the overlap constraint and minimal feature size of the AM process. It must be prevented that any supporting structures are needed to manufacture flexures. Damaging trough post processing affects the compliant properties of the optical mount.

- Supporting structures to improve the rigidity of compliant structures for post processing is needed. The compliant mechanism will deform due to work-tool forces without extra support.

The second research objective investigates the suitability of Topology Optimization for designing compliant mechanisms for optical mounts:

Optimization of compliant structures:

- Problem formulations are non-convex and highly dependent on the initial conditions such as required compliance and volume. Before optimizing a tailored compliant structure realistic constraints need to be take into account.
- Volume minimization is the best strategy for the design of optical mounts. A lower bound compliance constraint for compliance in the wanted displacement assures minimal compliance while an upper bounded compliance in the unwanted displacement assures minimal stiffness in the other DOFs.
- DOF of the optical mount should be separated into separated design spaces which individually can be optimized for a tailored compliance. These individual tailored compliant structures combined (build on top of each other) form the optical mount.

Optimization of compliant mechanisms:

- Compliant mechanisms can best be designed on a symmetry plane. Less constraints need to be formulated to control DOF which increases the computational speed and lowers intermediate density material of structures.
- Compliant mechanisms tend to be stiff when they need to meet the operational stiffness requirements. This constraint is less stiff when the mechanism can be fixated.
- Designing compliant mechanisms in 3D space brings in a lot of constraints. Per element (input, output) of the structure has 6 DOF to be properly constrained. The MMA optimizer cannot follow all of them. (3D mount case had 22 constraints).

Chapter 8

Recommendations and Outlook

8.1 Recommendations

The userfrienliness of flexure mounts currently available on the market (Siskiyou) can be improved by replacing the kinematic for a compliant (transmission) mechanism, reducing hysteresis in optics adjustments.

The angle of displacement of the adjustment screw is not in line with the displacement of the kinematic transmission, which creates friction. In order to prevent friction the adjustment screw should move in parallel with the contact surface of the adjustment mechanism.

The initial density in the design space is homogeneously distributed during all TO runs. Other layouts of the distribution can lead to new final designs. As seen in Example 4 in Chapter 5, compliance of a rigid body on a symmetry axis is limited in the direction of motion by a beam. A more logical solution is to design a flexure beam perpendicular to the symmetry axis to allow more compliance in de structure. A recommendation is to incorporate initial designs into the TO which brings the design to antother possible better design optimum.

Mesh dependency influences the outcome of the optimization. The optimization might converge to a completely different and more complex topology than the one obtained using the coarse mesh. Further investigation of the compliant mechanisms designs needs to be conducted to assure an optimal design.

The linear density filter and/or a combination of constraints introduces intermediate densities in the mechanisms. Projection methods from/or based on [48] can create discrete mechanisms. These methods work well on symmetry constrained TO, where unfavorable displacements are constrained by the symmetry axis. Erosion and dilution have more effect on free-moving rigid bodies where a slight change of density in the end violates compliance constraints. Deepak et. al. (Tu Delft 2016) developed a method where compliance maximization can be combined with volume minimization to reduce intermediate densities. I have implemented his method on the beam of Example 1 of chapter 5, with no successful results (due to relaxation of the objectives, no convergence was reached). I suggest a further and deeper research of his method for compliant mechanisms.

To assure the right stiffness in the design of an optical mount, TO compliance constraints are applied on the structure and mechanism to constrain minimal required stiffness. Further research can focus on Eigen-frequency constraints to assure minimal stiffness in the structure. Components in high precision optical systems needs to have at least a first Eigen-frequency of 100 Hz to be installed in the optical system.

Saxena and Mankame [49] proposed to discretize a design domain into a honeycomb pattern using hexagonal units. Each hexagonal unit is divided into two four-node finite elements for analysis. Since hexagonal units in the assembly only have edge-edge connection, point flexures can be completely eliminated.

This research did not take stress constraints into account. Current research is conducted on efficient stress constraint optimization. Since the current stress optimization is clustered into groups using a modified P-norm to decrease the number of stress constraints and thus the computational cost, it is not accurate for high local stresses such as in lumped compliant mechanisms. Still, high yield stresses need to be controlled in future compliant TO designs.

Process uncertainties from TO mesh to the final post processing steps have influence on the structure. Critical elements such as flexures are the most vulnerable parts of the design and change the function of the mechanism when they differ from the TO simulation geometry. TO can be extended by incorporating estimations such as from printing inaccuracies and beat blasting.

Flexures with an overhang cannot be supported with extra material since post processing will damage the delicate structures. A TO strategy will improve the design layout in such a way that no overhang occurs in flexures.

Support material can be utilized as usefull material for compliant mechanisms when post processing is needed (milling, drilling, etc). A TO strategy can determine the best layout of support material satisfying the post production constraints such as reactions forces of a drill.

Penalization in the SIMP method for TO is unnecessary, if intermediate densities can be manufactured. Following three possible implementations:

- New micro lattice structures can represent the structural properties of intermediate densities (figure 8.1). By interpolating the fictive greyscale result and replacing each finite element with a lattice structure representing the fictive Young's Modulus of each unitcell, a continuous merging of lattice structures can be achieved. Note: Material properties of low intermediate density cannot be represented by lattice structures.
- To some extent, the density of the manufactured component can be controlled by varying the processing parameters of SLM. The laser input power has a significant effect on the porosity of the part. Research has found that effective materials are produced above 60% porosity [50].
- Intermediate densities from the SIMP method could also be classified as materials with different densities. SLS and SLM AM are able to process several mixtures of metal powder with limitations [51].

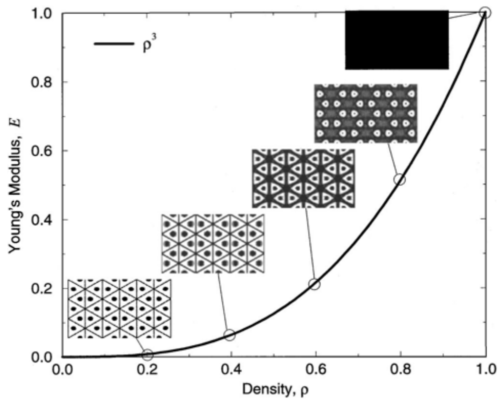

Figure 8.1: Micro structures realizing the material properties with $p = 3$ and $v = 1/3$ Reprinted from *"Material interpolation schemes in topology optimization"* by M. P.Bendsoe and O. Sigmund, Archive of Applied Mechanics, 1999

8.2 Outlook

Optical systems for the semiconductor industry will be more complex and will simultaneously have a limited surface area in semi-conductor equipment. Substantial space can be saved by integrating functions of several optical mounts into one single mount. Furthermore, the usability of the current mounts can be increased by placing the adjustable screws in a more convenient place for the user.

AM offers novel layouts for optical mounts, since taylor-made AM mounts can be specifically designed for applications where standard off-the-shelf mounts do not fit or do not meet the requirements. The performance results of the optical beam alignment module (figure 3.2) [7] are inspirational for further innovation of optical mounts.

TO is a powerful design tool for non-compliant (mostly convex) structures. Future works must prove if compliant TO is suitable for designing any compliant structure or mechanism. At the current state, complexity and non-standardized approaches limit the designer's possibilities of constructing proper compliant mechanisms.

AM is a suitable method for producing topology optimized structures. Strategies for AM needs to be implemented in TO in order to meet the manufacturing requirements of the process. Future topology optimized designs will very likely incorporate AM constraints, acknowledging TO as powerful design tool for AM.

Since tailored software tools are currently improving and since AM constraints can be incorporated during the design process, the design of parts for AM will be commercially more accessible in the future, accelerating the adoption of 3D printed parts in the equipment.

AM built simulations will be incorporated in CAD systems to predict production deformations in the manufacturing process that are related to thermal stress. The company Autodesk™ already developed a software tool (Nettfab Simulation®) available in beta-phase.

Currently, powder bed fusion is the mainstream AM technology for metals. The most prominent limitations are restricted overhang and a poor surface area. Innovation of AM techniques will decrease these limitations. For example, airplane manufacturer Boeing™ developed an

AM technique (patent publication number: US20160031156) without overhang constraints, which allows a metal part to grow in 3 dimensions by means of super magnets and lasers.

The cost per 3D printed metal part will further decrease due to scaling considerations as well as due to the integration of pre- and post processing in the AM batch process. Engineering companies will not likely move powder bed fusion technology AM production in-house, but outsource production to large manufactures since they have advantages from large scale production benefits [52].

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Poster presentation: Additive Manufacturing

Poster presentation at the Annual Engineering Conference KLA-Tencor (October 29th, 2015)

AM Design: *Topology Optimization*

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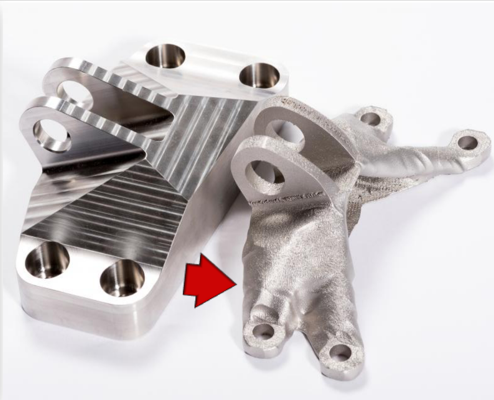
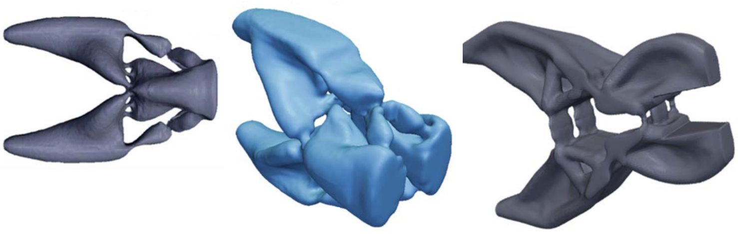
Workshop:
3D Printing for Designers
October 29th, 2015

The Reality of Designing for AM

- More design freedom and AM constraints results in higher complexity for engineer
- Benefits of AM can be used for optimal design
  - Material, Shape, Topology, Support structures
- Conventional designs are never optimal for AM
- Necessity for supporting ''design tool'' that takes advantages of AM

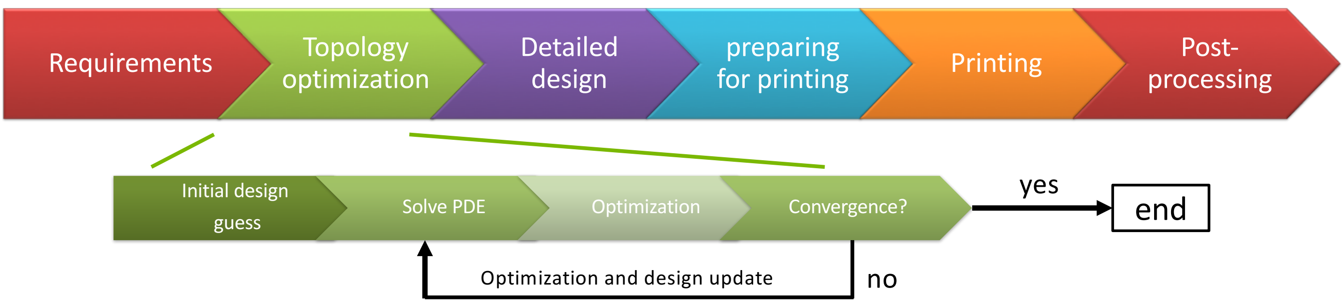


Topology Optimization as a tool for AM



- Software design tool
- Provides Optimal material layout for given function and constraints
- Part of constraints are related to AM
- AM suitable for manufacturing organic structures designed with TO

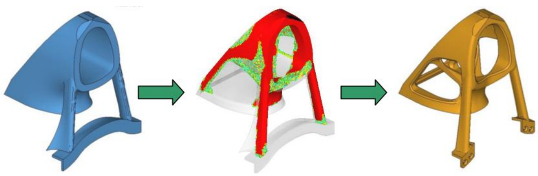
Design Process with Topology Optimization



Current Optimization tools for AM on the market

Altair Optistruct

- 1st phase: Topology Optimization
- 2nd phase: Adding and optimizing spatial density of lattice



Autodesk Within

- Optimization spatial density of lattice
- Optimization of skin thickness