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Analyse de la saisie d'objets déformables par des pinces adaptatives

NICOLAS MOUAZÉ
Département de génie mécanique

Thèse présentée en vue de l'obtention du diplôme de *Philosophiæ Doctor*
Génie mécanique

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POLYTECHNIQUE MONTRÉAL
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Cette thèse intitulée :

Analyse de la saisie d'objets déformables par des pinces adaptatives

présentée par **Nicolas MOUAZÉ**
en vue de l'obtention du diplôme de *Philosophiæ Doctor*
a été dûment acceptée par le jury d'examen constitué de :

Richard GOURDEAU, président
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Frédérick GOSSELIN, membre
Pierre RENAUD, membre externe

DÉDICACE

À mon parrain.

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RÉSUMÉ

L'évaluation des interactions entre systèmes robotiques et leurs environnements est un paramètre critique lors de la conception de ces systèmes, d'autant plus lorsque cela implique de la saisie. L'étude, ainsi que la commercialisation de pinces dites sous-actionnées se développent depuis quelques années, mais pour l'instant leurs utilisations se sont limitées à la saisie d'objet rigide. Lors de cette thèse la saisie d'objet déformable par des pinces sous-actionnées est considérée. L'objectif étant de proposer un modèle simple qui permet de simuler une telle interaction pour pouvoir ensuite réaliser des optimisations d'architectures de pince afin de maximiser leur avantage principal qui est l'adaptation mécanique. En effet, les pinces sous-actionnées adaptent mécaniquement (et donc sans contrôle complexe) leurs formes, c'est-à-dire la position de leurs phalanges à l'image des doigts humains, à la forme des objets qu'elles saisissent.

La méthode commune pour simuler les déformations d'un solide, la méthode classique des éléments finis, n'est pas adaptée lorsque l'on veut simuler rapidement des interactions, par exemple pour des applications temps réels ou des optimisations, de par sa complexité. Donc, dans cette thèse un modèle utilisant des points pour discrétiser la géométrie de l'objet reliés par des ressorts pour approcher la rigidité est choisi. L'objectif est de proposer un modèle complet doigts/objet permettant de simuler les interactions simplement et surtout rapidement, tout en gardant une certaine précision.

Dans cette thèse, deux approches sont présentées pour faire le lien entre le modèle de l'objet et celui des doigts. D'abord une approche énergétique, où en fonction du travail apporté par l'actionnement, on cherchera à minimiser l'énergie emmagasinée dans le système complet, puis une approche directe avec le principe fondamental de la statique. Dans ce dernier cas, il a fallu développer les formules connues de force de contact de doigt sous-actionné, afin de permettre le calcul lorsqu'il y a plusieurs contacts par phalange du doigt. Finalement, le second modèle qui présentait de meilleurs résultats que le premier a été utilisé afin d'optimiser une pince sous-actionnée afin de minimiser les déformations des objets qu'elle saisit. De plus, cette pince a été conçue afin de ne nécessiter aucune étape de montage lors de sa fabrication. Au final, lors de cette thèse un modèle numérique rapide permettant de calculer les interactions objets déformables/pinces mécaniques à été développé, et un prototype de pince optimisé grâce à ce modèle a été développé.

ABSTRACT

Evaluations of the interactions between robotic systems and their environments is a critical parameter when designing these systems, especially when it involves grasping. The analysis as well as the distribution of underactuated grippers have been developed for a few years, but currently their use is limited to grasping rigid objects. In this thesis, the interactions with a deformable object by underactuated grippers are considered. The goal is to propose a simple model which makes it possible to simulate such an interaction in order to be able to carry out optimizations of the clamp architecture. The objective of the optimization is to maximize the main advantage of underactuation, the mechanical adaptation. Indeed, the underactuated grippers mechanically adapt (therefore without complex control) their shapes, the position of their phalanges, to the shape of the objects they grasp.

The most common method to study the deformations of a solid, the finite element method, is not suitable when the goal is to quickly simulate interactions, for example for real-time applications or optimizations, due to its complexity. So in this thesis a model using points to discretize the geometry of the object and springs to approach the rigidity is chosen. The objective is to propose a complete finger/object model allowing fast simulations of interactions.

In this thesis, two approaches are presented to make the link between the models of the object and the fingers. First an energy approach, i.e. depending on the work provided by the actuation, the energy stored in the complete system is minimized to calculate the position of the system, then a direct approach with the fundamental principle of statics. In the latter case, it was necessary to develop the formulas for the contact force of the underactuated finger already developped, to calculate cases when several contacts per phalanx of the finger happen. Finally, the second model, which presented better results than the first one, was used to optimize an underactuated gripper to minimize deformation of the objects it grasps. Moreover, this gripper has also been designed to require zero assembly step during its manufacture.

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LISTE DES SIGLES ET ABRÉVIATIONS

ASME	American Society of Mechanical Engineers
DDL	Degré(s) de liberté
DOF	Degree(s) of freedom
IEEE	Institute of Electrical and Electronics Engineers
MEF	Méthode des éléments finis
MMR	Modèle masse-ressort
MPR	Modèle pseudo-rigide
PRBM	Pseudo-Rigid Body Model
PSM	Point-Spring Model

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CHAPITRE 1 INTRODUCTION

1.1 Problématique

La saisie d'objet est une action que toute personne réalise instinctivement. La main s'adapte sans réfléchir à la forme des objets que l'on souhaite attraper afin de ne pas les écraser. De leur côté, pour saisir, les pinces robotiques vont quant à elle généralement soit pincer, soit être adaptées à un objet, soit utiliser de systèmes complexes composés de moteurs combinés à des capteurs afin d'adapter leurs formes. Si l'on désire réaliser une pince, qui doit par exemple saisir des tomates, dont les caractéristiques varient d'un spécimen à l'autre (que ce soient leurs formes, leurs fermetés, etc.), les deux premières solutions ne sont pas envisageables afin de ne pas trop abîmer les légumes. Et cela est sans compter la difficulté supplémentaire que posent les possibles variations de leurs orientations. Si l'on considère l'utilisation de pinces robotiques sur une chaîne pour le tri de ces fruits, il faudra donc des systèmes potentiellement lourds, onéreux et encombrants. Les avantages des systèmes robotisés sont pourtant connus et ceux-ci sont déjà très présents dans l'industrie, que ce soit pour améliorer des tâches réalisées précédemment par un opérateur : augmentation de la productivité en accélérant les manipulations, répétabilité qu'une personne ne peut égaler même en prenant des pauses (qu'un robot ne prend pas), etc. De plus, ils rendent possibles des tâches auparavant dangereuses ou inaccessibles : ramassage de débris dans l'espace, actions sur des zones radioactives, opérations à très grandes ou très petites échelles, etc.

Afin d'améliorer leurs performances, des pinces construites avec des matériaux ou des architectures adaptatifs ont été développées. Parmi les technologies utilisées, les pinces dites sous-actionnées sont des systèmes avec un nombre d'actionneurs inférieur à leur degré de liberté (DDL). Ces mécanismes s'adaptent mécaniquement à la forme des objets à l'image du doigt humain. En outre, ils ont les avantages d'être moins chères, plus compacts et de nécessiter moins d'entretien que les systèmes classiques de par leur nombre réduit de moteur. Par ailleurs, leur capacité d'adaptation étant mécanique, leur système de commande se retrouve aussi simplifié, car il se limite souvent à une simple commande d'ouverture/fermeture. Plusieurs articles sur les mécanismes sous-actionnés présentent différentes architectures en analysant en particulier le mécanisme transmettant l'actionnement. Certains auteurs ont quant à eux travaillé sur la modélisation des forces de contact, ce qui a conduit aussi à l'étude de la stabilité de la saisie. En revanche, les études ont pour l'instant été limitées aux cas où les objets saisis sont considérés rigides. En reprenant l'exemple précédent, concevoir une pince pour saisir des tomates en considérant ces dernières rigides ne sera pas optimal

et cela entraînera de la perte. Pour une application où les dégâts auraient de plus fâcheuses conséquences, par exemple la conception d'outils chirurgicaux pour la saisie de tissu humain, l'hypothèse d'objet rigide n'est plus possible. Cette thèse cherche à répondre à ce manque : l'étude des interactions entre les pinces sous-actionnées et des objets déformables.

Pour étudier les interactions entre solides et en particulier les déformations plusieurs outils existent. La méthode des éléments finis est sans doute la plus connue. Elle est précise, mais aussi généralement lente et demandant beaucoup de ressources informatiques, en plus de ne pas être adaptée pour l'optimisation de mécanisme. Dans le domaine des effets visuels un modèle alternatif est utilisée, le modèle masse-ressort (MMR). Des masses et des ressorts vont approcher la géométrie d'un objet ainsi que la rigidité de son matériau afin de simuler ses déformations. Cette méthode est choisie pour sa rapidité et son adaptabilité, et peut donc être une alternative intéressante aux éléments finis pour l'optimisation de mécanisme. C'est cette méthode dont nous allons nous servir ici.

1.2 Objectifs

Pour faire suite aux travaux déjà réalisés sur les doigts sous-actionnés, l'objectif de cette thèse est d'étudier leurs interactions avec des objets déformables, ce qui malgré les nombreuses publications sur le sous-actionnement manque dans la littérature. Pour cela, il faudra mettre en place un modèle numérique qui sera comparer à des résultats expérimentaux. De plus, une contrainte principale a été posée au début du doctorat : trouver un modèle simple afin d'obtenir des simulations rapides qui permettront par la suite l'optimisation d'architectures de pince. Le modèle devra aussi être adaptable à plusieurs mécanismes de transmission de pince avec un passage rapide entre plusieurs configurations (transmission, mais aussi nombre de doigt/phalange, etc.) afin de les comparer. L'intérêt des doigts sous-actionnés étant leur adaptabilité mécanique, l'aboutissement de la thèse sera l'optimisation d'une pince à partir du modèle afin que celle-ci déforme le moins possible un objet en le saisissant et s'adapte au maximum.

1.3 Plan de la thèse

Cette thèse est rédigée par articles. Après cette introduction, une revue de la littérature est présentée afin de situer les travaux réalisés lors de ce doctorat par rapport à ceux existant. Ce chapitre est suivi de la méthodologie et des contributions apportées. Ensuite, les chapitres suivants sont des articles publiés ou soumis lors du doctorat, ainsi que des résultats complémentaires à ces chapitres. Finalement une discussion des résultats ainsi qu'une conclusion apportent des pistes d'amélioration future.

CHAPITRE 2 REVUE DE LITTÉRATURE

Afin de positionner cette thèse par rapport aux études existantes, une revue de la littérature est présentée dans ce chapitre pour résumer l'état de l'art dans le domaine des préhenseurs et en particulier des préhenseurs adaptatifs sous-actionnés. Ensuite, les mécanismes compliants souvent combinés aux doigts sous-actionnés seront introduits, puis les procédés de modélisation d'objet déformable. Enfin des exemples d'applications possibles de préhenseur seront présentés.

2.1 Les préhenseurs

Un préhenseur est un outil qui permet de saisir un objet entre ses doigts. La main humaine en est bien sûr un des exemples le plus immédiat, mais plus généralement on retrouve aussi ce terme en mécanique. On l'utilise que ce soit pour parler par exemple de simples pinces plates, mais aussi des pinces robotisées qui peuvent être montées sur des robots industriels, ainsi que des ventouses pneumatiques.

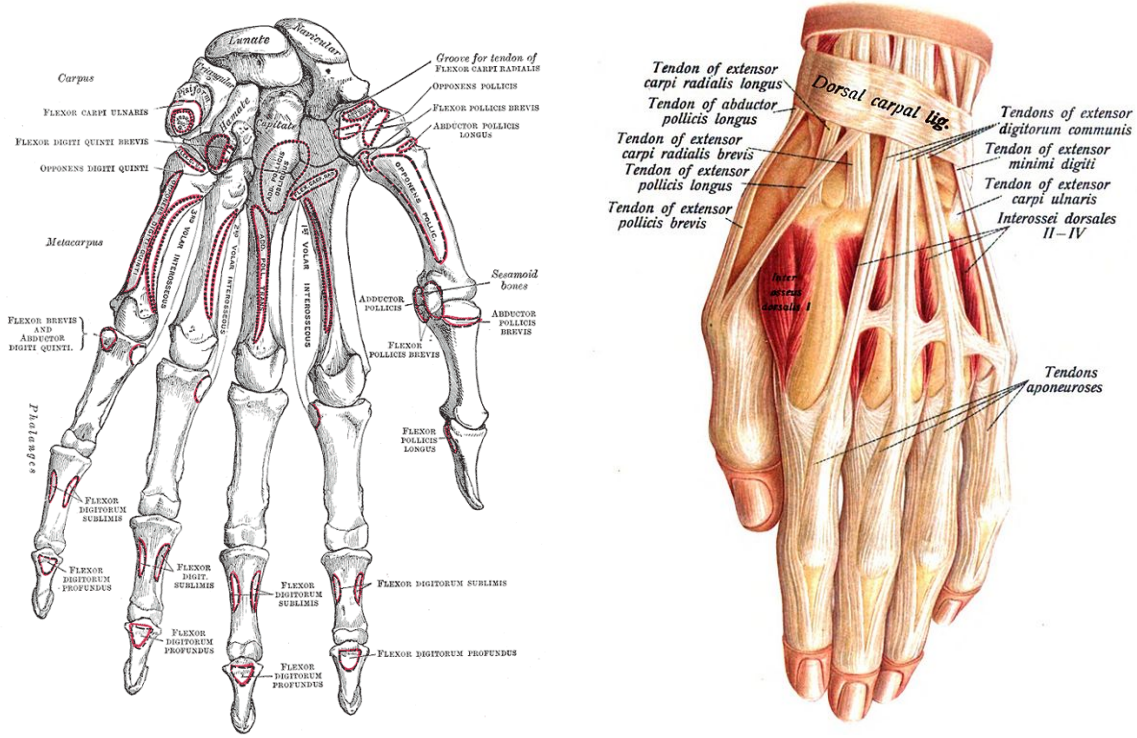
2.1.1 La main humaine

Atteindre les performances de la main humaine, en particulier sa dextérité, est souvent l'objectif ultime lors de la conception d'un préhenseur artificiel [1]. La figure 2.1(a) tirée de [2] illustre son squelette, son architecture, sur lequel on peut observer la séparation entre les trois parties qui la constitue. D'abord, le poignet qui est constitué du carpe en haut de la figure, ensuite la paume au centre avec les os du métacarpe et finalement les doigts. Chaque doigt a trois phalanges, excepté le pouce qui n'en a que deux. En observant seulement le squelette, les os du métacarpe peuvent être confondus avec les phalanges, mais ceux-ci font partie de la paume et non des doigts. Par analogie avec la mécanique, les os peuvent être considérés comme les membrures de la main, tandis que les articulations sont réalisées par l'imbrication des os entre eux, ainsi qu'aux ligaments qui les retiennent entre eux. Ces derniers sont là pour éviter les mouvements anormaux pouvant entraîner des luxations, des déboîtements d'une articulation. La forme des os va contraindre les mobilités possibles entre chaque os. Une façon de différencier la limite entre la paume et les doigts est d'examiner la forme des métacarpes au niveau de la zone en contact avec la phalange proximale. Celle-ci est plus arrondie que les zones des phalanges au même niveau (en contact avec l'os suivant). En effet, les articulations entre la paume et les doigts permettent des mouvements suivant le

plan de la paume, mobilités qui n'existent pas entre les phalanges. On parlera d'abduction et d'adduction, mouvements qui permettent de rapprocher ou d'écarter les doigts.

Les termes de phalanges distales, proximales et intermédiaires sont généralement utilisés pour parler des phalanges les plus distantes de la paume, les plus proches et enfin celles centrales respectivement. Sur la figure 2.1(b) (tirée de [3]) est illustrée une partie du système qui actionne le doigt, les tendons. Ceux-ci servent à transmettre les forces appliquées par les muscles aux os. Couplés aux gaines tendineuses reliant les phalanges, ils peuvent être vus comme l'équivalent mécanique d'un système mécanique de câbles et de poulies. Les tendons fléchisseurs situés du côté palmaire de la main vont servir à plier les doigts quand ils seront actionnés, tandis que les tendons extenseurs du côté dorsal que l'on peut observer sur la figure vont servir à les étendre. Le tendon interosseux lui n'est pas relié à un muscle, il est passif et seul son élasticité va engendrer des efforts. Il va aider soit pour la flexion soit pour l'extension en fonction de sa position par rapport à l'axe de l'articulation phalange proximale/phalange intermédiaire et cette position va changer en fonction de la configuration du doigt. Par exemple si le doigt est plus proche d'une position fermée, le tendon va aider la flexion et augmenter les forces qui seront appliquées potentiellement lors d'une éventuelle saisie d'un objet. D'autres tendons sont présents dans la main et servent aux autres mouvements de la main comme l'abduction et l'adduction déjà évoqués, mais dans cette thèse, nous nous attarderons seulement sur les mouvements de flexion et d'extension. Afin d'optimiser ces actions, la main possède aussi des capteurs dont évidemment plusieurs pour la perception du toucher. La main peut par exemple capter la température des objets qu'elle touche, leurs rugosités, etc. Ceci permet de sentir lorsqu'un objet glisse de notre main, s'il se déforme, etc.

La main et en particulier les doigts ont beaucoup été étudiés et même s'il reste encore des choses à comprendre sur leur fonctionnement, ils ont été beaucoup copiés [4]. L'objectif étant par exemple de créer des prothèses de main pour les personnes amputées ressemblant le plus possible aux membres manquants, même si cela n'inclut pas forcément la transmission des sens. La figure 2.2 tirée de [5] illustre d'ailleurs une prothèse de main du *Science Museum* à Londres datant du $19^{éme}$ siècle. À cela s'ajoute les études pour créer des préhenseurs pouvant réaliser des actions que l'homme ne peut pas lui-même faire, que ce soit pour des raisons environnementales (opérations dans un corps humain, à distance, dans une zone avec de fortes radiations nucléaires) ou afin de compenser ses propres limites. Parmi ces derniers, on peut citer la vitesse de ses mouvements, les possibles tremblements qui peuvent entraîner des perturbations, l'impossibilité de travailler à certaines échelles (systèmes microscopiques [6] ou à l'inverse trop grands), etc.



(a) Squelette de la main humaine (b) Tendons d'un doigt humain

Figure 2.1 La main humaine (images libres de droits tirées de [2] et [3])

Figure 2.2 Main artificielle (image libre de droits tirée de [5])

2.1.2 Les préhenseurs artificiels

Dans cette section, seront présentés différents préhenseurs. Ceux-ci ont pour fonction de saisir les objets en appliquant une force, plus ou moins forte sur l'objet. Les systèmes se servant de crochet, de mécanismes d'aspiration/magnétique ou encore de surfaces adhésives ne seront donc pas discutés ici.

Mécanismes à membrures rigides pleinement actionnés

La *Shadow Hand Lite* illustrée sur la figure 2.3 est un second exemple de préhenseur artificiel qui copie la main humaine [7]. Avec treize DDL l'objectif de la compagnie *The Shadow Robot Compagny* qui la commercialise était de créer une main artificielle dont la dextérité est la plus proche possible de la main humaine. Ce système illustre l'approche le plus intuitive pour s'en approcher, pour chaque phalange du squelette humain un corps rigide le remplace, et pour chaque DDL, correspondant au nombre de phalanges, un moteur couplé à des capteurs de position est utilisé pour le contrôler. Si le prix, le poids, ainsi que le volume ne sont pas des contraintes prises en compte lors de la conception d'un préhenseur, l'utilisation d'un moteur par DDL couplé à des capteurs de position permet d'obtenir un outil très précis tout en étant relativement simple et direct à mettre en œuvre. Par contre, dans certains environnements ces contraintes sont à prendre en compte. Le prix si l'on désire un grand nombre de pinces par exemple ou encore le volume et le poids si la pince est utilisée dans un espace restreint. On peut d'ailleurs noter que la version classique de la *Shadow hand* comporte cinq doigts, tandis que la version *Lite* (celle illustrée ici) n'en comporte que quatre. C'est un compromis qui a aussi été observé sur d'autres mains robotiques [8]. De plus, la précision se fait aussi au détriment de contrôles complexes, qui nécessitent des rétroactions de la part des capteurs.

Mécanismes à membrures rigides sous-actionnés

Les pinces illustrées sur les figures 2.4(a) et 2.4(b) sont des exemples de pinces ayant une architecture adaptatives tirées respectivement de [9] et [10]. Ces pinces adaptent leurs formes à leurs environnements en ayant moins de moteurs que de DDL. On parlera d'architecture sous-actionnée, car à la différence des mécanismes pleinement actionnés précédent ils ont moins d'actionneur que de DDL. Ces mécanismes permettent donc tout en gardant un même nombre de DDL de créer des systèmes plus légers, plus compacts et moins onéreux. La section 2.2 de ce manuscrit va présenter plus en détail les pinces sous-actionnées. Parfois, ces mécanismes utilisent des matériaux souples afin de créer des mobilités, mais leurs DDL restent quantifiables.

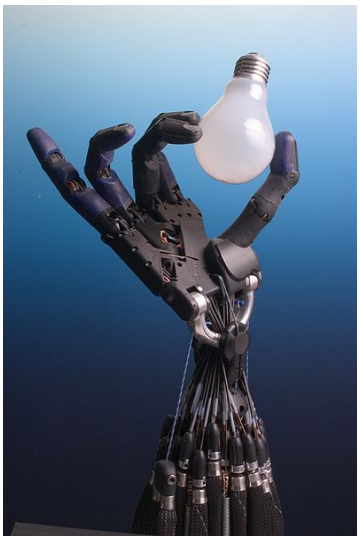
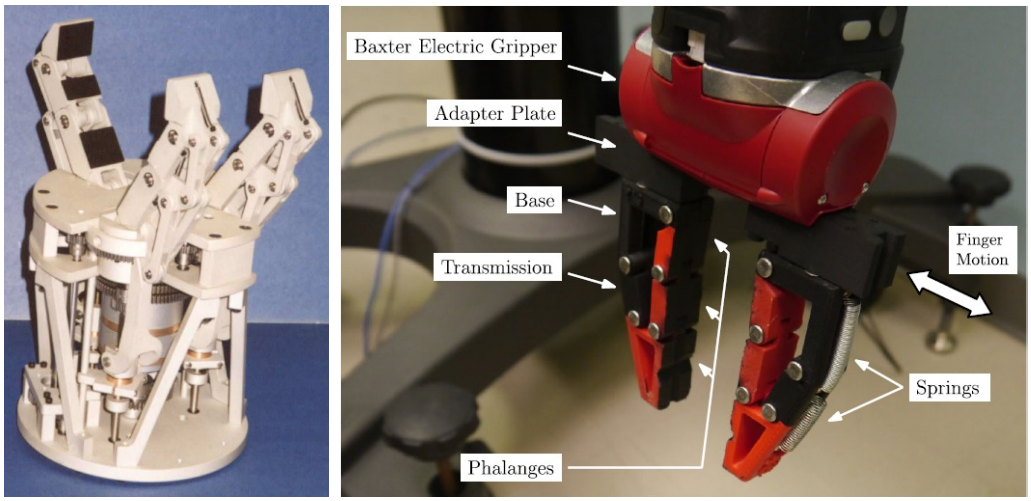


Figure 2.3 *Shadow Dextereous hand lite*, main robotisée pleinement actionnée (image libre de droits tirée de [7])



(a) (b)

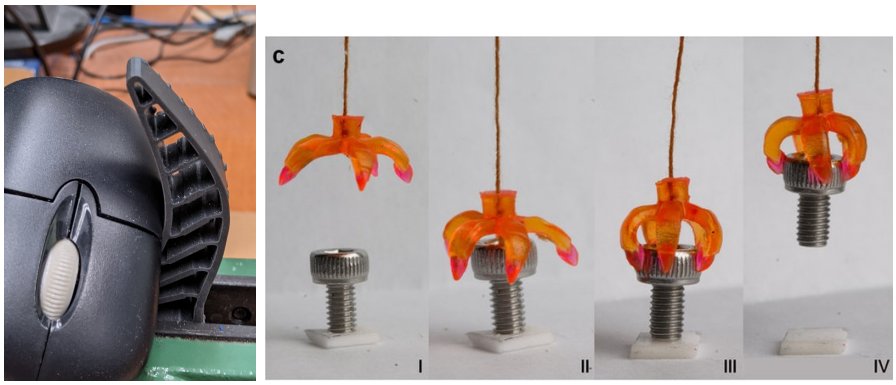
Figure 2.4 Pinces sous-actionnées (tirées de [9] et [10] avec l'accord du professeur Lionel Birglen)

Mécanismes à membrures souples

Enfin, contrairement aux pinces précédentes dont les phalanges sont rigides, une troisième approche consiste à utiliser des matériaux souples, c'est ce que l'on appelle les *soft grippers* (ou pinces souples) qui sont une sous-partie de la *soft robotic* [11]. Dans ce cas là, vu que l'intégralité de la surface des doigts va fléchir, il y a une infinité de DDL. Les pinces présentées sur la figure 2.5 sont des exemples de conception à partir de matériau souple (la figure (b) a été tirées de [12]. On peut observer qu'il s'adapte à la souris qui appuie dessus. Ce dernier exemple illustre une architecture que l'on retrouve beaucoup dans la littérature actuellement qui contrairement aux pinces présentées précédemment va plutôt se rapprocher à un tentacule des poulpes, avec une "infinité" de DDL, pour garder l'analogie avec un animal. Par contre les forces qu'elles peuvent développer sont encore faibles, encore une fois à l'image du poulpe qui doit aussi se servir de ventouse pour assurer la stabilité de ses saisies [13]. Ce type d'architecture est sûrement ce que l'on retrouve le plus en *soft robotic*, en plus des pinces sur lesquelles sont montés des blocs mous sur la surface des phalanges rigides pour épouser la forme de l'objet à son contact. Pour ces derniers, on parlera de *soft fingertip*.

Pour les mécanismes type *soft grippers*, leurs simulations se font majoritairement à l'aide de la méthode éléments finis. De plus, celles-ci se limitent généralement à étudier les déformations des doigts [14]. L'environnement, en particulier un objet à saisir, est rarement pris en compte et quand celui-ci l'est, il est considéré rigide et fixe [15–17]. L'objectif principal de ces études éléments finis est donc l'étude des mouvements du doigt pour ainsi optimiser le mouvement, les matériaux ou la fabrication [18, 19]. C'est pourquoi bien souvent après avoir réalisé des simulations, des expériences avec des objets du quotidien sont menées [20]. C'est le cas dans [21], où la fonction du mécanisme qui est l'objet de l'article est pourtant la saisie de nourriture. Le mécanisme est d'abord étudié avec une analyse éléments finis. Une première fois pour observer le mouvement du doigt et une seconde fois pour le mouvement du fluide qui passe dans le doigt pour le plier ou déplier. Ce n'est qu'une fois ces simulations réalisées que des expériences comprenant des interactions avec de la nourriture sont présentées. Les exemples d'articles illustrent bien la difficulté actuelle de modéliser un *soft gripper* interagissant avec un objet déformable.

Parfois, la démarcation n'est pas très nette entre les approches *soft grippers* et pinces sous-actionnées. Ces derniers utilisent des architectures compliantes tandis que l'autre des matériaux souples, mais dans les deux cas moins de moteurs sont utilisés que de DDL. Dans [13] les deux types sont réunies sous le terme *soft grippers*, puis séparés en *non-continuum bending-type* et *continuum bending-type* ou *continuum twisting-type*. Tout comme les pinces sous-actionnées les configurations des *soft fingers* vont dépendre de l'actionnement, mais aussi de



(a) (b)

Figure 2.5 *Soft grippers* (l'image (b) est tirée de [12] et libre de droits)

l'environnement. Ainsi, elles peuvent bouger, même sans variation de l'actionnement. Elles vont donc aussi s'adapter à la forme des objets, et cela va permettre d'appliquer des forces sur une surface plus grande et donc de limiter la concentration de force en un point localisé. Ces deux approches sont donc adaptées pour la saisie d'objet fragile et donnent des solutions à la création de préhenseur lorsque la première ne permet pas d'obtenir de système suffisamment compact, adapté à l'environnement, trop cher, etc. Lors de la conception de prothèse par exemple, que ce soit pour les mains, mais aussi d'autres membres du corps [22, 23], il est essentiel que la prothèse ne soit pas trop lourde et encombrante. Dans ce cas, il est important de faire un compromis entre plusieurs aspects, dont la précision, la dextérité, l'apparence, le prix, le confort ou encore la durabilité. Étant aussi de plus en plus facile à fabriquer grâce au développement des imprimantes 3D les mécanismes utilisant des matériaux souples sont de plus en plus développés [11].

Les domaines d'application ayant un intérêt dans le développement de ces pinces sont donc multiples. Elles sont par exemple montées comme outil sur des robots industriels [24, 25], sur des drones [26], ou encore simplement servent pour du divertissement sur des robots comme ceux utilisés par Disney ou Ford [27]. Dans l'industrie bien sûr, mais aussi dans des environnements moins accessibles. Le *Light-Duty Utility Arm* (voir figure 2.6) est par exemple conçu pour être utilisé dans des environnements radioactifs [28], tandis qu'il existe plusieurs pinces utilisées en chirurgie, actionnées soit par des robots comme le Da Vinci, soit directement par le praticien. En restant dans le domaine médical, ces pinces peuvent être montées comme prothèse [29] pour remplacer un membre ou encore pour aider à la rééducation de la main. En effet, les *soft grippers* sont par exemple utilisés sous forme de gant [30] que le patient met afin d'accompagner ses mouvements.

Si l'on parle de prothèse ou de gant, on peut noter que les préhenseurs artificielles peuvent

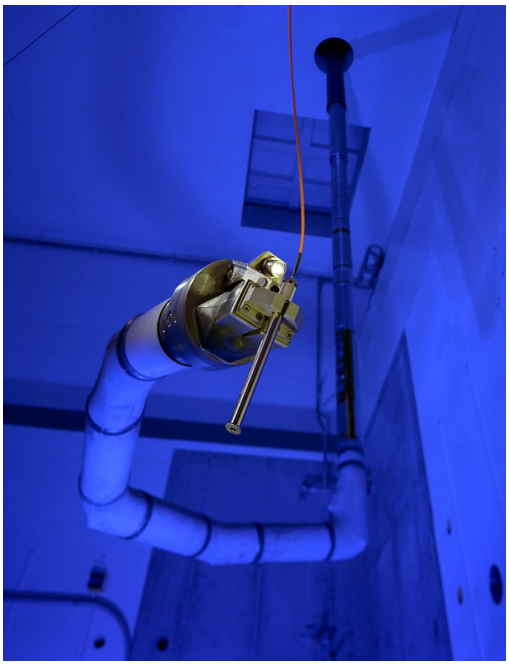


Figure 2.6 *Light-Duty Utility Arm* (image libre de droits tirée de [31])

être aussi combinés à des capteurs afin de permettre un retour d'informations à l'image du toucher des doigts. Ceci est réalisé à l'aide de système haptique intégré qui communique une information tactile à l'utilisateur. Les capteurs peuvent être par exemple des capteurs de pressions [32] ou de température sur le bout des doigts [33] qui vont envoyer un signal au porteur. Une difficulté est que ce signal doit être interprétable par l'utilisateur. Des vibrations peuvent par exemple indiquer suivant leurs intensités la température de ce que le doigt touche. Ces capteurs sont une option pour s'assurer qu'une pince n'écrase pas un objet [34], mais cela implique un contrôle soit numérique soit de l'opérateur ainsi que l'ajout d'un capteur au système. Des exemples de systèmes permettant de renvoyer des informations sur le contact seront présentés dans la section 2.2.4.

2.2 Doigts Sous-actionnés

2.2.1 Principe

Le *soft gripper* du professeur Shigeo Hirose est souvent cité comme le premier doigt dit sous-actionné [35], mais le sous-actionnement remonte au moins jusqu'aux travaux de Leonardo Da Vinci dans le Codex Atlanticus qui présentait déjà cette architecture sur des ailes articulées [36]. Un système sous-actionné est défini comme un mécanisme avec un nombre de DDL supérieur à son nombre d'actionneurs à l'inverse d'un mécanisme pleinement ac-

tionné, voire sur-actionné [37]. Ces systèmes sont constitués d'un système de transmission, ainsi que d'élément(s) passif(s) afin de contraindre statiquement le mécanisme. Le mécanisme de transmission distribue aux p sorties du système que sont les phalanges les couples ou forces d'actionnement, dont le nombre est au maximum $p-1$. La figure 2.7 illustre plusieurs doigts sous-actionnés avec des mécanismes de transmission différents (tirés de [37]). Ces derniers peuvent être des mécanismes à membrures [38], des mécanismes de transmission à engrenage [39] ou encore des mécanismes utilisant des tendons et des poulies [40, 41]. Ces derniers sont assez fréquents et considérés comme plus efficaces, tout en étant compacts, mais ne permettant pas d'appliquer autant de force qu'un mécanisme à membrure [42]. Si des forces de contact importantes sont nécessaires et pour un doigt comportant un nombre élevé de phalange les mécanismes à membrure seront à privilégier, typiquement les mécanismes quatre-barres ou cinq-barres. Ils sont eux aussi fréquents. Les mécanismes à engrenages sont quant à eux plus rares, car présentant des performances moindres. Les éléments passifs courants sont les ressorts placés au niveau des liaisons, parfois combinés avec des butées. Les ressorts peuvent être remplacés par des tendons élastiques appelés antagonistes car opposés aux tendons actionneurs des systèmes [43].

Les mécanismes sous-actionnés ont l'avantage principal de pouvoir s'adapter mécaniquement à leurs environnements à l'image de la main humaine saisissant un objet de forme inconnue sans que l'on y réfléchisse. En effet sans capteur ou commande complexe, les mécanismes sous-actionnés, comme l'illustre la pince présentée sur la figure 2.8 tirée de [44], vont s'adapter à la forme de l'objet qu'ils saisissent. On parle d'adaptation mécanique, voire parfois d'intelligence mécanique [45]. Les doigts de la pince présentée sur la figure sont seulement actionnés linéairement, c'est-à-dire un moteur au niveau de la base fait se rapprocher les bases des doigts dans un mouvement de translation. D'ailleurs, comme le doigt en lui-même n'est pas actionné (seulement sa base) on peut parler de doigts non-actionnés. Ensuite, lors du contact avec la roche, les trois phalanges des doigts alignées lorsqu'il n'y a pas de contact vont épouser la forme de la roche. On peut observer d'ailleurs que la prise n'est pas symétrique. Ce type de saisie s'appelle une saisie enveloppante ou préhension en force. Lorsque un objet est saisi par le bout des doigts, on parle de saisie de précision ou de pincement. Les autres avantages du sous-actionnement sont évidemment la limitation du nombre de moteurs qui entraîne des coûts réduits, car il y a moins de composants, mais aussi moins d'entretien (lubrification). Il est aussi possible de concevoir des systèmes plus compacts, plus légers et avec des contrôles simplifiés, car le mouvement des phalanges n'est pas à coordonner grâce à un programme ou à contrôler en boucle fermée. L'un des inconvénients des systèmes sous-actionnés est que le contrôle des DDL indépendamment est impossible et que l'on ne peut prévoir la position des doigts fermés si l'on ne connaît pas l'objet et sa position. On retrouve dans le tableau 2.1

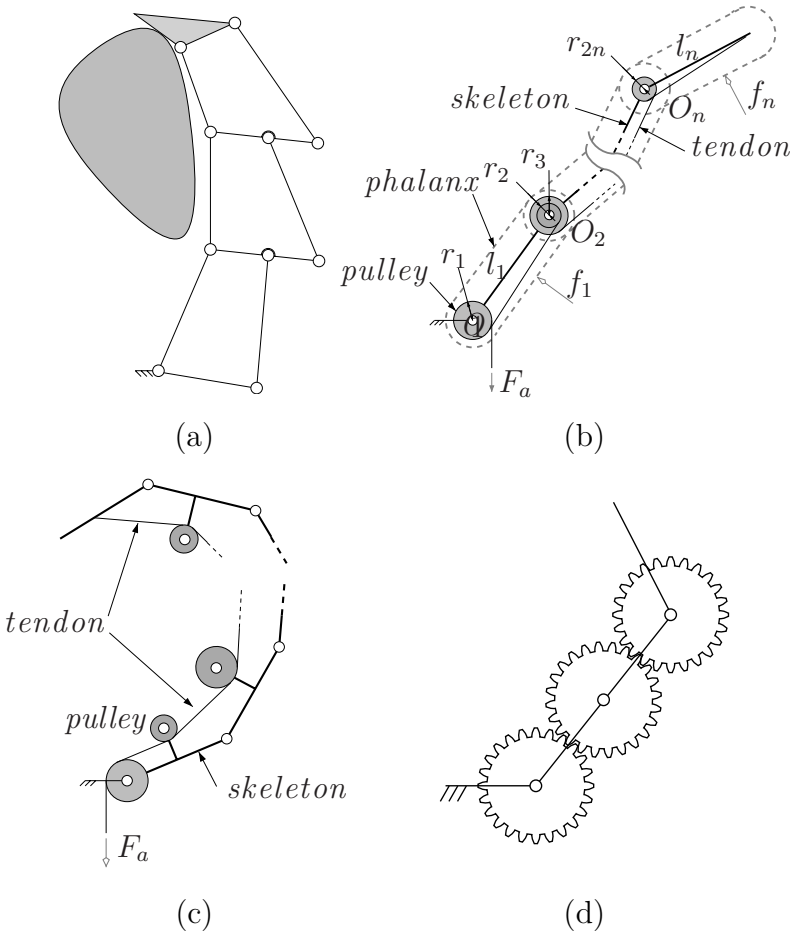


Figure 2.7 Mécanisme de transmission : (a) à membrure, (b) à tendon, (c) de Da-Vinci et (d) à engrenage (reproductions avec l'autorisation du professeur Lionel Birglen tirées de [37])

les avantages et inconvénients des doigts sous-actionnés par rapport aux doigts pleinement actionnés et aux *soft grippers*. Il est noté "oui/non" dans la catégorie "Fortes forces de saisie", car suivant la conception du mécanisme de transmission, un doigt peut soit beaucoup s'adapter à la forme de l'objet et potentiellement ne pas appliquer assez de force sur un objet pour le saisir ou à l'inverse pouvoir appliquer une force plus conséquente, mais avec une moins bonne adaptation.

2.2.2 Modélisation de la saisie d'objet rigide

La saisie d'un objet par des doigts sous-actionnés se fait en plusieurs étapes. La figure 2.9 tirée de [46] illustre le mouvement usuel. Le doigt présente deux phalanges, donc deux DDL, tandis que l'objet est infiniment rigide et fixe. Un moteur est situé au niveau de l'articulation entre la base et la membrure inférieure pour appliquer un couple (indiqué par la flèche verte) qui est transmis aux phalanges grâce à un mécanisme de transmission quatre-barres. Dans un premier temps, quand un couple d'actionnement va être appliqué seulement la phalange proximale va s'approcher de l'objet, le ressort retenant tout mouvement de la seconde phalange. Cette étape pendant laquelle le doigt s'approche de l'objet, mais ne le touche pas encore est communément appelée la phase de *preshaping*, voire plus rarement de *pregrasping*. Ensuite, une fois la première phalange en contact avec l'objet, avec l'augmentation du couple la seconde phalange va elle aussi venir en contact, car la première est bloquée. L'actionnement va être suffisant pour étirer le ressort. Il y a donc une adaptation et le doigt épouse bien la forme de l'objet. Plus il y aura de phalange, meilleur sera l'adaptation.

Si l'on suppose que les forces de contact sont faibles lorsque seulement la phalange proximale touche l'objet, ce qui est généralement voulu puisque l'on veux une adaptation, cette hypothèse n'est plus possible quand toutes les phalanges sont en contact et que le couple continue d'augmenter. Malgré l'adaptation, les forces de contact peuvent être importantes. Afin d'étudier les forces que les pinces sous-actionnées appliquent à leur environnement, plusieurs équations ont été développées afin de lier les couples d'entrées de pince aux forces de contact. La figure 2.10 tirée de [37] présente le modèle général d'un doigt sous-actionné. Celui-ci a n phalanges, chacune associées à un angle interphalange θ. Les forces représentées sont les forces de contact de chaque phalange situées à une distance notée k. Avec $\boldsymbol{f}_i$ la composante normale de la force de contact de la $i^{\hat{e}me}$ phalange et $\boldsymbol{f}_{ti}$ la composante tangentielle. Dans [37], suivant des hypothèses de mouvements lents et en négligeant la gravité l'équation 2.1 est présentée à la suite d'une analyse quasi-statique. Elle relit le vecteur des forces de contact $\boldsymbol{f}$ au vecteur des couples d'entrée $\boldsymbol{t}$ en fonction de l'architecture du doigt (les raideurs des ressorts et la géométrie du doigt) et des distances de contacts sur les phalanges.

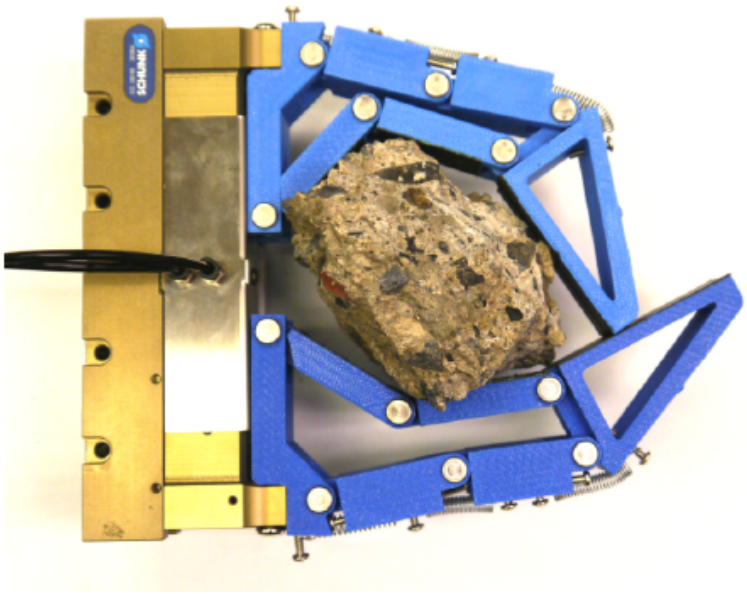


Figure 2.8 Pince sous-actionnée enveloppant une roche (reproduction avec autorisation du professeur Lionel Birglen tirée de [44])

Tableau 2.1 Comparaison des doigts sous-actionnés avec les doigts pleinement actionnés et les *soft grippers*

	Mécanismes pleinement actionnés	sous-actionné	*Soft grippers*
Précis	oui	non	non
Fortes forces de saisie	oui	oui/non	non
Modélisation simple	oui	oui	non
Adaptation sans capteur	non	oui	oui
Fabrication rapide	non	oui	oui
Entretien limité	non	oui	oui
Prix réduit	non	oui	oui
Volume/Poids réduits	non	oui	oui

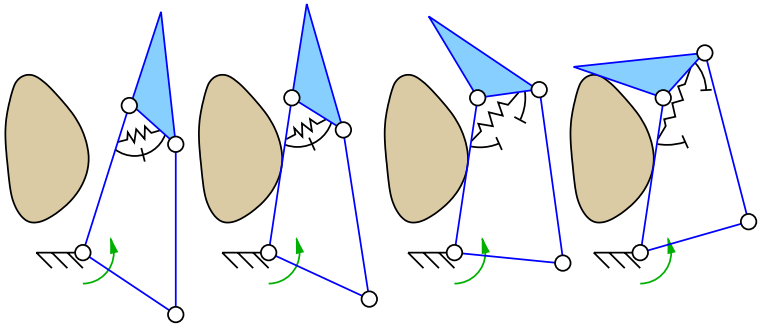


Figure 2.9 Séquence de fermeture d'un doigt sous-actionné autour un objet (reproduction avec autorisation du professeur Lionel Birglen tirée de [46])

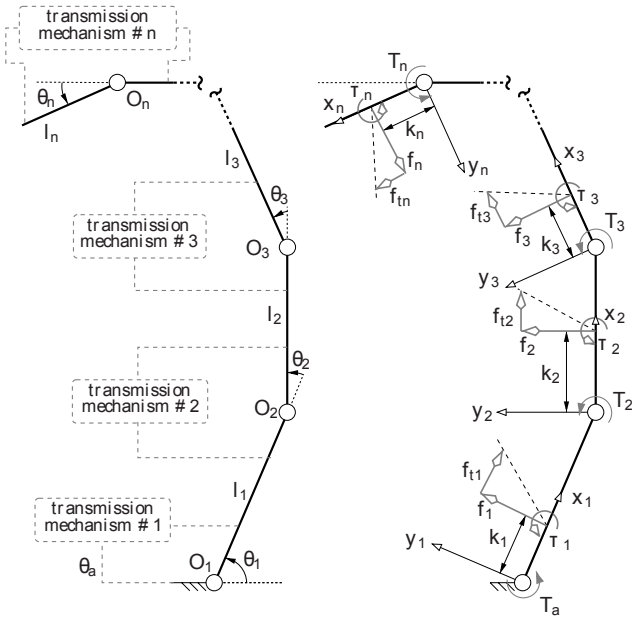


Figure 2.10 Modèle d'un doigt sous-actionné (reproduction avec autorisation du professeur Lionel Birglen tirée de [37])

$$\boldsymbol{f} = \boldsymbol{J}^{-T}\boldsymbol{T}^{-T}\boldsymbol{t}, \tag{2.1}$$

avec

$$\boldsymbol{f} = \begin{bmatrix} f_1 \\ \vdots \\ f_p \end{bmatrix}. \tag{2.2}$$

Le vecteur $\boldsymbol{t}$ comprend les forces en entrée du système, classiquement le couple d'actionnement ainsi que les forces dues aux ressorts. $\boldsymbol{J}$ est la matrice jacobienne qui dépend de la configuration du doigt et des distances de contact et est définie par :

$$\boldsymbol{J} = \begin{bmatrix} \boldsymbol{r}_{11}^T\boldsymbol{x}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T\boldsymbol{x}_2 & \boldsymbol{r}_{22}^T\boldsymbol{x}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1p}^T\boldsymbol{x}_p & \boldsymbol{r}_{2p}^T\boldsymbol{x}_p & \boldsymbol{r}_{3p}^T\boldsymbol{x}_p & \dots & \boldsymbol{r}_{pp}^T\boldsymbol{x}_p \end{bmatrix} - \boldsymbol{\mu} \begin{bmatrix} \boldsymbol{r}_{11}^T\boldsymbol{y}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T\boldsymbol{y}_2 & \boldsymbol{r}_{22}^T\boldsymbol{y}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1p}^T\boldsymbol{y}_p & \boldsymbol{r}_{2p}^T\boldsymbol{y}_p & \boldsymbol{r}_{3p}^T\boldsymbol{y}_p & \dots & \boldsymbol{r}_{pp}^T\boldsymbol{y}_p \end{bmatrix}, \tag{2.3}$$

avec $\boldsymbol{r}_{ki}$ le vecteur allant du point A_k au $i^{ème}$ point de contact, $\boldsymbol{x}_i$ le vecteur unitaire le long de la $i^{ème}$ phalange allant vers la phalange distale et $\boldsymbol{y}_i$ le vecteur défini par $\boldsymbol{y}_i = \boldsymbol{z} \times \boldsymbol{x}_i$. On retrouve ces notations sur la figure 2.10. $\boldsymbol{\mu}$ est une matrice diagonale dont les coefficients lient les forces normales aux forces tangentielles correspondantes. La deuxième partie de l'équation modélise la friction et assumer que le frottement est négligeable est une hypothèse classique lors d'étude de pinces sous-actionnées, car une hypothèse conservatrice. En effet, la friction aidant à garder un objet dans une prise, les cas sans frottement constituent donc les pires cas pour la saisie. La matrice jacobienne est donc généralement calculée avec seulement la première partie. Finalement $\boldsymbol{T}$ est la matrice de transmission qui comme son nom l'indique dépend du mécanisme de transmission. Cette matrice est développée pour plusieurs architectures dans [37], dont les mécanismes tendons/poulies, à engrenages, mécanismes de Da Vinci, etc. Cette matrice peut s'obtenir en trouvant la relation entre $\boldsymbol{w}_a$ le vecteur des dérivées des angles du vecteur des couples d'entrée et $\dot{\boldsymbol{\theta}}$ le vecteur des dérivées des angles interphalanges :

$$\boldsymbol{w}_a = \boldsymbol{T}\dot{\boldsymbol{\theta}}. \tag{2.4}$$

À partir de cette équation, la simulation de la saisie par un doigt est donc possible si l'on suppose que l'objet est rigide et en connaissant certaines informations sur ce dernier comme sa forme et sa position. Cette formule est applicable à tout type de transmission, mais est restreinte aux cas où il n'y a pas plus d'un contact par phalange. Finalement, à la connaissance

de l'auteur, un tel modèle des forces de contact n'a pas été développé pour tous les préhenseurs artificielles, en particulier certains *soft grippers*. En effet, pour les doigts où les déformations sont sur l'ensemble de la longueur du doigt (ressemblant plus à des tentacules) cela est généralement difficile de trouver une relation entre l'entrée des systèmes et les forces de contact. Ces derniers sont donc généralement seulement étudiés grâce à la méthode des éléments finis, ce qui les rend plus complexes à étudier que les doigts sous-actionnés.

2.2.3 Analyse de la saisie d'objet rigide

La modélisation des forces de contact a amené à l'évaluation de la saisie par des doigts sous-actionnés. En effet, l'adaptation des doigts sous-actionnés n'assure pas forcément une bonne saisie. Si l'on prend l'exemple d'un doigt qui ne touche un objet fixe qu'avec sa phalange distale, l'une des possibilités lorsque le couple d'actionnement va ensuite encore augmenter est que cette phalange distale glisse sur l'objet vers la paume. Jusqu'à ce que l'objet touche le bout du doigt et s'échappe de la prise. Dans ce cas-là, la saisie sera mal assurée. Un autre exemple de cas à étudier est lorsque des forces extérieures peuvent venir éjecter l'objet si les forces de saisie ne sont pas assez importantes. La stabilité de la saisie est un des éléments clef de la conception de pince sous-actionnée. Plus globalement plusieurs critères afin d'évaluer la qualité d'une saisie ont été définis dans la littérature.

Les équipes de Domenicco Prattichizzo et Antonio Bicchi présentent dans leurs articles ce qu'ils nomment les synergies. Ce sont toutes les positions que prend la main humaine pour saisir des objets [47]. Pour ces auteurs les configurations que doivent prendre les pinces sous-actionnées doivent se rapprocher de ces synergies. Dans le cas où l'on souhaite créer des prothèses cela peut-être intéressant d'avoir des systèmes qui prennent les mêmes configurations que les doigts, afin que l'utilisateur s'habitue plus facilement et rapidement à son mécanisme. En revanche, les pinces sous-actionnées n'étant pas forcément des copies de main humaine ou n'ayant pas pour but de la remplacer, l'étude des synergies n'implique pas forcément qu'elles soient aussi adaptées à toutes les pinces. C'est pourquoi, l'évaluation de la saisie passe généralement par l'étude des forces de contact. C'est le cas dans [46], où une étude des forces de contact et de la stabilité est réalisée avec l'équation 2.1 présentée précédemment, en particulier pour les cas où la phalange distale est en contact tandis que les autres phalanges ne le sont pas (leurs forces de contact sont négatives). Cela a pour but d'éviter les conceptions où une éjection de l'objet est possible. Aussi, l'étude des forces de contact permet d'évaluer l'amplitude de taille d'objet qu'une pince peut saisir, ainsi que la distribution des forces de contact possible afin d'adapter l'architecture de la pince, en particulier en faisant le bon choix sur la géométrie du mécanisme de transmission [42]. Dans [48], la qualité de la

saisie d'une force est définie suivant la valeur de la force maximale que l'on peut appliquer sur un objet saisi sans qu'il s'éjecte d'une prise. Cette valeur est d'abord calculée théoriquement grâce à un modèle numérique, mais ensuite aussi testée à l'aide d'un banc d'essai. Ce qui leur permettra par la suite d'améliorer une architecture en utilisant une phalange distale ayant une surface courbée pour mieux saisir [49].

La manière la plus répandue d'évaluer les capacités d'une pince reste finalement par l'expérimentation [50–60]. Cela peut-être pour trouver la configuration qui permettra d'appliquer le plus de force [61] ou encore augmenter la taille des objets que la pince peut saisir [62]. Il n'est pas rare de voir à la fin d'article sur le sujet, des saisies de divers objets de la vie quotidienne pour tester et illustrer l'efficacité d'une pince. Cette méthode est directe, mais est très lente et donc non adaptée pour de l'optimisation si l'on veut tester des milliers de configurations.

2.2.4 Amélioration des architectures

Dans les sections précédentes, les doigts sous-actionnés ont été présentés ainsi que des méthodes pour évaluer leurs efficacités et ainsi les optimiser. L'évaluation permet les discussions sur les architectures de doigt (phalanges, transmission et éléments passifs), en se concentrant majoritairement sur les mécanismes de transmission, ou encore sur la méthode d'actionnement. Des articles se sont intéressés à d'autres aspects des systèmes complets de pince. En particulier la combinaison des doigts, mais aussi à d'autres fonctions qui pourraient être réalisées par les pinces en passant par un travail sur les surfaces de contact. Nous allons voir rapidement dans cette section quelques axes de recherches sur les doigts sous-actionnés, puis conclure sur l'une des limites de ces études qui a amené à ce travail.

Nombre de doigts

En effet, les pinces ne sont généralement pas composées d'un seul doigt, le cas général étant avec deux doigts, dont au moins l'un est mobile. Ces pinces sont adaptées pour un nombre limité de forme d'objet, dans d'autres cas l'ajout de doigts supplémentaires va améliorer la saisie. Cela est le cas aussi pour une personne par exemple lors de la saisie d'une sphère, alors que pour un cylindre deux doigts devraient suffire s'il n'est pas trop lourd. Plus une pince a de doigt, plus facile devrait être la saisie pour des objets de formes complexes. D'ailleurs, dans [63] pour saisir un cube puis une balle de base-ball un prototype d'une pince sous-actionnée utilise huit doigts. En complément de l'ajout de doigt, lorsque l'on saisit une sphère il arrive que l'on ait besoin d'écarter ou de rapprocher les doigts afin d'améliorer sa prise. C'est pourquoi les architectures de pinces à plus de deux doigts incluent souvent des moteurs supplémentaires afin de réaliser les mouvements d'abduction et d'adduction [64].

Ces mouvements permettent de manipuler les objets saisis.

Retour d'information

Les systèmes sous-actionnés ayant généralement un seul moteur situé au niveau de la base du mécanisme. Cela entraîne une répartition plus facile du poids sur un robot complet que pour les doigts pleinement actionnés. Ces derniers afin d'adapter leurs positions doivent être dotés de capteurs et peuvent donc renvoyer des informations. Généralement sur les couples d'actionnement et la position des moteurs, ce qui donne la position des phalanges, et parfois sur la position des contacts [65]. Une majorité des pinces sous-actionnées ne renvoient aucune information, car leur contrôle est en boucle ouverte. Parfois des capteurs peuvent aussi être intégrés sur les surfaces des doigts sous-actionnés afin de retourner des informations tactiles qui ne serviront pas forcément la commande [66]. Cela va à l'encontre de la philosophie du sous-actionnement qui est de réduire le nombre d'éléments et de profiter de l'avantage mécanique de l'architecture. Dans [67], les variations de couple demandé pour actionner un doigt sous-actionné sont observées lors d'interactions avec et sans objet. Il a été observé que lorsqu'un contact est réalisé entre le doigt et un objet, le couple demandé va être plus important que s'il n'y avait pas de contact. En fonction de la différence mesurée entre les deux, il est donc possible de déterminer à quel endroit est le point de contact avec la phalange et donc de créer un capteur de contact. Le capteur étant placé au niveau du moteur, loin de l'effecteur doigt, on parle de capteur proprioceptif de manière analogue à la proprioception du corps humain. En effet, les muscles et les tendons réagissent à des mouvements internes du corps, ce qui permet à une personne de connaître les positions de son propre corps. Cette caractéristique permet de garder un système ayant une grande partie de sa masse répartie au niveau de la base, moteur et capteur, ce qui est intéressant pour une application de prothèse ou de robot industriel, contrairement aux capteurs précédemment évoqués sur le bout des doigts. De plus, limiter le nombre d'éléments sur une pince permet d'avoir des pinces plus compactes. Pour des applications, médicales entre autres, plus la taille de la pince sera réduite plus elle sera adaptée. Dans l'exemple précédent, la proprioception est adaptée à un système tendon/poulie, mais on la retrouve aussi dans d'autres architectures. Dans [68], les doigts fléchissent et se ferment sous l'augmentation du volume d'un fluide dans leur membrane et c'est grâce à des mesures des propriétés de ce fluide que le contact est ensuite évalué.

Manipulabilité

Jusqu'à maintenant les systèmes présentés n'ont eu qu'une fonction : la saisie. Pourtant, lorsque l'on saisit un objet, on peut avoir besoin de le manipuler que ce soit pour améliorer

notre saisie ou tout simplement pour l'utiliser, le repositionner, etc [69]. La manipulabilité comprend la possibilité de repositionner et/ou réorienter un objet avec dextérité dans la main [70]. Les points importants pour assurer une bonne manipulation sont la coordination entre les doigts et la possibilité de faire changer l'orientation l'objet par rapport à la paume l'objet, tout en assurant comme pour la saisie la stabilité. Dans [71], deux mécanismes secondaires montés sur deux doigts permettent par exemple de faire rouler l'objet entre les doigts. L'un des doigts est muni d'un système de courroie au niveau de la surface de ses phalanges, tandis que la surface du doigt opposé est munie de rouleaux passifs. Le système de courroie peut être actionné afin de faire rouler l'objet. Ce système permet de corriger les problèmes de stabilité évoqués dans la partie 2.2.3. On peut imaginer un système complet qui évalue les distances de contact lors d'une interaction, puis estime l'évolution de ses configurations. Si une position instable est prévue, manipuler l'objet pourra permettre d'éviter toute éjection possible de celui-ci.

Saisies particulières

D'autres doigts ont aussi été conçus pour se spécialiser plus dans les saisies particulières. Ces contraintes supplémentaires sont par exemple de maintenir les phalanges distales parallèles [72] ou encore de pouvoir saisie un objet plat posé sur une surface lisse [73]. Une carte de jeu sur une table par exemple, là où une personne utiliserait ses ongles. Dans [74], une architecture à deux mécanismes de transmission est présentée. Le premier mécanisme transmet l'actionnement aux deux premières phalanges pour un mouvement classique comme ceux vu précédemment, tandis que le second transmet le même actionnement à une membrure située au bout du doigt afin que celle-ci glisse sur les surfaces planes. De plus, la pince peut porter des objets lourds du bout des doigts. De la même manière, des pinces sous-actionnées peuvent être couplées à des mécanismes bistables. Dans [75], des pinces sont montées sur un drone afin qu'il s'accroche à des branches et recharge ses batteries. Afin de ne pas consommer d'énergie lors de la recharge ou pendant le vol, le mécanisme bistable permet de garder stables deux positions de la pince, ouverte ou fermée, sans besoin d'énergie. Enfin, les surfaces de contact des phalanges ont aussi bien évidemment été étudiées afin d'améliorer les systèmes pour par exemple augmenter le frottement ou encore épouser encore plus la forme de l'objet avec des surfaces déformables et ainsi augmenter encore plus l'adaptabilité [76]. Ceci tend vers les *soft grippers* avec zéro membrure rigide.

2.2.5 Applications et limites

Finalement, les mécanismes sous-actionnés ayant des avantages par rapport aux mécanismes *classiques* pleinement actionnés, ceux-ci vont être plus adaptés dans certains domaines. Dans le cas des prothèses par exemple, un système comportant beaucoup de moteur peut devenir rapidement lourd, encombrant et/ou onéreux. Un compromis peut être fait en limitant le nombre de doigts, comme on l'a vu dans l'exemple de la figure 2.4(a), mais cela limite aussi le nombre de DDL. Le sous-actionnement est donc une alternative pour augmenter la mobilité, même si le mouvement n'est pas totalement contrôlable [77]. Si l'on compare aux mouvements du doigt humain, le mouvement de nos phalanges de façon indépendante n'est de toute façon pas possible. Sans bloquer les phalanges proximale et intermédiaire d'un doigt il est par exemple impossible de bouger une phalange distale seule. En plus de prothèse pour remplacer un membre perdu, plusieurs doigts sous-actionnés ultra légers et compacts se montent directement sur la main humaine afin de guider ou de renforcer les mouvements de celle-ci [78, 79], afin par exemple d'accélérer une rééducation.

Dans le secteur agroalimentaire, certaines opérations nécessitent de déplacer des aliments fragiles un à un et donc de les saisir. C'est le cas pour une opération de tri. Malheureusement deux fruits ou deux légumes de la même espèce n'étant pas de même taille ou forme, la saisie de ceux-ci par une pince robotisée pleinement actionnée implique soit que l'aliment va subir une certaine force soit un système complexe de la pince avec capteurs et boucle de commande, afin de s'adapter et ainsi ne pas entraîner de dommage. L'utilisation d'une pince sous-actionnée qui s'adapte mécaniquement à la forme de façon passive est une alternative intéressante pour ne pas perdre de la marchandise ou investir dans des pinces complexes, plus lentes. De la même manière, mais avec des conséquences plus graves, dans le domaine médical, là où un chirurgien va utiliser ses mains pour saisir un organe fragile car les pinces pourraient endommager celui-ci, les pinces sous-actionnées pourrait être utilisées soit comme outil à main [80], soit installées sur un robot type Da Vinci. Actuellement les pinces chirurgicales sont utilisées en pinçant le tissu humain que l'on veut saisir. Elles comportent parfois des dents afin de pouvoir mieux saisir et ainsi réduire des forces de pincement, mais les dents sont aussi responsables de microlésions qui augmentent généralement le temps qu'un patient passe à l'hôpital pour récupérer. L'adaptation pourrait être utile, tout en gardant le mécanisme assez petit pour être le moins invasif possible. Finalement, dans le domaine industriel, ces doigts sont très adaptés aux robots collaboratifs car en effet en cas de collision avec une personne les doigts pourraient s'adapter afin de ne pas occasionner de dommage.

Dans les sections précédentes toutes les études sur les doigts sous-actionnés présentés portaient sur des interactions où l'on supposait un objet à saisir rigide, à la connaissance de

l'auteur, aucun article ne s'est intéressé à la saisie d'objet déformable. Ce qui au vu des applications présentées serait intéressant. Dans ce manuscrit, on appellera un objet déformable un solide avec une faible rigidité qui se déformera lorsque soumis à des forces externes.

2.3 Mécanismes compliants

Les mécanismes sous-actionnés comprenant des ressorts afin de contraindre statiquement le système, leur architecture est parfois réalisé à l'aide de lamelle flexible à la place de liaison rotoïde. On parle alors de liaisons compliantes [81, 82]. Les mécanismes compliants sont des systèmes qui utilisent la flexibilité des matériaux pour créer des mobilités. Cette technique se retrouve en origami, l'art du pliage de papier japonais, où les pliures vont servir d'articulations et les faces intactes de membrures afin de créer une sculpture à partir d'une simple feuille [51, 83]. La méthode la plus simple afin de réaliser une articulation compliante est de réaliser un amincissement de matière à un endroit localisé du système. Lorsque soumis à des forces internes au système, l'amincissement va fléchir et approcher le mouvement dans une plage de déplacement raisonnable une liaison rotoïde. Cette géométrie de liaison s'appelle communément une compliance localisée. La figure 2.11 présente une pince réalisée en une seule pièce au repos, puis actionnée [80]. Sur cette figure, on peut observer des liaisons compliantes localisés qui fléchissent et permettent ainsi de bouger et de saisir le cylindre central. Le prototype illustré n'a pas nécessité de montage lors de sa fabrication et tient dans la paume d'une main, ce qui aurait été compliqué avec des liaisons classiques. À l'inverse dans le cas d'une compliance distribuée, une seconde technique aussi très présente dans la littérature, des poutres minces sont utilisées et la déformation se fera sur une longueur de poutre plus importante (voir l'exemple de la figure 2.12 tirée de [84]). Dans ce cas, l'approximation à l'aide d'une seule liaison rotoïde n'est plus réaliste, mais la plage de déplacement possible sera potentiellement plus grande, car moins de contraintes seront concentrées dans une petite zone. Pour le même angle de rotation que celui illustré sur la figure, un amincissement de matière à la place de la poutre flexible inférieur aurait amené l'acrylique (le matériel de la poutre) dans le domaine plastique ou l'aurait cassé. La conception de mécanisme compliant n'étant pas autant maîtrisé que celui des systèmes à corps rigides, un modèle équivalent a été mis en place afin de les étudier à l'aide de systèmes à membrures rigides avec des liaisons classiques, c'est le modèle pseudo-rigide (MPR) [81, 85]. Les liaisons compliantes sont modélisées par une ou plusieurs liaisons rotoïdes liant des membrures rigides combinées à des ressorts angulaires au niveau des articulations afin d'approcher l'élasticité du matériel. On a déjà évoqué le modèle pour une liaison localisée (une liaison rotoïde et une ressort), la poutre inférieure de la figure 2.12 va quant à elle correspondre à une combinaison de mem-

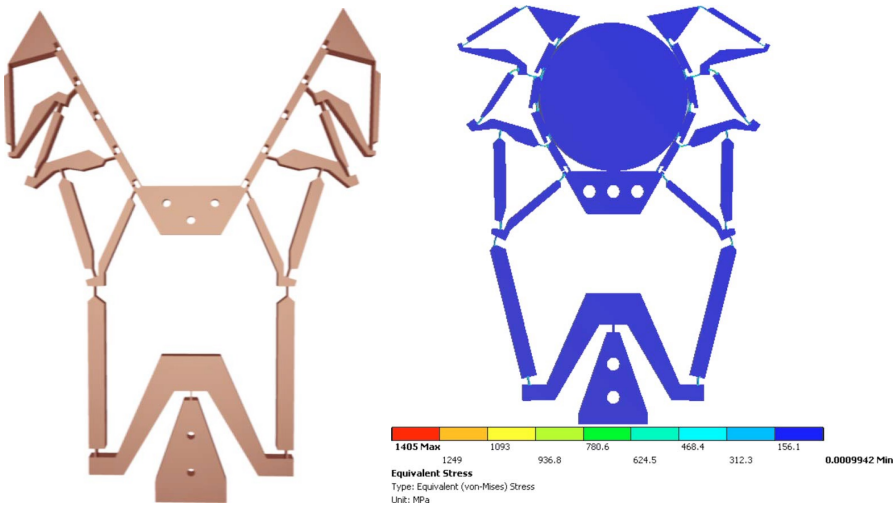


Figure 2.11 Pince compliante (reproduction avec autorisation du professeur Lionel Birglen tirée de [80])

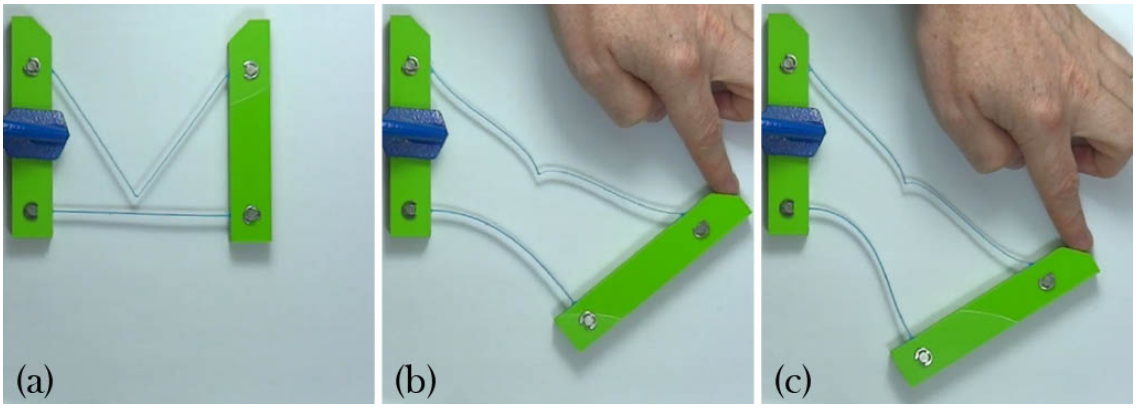


Figure 2.12 Liaisons distribuées (tirée de [84])

brures rigides et deux liaisons rotoïdes combinés à des ressorts, par exemple. Les géométries compliantes sont souvent couplées au sous-actionnement, premièrement car cela permet de réduire encore plus les composants, mais aussi car les deux techniques se combinent facilement. En effet on le rappelle des ressorts sont généralement utilisés en parallèle des liaisons rotoïdes pour contraindre statiquement les mécanismes sous-actionnés, cela correspond donc aux MPR des mécanismes compliants. Il est à noter que l'on peut trouver des conceptions dans lesquelles il est possible de faire varier la raideur des liaisons compliantes en jouant sur la géométrie de la liaison [86–88].

Les articulations compliantes ont l'intérêt d'être réalisables à des échelles réduites non-atteignables actuellement avec des liaisons classiques, ce qui fait qu'elles sont utilisées pour faire des micro-systèmes électromécaniques (MEMS) et ainsi créer des systèmes compacts ou avec plus de composants (téléphones, ordinateur, etc.). La figure 2.13 illustre par exemple un instrument de mesure du glucose compliant de $1,3$ par $0,36mm$ (tirée de [89]). De plus, les phases de fabrication et d'assemblage sont réduites, le nombre de pièces constituant le mécanisme étant inférieur à un mécanisme classique équivalent. Le mécanisme à membrures rigides équivalent à celui présenté sur la figure 2.11 nécessiterait par exemple l'assemblage de vingt-sept membrures à l'aide de trente-trois liaisons. Aussi, le développement des imprimantes 3D permet d'imaginer d'imprimer des mécanismes compliants prêts à l'emploi dès la fin de l'impression.

Finalement, l'absence de liaison classique entraîne aussi l'absence de jeu, ainsi que de friction interne au système et donc la lubrification n'est plus nécessaire. Pour une utilisation médicale où l'on ne souhaite pas qu'un fluide extérieur vienne dans un corps, cette propriété est intéressante. L'usure sera par contre toujours un point à prendre en compte, il faudra donc avoir une attention particulière aux limites des plages de mouvements. De plus, les architectures de ces mécanismes sont plus difficiles à concevoir. En effet, des analyses par éléments finis

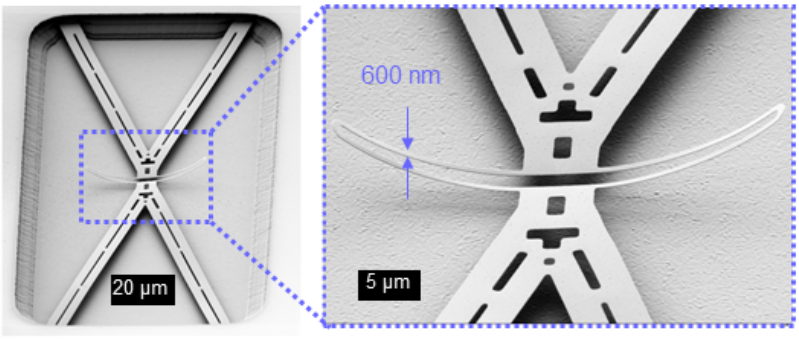


Figure 2.13 BioMEMS compliant (image libre de droits tirée de [89])

sont souvent nécessaires lors de la conception pour s'assurer que l'on reste dans les limites élastiques du matériau. Il est aussi encore compliqué de réaliser des mécanismes comprenant des liaisons prismatiques. En effet, ces dernières n'ont pas encore d'équivalent immédiat en version compliante. Pour l'instant les mécanismes quatre-barres RRRR compliants sont ce qu'il y a de plus simple afin de s'en approcher [84].

2.4 Modélisation d'objets déformables

Lors de la conception d'une pince, les interactions avec les objets qui seront saisis sont un élément critique à évaluer afin d'assurer leurs intégrités. Pour les modéliser, on peut séparer le problème en trois parties : la modélisation des déformations de l'objet en fonction des forces extérieures, la modélisation de la pince et donc des forces de contact en fonction de l'actionnement et finalement l'interaction entre les deux systèmes. Si les forces appliquées sont faibles par rapport à la raideur de l'objet, les déformations peuvent être négligées, c'est d'ailleurs une hypothèse commune dans la littérature sur les doigts sous-actionnés. Pourtant, cela ne convient pas dans tous les cas d'application de saisie. On peux aussi trouver des exemples d'études de saisie d'objet déformable qui se limitent à des cas de pincée avec deux points de contact localisés [90, 91]. Ces modèles ne s'appliquent donc pas aux cas des saisies enveloppantes qui sont pourtant préférées pour leur stabilité. Dans cette section, nous allons évoquer des techniques utilisées afin de simuler la saisie d'objets déformables de forme quelconque.

Dans le cadre de cette thèse, les pinces ont pour objectif de saisir des objets sans les déformer de manière importante, on considérera donc que le comportement des solides restera dans le domaine élastique. La modélisation des déformations va s'appuyer sur des systèmes d'équations généralement difficiles à résoudre de manière directe. Des approximations ainsi que des discrétisations sont donc généralement combinées afin de résoudre ces systèmes. La technique la plus connue afin de répondre à ce problème est la méthode des éléments finis (MEF). Avec cette méthode, on va chercher une solution approchée à un problème aux dérivées partielles grâce à une discrétisation d'un domaine, puis à la simplification du problème sur chaque élément. On va passer d'équations quelconques potentiellement non-linéaires à des équations linéaires plus faciles à résoudre. Elle est précise, mais complexe et nécessite généralement beaucoup de temps de calcul [92, 93]. Avec cette technique, d'abord les solides sont discrétisés en sous-éléments simples délimités par des nœuds. On parle alors de maillage. Ces éléments sont connectés entre eux et peuvent être des triangles, des quadrilatères, des tétraèdres, etc. La figure 2.14 est un exemple de maillage utilisé pour simuler un genou.

Ensuite, pour chaque petit élément créé, les éléments finis, les équations décrivant les carac-

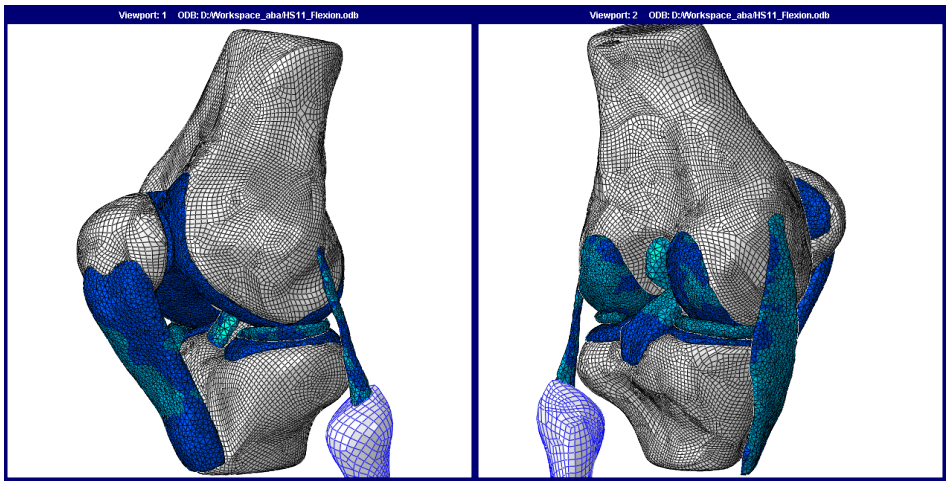


Figure 2.14 Modèle d'un genou discrétisé en éléments finis (image libre de droit tirée de [94])

téristiques physiques du solide complet difficilement résolubles vont être approchées par des équations linéaires. Celles-ci seront plus faciles à résoudre afin de calculer les déplacements et les contraintes à partir de conditions limites. Cette méthode revient à résoudre le système matriciel reliant $\boldsymbol{u}$ le vecteur des déplacements aux nœuds et $\boldsymbol{F}$ le vecteur des actions extérieures, c'est-à-dire :

$$\boldsymbol{K}\boldsymbol{u} = \boldsymbol{F}, \tag{2.5}$$

avec $\boldsymbol{K}$ la matrice de raideur globale. Cette dernière s'obtient en assemblant des matrices de raideur élémentaires qui comme leur nom l'indique sont caractéristiques des propriétés mécaniques des sous-éléments. Cette équation linéaire remplace des équations complexes. Après cette étape d'assemblage et tout en prenant en compte les conditions limites du problème, on peut trouver une solution approchée pour le solide complet. Pour avoir des résultats précis, il sera donc nécessaire de mailler l'objet finement, mais augmenter le nombre d'éléments de la discrétisation augmente aussi les temps de calcul. Un maillage trop grossier va à l'inverse mener à des résultats erronés ou à des problèmes de convergence et donc à l'arrêt de la simulation. Il y a donc un compromis à trouver entre résultat précis et temps de calcul pour les simulations par éléments finis.

La figure 2.16 illustre un exemple de simulation avec la MEF : une collision entre une voiture et un mur afin d'observer les déformations. Les exemples d'applications de la MEF sont nombreux et ceux-ci ne s'arrêtent pas qu'à la mécanique du solide. En ce qui concerne la saisie d'objet déformable, deux domaines d'application déjà évoqués précédemment viennent à l'esprit : l'industrie agro-alimentaire et les opérations chirurgicales. Dans l'agro-alimentaire, l'étude de la déformation de fruits ou de légumes soumis à des contraintes est importante

à évaluer car si l'on souhaite les manipuler à l'aide de robots ils pourront être endommagés avant leurs commercialisations [95]. En effet une déformation sur une banane ou une tomate par exemple va créer une tâche noire quelques jours après, ce qui la rendra plus difficile à vendre. C'est pourquoi la MEF a été utilisée dans [96] pour simuler la déformation d'une tomate en fonction d'une force qu'on lui applique et ainsi trouver une relation entre le volume des lésions internes et le pourcentage de déformation. Dans [97], c'est un pamplemousse qui est étudié et ses déformations sont évaluées en fonction de la position des forces appliquées sur le fruit. On peut imaginer se servir de ces résultats afin de concevoir la commande de pinces pleinement actionnées pour une chaîne de production afin d'assurer des saisies douces du fruit.

Des études semblables ont été réalisées pour la chirurgie sur les tissus humains. Dans ce cas-là, les déformations pourraient avoir des conséquences bien plus dramatiques que pour les légumes [98], il faut donc aussi évaluer en amont les possibles dommages qu'une saisie peut entraîner. Dans [99], différents modèles éléments finis en deux ou trois dimensions sont utilisés afin d'analyser un foie soumis à de la compression. Cette simulation correspond à une opération de saisie à l'aide de pinces chirurgicales classiques qui vont pincer les tissus humains pour les saisir et donc les comprimer. Ces mêmes auteurs ont aussi évalué les contraintes dans le foie lors d'une saisie en fonction de la géométrie de la surface des pinces afin d'optimiser celle-ci [100]. Les effets de dents placées sur la surface pour améliorer la stabilité de la prise sur la déformation sont simulés et ainsi leurs géométries peuvent être optimisées. Il est à noter dans cet article que la précision et le temps de calcul entre des modèles complets des solides et des modèles ne considérant que des membranes fines sont comparés. En effet la méthode présentant des temps de calcul longs, des simplifications sont souvent recherchées afin d'obtenir des résultats satisfaisants en moins de temps [101]. L'une des techniques pour simplifier le modèle est de réduire le nombre de nœuds du maillage, ainsi des applications temps réels sont potentiellement possibles [102].

Comme évoqué précédemment, les éléments finis peuvent être de plusieurs types. Les éléments peuvent être en une, deux ou trois dimensions selon l'application. Les éléments en une dimension sont typiquement les barreaux et les poutres et ils peuvent reprendre ou non certaines contraintes selon les directions de celles-ci. Les barreaux par exemple ne transmettent que les efforts axiaux, tandis que les poutres ne vont reprendre que les efforts de traction, torsion et flexion. Ces éléments simplifiés vont permettre de calculer le comportement global de structures rapidement, par exemple pour des ponts [103], mais il ne sera pas possible d'obtenir des résultats pour des zones localisées. Les éléments en deux dimensions ou trois dimensions donneront des informations plus précises, mais nécessiteront généralement plus de noeuds et d'éléments en plus de reprendre des efforts plus variés et donc plus de temps de calcul est

requis. Les éléments en deux dimensions sont souvent utilisés quand les pièces à étudier ont une épaisseur négligeable par rapport aux autres dimensions. Le choix des éléments à utiliser pour un problème va donc être une étape importante de la méthode des éléments finis et va se faire suivant les types de résultats que l'on souhaite obtenir ainsi qu'avec les contraintes que l'on a (des délais par exemple).

Les exemples présentés utilisant les éléments finis portaient plutôt sur la modélisation des objets saisis ou à saisir, mais la méthode s'applique aussi du côté du préhenseur. Par exemple pour les doigts sous-actionnés compliants [105] ou encore les *soft grippers* [106–108]. À la connaissance de l'auteur peu d'articles présentent des analyses de systèmes objets et pinces sous-actionnées complets à l'aide d'éléments finis. Les contraintes appliquées sur les objets sont généralement simplifiées. Dans les exemples présentés précédemment l'outil, la pince, étaient généralement modélisée par deux surfaces qui viennent compresser l'objet, voir seulement deux forces. À cause de sa complexité et de ses temps de calcul, la MEF n'est pas adaptée pour l'analyse rapide et complête des interactions entre doigts adaptatifs et objets déformables.

La MEF est la technique la plus commune, mais d'autres techniques existent encore bien que moins répandues pour étudier les problèmes de saisie et les déformations de l'objet. On peut évoquer parmi celle-ci d'autres méthodes avec maillage (méthode des différences finies et des volumes finis) ainsi que des méthodes sans maillage [110]. L'hydrodynamique des particules lissées [111] est une de ces dernières et est utilisée en mécanique des fluides, mais aussi en mécanique du solide. Dans le cas des méthodes sans maillage, les nœuds ne sont pas connectés entre eux, mais les interactions entre eux seront prises en compte. L'absence de connexion facilitent les études de déformations lorsque les limites des solides à étudier sont complexes et évolutives, telles que dans les cas de propagations de fissure ou encore de fragmentation.

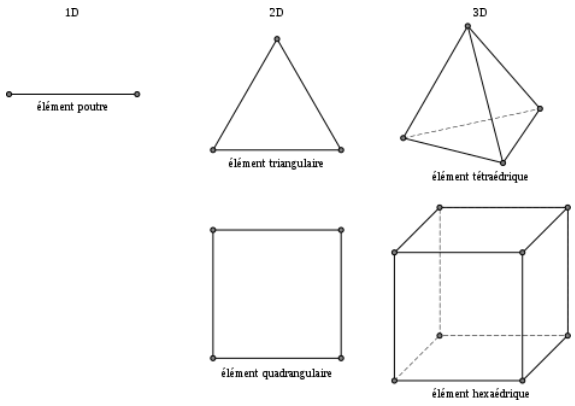


Figure 2.15 Exemples de types d'éléments finis (image libre de droit tirée de [104])

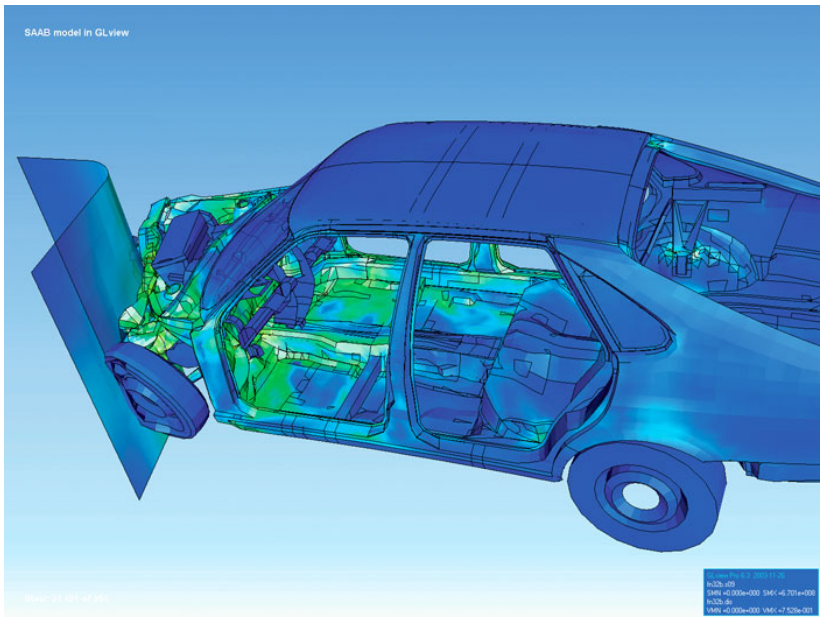


Figure 2.16 Étude par éléments finis d'une collision (image libre de droit tirée de [109])

Par contre les temps de calcul de ces méthodes ne sont pas forcément meilleur qu'avec la MEF [112]. Le temps gagné par l'abscence de maillage étant compensé par la définition des limites. Il existe encore aussi des méthodes hybrides combinant méthode sans maillage et éléments finis [113]. Du côté des pinces, pour les *softfingers* d'autres méthodes que la MEF sont utilisées, comme le modèle de Cosserat [114] pour calculer les forces de saisie sur un objet rigide ou encore en utilisant le modèle pseudo-rigide évoqué précédemment pour revenir à un cas de mécanisme à membrures rigides "classiques". Avec le modèle de Cosserat, un objet souple est approché par un empilement infini de micro-solide plat le long d'une courbe décrivant l'objet [115, 116]. On aura par exemple pour un doigt une courbe allant de la base jusqu'au bout du doigt. Pour chaque abscisse de cette courbe une section perpendiculaire sera un micro-élément du modèle. Ces derniers seront décrits par leur position et leur orientation en fonction de leur abscisse de long de la courbe. Avec cette méthode, des équations pour les micro-éléments permettront aussi de calculer la déformation de l'objet. Dans les cas présentés précédemment, l'évaluation d'un système complet déformable est à la connaissance de l'auteur encore une fois rarement présent dans la littérature.

Une autre méthode alternative à la MEF classique que l'on retrouve avec des problèmes de saisie est le modèle masse-ressort. La figure 2.17 illustre l'utilisation de ce modèle. Comme son nom l'indique, un objet est modélisé grâce à des points (en rouge sur la figure), pour approcher la géométrie (en bleue), reliés par des ressorts qui eux miment la flexibilité du matériau. La masse est finalement répartie sur chaque point. Cette technique est très répandue dans le

domaine des effets spéciaux et de l'animation, car rapide et simple, à l'inverse de la MEF. Elle est ainsi utilisée pour les applications temps réel [117], même si sa précision n'est pas parfaite. Plusieurs équations existent, mais les valeurs de raideurs des ressorts des MMR sont généralement calculées en fonction de la géométrie de la discrétisation, c'est-à-dire les aires délimitées par les points de masse et des propriétés du matériau [118,119]. Les discrétisations sont généralement des triangles et des carrés en deux dimensions.

Dans l'optique d'optimiser des pinces sous-actionnées pour la saisie d'objet déformable et donc de simuler et comparer un grand nombre de configurations, le MMR semble être une alternative de choix à la MEF, exigeante en capacité de calcul.

2.5 Conclusion

Dans cette revue de littérature, les avantages des pinces sous-actionnées ont été mis en avant, ainsi que quelques axes de recherche sur le sujet. Bien qu'il y ait de nombreuses voies afin d'optimiser ces pinces comme l'a montré la section 2.2.4, en particulier pour utiliser au mieux l'adaptation mécanique, aucune étude n'a encore porté sur les interactions avec des objets déformables. En effet, les interactions ont toujours été considérées avec des objets rigides. C'est une hypothèse qui peut se vérifier dans certains cas, mais qui a ses limites dans certains domaines : le médical par exemple. C'est pourquoi l'objectif central de cette thèse est d'étudier les interactions pinces sous-actionnées - objets déformables, en particulier pour l'optimisation des pinces. Dans la prochaine section, nous allons voir la méthodologie développée lors de cette thèse.

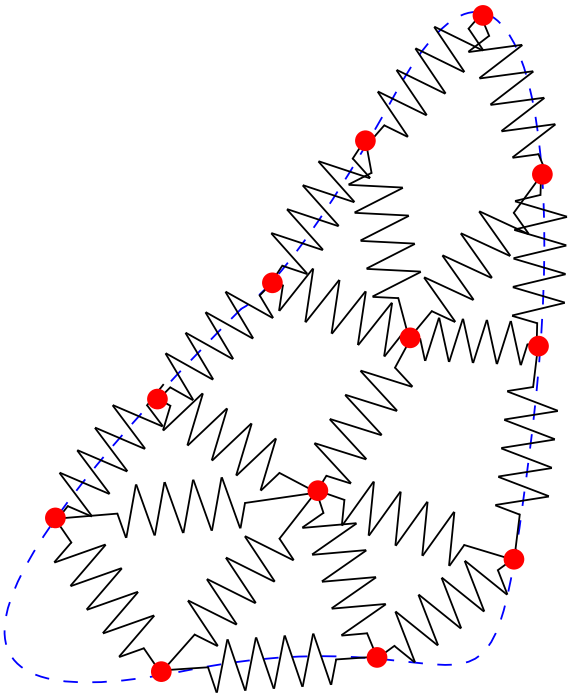


Figure 2.17 Modèle masse-ressort

CHAPITRE 3 MÉTHODOLOGIE ET ORGANISATION

L'élément central de ce doctorat est la modélisation des interactions entre objets déformables et pinces sous-actionnées pour l'optimisation de la saisie en minimisant les déformations de l'objet. La première étape a donc été de trouver des modèles adaptés à notre problématique. De par sa simplicité de mise en œuvre, le MMR en a fait un bon candidat pour le modèle de l'objet. Tandis que le MPR est déjà une approche bien connue pour modéliser les doigts compliants sous-actionnés. Deux approches ont ensuite été utilisées pour faire le lien entre les forces et déformations des pinces et d'objets déformables.

La première approche fait intervenir les énergies potentielles élastiques ainsi que le travail des forces d'actionnement [120]. Un article sur le modèle qui a été réalisé grâce à cette approche a été publié lors de l'*ASME International Design Engineering Technical Conferences and Computers and Information in Engineering Conference* en 2018 et s'intitule *Deformation analysis of a compliant underactuated fingergrasping a soft object*. Il constitue le chapitre 4 de ce manuscrit. Le modèle est présenté, suivi par sa validation à l'aide de comparaisons entre des simulations du modèle et des résultats obtenus avec un logiciel de simulation dynamique, ainsi que sa vérification à l'aide d'un logiciel d'analyse par éléments finis. Finalement, un exemple d'application illustre l'utilité d'un tel modèle. À la suite de cet article des comparaisons ont été réalisées avec des résultats expérimentaux et bien que les simulations étaient relativement proches des mesures prises avec le prototype, un faible écart entre les deux était tout de même observable.

Une seconde approche a donc été approfondie, cette fois en se servant directement du principe fondamental de la statique [121]. Pour cela le modèle présenté dans la revue de littérature tiré de [37] a été employé, mais il n'a pas été applicable directement. En effet celui-ci n'est valable que dans les cas où chaque phalange n'a qu'un point de contact avec l'objet et dans les cas où les objets sont déformables, plusieurs points de contact entre l'objet et le doigt sont à prévoir sur les phalanges. Une méthode a été développée afin de ne plus avoir cette limite de contact en séparant virtuellement les membrures avec plusieurs points de contact par des liaisons rotoïdes combinées à des ressorts très raides. Avec cette approche, un article a aussi été publié dans *Journal Mechanisms Robotics* intitulé *Deformation Modeling of A Compliant Robotic Finger Grasping Soft Object*. Celui-ci est présenté dans le chapitre 5. Les mêmes comparaisons ont été réalisées afin de valider et vérifier le modèle numérique. La deuxième approche présentant de meilleurs résultats a finalement été choisie pour une application d'optimisation d'architecture de pince sous-actionnée afin de déformer au minimum un objet

cylindrique. Cette application a été présentée dans l'article *Bistable Compliant Underactuated Gripper for the Gentle Grasp of Soft Objects* soumis au journal *Mechanism and Machine Theory* et qui constitue le chapitre 6 de cette thèse.

3.1 Contributions

Les contributions de cette thèse sont :

- Deux modèles numériques pour estimer les interactions entre des doigts mécaniques et des objets déformables. L'une en utilisant la conservation de l'énergie, l'autre l'équilibre statique des forces ;
- Application de l'équation des forces de contacts de doigts sous-actionnés dans le cas où il y a plus d'un contact par phalanges. Ce qui n'était pas possible précédemment ;
- Optimisation de l'architecture d'une pince sous-actionnée pour la saisie d'objets mous à l'aide du modèle numérique.

3.2 Publications

Les publications issues des travaux de recherche effectués sont les suivantes :

- N. Mouazé and L. Birglen, "Deformation analysis of a compliant underactuated finger grasping a soft object," dans *ASME 2018 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference.* Volume 5A : 42nd Mechanisms and Robotics Conference. Quebec City, Quebec, Canada. 26–29 août, 2018. V05AT07A001. ASME. DOI : 10.1115/DETC2018-85226
- ———————, "Deformation Modeling of A Compliant Robotic Finger Grasping Soft Object," *Journal of Mechanisms and Robotics.* Février, 2021 ; 13(1) : 011009. DOI : 10.1115/1.4047988
- ———————, "Bistable Compliant Underactuated Gripper for the Gentle Grasp of Soft Objects," soumis au journal *Mechanism and Machine Theory.* 28 juillet, 2021

D'autres publications ont été soumises et acceptées lors de ce doctorat liées à des sujets connexes. L'article de journal [122] publié en 2019 dans *Transactions of the Canadian Society for Mechanical Engineering* et qui a fait suite à un article de conférence du même nom présenté la même année lors du Symposium du *Canadian Committee for the Theory of Machines and Mechanisms* [84] présente un mécanisme à topologie variable compliant réalisable en un seul élément passant d'un mouvement de rotation à un mouvement de translation. Un second

article de conférence [123] accepté en 2020 par l'*U.S. Committee for the Theory of Machines and Mechanisms* pour le *Symposium on Mechanical Systems and Robotics* intitulé *Twisting String Actuation with Noncircular Wrapping Rods* présente des travaux sur les mécanismes d'actionnement dit *Twisting String Actuation* qui ont l'avantage d'être légers et compacts. Celui-ci a aussi été suivi d'un article de journal sur le même sujet publié dans *International Journal of Mechanisms and Robotic Systems* [124]. Finalement, en 2021 des évaluations du second modèle présenté dans cette thèse pour simuler les interactions objet/pince ont été réalisées et présentées dans l'article intitulé *Point-Spring Modeling of the Grasp of a Deformable Object by an Angular Gripper* [125]. Cet article de conférence a été présenté lors du Symposium du *Canadian Committee for the Theory of Machines and Mechanisms* tenu en ligne en 2021.

— N. Mouazé and L. Birglen, "Preliminary design of a revolute to prismatic morphing compliant joint," *Proceedings of the CCToMM Mechanisms, Machines, and Mechatronics M3 Symposium*, Montréal, QC., Canada, 15-16 mai, 2019.

— ———————, "Preliminary design of a revolute to prismatic morphing compliant joint," *Transactions of the Canadian Society for Mechanical Engineering*. 44(4) : 481-491, 2019, NRCResearchPress.

— ———————, "Twisting String Actuation with Noncircular Wrapping Rods," *Proceedings of the 2020 USCToMM Symposium on Mechanical Systems and Robotics*, 126-136, Rapid City, South Dakota, USA, 14-16 mai, 2020.

— ———————, "Non-cylindrical wrapping rods and compliant adapters for multi-mode twisting string actuation," *International Journal of Mechanisms and Robotic Systems*, 5 :1-2, 93-110, 2021, Inderscience Publishers (IEL).

— ———————, "Point-Spring Modeling of the Grasp of a Deformable Object by an Angular Gripper," *Proceedings of the 2021 CCToMM Symposium on Mechanical Systems and Robotics*, 3-4 juin, 2021.

CHAPITRE 4 ARTICLE 1 : DEFORMATION ANALYSIS OF A COMPLIANT UNDERACTUATED FINGER GRASPING A SOFT OBJECT

Cet article a été présenté lors de la conférence avec comité de lecture ASME 2018 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference (IDETC/CIE 2018), disponible en ligne depuis le 2 Novembre 2018.

Nicolas Mouazé et Lionel Birglen

Abstract

In the literature, many models of compliant fingers grasping rigid objects have been extensively discussed. However, when the objects are themselves deformable, as in many practical cases, the effect of compliant underactuated fingers onto these soft objects is generally not addressed due to the complexity of the model required for accurate results. This paper aims at addressing this issue by proposing to simulate deformations using a simple mass-spring model. This model discretizes the object similarly to how a pseudo-rigid body technique usually approximates the compliant finger. Comparisons between simulations using the proposed model and finite element analyses demonstrate that for a significant range of deformations our approach offers an efficient and accurate approximation while less computationally intensive.

4.1 Introduction

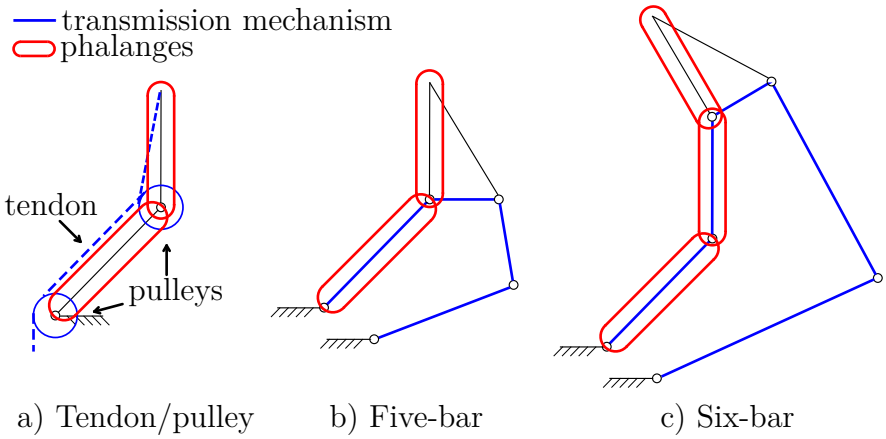
An underactuated finger [37] is constituted by a serial arrangement of n links (the phalanges) actuated by a maximum of $n-1$ actuator(s) and completed by a transmission mechanism. Usually, two or three phalanges are used and driven by a single actuator. In order to distribute the actuator torque to every phalanx, transmission mechanisms are required and typically combined with passive elements such as springs with mechanical limits to statically constrain the device. For example, a tendon/pulley mechanism is used in [40] (illustrated in Fig. 4.1a), but a five-bar as in [37] (cf. Fig. 4.1b) or a six-bar mechanism [38] (cf. Fig. 4.1c) are also possible. Underactuated mechanisms in grasping have many advantages. First of all, having fewer actuators than degrees of freedom (equal to the number of phalanges in our case) considerably simplifies control and diminishes costs. Secondly, an underactuated finger will adapt to the shapes of grasped objects mechanically with a natural enveloping motion similar to the one of a human finger. This attractive feature is obtained without any sensor

or control. Indeed, it has been noticed that underactuated fingers mimic the adaptation of human fingers when grasping [48, 105, 126]. Compared to underactuated fingers, typical mechanical grippers and pliers have a simplistic closing motion and produce pinch grasps which can damage fragile objects due to the concentrated pressure at the few contact points.

The actual grasp sequence of an underactuated finger can be separated in several steps. During the first phase, the finger is not in contact with the object and the phalanges approach the latter (this phase is sometimes referred to as pregrasping or preshaping). Then, in a second step, one of the phalanges makes contact with the object. From this situation, when increasing the actuation force or torque several motions can occur [37, 42, 127] generally leading to stable grasps if the finger is properly designed. Grasping can improve with more phalanges in contact yielding a more homogeneous distribution of contact forces which allows for a more delicate hold.

Underactuated fingers can be built using compliant materials and are thusly referred to as compliant underactuated fingers [128–130]. Relying on compliance usually simplifies manufacturing, makes lubrication unnecessary, and is an attractive alternative to traditional machine design in many cases. In [128] for example, every joint of an underactuated finger model (two phalanges with a four-bar transmission mechanism) were coupled with springs in order to obtain a pseudo-rigid body model of a compliant finger using living hinges. Then, in [80], a compliant gripper (two fingers with five phalanges both) in Nitinol was presented as a first step toward a surgical tool using a similar design. Lumped compliance [131, 132] is the closest model to rigid mechanisms and the most popular in the literature but the use of distributed compliance has also been proposed [133]. However, to the best authors' knowledge, although many compliant underactuated finger models exist, the issues arising during the grasp of soft objects has never been studied.

Anticipating the deformation of a soft object is an important criterion to design a finger or a clamp in order to mitigate possible local damages [134, 135]. Controlling contact forces according to stiffness is also necessary for health critical applications such as surgery [136, 137]. In the literature, various models have been discussed to accurately represent soft bodies, e.g. [138, 139], and the best-known method is probably the finite element analysis (FEA) which is very accurate but generally quite computationally intensive and thus, slow to produce results [92, 93]. Conversely, the mass-spring system (MSS) model, commonly used in visual special effects for instance [140], is faster and much simpler at the cost of more limited accuracy. With MSS, objects are discretized into a set of mass points connected by springs [118] and in order to estimate these springs' stiffnesses from a known material and geometry, several techniques were developed [119, 141, 142]. Amongst the latter, the Van



a) Tendon/pulley b) Five-bar c) Six-bar

Figure 4.1 Examples of underactuated finger design

Gelder model [118] uses a 2D triangular geometry, or "meshing," and proposes semi-empirical formulas for these stiffnesses depending on areas, lengths, and material properties.

In this paper, a MSS model is used to simulate the grasp of a compliant underactuated finger in order to observe the resulting deformations of the seized soft objects and the results are shown to be comparable to FEA with a surprising accuracy. A numerical model will be first detailed and validated with multi-body dynamic software simulations and then, FEA. Subsequently, to illustrate applications of this model, the MSS model will be used to simulate soft objects with various Young's modulus and quantify penetration of the grasper inside these objects.

4.2 Modeling

The models of underactuated finger and semi-circular objects considered in this paper are presented in Fig. 4.2. This model represents half the gripper which is assumed symmetrical. The illustrated transmission mechanism is a five-bar mechanism, distributing the actuation torque T_a to the two phalanges.

During the grasp, the velocities of the links are assumed small and thus, kinetic energy is neglected compared to the elastic potential energy stored by the springs. Gravity is assumed negligible as well. Therefore, a quasi-static analysis of the grasp is considered sufficient.

To model the finger's living hinges, revolute joints are assumed following the model presented in [82]. For small to moderate angular displacements, the rotation between the two rigid links connected by these revolute joints can approximate bending in the associated living hinges.

The masses and springs discretizing objects in triangles according to the Van Gelder model

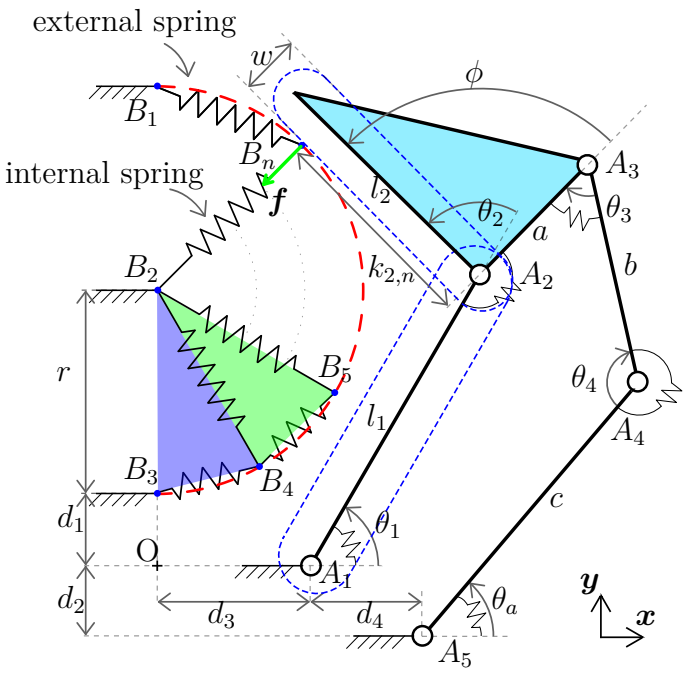


Figure 4.2 Model of a semi-circular object using a mass-spring system and a compliant underactuated finger with a five-bar transmission mechanism

are illustrated in Figs. 4.3 and 4.4. Since gravity is assumed negligible, the actual magnitudes of the masses is not relevant in our model. The springs' stiffnesses on the other hand are critical and can be calculated with [118]:

$$K_j^{obj} = \sum_{i=1}^{n_t} \frac{E}{1+\nu} \frac{A_{T_i}}{c_i^2} + \frac{E\nu}{1-\nu^2} \frac{a_i^2 + b_i^2 - c_i^2}{8A_{T_i}} \tag{4.1}$$

for $j = 1, ..., p$, where p is the total number of springs in the model, n_t is the number of triangles contiguous to the j^{th} spring, A_{T_i} are the areas of these triangles, and a_i, b_i, c_i are the lengths defining the triangles T_i (see Fig. 4.4). The internal springs of a discretized semi-cylinder, as shown in Fig. 4.2, are each contiguous to $n_t = 2$ triangles, whereas external springs are contiguous to $n_t = 1$ triangle. Finally, E is the Young's modulus of the object material and ν is its Poisson's ratio.

In our model, the origin is always assumed at point O, shown in Fig. 4.2. From its resting state, i.e. with a zero actuation torque, the finger is flexed by an incrementally increasing T_a and for each of these increments, the new position of the finger is calculated by minimizing the variation of the potential energy of the model from the previous state. The degrees of freedom in our model, determined by this minimization, are $\theta_1, ..., \theta_n$ i.e. the relative angles between the phalanges, $k_{j,m}$ the position of contact if the m^{th} mass point is in contact with

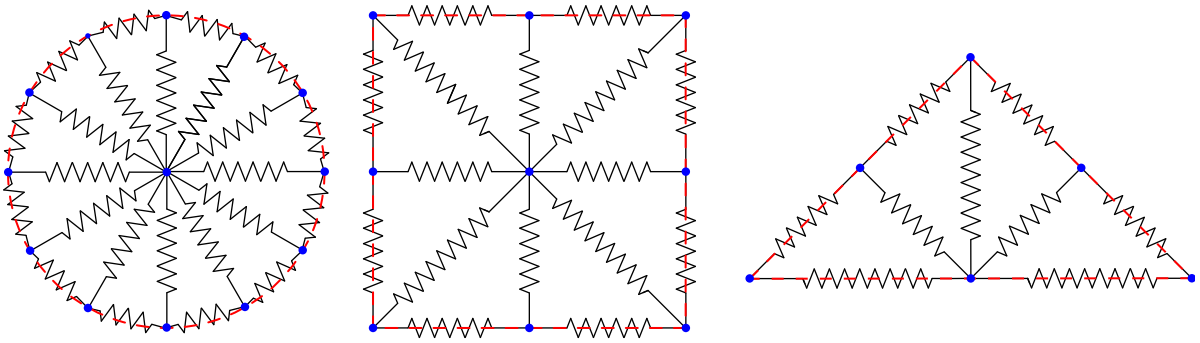


Figure 4.3 Discretization of common shapes for Van Gelder models

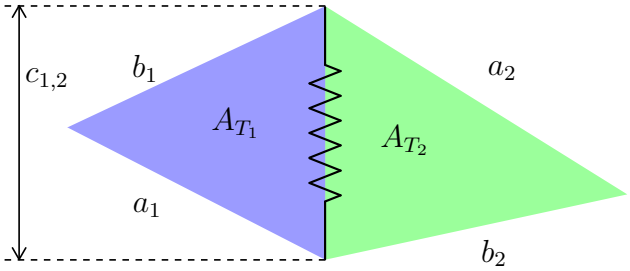


Figure 4.4 Nomenclature of triangular Van Gelder models

the j^{th} phalanx and $\boldsymbol{r_{OB_i}}$ the positions of the i^{th} mass if the latter is not touching the finger. The corresponding potential energy not only include the one stored by the springs in the finger, but also the potential energy from the springs modelling the object. The total virtual work δW of all the associated forces and torques is:

$$\delta W = T_a \delta\theta_a + \sum_{i=1}^{n_f} T_i^f \delta\theta_i + \sum_{j=1}^{n_o} T_j^o \delta x_j \tag{4.2}$$

with T_a the actuation torque, θ_a the associated actuation angle, n_f is the number of joint in the finger, $T_i^f = -K_i^f(\theta_i - \theta_{0i})$ is the torque at the i^{th} joint of the finger due to its spring, K_i^f is the stiffness of this spring while θ_i and θ_{0i} are the corresponding angle at torque T_a and the initial position respectively. Similarly, n_o is the number of springs used to model the object, $T_j^o = -K_j^o(x_j - x_{0j})$ is the force due to the j^{th} spring of the Van Gelder model, K_j^o is the stiffness of the latter j^{th} spring while x_j and x_{0j} are the corresponding lengths at torque T_a and rest respectively. From all the previous variables, all other geometrical parameters representing the grasp can be established, i.e.:

$$\boldsymbol{r_{OA_2}} = \boldsymbol{r_{OA_1}} + \begin{bmatrix} l_1 \cos(\theta_1) \\ l_1 \sin(\theta_1) \end{bmatrix}, \tag{4.3}$$

$$\boldsymbol{r_{OA_3}} = \boldsymbol{r_{OA_2}} + \begin{bmatrix} a\cos(\sum_{p=1}^{2}\theta_p + \phi) \\ a\sin(\sum_{p=1}^{2}\theta_p + \phi) \end{bmatrix} \tag{4.4}$$

where $\boldsymbol{r_{AB}}$ is the vector from point A to point B and with a the length between A_2 and A_3. One can also define b the length between A_3 and A_4, c the length between A_4 and A_5, d the length between A_2 and A_4, e the length between A_3 and A_5, w the phalanx width, and l_j the length of the j^{th} phalanx. Except d and e which were omitted to improve legibility, all these parameters are illustrated in Fig. 4.2.

Using Eqn. (4.3) and (4.4), $\boldsymbol{r_{OA_4}}$ is then calculated by finding the intersections of the two circles centered in A_5 and A_3, and with radii c and b respectively. From the two intersections found, A_4 is chosen as the closest to his previous position before the latest increment of the actuation torque. From the positions of these points, all the other angles in the finger can be calculated, namely:

$$\theta_3 = \arccos\left(\frac{a^2 + b^2 - d^2}{2ab}\right) \tag{4.5}$$

$$\theta_4 = \arccos\left(\frac{b^2 + c^2 - e^2}{2bc}\right) \tag{4.6}$$

$$\theta_a = \arctan2\left(\boldsymbol{y}^T\boldsymbol{r_{A_5A_4}}, \boldsymbol{x}^T\boldsymbol{r_{A_5A_4}}\right) \tag{4.7}$$

where $\boldsymbol{x}$ and $\boldsymbol{y}$ are the vectors along the $\boldsymbol{x}$ and $\boldsymbol{y}$ axes of the frame respectively. Eqn. (4.5), (4.6) and (4.7) are then used to compute the virtual work of the different torques applied to the finger and thus, the potential energy stored in the latter the finger due to its springs, i.e. the first and second parts of the right-hand side in Eqn. (4.2).

When a contact exists, if the m^{th} mass point is touching the j^{th} phalanx (this is valid with even more than two phalanges), one has:

$$\boldsymbol{r_{OB_m}} = \begin{bmatrix} s_j + k_{j,m}\cos(\theta_j) + d_3 + \frac{w}{2}\cos(\sum_{p=1}^{j}\theta_p + \frac{\pi}{2}) \\ t_j + k_{j,m}\sin(\theta_j) + \frac{w}{2}\sin(\sum_{p=1}^{j}\theta_p + \frac{\pi}{2}) \end{bmatrix} \tag{4.8}$$

where $s_1 = 0$ and $t_1 = 0$, if $j = 1$. For $j > 1$:

$$s_j = \sum_{p=1}^{j-1} l_p \cos(\sum_{q=1}^{p}\theta_q), \tag{4.9}$$

$$t_j = \sum_{p=1}^{j-1} l_p \sin(\sum_{q=1}^{p}\theta_q). \tag{4.10}$$

From all the positions of the mass points, the lengths x_i can be obtained and used in

Eqn. (4.2). Finding the minimum value of the potential energy is finally realized through a numerical optimization (in MATLAB), in which the positions of the finger and deformations of the object can be obtained depending on the torque applied on the finger. This minimization process is a direct search method which uses the Nelder-Mead simplex algorithm.

4.3 Comparison with a multi-body software simulation

Figure 4.5 shows comparisons between the numerical values of the angles between the phalanges of a compliant underactuated finger as obtained from the method presented in this paper and dynamic simulations with a commercial package (MSC ADAMS). Material properties used for this simulation are these of Polypropylene for the finger (Young Modulus E and Poisson's ratio ν). The grasped object was defined as a semi-circle, fixed on its flat side and discretized in n_{div} triangles, and thus with $n_{div} + 2$ mass points, with stiffnesses corresponding to stainless steel in order to approximate an infinitely stiff object. The relative angles θ_i are compared in the figure depending on the magnitude of the actuation torque. During the preshaping, when no contact is established between the finger and the object, these angles are identical. In Fig. 4.5, the first contact happens around $T_a = 4{,}740N \cdot m$ for the dynamic simulation and $T_a = 5{,}520N \cdot m$ for the numerical simulation, leading to a slight shift of both angles. This slight discrepancy between the value of T_a at the moment of first contact can be explained by the fact that in the numerical simulation the vertices of the MSS model are represented by a point, i.e. an infinitesimal element, whereas in the dynamic simulation, connecting elements between springs must be nonzero volume bodies (small spheres were used). Therefore, when the centers of masses are placed at the same location, these spheres will make contact slightly before the value predicted by the numerical model geometry. Nevertheless the angles in Fig. 4.5 show a close correspondence between the results of our model and the software. Parameters of the simulation are listed in Tab. 4.1. Subscripts K_{ext} and K_{int} indicate the stiffnesses of the external and internal springs of the semi-cylinder respectively.

4.4 Comparison with a finite element analysis

In order to further establish the accuracy of our model, a comparison with a FEA software was also conducted. In our numerical model, the object considered is again a semi-circle and the springs' stiffnesses are calculated based upon the Young's modulus and Poisson's ratio of the material. The same parameters were used to construct the FEA model of the finger (see Tab. 4.2).

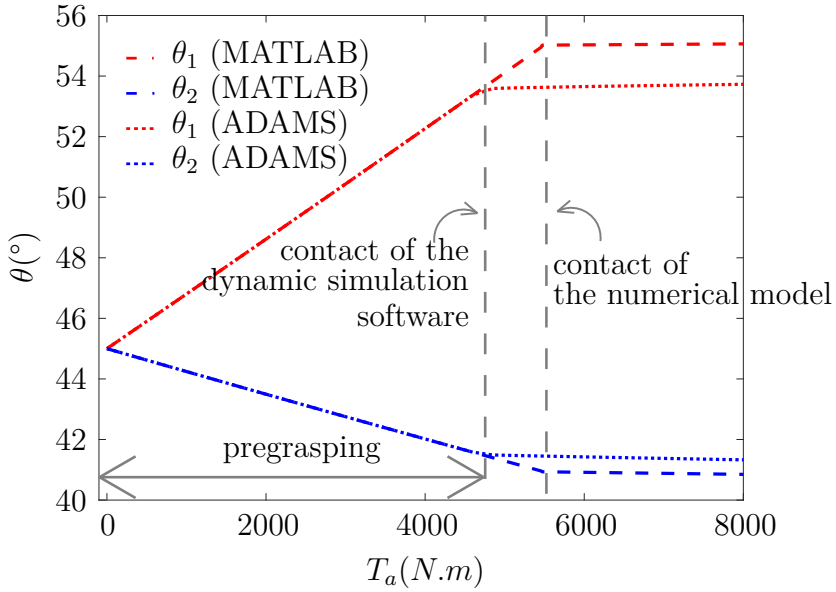


Figure 4.5 Comparison of the relative angles in the finger between the numerical simulation from a commercial software

Table 4.1 Parameters of the simulation presented in Fig. 4.5 (all lengths are in meters, angles in degrees, stiffnesses in kilonewtons per meter)

l_1	l_2	a	b	c	d_2	d_3	d_4	w
1	1	0.76	0.41	1.58	0.5	0.5	0	0.1

θ_{01}	θ_{02}	K_i^f	d_1	r	K_{ext}^{obj}	K_{int}^{obj}	n_{div}
45	45	12.4	0.2	1	81.9	55.9	5

Figure 4.6 shows the final position of the finger when actuated, as obtained with the FEA software (ANSYS). Initial comparisons between the relative angles from the numerical simulation and the finite element simulation showed a small gap between the curves when the torque increases, illustrating the fact that the accuracy of approximating a compliant mechanism with a PRBM is not perfect when the displacement grows. Using PRBM empirical formulas as initial estimates, spring' stiffnesses in the finger were adjusted to achieve the best fit between the curves. Figure 4.7 illustrates the relative angles in the finger as functions of T_a both from the FEA and from our model when no contact occurs, after adjustment.

In a second set of simulations an object was included (see Figs. 4.8 to 4.11). A rigid object is first used, and a soft one later on. The Young's modulus of the soft object is chosen at $10^5 Pa$ (with also $d_1 = 0.1m$, $r = 1.15m$ and $n_{div} = 16$). For both rounds of simulations,

the adjusted stiffnesses in the finger are used and the stiffnesses in the object are also fine tuned from the values obtained by using Eqn. (4.1). Table 4.3 shows the final stiffnesses used. As in the previous section, subscripts K_{ext} and K_{int} indicate the stiffnesses of the external and internal springs of the semi-cylinder respectively, while the superscripts K^R and K^S correspond to the rigid and soft cases respectively. Figures 4.8 and 4.10 show results of the finite element analysis in both cases.

In the rigid object case in Fig. 4.9, the discrepancy (small in terms of absolute magnitude) between the curves can be explained by the discretization of the object, as decreasing the number of triangles lowers the accuracy of this simulation. Results with the soft object are quite good on the other hand even for a large range of actuation torques, see Figs. 4.10 and 4.11.

4.5 Application: estimating non-damaging actuation torque

From the numerical model presented in this paper, many useful information can be obtained. For instance, one of the main purposes of underactuated fingers is to adapt to the shapes of grasped objects in order to distribute the contact forces over many phalanges. With this objective in mind, the developed model of a soft object and grasper can simulate and predict the deformation and penetration of the finger in the object. In Fig. 4.12, such penetrations of the finger in objects with different Young's modulus and a variable magnitude of the actuation torque are presented. The finger is assumed to be made of polypropylene, and its geometrical parameters are shown in Tab. 4.2. The Young's modulus of the object ranges from very small ($10^2 Pa$ for fat) to large values ($10^{11} Pa$ for Titanium). Several materials are indicated in the figure to illustrate the range of materials considered [143]. The Poisson's ratio is chosen equal to 0.3 in all cases. The penetration percentage is defined as the maximum distance between the deformation of the surface points with respect to their original position divided by the

Table 4.2 Parameters of the simulation presented in Figs. 4.6, 4.8 and 4.10 (all lengths are in meters, angles degrees)

l_1	l_2	a	b	c	d_2	d_3
1	1	0.76	0.41	1.58	0.5	0.5
d_4	θ_{01}	θ_{02}	d_1	r	n_{div}	w
0	45	45	0.1	1.15	16	0.1

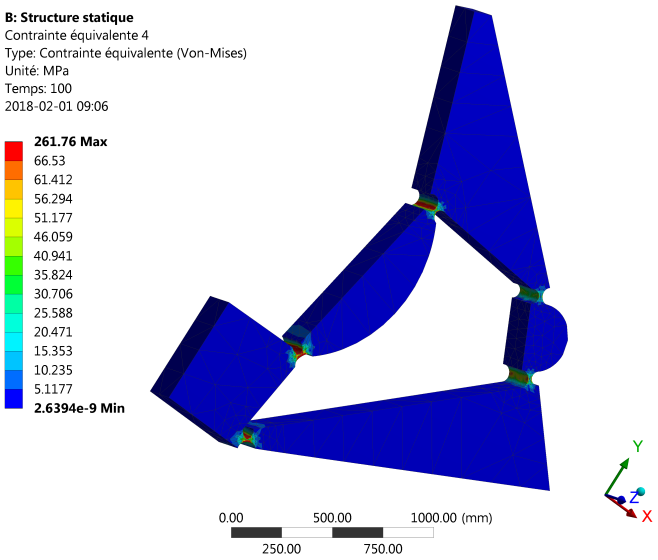


Figure 4.6 Compliant finger model

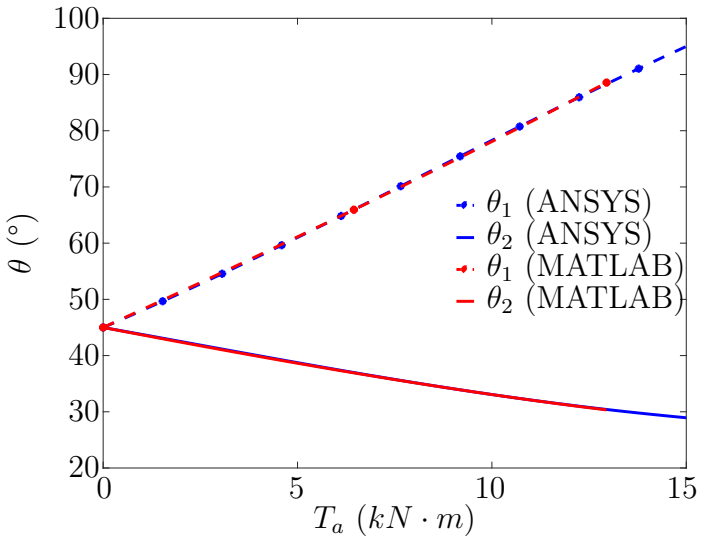


Figure 4.7 Comparison between the relative angles from our model and the finite element simulations

Table 4.3 Spring stiffnesses $(kN \cdot m)$

K_i^f	K_{ext}^R	K_{int}^R	K_{ext}^S	K_{int}^S
7.0404	216.14	14.1	111.99	7.31

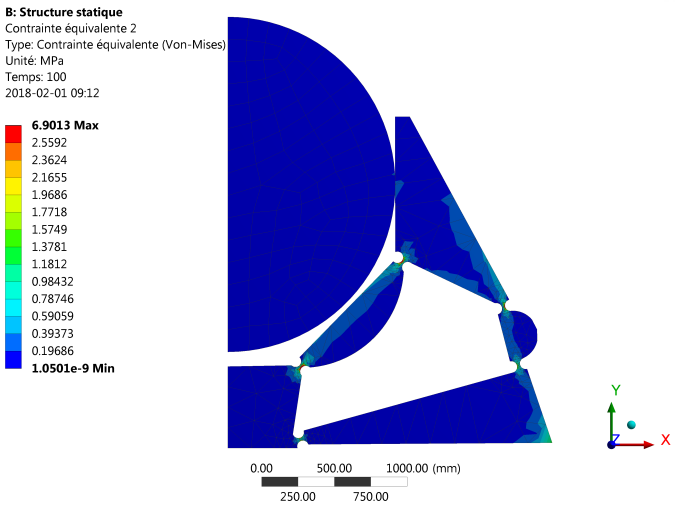


Figure 4.8 Compliant finger model with a rigid object

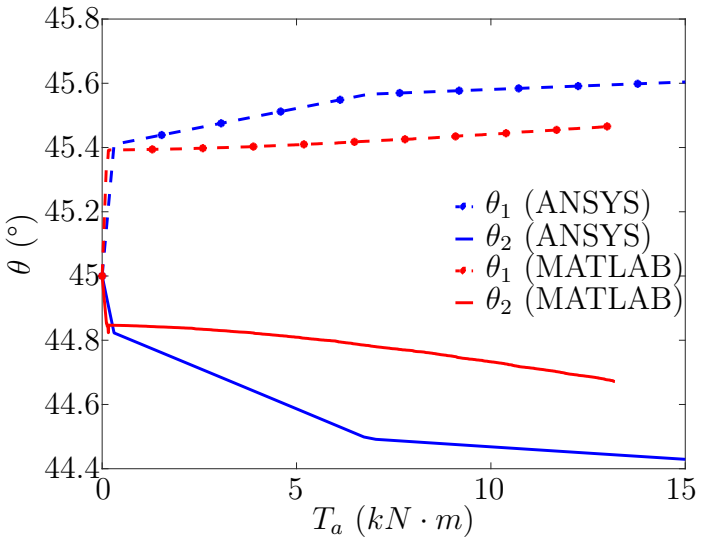


Figure 4.9 Comparison between the relative angles from our model and the finite element simulations with a rigid object

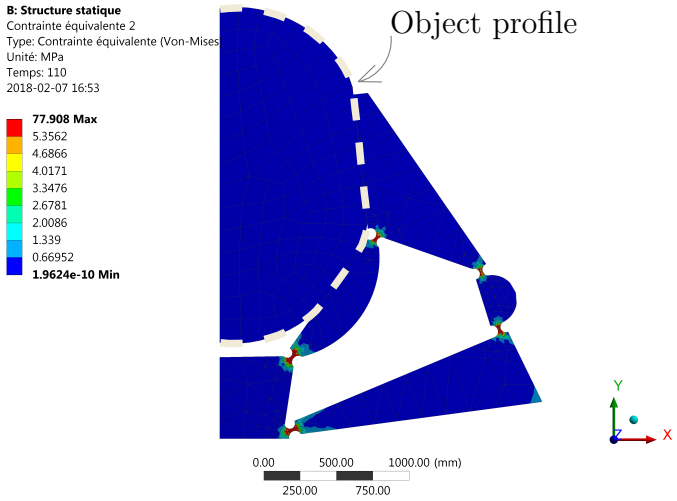


Figure 4.10 Compliant finger model with a soft object

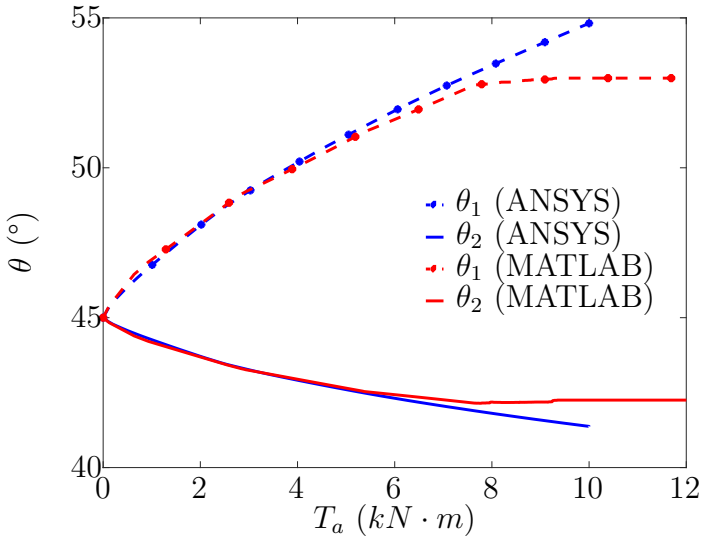


Figure 4.11 Comparison between the relative angles from our model and the finite element simulations with a soft object

initial radius. A penetration threshold is arbitrarily chosen at 5% to illustrate how the figure can be used to set a maximal non-damaging torque for the finger depending on the material of the object.

4.6 Discussion

In all the conducted simulations with our model, until the actuation torque reaches a certain magnitude (around $2.1N \cdot m$ in the previous Section), no phalanges touch the object regardless of the object material. With an input torque of $2.25N \cdot m$ and objects with Young's modulus from 10^2 to $10^4 Pa$, the calculated penetrations are almost constant. The stiffness of the object is negligible for the generated contact forces and the positions of the finger can be approximated as if there was not any object at all for such soft materials. With an input torque lower than $4N \cdot m$ and a Young's modulus for the object greater than $10^8 Pa$, the penetrations are almost 0% and in that case, the object can be considered rigid compared to the finger.

Finally, for input torques between $2.25N \cdot m$ and $4N \cdot m$ and Young's modulus inside the extreme values discussed before, one can efficiently evaluate with the proposed model the minimum Young's modulus of the object where the finger will not penetrate of a value greater than a set limit. For instance, with an input torque of $2.25N \cdot m$, the finger will not reach a position where the penetration of any material exceeds 5% in the illustrated example. For an object with a Young's modulus of $10^5 Pa$, the maximum input torque for which the penetration is lower than 5% is $2.75N \cdot m$, using again the parameters of the example.

It should be noted that the number of triangles used in the object model impacts the accuracy of the results and the computation time. Increasing this number in order to improve the quality of the approximation results in an increasing number of springs to model and evaluate. This also makes our model actually become a FEA in all but name. Conversely, an insufficient number of divisions will lead to only a rough approximation. In the FEA simulations illustrated in Figs. 4.8 and 4.10, the semi-cylinder is discretized with 210 elements and 1,320 nodes and the finger with 3,058 elements and 6,408 nodes. In comparison, 16 triangular elements and 18 nodes discretized the object in the MSS model while the finger uses a PRBM. The computing time of the finite elements analyses on a standard computer is on average 2.5 hour whereas our numerical simulations take 1.2 hour to compute with a code not optimized for speed.

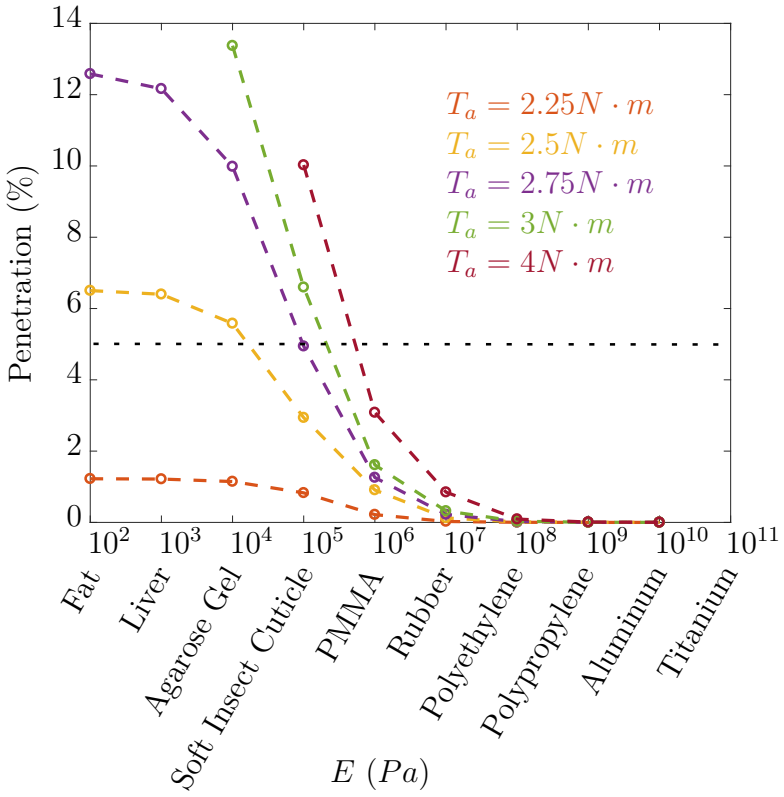


Figure 4.12 Simulation of the penetration in an object by a compliant underactuated finger

4.7 Conclusion

This paper presented a model for a compliant underactuated gripper grasping a soft object. In order to evaluate and limit possible damages on this object and to take full advantage of the adaptive behavior of the finger, this model was developped using a MSS methodology to simulate the resulting deformations. From this model, numerical simulations were firstly validated with dynamic simulations, then with FEA, to ensure accuracy. Finally, an application of this model was presented to illustrate its usefulness for the design and possibly force control of adaptive grippers.

4.8 Résultats complémentaires

Des résultats complémentaires à l'article présenté dans ce chapitre non publiés sont présentés dans cette section.

4.8.1 Comparaisons complémentaires

Le modèle proposé dans ce chapitre est extensible à d'autres géométries d'objet, par exemple un cylindre complet seulement attaché au niveau de son centre et donc, libre de tourner et/ou de se compresser verticalement, voir la figure 4.13 pour un exemple. Un doigt différent a aussi été simulé dont les paramètres sont présentés dans le tableau 4.4, où d_5 est la distance horizontale entre le centre du cylindre et son origine O. La figure 4.14 présente les résultats d'une comparaison entre les résultats d'une simulation éléments finis et le modèle. La simulation montre une saisie enveloppante (la seconde phalange se ferme sur l'objet pendant que la première est presque à sa position initiale à cause du contact). La comparaison illustrée met en évidence des bruits pour les résultats obtenus avec le modèle proposé, en particulier pour la position de la seconde phalange. Ce bruit de mesure apparaît après une forte déformation de l'objet. Son origine pourrait venir de la difficulté de l'algorithme à trouver une solution pour minimiser l'équation (4.2) quand les déformations deviennent importantes, c'est-à-dire quand plus de points sont en contact. Finalement, les simulations de saisie par le bout des doigts d'objets carrés présentées figure 4.15 et 4.16 sont, de la même manière que dans le cas des prises enveloppantes, très proches des résultats obtenus avec les simulations éléments finis. Les paramètres des simulations sont listés dans le tableau 4.5. Les variables d_2 et d_5 sont les distances entre O et le milieu de l'objet tandis que l est la longueur d'un côté du carré. À l'état initial, la partie inférieure du carré est fixée parallèle à $\boldsymbol{x}$.

Tableau 4.4 Paramètres de la simulation présentée sur la figure 4.13 (toutes les longueurs sont en mètre, les angles en dégrée, les raideurs en $N \cdot m$)

l_1	l_2	a	b	c	d_2	d_3
1	1	0.53	0.73	1.43	0.5	0.5

d_4	w	θ_{01}	θ_{02}	d_1	r	n_{div}
0	0.1	45	45	0.4	0.35	32

d_5	K_i^f	K_{ext}^{obj}	K_{int}^{obj}
0.682	195	$144 \cdot 10^3$	$549 \cdot 10^3$

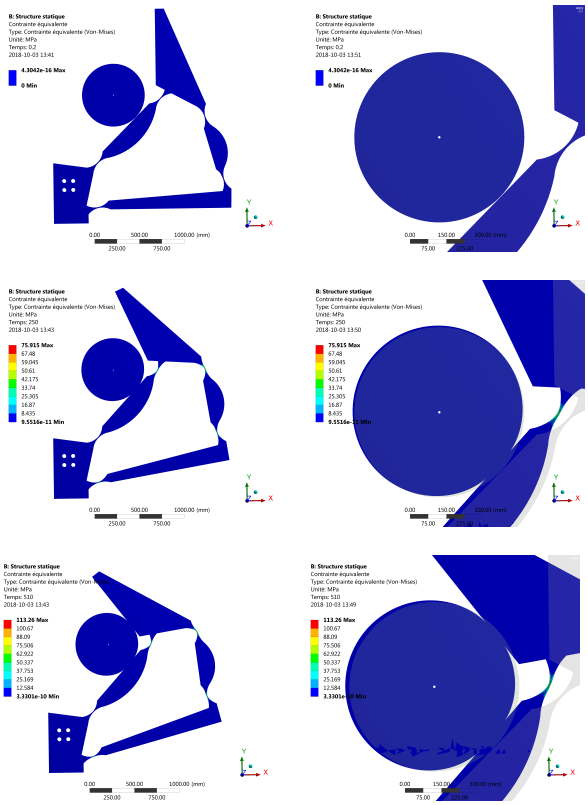


Figure 4.13 Prise enveloppante d'un cylindre illustrant l'adaptation des doigts aux déformations de l'objet

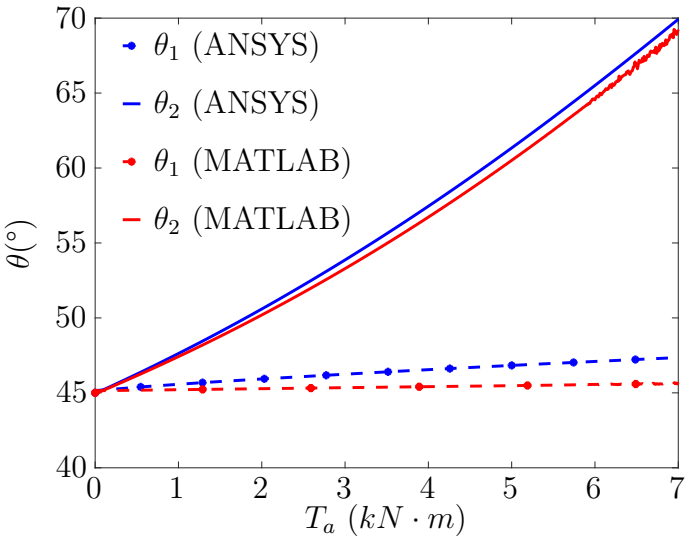


Figure 4.14 Comparaison entre les angles inter-phalanges simulées avec le modèle et avec des analyses éléments finis lors de la saisie enveloppante d'un objet déformable

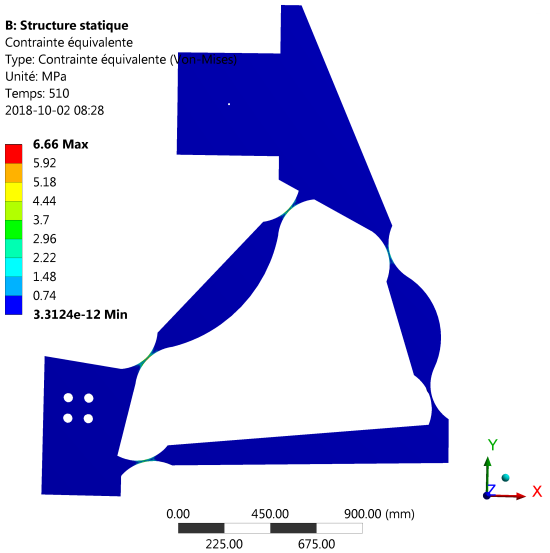


Figure 4.15 Saisie du bout des doigts d'un objet carré

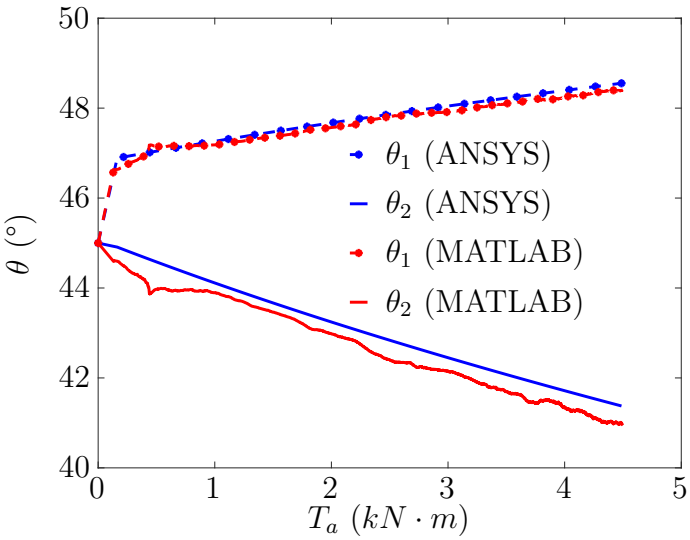


Figure 4.16 Comparaison entre les angles inter-phalanges simulées avec le modèle et avec des analyses éléments finis pour un objet carrée

Tableau 4.5 Paramètres de la simulation illustrée sur la figure 4.15 (toutes les longueurs sont en mètre, les angles en dégrée, les raideurs en $N \cdot m$)

l_1	l_2	a	b	c	d_2	d_3
1	1	0.53	0.73	1.43	0.5	0.5
d_4	w	θ_{01}	θ_{02}	d_1	l	n_{div}
0	0.1	45	45	1.25	0.5	32
d_5	K_i^f		E^{obj}		ν^{obj}	
0.87	195		$10^6 Pa$		0.31	

Dans le cas de la saisie de l'objet carré, on peut observer un écart plus significatif pour la position de phalange distale que pour la phalange proximale entre mesures expérimentales et résultats numériques. La phalange distale étant la seule en contact, cet écart pourrait s'expliquer par l'angle entre la surface du carré et celle de la phalange distale. En effet, au repos les deux surfaces sont parallèles, mais lors du contact la phalange distale a déjà tourné d'un petit angle. Dans la modélisation numérique, du modèle, une fonction va évaluer s'il y a contact ou non entre un point de l'objet et une phalange suivant la distance entre les deux, une distance minimale est à fixer en fonction de l'incrément du couple d'actionnement. Si un pas important d'actionnement est utilisé, la distance est fixée en conséquence afin que le doigt ne rentre pas dans l'objet. Si un contact est détecté la position du point concerné sera sur la phalange correspondante. Dans le cas du carré, plusieurs points se situent sur la surface verticale, il est donc possible que plusieurs points soient considérés en contact dès le premier contact. Le programme pourrait donc coller directement la phalange à toutes la surface verticale de l'objet à cause de la tolérance fixée lors de cette simulation ce qui expliquerait cet écart qui reste presque constant.

4.8.2 Banc de test

Afin de valider le modèle numérique, un banc de test simple a été conçu afin de mesurer les déformations lors de l'interaction entre un doigt et un cylindre déformable. Du silicone venant du commerce (Smooth-On, Ecoflex$^{\text{TM}}$ 00-30) généralement utilisé pour des effets spéciaux est versé dans des moules afin de créer l'objet. D'après [160, 161], le module de Young des objets fabriqués varie entre 27 et $125kPa$ et leur coefficient de Poisson est égal à 0.43. Des tiges cylindriques fixent l'objet à la base grâce à des ouvertures comme le montre le modèle CAO illustré figure 4.17. Une plaque d'acrylique transparente découpée au laser avec des

ouvertures qui serviront aussi de référence a été utilisée comme base de support et afin d'observer les déformations. La même technique de découpe a aussi été utilisée pour créer un doigt compliant correspondant au modèle de la figure 4.2 et réalisé dans une plaque de $7mm$ de Delrin, correspondant à l'épaisseur de l'objet. Finalement, un appareil photo fixé au-dessus du montage est utilisé pour prendre des photos et enregistrer les saisies à plusieurs positions pour comparer les résultats expérimentaux avec les simulations numériques.

4.8.3 Comparaison avec des résultats expérimentaux

Des défauts de fabrication sont survenus lors de la découpe laser de la plaque de Delrin. L'épaisseur de la plaque était de $7mm$ et le laser était calibré avec le haut de la plaque, donc lorsque le laser pénètre dans le matériau, à cause de la chaleur et avec le faisceau pas parfaitement focalisé, cela à crée des écarts entre la géométrie voulue et la finale peuvent être observables en particulier vers le bas de la plaque. Ces imperfections en plus des approximations sur la géométrie des articulations compliantes ont été évaluées en mesurant différentes positions du doigt. Le centre de rotation des liaisons rotoïdes est directement corrigé à l'aide du modèle numérique et, de la même manière, les valeurs de raideur des ressorts sont estimées par une valeur moyenne d'épaisseur mesurée des deux côtés du mécanisme.

Des saisies d'objets déformables sont illustrées figures 4.18 and 4.19 en utilisant le montage décrit dans la section précédente. Des comparaisons entre les simulations numériques et les données de résultats expérimentaux sont données sur les figures 4.20, 4.21 et 4.22, sans objet, pour une prise enveloppante et lors d'une saisie par le bout des doigts respectivement. Il est à noter que sur ces figures, l'axe x donne la variation de l'angle actionné à partir de sa valeur à l'état initial au lieu du couple d'actionnement. Ce changement de variable implique qu'il n'est plus nécessaire de mesurer expérimentalement le couple d'actionnement qui peut être parfois difficile à obtenir de façon précise. Sur la figure 4.18, à partir de l'état initial (figure 4.18(a)) le doigt se rapproche de l'objet jusqu'à ce qu'un contact se fasse sur la phalange proximale (figure 4.18(b)). Ensuite, grâce à l'adaptation mécanique due au sous-actionnement du doigt la seconde phalange va aussi commencer à tourner jusqu'à ce qu'elle soit aussi en contact avec l'objet. Lors de cette phase, la première phalange est presque immobile : elle augmente sa pénétration dans l'objet. En effet, lors de la séquence de fermeture l'objet est nettement déformé par le doigt. Dans le cas de la saisie par le bout des doigts illustrée figure 4.19(b) la déformation était encore plus importante, car le contact se concentre sur une plus petite zone de l'objet.

Le petit écart entre les positions simulées et mesurées du doigt à la figure 4.20 peut être expliqué par les effets non modélisés comme le frottement sur le support les tolérances de

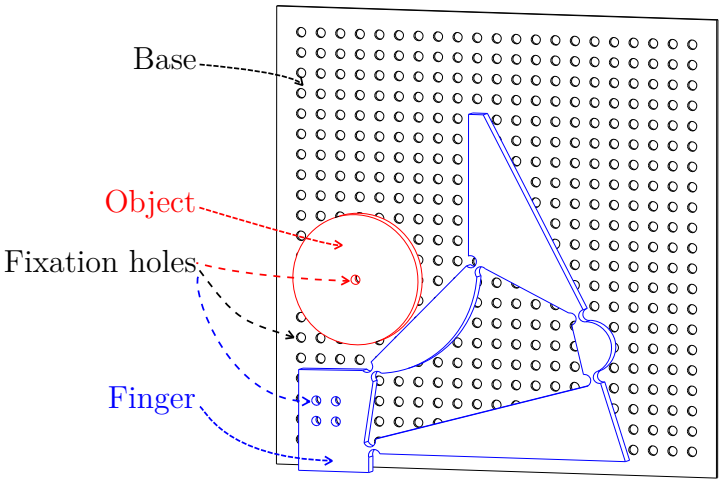


Figure 4.17 Banc de test expérimental

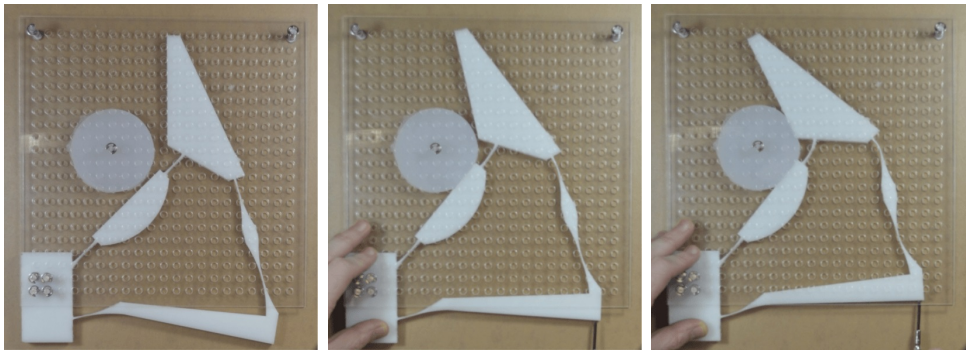
fabrication, en particulier l'épaisseur des articulations compliantes créées par découpe laser. Pourtant, l'allure des courbes sur les figures 4.21 et 4.22 sont toutefois proches. On observe à la figure 4.22 deux phases dans le mouvement du prototype par rapport à l'objet : d'abord le *preshaping*, autrement dit la phase d'approche du doigt, puis une phase de contact entre la phalange distale et l'objet. Ce qui explique les deux régions de pentes différentes que l'on peut observer. Ces deux étapes se retrouvent sur les résultats obtenus avec le modèle et par expérience, mais le passage de l'un à l'autre ne se fait pas au même angle. Cet écart pourrait s'expliquer par les imperfections du prototype venant de sa fabrication, ainsi que des erreurs de mesures. Ces dernières seront discutés dans le chapitre 6 pour d'autres prototypes.

Il est à noter que la transmission du doigt testé et sa géométrie ont été choisies de manière arbitraire. Le modèle est aussi adaptable à d'autres architectures, mais aucune expérience n'a été réalisée pour les valider. D'autres situations présentant des cas particuliers limites, seront à tester pour valider le modèle complètement. Pour cela il faudra donc aussi identifier en

Tableau 4.6 Paramètres de la comparaison présentée figure 4.20 (toutes les longueurs sont en mètre, les angles en dégrée, les raideurs en $N \cdot m$)

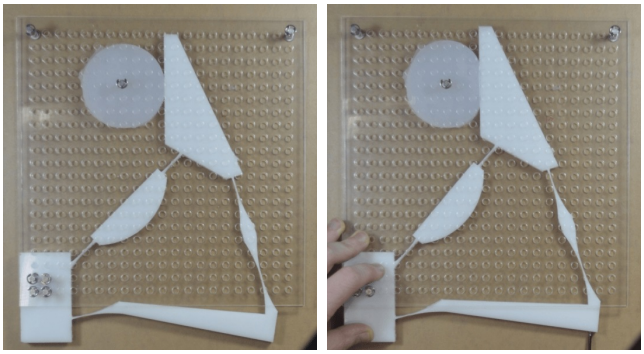
l_1	l_2	a	b	c	d_2
1	1	0.53	0.73	1.43	0.5

d_3	d_4	w	θ_{01}	θ_{02}	K_i^f
0.5	0	0.1	45	45	171



(a) (b) (c)

Figure 4.18 Saisies enveloppantes expérimentales d'un objet déformable



(a) (b)

Figure 4.19 Saisie enveloppante expérimentale d'un objet déformable avec contact sur la phalange distale

Tableau 4.7 Paramètres de l'objet simulé présentée figure 4.21 (toutes les longueurs sont en mètre, les angles in dégrée, les raideurs en $kN \cdot m$)

d_1	r	n_{div}	d_5	K_{ext}^{obj}	K_{int}^{obj}
0.48	0.32	16	0.48	33.5	2.04

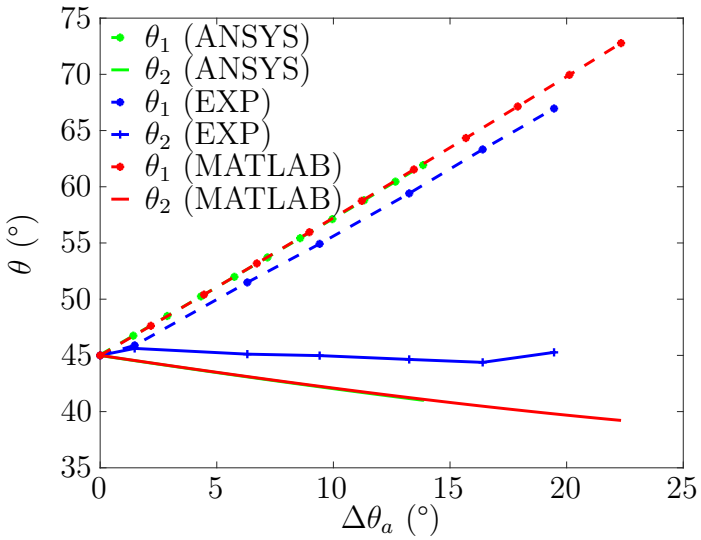


Figure 4.20 Comparaison entre les angles inter-phalanges simulées avec le modèle et lors de mesures expérimentales (sans objet)

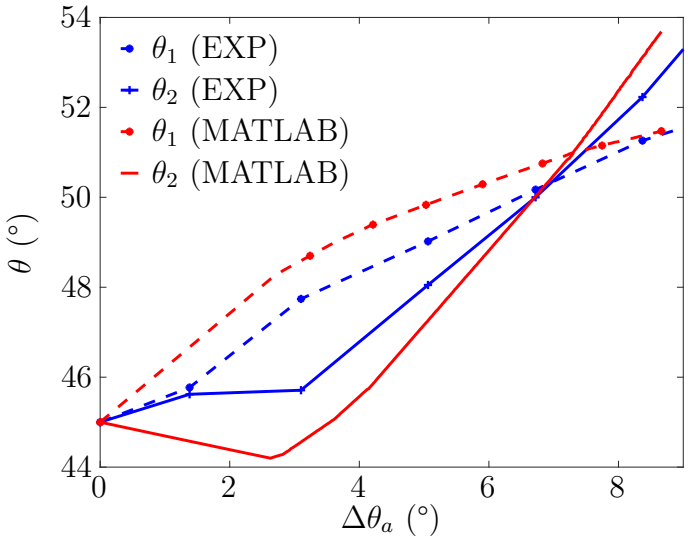


Figure 4.21 Comparaison entre les angles inter-phalanges simulées avec le modèle et lors de mesures expérimentales illustrées figure 4.18 (saisie enveloppante)

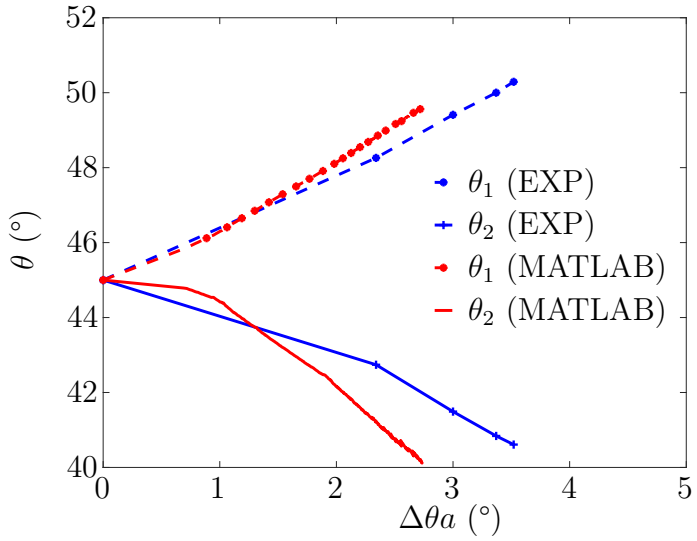


Figure 4.22 Comparaison entre les angles inter-phalanges simulées avec le modèle et lors de mesures expérimentales illustrées à la figure 4.19 (saisie de pincée)

Tableau 4.8 Paramètres de la simulation présentée figure 4.22 (toutes les longueurs sont en mètre, les angles en dégrée, les raideur en $kN \cdot m$)

d_1	r	n_{div}	d_5	K_{ext}^{obj}	K_{int}^{obj}
0.98	0.32	16	0.82	33.5	2.04

premier lieu ces cas particuliers afin de les éprouver et ainsi apporter les correctifs appropriés, si possible, pour obtenir au final un modèle efficace dans un maximum de situations possibles. Un cas limite pourrait par exemple être une situation où les raideurs de l'objet sont très grandes (ou très petites) en comparaison de celles de la pince.

CHAPITRE 5 ARTICLE 2 : DEFORMATION MODELING OF A COMPLIANT ROBOTIC FINGER GRASPING A SOFT OBJECT

Cet article a été publié dans le *Journal of Mechanisms and Robotics*, disponible en ligne depuis le 7 octobre 2020.

Nicolas Mouazé et Lionel Birglen

Abstract

In this paper, a model is shown to predict the simultaneous deformations occurring when compliant robotic fingers are grasping soft objects. This model aims at providing an accurate estimation of the penetration, internal forces, and deformed shapes of both these fingers and the objects. A particular emphasis is placed on the case when the finger is underactuated but the methodology discussed in this paper is general. Usually in the literature, underactuated fingers are modeled and designed considering their grasps of rigid object because of the complexity associated with deforming objects. This limitation severely hinders the usability of underactuated grippers and either restricts them to a narrow range of applications or requires extensive experimental testing. Furthermore, classical models of underactuated fingers in contact with objects are typically applicable with a maximum of one contact per phalanx only. The model proposed in this paper demonstrates a simple algorithm to compute a virtual subdivision of the phalanges which can be used to estimate the contact pressure arising when there are contacts at many locations simultaneously. This model also proposes a computationally efficient approximation of isotropic soft objects. Numerical simulations of the proposed model are compared here with dynamic simulations, finite element analyses, and experimental measurements which all shows its effectiveness and accuracy. Finally, the extension of the model to other types of underactuated fingers, standard grippers, and fully actuated robotic fingers as well as potential applications is discussed and illustrated.

5.1 Introduction

Underactuated mechanisms are defined as mechanical systems with less actuators than degrees of freedom (DOF) [37]. For an underactuated finger with p DOF, which is also typically the number of phalanges of this finger, a transmission mechanism with a maximum of $p-1$ motors (usually a single one) drive the system. This transmission mechanism distributing

the actuation can be implemented in practice in many different ways, most commonly using gears, tendons and pulleys, or linkages. In addition to this transmission, passive elements are also required to statically constrain the mechanism since the number of actuator(s) is by definition not sufficient for this. These passive elements are usually simple mechanical springs embedded in certain joints of the finger, making the latter intrinsically compliant and behaving similarly to fully actuated robotic fingers in impedance mode control. Switching to an underactuated design for a robotic finger is one the available solutions to simplify this finger by replacing one or several actuators by an often lighter, simpler, and cheaper transmission mechanism. Furthermore, the resulting closing sequence of the underactuated finger can be made remarkably similar to the natural motion of human fingers when they seize an object and lead to a mechanical shape adaptation for the robotic grasper using the fingers. This capability of underactuated fingers to adapt their postures to the shapes of objects they are in contact with, without any sensor or closed-loop control is very attractive in many practical scenarios [48, 126].

The intrinsic compliance of underactuated finger does not necessarily comes from the addition of a specific machine element. Indeed, the fingers can be compliant mechanisms namely a class of mechanisms where joints are approximated by the local deformation of the linkage when actuated. Typical internal force arising in the mechanism will produce a bending at locations where the parts are thinned out which, for at least a small range of motion, efficiently approximates a revolute joint combined to a spring. This design approach is also very commonly used to reduce weight, fabrication time, and price [81,82]. Noticing the small usable range of travel of this type of compliant joints, increasing the size of the flexural portion of the compliant mechanical part to a significantly larger size has been proposed and is often referred to as distributed compliance. In the latter case, the resulting thin beam of the mechanism cannot usually be modeled with a single revolute joint but a series of them. However, in [144], a pseudorigid model was introduced to approximate the motion of long compliant beams with a series of rigid links, revolute joints, and springs depending on its load and it was shown to produce excellent results despite its minimal number of parameters. Since compliance was already required with underactuated fingers, the combination of the latter with flexural beams was naturally proposed in the literature and led to the introduction of lightweight, cheap, and easy to manufacture adaptive fingers, see for instance [129, 130].

Contact models between a robotic tool, underactuated or not, and the objects it is intended to grasp are critical during the development phase of that tool [145,146]. However, optimizations of underactuated fingers as presently found in the literature have only mostly focused on the cases where the fingers are grasping rigid objects, e.g. [147], or based on experimental results only. This limitation happens to be true whether the grasp was of the enveloping type, also

known as a power grasp, or if a precision (pinch) grasp was used. However, as underactuated fingers and hands are now more and more popular in practical applications, their range of applications has been extended to tasks where they are used in conjunction with soft objects. In this paper, soft objects refer to structural pieces of materials with relatively low stiffnesses which will deform easily and significantly in response to an externally applied force. One can think of the food and produce industry which requires efficient, fast, and adaptable end-of-arm tooling to pick and process items without local bruising. Additionally, there also exist soft fingers often referring to a distinct class of underactuated fingers where the deformation of the latter happens along the continuum of the finger bodies and not, conversely to more classic underactuated fingers, at a discrete set of locations. These fingers are often constituted by a series of connected asymmetric inflatable rubber chambers. Their models are usually more complex to establish and mainly rely on finite element methods, piecewise constant curvature modeling or Cosserat approximation [148–150]. Literature dealing with soft robotic tool also frequently focus on the deformation of the gripper, particularly soft fingertips [151–153] or material selection and fabrication processes [18, 19] but the interaction of soft fingers with soft objects is not commonly modeled [154–157]. If these interactions are considered, they are typically limited to a case where the robotic finger is pinching soft objects at only two contacts points or small areas [90, 91, 158]. However, for securely manipulating an object, power grasps are preferred due to being force closed. Power grasps also distribute the contact forces over a larger area and therefore, decrease overall contact pressure, a desirable trait when the object is very fragile for example in surgical applications [136]. The *a priori* estimation of the resulting deformation during a grasp is therefore an essential criterion during the design of a gripper when soft objects are the preferred targets [134, 135].

Among the available models to evaluate the deformations of soft bodies, a trade-off between precision and time/memory has to be made to maximize efficiency [138,139]. A finite element analysis (FEA) is for instance a very accurate and well-known technique to calculate the deformations of soft bodies [92] but it is also generally prohibitively time consuming and computationally intensive. Because of this issue algorithms have been proposed to reduce the complexity of FEA, e.g. in [159], but this approach if efficient remains complex to implement and in that reference, relies heavily on offline simulations. A close relative to FEA but less accurate technique is the classical mass-spring system models (MSPM), or point-spring models (PSM) when masses are not relevant, to evaluate deformation. That latter technique is quicker and requires far less time and memory [118]. Using PSM, objects are discretized with points connected by springs to approximate their overall shapes and softness respectively, as illustrated in Fig. 5.1 with the approximation of a circle, a square and a triangle. As will be shown, with a reasonable number of points, significantly smaller than typical FEA, this

model can efficiently capture the deformations of soft objects. The real difficulty of this model is to obtain a reasonably accurate value of the stiffnesses of the springs. To this aim, several formulae have been proposed in the literature, e.g. in [118,119]. For example, in [118] the approximations of these stiffnesses are obtained from the initial geometry of the object and and its material properties. The resulting model favorably compares to FEA considering the small computation burden of PSM.

In [120], a preliminary version of this paper, the model of a compliant underactuated finger grasping a soft object was presented, similar to the one illustrated in Fig. 5.2. Despite the fact that the model is not as accurate as the results from a FEA, it was shown to be fast and its parameters are quickly adjustable making it adapted to optimization problems where numerous iterations are required. In the current paper, a novel extended model is presented to improve the one from [120]. This model is first presented and then, validated with multibody dynamic software simulations and finite element analyses. Then, experimental measurements with a test bench are used to confirm its accuracy. Finally, how the model can be adapted to other types of fingers will be shown and potential application examples discussed.

5.2 Modeling

In this section, a model is proposed to efficiently evaluate the deformations of soft objects grasped by an underactuated compliant finger. This model relies on a few hypotheses. First and foremost, a stable grasp is assumed to have been achieved, namely that the object is actually seized by the grasper and has not escaped the latter during the closing sequence. Secondly, both the object and the finger are assumed planar. Thirdly, the motions of the finger and of the object deformations are assumed slow: the kinetic energies of the pair are thus neglected compared to the potential energy stored in the springs of the model. Gravity is also neglected in comparison to the grasp forces. Thus, quasi-static conditions are assumed. Finally, as a first approximation the contact forces produced by the finger onto the object are assumed perpendicular to the surface of the finger. In practice, frictionless contacts often constitute a worst case scenario in order to achieve stable grasps with robotic fingers and are thus of interest to their designers since such a scenario is both simpler and conservative. Extension of the model when frictional surfaces need to be considered will be discussed in Section 5.4.

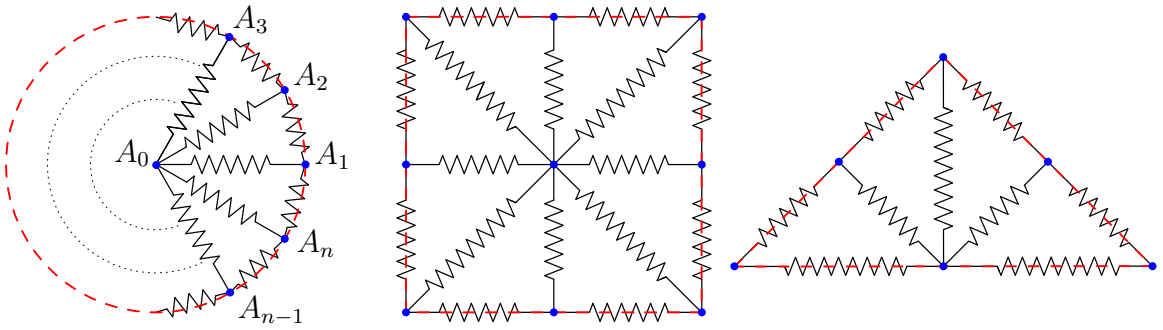


Figure 5.1 Discretization of common shapes for Van Gelder models

5.2.1 Object model with a point-spring system

Figure 5.1 illustrated three planar Van Gelder model of soft objects with simple geometries [118]. The leftmost model consists in a circle whose outer contour is discretized into n points and as many springs. An additional central point is connected to these contour points with another series of springs thereby dividing the object into triangles in accordance with Van Gelder's proposed methodology. The stiffnesses of the springs can also be approximated using the formulae listed in [118] and based on both the geometries of these triangles and material properties of the object. When a grasp force is applied, the equations relating the internal forces (i.e. the forces acting at the object's points) and displacements of each of the springs can be obtained analytically as presented in the next section. This technique can be generalized to any planar object which can be approximated by a radially symmetric shape with a fixed central point and n external points defining its outer shell.

Analysis of a single spring

Let us consider a single linear spring defined by its endpoints, as illustrated in Figure 5.3. Both the endpoints of this spring can move in the plane and, conversely to typical finite element algorithms, the displacement of these points is generally neither small nor in the direction of the spring. The variation of the length of the spring (shown in bold lines) is presented for a displacement from its initial position at rest (defined by points A_0 and B_0) to a position i (defined by points A_i and B_i). The length at rest of the spring is denoted l^0 while Δl is the variation of that length under the defined displacement. The angles α_0 and α_i are defined between the $\boldsymbol{x}$-axis of a fixed reference frame and the line of action of the springs respectively at rest and after deformation. Finally, $\boldsymbol{u}_A$ and $\boldsymbol{u}_B$ are the displacement vectors of points A_0 and B_0 respectively, from rest to position i. All these parameters are illustrated in Fig. 5.3. The variation of the spring length can be expressed depending on the

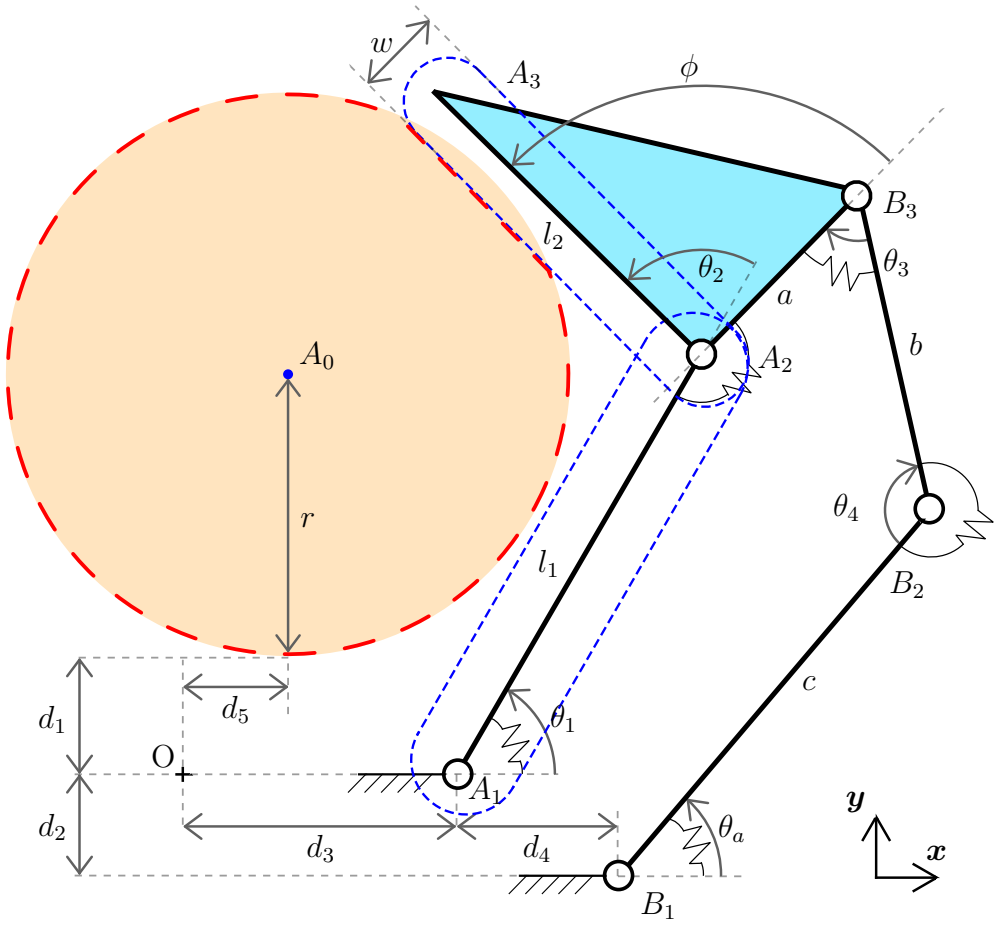


Figure 5.2 Model of a circular object (dashed red line) deformed by a precision graps with a compliant underactuated finger using a five-bar transmission mechanism

position change of its endpoints as:

$$\Delta l = \boldsymbol{a}^T \boldsymbol{u} + l^0 b \tag{5.1}$$

with :

$$\boldsymbol{a} = \begin{bmatrix} \cos(\alpha_i) & \sin(\alpha_i) & -\cos(\alpha_i) & -\sin(\alpha_i) \end{bmatrix}^T, \tag{5.2}$$

$$b = 1 - \cos(\alpha_i - \alpha_0), \tag{5.3}$$

$$\boldsymbol{u} = \begin{bmatrix} \boldsymbol{u}_A^T & \boldsymbol{u}_B^T \end{bmatrix}^T. \tag{5.4}$$

From Eqn. (5.1) and using Hooke's law, the vector $\boldsymbol{f} = \begin{bmatrix} f_x^A & f_y^A & f_x^B & f_y^B \end{bmatrix}^T$ of all the components of the forces at the endpoints of the spring (the superscripts A or B indicate which one) can be projected along the $\boldsymbol{x}$ and $\boldsymbol{y}$ axes of the reference frame (again using subscripts

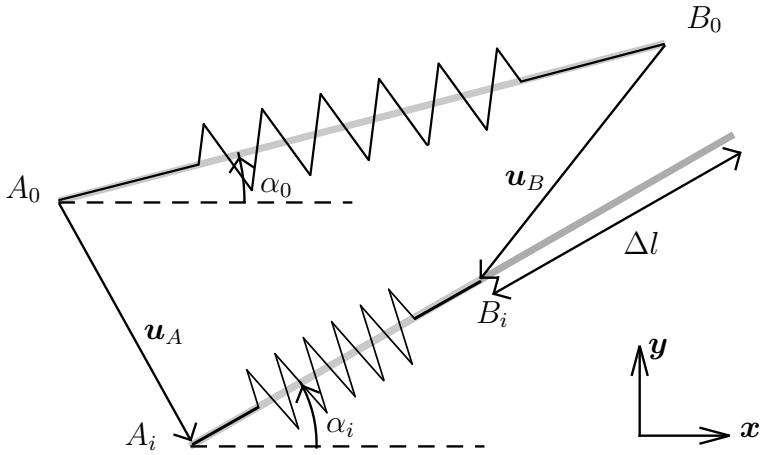


Figure 5.3 Displacement of a spring from position 0 to i

x or y to indicate which axis), namely one has:

$$\boldsymbol{f} = -k_{AB}(\boldsymbol{a}\boldsymbol{a}^T\boldsymbol{u} + \boldsymbol{a}l^0 b) \tag{5.5}$$

where k_{AB} is the spring stiffness. One should notice that the determinant of $\boldsymbol{a}\boldsymbol{a}^T$ is always equal to zero. Therefore, as this matrix is always singular, if one is looking for the displacement $\boldsymbol{u}$ created by a known applied force $\boldsymbol{f}$, Eqn. (5.5) cannot be solved directly. To solve this issue, the coordinates of one of the two points must be fixed for instance.

Then, matrix $\boldsymbol{S}_{AB}$ can be defined as:

$$\boldsymbol{S}_{AB} = -k_{AB}\begin{bmatrix} \cos(\alpha_i)^2 & \cos(\alpha_i)\sin(\alpha_i) \\ \cos(\alpha_i)\sin(\alpha_i) & \sin(\alpha_i)^2 \end{bmatrix}. \tag{5.6}$$

and comparing with the previous equation, one can write:

$$-k_{AB}\boldsymbol{a}^T\boldsymbol{a} = \begin{bmatrix} \boldsymbol{S}_{AB} & -\boldsymbol{S}_{AB} \\ -\boldsymbol{S}_{AB} & \boldsymbol{S}_{AB} \end{bmatrix}. \tag{5.7}$$

The previous term defines a stiffness matrix and its expression is commonly found in the literature about linear finite element algorithms but it should be noted that it neglects the second part of Eqn. (5.5) and in this work this second part cannot be ignored since large deformations are anticipated with soft objects.

Generalization to a n-points contour

The methodology used in the previous section to relate the force and displacement of a single spring can be generalized to the springs used in the Van Gelder model of soft objects as shown in Fig. 5.1. In the subsequent parts of this paper, focus will be placed on the circular (leftmost in Fig. 5.1) model of an object but it could be extended to the other geometries. This circular object is discretized with n points defining the outer contour and one fixed point inside the object body, usually for symmetry purpose its center. This central point will be indicated with a 0 subscript and the contour points will use subscripts going from 1 to n. The resulting force applied on the i^{th} contour points connected by springs to the central and two adjacent contours points (noted $i+1$ and $i-1$) is related to the displacements of the points by:

$$\begin{aligned}\begin{bmatrix} f_x^i \\ f_y^i \end{bmatrix} = &\begin{bmatrix} -\boldsymbol{S}_{i,i-1} \\ \boldsymbol{S}_{i,0} + \boldsymbol{S}_{i,i-1} + \boldsymbol{S}_{i,i+1} \\ \boldsymbol{S}_{i,i+1} \end{bmatrix}^T \begin{bmatrix} \boldsymbol{u}_{i-1} \\ \boldsymbol{u}_i \\ \boldsymbol{u}_{i+1} \end{bmatrix} \\ &+ \boldsymbol{P}_i \begin{bmatrix} k_{i,0} b_{i,0} l_{i,0}^0 \\ k_{i,i-1} b_{i,i-1} l_{i,i-1}^0 \\ k_{i,i+1} b_{i,i+1} l_{i,i+1}^0 \end{bmatrix}\end{aligned} \tag{5.8}$$

where $\boldsymbol{P}_i$ is the matrix:

$$\boldsymbol{P}_i = \begin{bmatrix} \cos\alpha_{i,0} & \cos\alpha_{i,i-1} & \cos\alpha_{i,i+1} \\ \sin\alpha_{i,0} & \sin\alpha_{i,i-1} & \sin\alpha_{i,i+1} \end{bmatrix}. \tag{5.9}$$

Thus, combining Eqn. (5.8) for every contour points the relationship between $\boldsymbol{f}_o$ the vector of internal forces and $\boldsymbol{u}$ the vector of displacements of all the object points can be obtained, namely:

$$\boldsymbol{f}_o = \boldsymbol{S}\boldsymbol{u} - \boldsymbol{P}\boldsymbol{k} \tag{5.10}$$

with $\boldsymbol{u}$, $\boldsymbol{f}_o$ and $\boldsymbol{k}$ are vectors defined as:

$$\boldsymbol{u} = \begin{bmatrix} \boldsymbol{u}_1^T & \dots & \boldsymbol{u}_i^T & \dots & \boldsymbol{u}_n^T \end{bmatrix}^T \tag{5.11}$$

$$\boldsymbol{f}_o = \begin{bmatrix} f_x^1 & f_y^1 & \dots & f_x^i & f_y^i & \dots & f_x^n & f_y^n \end{bmatrix}^T \tag{5.12}$$

$$\boldsymbol{k} = \begin{bmatrix} k_{1,0}b_{1,0}l^0_{1,0} \\ k_{1,n}b_{1,n}l^0_{1,n} \\ k_{1,2}b_{1,2}l^0_{1,2} \\ \vdots \\ k_{i,0}b_{i,0}l^0_{i,0} \\ k_{i,i-1}b_{i,i-1}l^0_{i,i-1} \\ k_{i,i+1}b_{i,i+1}l^0_{i,i+1} \\ \vdots \\ \\ k_{n,0}b_{n,0}l^0_{n,0} \\ k_{n,n-1}b_{n,n-1}l^0_{n,n-1} \\ k_{n,1}b_{n,1}l^0_{n,1} \end{bmatrix} \tag{5.13}$$

In the previous equation, $b_{i,j}$ and $l^0_{i,j}$ are the parameters, defined similarly to the previous section, corresponding to the spring whose endpoints are A_i and A_j, see the righthand side of Eqn. (5.5). The matrices $\boldsymbol{S}$ and $\boldsymbol{P}$ in Eqn. (5.10) are defined by:

$$\boldsymbol{S} = \begin{bmatrix} \boldsymbol{Q}_1 & \dots & \boldsymbol{Q}_n \end{bmatrix}^T \tag{5.14}$$

with:

$$\boldsymbol{Q}_i = \begin{bmatrix} \boldsymbol{0} & \dots & \boldsymbol{0} & -\boldsymbol{S}_{i,i-1} & \boldsymbol{S}_{i,0} + \boldsymbol{S}_{i,i-1} + \boldsymbol{S}_{i,i+1} & -\boldsymbol{S}_{i,i+1} & \boldsymbol{0} & \dots & \boldsymbol{0} \end{bmatrix}^T \tag{5.15}$$

using the definition of Eqn. (5.6) and also with:

$$\boldsymbol{P} = \begin{bmatrix} \boldsymbol{P}_1 & \boldsymbol{0} & \dots & \dots & \boldsymbol{0} \\ \boldsymbol{0} & \ddots & \ddots & & \vdots \\ \vdots & \ddots & \boldsymbol{P}_i & \ddots & \vdots \\ \vdots & & \ddots & \ddots & \boldsymbol{0} \\ \boldsymbol{0} & \dots & \dots & \boldsymbol{0} & \boldsymbol{P}_n \end{bmatrix}. \tag{5.16}$$

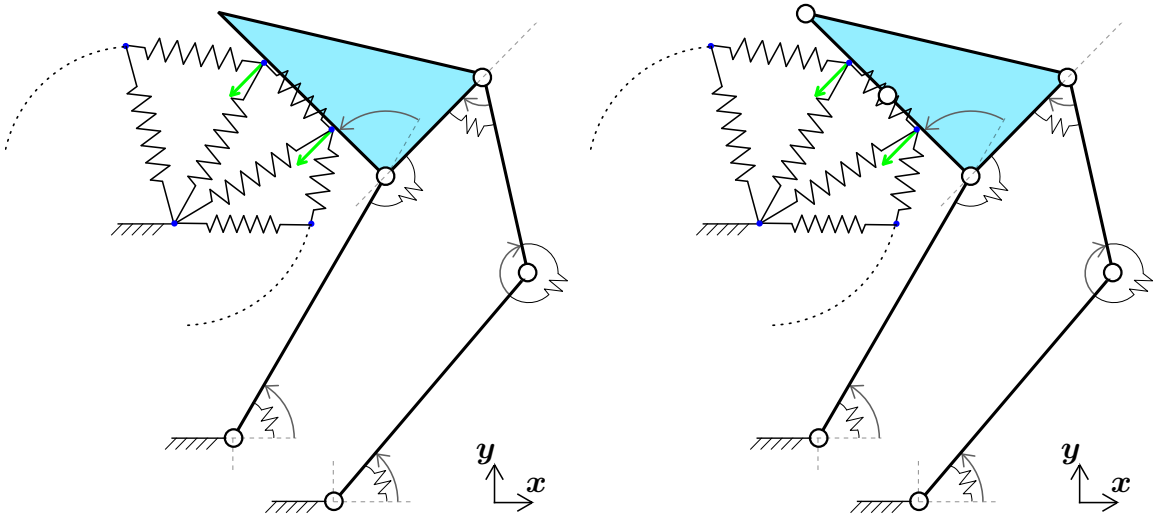
From Eqn. (5.10), the displacements of all the contour points (nodes) can be calculated from the forces applied at these points. In the next section, the relationship between the finger actuation torque and these contacts forces will be detailed in order to obtain the input for this equation and complete the model.

5.2.2 Compliant finger model

Figure 5.2 showed a linkage-driven underactuated finger grasping a circular object with a linear contact along the distal phalanx. This finger will be used as a primary example but Section 5.4 will show how to modify the model to include other type of fingers. As the finger of Fig. 5.2 is assumed to have small living hinges at all joints, a classic approximation is used here where each of these hinges is modeled by a revolute joint combined with a constant stiffness rotational spring [81,82]. In this illustrated case, the magnitude of the contact forces cannot be evaluated with the methodology presented in [37] because the latter assumed only a single point of contact at each phalanx. This hypothesis is clearly not valid when the contact is of non-negligible size as one expects when grasping a soft object or equivalently, when there are multiple simultaneous contact points with a phalanx. This issue can be solved by virtually dividing the phalanx of the fingers thereby increasing the number of independent equations obtained from the kinetostatic model (see [37]) to a number matching the number of variables.

This approach is illustrated in Figs. 5.4(a)-5.4(b) where the distal phalanx of the finger is in contact with the object at two points. That phalanx is then subdivided (see Fig. 5.4(b)) to virtually increase the number of DOF of the finger and simultaneously, model its structural compliance. Each contact is then attributed to a single virtual phalanx and the methodology presented in [37] can be adapted to yield an analytical expression of the two contact force magnitudes. In order to separate one phalanx with c contacts, $c-1$ revolute joints are thus created between these contact points. In this paper, the joints are arbitrarily chosen equidistant to two consecutive contacts points. For the distal phalanx, if there is more than one contact, an additional joint is required for the number of contact to match the number of DOF of the mechanism and this joint is added at the fingertip. This process of virtually dividing the phalanges into as many elements as there are contact points can be repeated as the grasp progresses and penetration increases into the object. Each supplementary joint will create an additional DOF and is combined with a spring. Commonly, these stiffnesses are greater than these of the living hinges but the methodology is general and could be applied to any stiffness ratios.

Fig. 5.5 shows the considered two-phalanx underactuated finger with m and q virtual divisions of the proximal and distal phalanges. These numbers are also equal to the number of contacts on the corresponding phalanx. If there is no contact, there is no division and the phalanges stay whole. In the following, the A points shown in Fig. 5.5 have a subscript going from one to p, p being the number of virtual phalanges ($p = m + 1$ if the distal phalanx has less than one contact or else $p = m + q + 1$) in order to simplify notations. Around each joint



(a) (b)

Figure 5.4 Virtual division of a finger: from (a) two contacts on the distal phalanx to (b) one contact per virtual phalanges

at a point A_i an angle θ_i is introduced corresponding to the relative angle between the two connected virtual phalanges as well as a rotational spring with a stiffness indicated by k_{A_i}. For the discretization of the object, n_c and n_{nc} are the numbers of contour points in contact and not in contact with the finger respectively ($n = n_c + n_{nc}$). Similarly, p_c and p_{nc} are the numbers of virtual phalanges in contact and not in contact with the object respectively ($p = p_c + p_{nc}$). Using the proposed division technique, p_c is obviously equal to n_c. From [37], the relationship between interphalanx torques and input (actuation and spring) torques is obtained, i.e.:

$$\boldsymbol{f}_p^T \boldsymbol{J} = \boldsymbol{t}^T \boldsymbol{T}, \tag{5.17}$$

where $\boldsymbol{f}_p \in \mathbb{R}^{p\times 1}$ is the vector of the contact force magnitdudes, i.e.:

$$\boldsymbol{f_p} = \begin{bmatrix} f_1 \\ \vdots \\ f_p \end{bmatrix} \tag{5.18}$$

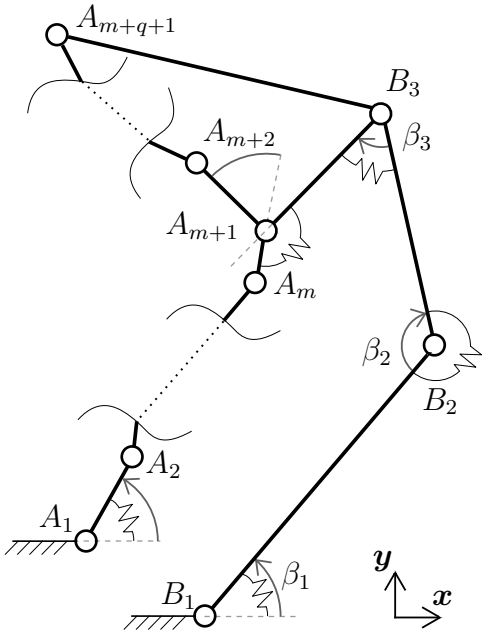


Figure 5.5 Virtual phalanx division

and $\boldsymbol{J}$ ($\in \mathbb{R}^{p\times p}$) the Jacobian matrix of the finger, namely:

$$\boldsymbol{J} = \begin{bmatrix} \boldsymbol{r}_{11}^T\boldsymbol{x}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T\boldsymbol{x}_2 & \boldsymbol{r}_{22}^T\boldsymbol{x}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1p}^T\boldsymbol{x}_p & \boldsymbol{r}_{2p}^T\boldsymbol{x}_p & \boldsymbol{r}_{3p}^T\boldsymbol{x}_p & \dots & \boldsymbol{r}_{pp}^T\boldsymbol{x}_p \end{bmatrix} \tag{5.19}$$

with $\boldsymbol{r}_{ki}$ the vector from point A_k to the i^{th} contact point and $\boldsymbol{x}_i$ the unit vector along the i^{th} phalanx in the direction of the distal end. The vector of the input torques $\boldsymbol{t}$ ($\in \mathbb{R}^{4+p\times 1}$) is defined as:

$$\boldsymbol{t} = \begin{bmatrix} T_a \\ T_{B_1} = -k_{B_1}\Delta\beta_1 \\ T_{B_2} = -k_{B_2}\Delta\beta_2 \\ T_{B_3} = -k_{B_3}\Delta\beta_3 \\ T_{A_1} = -k_{A_1}\Delta\theta_1 \\ \dots \\ T_{A_p} = -k_{A_p}\Delta\theta_p \end{bmatrix} \tag{5.20}$$

with T_a the finger actuation torque and β_1, β_2 and β_3 the transmission mechanism angles associated with points B_1, B_2 and B_3 respectively. The torque at point P due to a spring with a stiffness k_P can be expressed by $T_P^f = -k_P^f\Delta\alpha$, while $\Delta\alpha$ is the variation of the

angles α between the initial position and the one obtained for the input torque T_a, also identified by the actuation angle β_1. Finally, the matrix $\boldsymbol{T}$ ($\in \mathbb{R}^{4+p\times p}$) is referred to as the Transmission matrix and characterizes the transmission mechanism. It is obtained by expressing the following equation:

$$\begin{bmatrix} \dot{\beta}_1 = \dot{\beta}_a \\ \dot{\beta}_1 \\ \dot{\beta}_2 \\ \dot{\beta}_3 \\ \dot{\theta}_1 \\ \vdots \\ \dot{\theta}_p \end{bmatrix} = \boldsymbol{T} \begin{bmatrix} \dot{\theta}_1 \\ \vdots \\ \dot{\theta}_p \end{bmatrix} \text{ with } \boldsymbol{T} = \begin{bmatrix} a_1 & a_2 & \dots & a_p \\ a_1 & a_2 & \dots & a_p \\ b_1 & c_2 & \dots & b_p \\ c_1 & b_2 & \dots & c_p \\ 1 & 0 & \dots & 0 \\ 0 & 1 & \dots & 0 \\ \vdots & \ddots & \ddots & \\ 0 & \dots & 0 & 1 \end{bmatrix}. \tag{5.21}$$

To obtain the coefficients of the Transmission matrix, one needs to find the relationship between the derivative of the angles in the transmission mechanism and the derivative of the interphalanx angles which can be conveniently obtained by expressing the loop closure equation in terms of screw coordinates, namely:

$$\sum_{i-1}^{p} \dot{\theta} \boldsymbol{\xi}_{0,A_i} = \sum_{j=1}^{3} \dot{\beta} \boldsymbol{\xi}_{0,B_i} \tag{5.22}$$

where $\boldsymbol{\xi}_{0,C}$ stands for the zero-pitch twist of a revolute joint along the $\boldsymbol{z}$-axis and passing through point C. Using zero-pitch wrenches $\boldsymbol{\zeta}_{0,\boldsymbol{s}_{23}}$, $\boldsymbol{\zeta}_{0,\boldsymbol{s}_{13}}$ and $\boldsymbol{\zeta}_{0,\boldsymbol{s}_{12}}$ (where $\boldsymbol{s}_{ij}$ is the vector from B_i to B_j, i.e. the direction of the wrench), the elements from matrix $\boldsymbol{T}$ can be obtained (for $i = 1 \dots p$) as:

$$a_i = \frac{\boldsymbol{\zeta}_{0,\boldsymbol{s}_{23}} \circ \boldsymbol{\xi}_{0,A_i}}{\boldsymbol{\zeta}_{0,\boldsymbol{s}_{23}} \circ \boldsymbol{\xi}_{0,B_1}}, \tag{5.23}$$

$$b_i = \frac{\boldsymbol{\zeta}_{0,\boldsymbol{s}_{13}} \circ \boldsymbol{\xi}_{0,A_i}}{\boldsymbol{\zeta}_{0,\boldsymbol{s}_{13}} \circ \boldsymbol{\xi}_{0,B_2}}, \tag{5.24}$$

$$c_i = \frac{\boldsymbol{\zeta}_{0,\boldsymbol{s}_{12}} \circ \boldsymbol{\xi}_{0,A_i}}{\boldsymbol{\zeta}_{0,\boldsymbol{s}_{12}} \circ \boldsymbol{\xi}_{0,B_3}} \tag{5.25}$$

where $\circ$ stands for the reciprocal product of screws.

Eqn. (5.17) provides a relationship between the input torques and contact forces created by the finger whereas Eqn. (5.10) gives one between the forces applied on contour points of the PSM and displacements of these contour points. These two equations are clearly related as these forces are the same (except for the sign) but one needs to express them in the same

frame. To this aim, two projection matrices $\boldsymbol{D}_x$ and $\boldsymbol{D}_y$ can be defined as function of the position of the contact points on the phalanges and the orientation of the latter:

$$\boldsymbol{f}_p = \boldsymbol{D}_x \boldsymbol{f}_o, \tag{5.26}$$

$$\boldsymbol{f}_p = \boldsymbol{D}_y \boldsymbol{f}_o. \tag{5.27}$$

With d_{ij}^x and d_{ij}^y (with $i = 1 \dots p$ and $j = 1 \dots 2n$) the elements of matrix $\boldsymbol{D}_x$ and $\boldsymbol{D}_y$ respectively, one has:

$$d_{ij}^x = \begin{cases} \frac{-1}{\cos(\sum_{q=1}^{i} \theta_q) + \frac{\pi}{2}} & \text{if j is odd and if the } i^{th} \text{ phalanx} \\ & \text{touches the } (j+1)/2 \text{ point,} \\ 0 & \text{otherwise.} \end{cases} \tag{5.28}$$

$$d_{ij}^y = \begin{cases} \frac{-1}{\sin(\sum_{q=1}^{i} \theta_q) + \frac{\pi}{2}} & \text{if j is even and if the } i^{th} \text{ phalanx} \\ & \text{touches the } j/2 \text{ point} \\ 0 & \text{otherwise.} \end{cases} \tag{5.29}$$

When $\cos(\sum_{q=1}^{i} \theta_q) + \frac{\pi}{2}$ or $\sin(\sum_{q=1}^{i} \theta_q) + \frac{\pi}{2}$ are equal to zero, a representation singularity, the system of reference axes can be tilted by an arbitrary angle in order to calculate $\boldsymbol{D}_x$ and $\boldsymbol{D}_y$. These two projection matrices relate $\boldsymbol{f}_p$ the vector of contact force and $\boldsymbol{f}_o$ the vector of forces in the object. Finally, combining Eqn. (5.26) and (5.27) with Eqn. (5.17) and (5.10), one obtains:

$$(\boldsymbol{S}\boldsymbol{u} - \boldsymbol{P}\boldsymbol{k})^T \boldsymbol{D}_x^T \boldsymbol{J} - \boldsymbol{t}^T \boldsymbol{T} = \boldsymbol{0}^T \tag{5.30}$$

$$(\boldsymbol{S}\boldsymbol{u} - \boldsymbol{P}\boldsymbol{k})^T \boldsymbol{D}_y^T \boldsymbol{J} - \boldsymbol{t}^T \boldsymbol{T} = \boldsymbol{0}^T. \tag{5.31}$$

The system of equations obtained from Eqn. (5.30) and (5.31) gives $2p_c$ equations for the phalanges in contact with the object and $2p_{nc}$ additional equations the other phalanges. In the latter case the same equation will appear in the system twice and therefore, only p_{nc} equations are independent. In addition to Eqn.(5.30) and (5.31), the sum of all the forces acting on the object points not in contact with the finger must be equal to zero as well as the projections of these forces in the reference frame. Thus, the equations associated with the overall static equilibrium of both the finger and the object can be obtained by multiplying

$\boldsymbol{f}_o$ by a diagonal matrix $\boldsymbol{D}_z$ with diagonal elements defined by d_i^z (with $i = 1...2n$) such as:

$$d_i^z = \begin{cases} 1 & \text{, if i is odd and point } \frac{i+1}{2} \\ & \text{has no contact with the finger,} \\ 1 & \text{, if i is even and point } \frac{i}{2} \\ & \text{has no contact with the finger,} \\ 0 & \text{, otherwise.} \end{cases} \tag{5.32}$$

and finally

$$\mathbf{0} = \boldsymbol{D}_z(\boldsymbol{S}\boldsymbol{u} - \boldsymbol{P}\boldsymbol{k}). \tag{5.33}$$

The diagonal coefficients of $\boldsymbol{D}_z$ are equal to zero if multiplied by a non-zero force (i.e. forces created by the finger on the object) and equal to one if multiplied by a zero force, e.g. points of the object not in contact with the finger. Finally, Eqn. (5.33), (5.33) and (5.31) yield $2n_{nc} + 2p_c + p_{nc}$ independent equations for $2n_{nc} + p + p_c$ unknowns. Therefore, the system can be solved and a unique solution found. Combining Eqn. (5.30), (5.31) and (5.33), the motion of the finger and the deformations of the point-spring object can be simulated. Incrementally increasing the input torque, starting from the a rest configuration where this input torque is zero, numerically minimizing the difference between the left- and the righthand side of Eqn (5.30), (5.31) and (5.33) allows to obtain the angles of the finger and thus, the coordinates of all the points defining the grasp. In the numerical examples of this paper, this minimization is implemented in Matlab with a Nelder-Mead simplex algorithm.

5.3 Validation

5.3.1 Software simulations

Fig. 5.6 illustrates the comparisons between two grasp simulations performed by a commercial multi-body dynamic simulation software (MSc ADAMS) and with the numerical model proposed in this paper. The angles of the finger phalanges are computed and plotted using both methods as they completely define the grasp. All the parameters of these simulations are listed in Tab. 5.1 with reference to Fig. 5.2. One should note that there is an additional parameters, namely r_{sphere}, which is the size of sphere used as contour points of the object in MSc ADAMS. If these spheres have a radius that is too small (it should be zero in the spring-point theoretical model), MSc Adams often fails to detect contact occurrences leading to stability issues. Thus, our numerical simulations were also slightly changed to take this radius into account but it should be noted that this is only due to numerical issues in MSc Adams. The weight of the object is not considered in the simulations as gravity is assumed

perpendicular to the plane containing both the finger and the object. Furthermore, a suitably refined discretization of the object has been chosen in order to avoid excessively large displacements of a single node. Phalanx lengths are typically designed for their surfaces to contact objects along a significant area which would conflict with a model using only a single point of contact. Another benefit of a relatively fine discretization is that configurations where one spring of the model would overlap another are avoided. Fig. 5.6(a) illustrates a simulation of the deformation of an object made of a very soft material ($E = 10^3 Pa$) whereas the object simulated in the second example in Fig. 5.6(b) is a hundred times stiffer ($E = 10^5 Pa$). As can be seen, the differences between our model and MSc Adams are tiny which validates the mathematical formulation of the previous section. The proposed model can then be compared with finite element simulations as illustrated in Fig. 5.7. This time, the discretized contour of our model is made of points and not spheres. Again, the detailed parameters of the proposed model corresponding to the FEA are listed in Tab. 5.2. It is also quite clear from the results that the model proposed in this paper can clearly approximate the deformation of the object and the finger to an excellent degree for a wide range of motion and material properties. Indeed, the latter ranges from a Young Modulus of 10^5 to 10^8 Pa. The numerical model shortens the computing times of the deformations simulations. Finite element results in Fig 5.7(a) required 234 minutes of computing time, whereas the corresponding numerical model was finished in three minutes using the same computer. Average computing times are therefore dramatically reduced with the simple and adaptable model provided in this paper.

5.3.2 Experiments

In order to have the final word on the validity of the proposed model, experiments were also conducted. A simple test setup was designed to estimate the deformation during the grasp of compliant underactuated finger and a soft cylinder. A commercial silicone compound (Smooth-On, Ecoflex^{TM} 00-30) commonly found for special effect purposes was used to build the object by pouring the compound into molds. The Young's modulus of the resulting object ranges from 27 to $125kPa$ according to the literature [160, 161] and its Poisson ratio is 0.43. The compliant underactuated finger was 3D printed in polylactic acid plastic (PLA) and $0.8mm$ thick Delrin sheets were inserted in these 3D printed parts to create compliant living hinges. Cylinder rods attach the object and the finger to the ground through holes at adjustable locations. The CAD model of the setup is illustrated in Fig. 5.8. A transparent acrylic plate was used as the supporting ground in which a laser cut grid of holes was cut for registration. Finally, a camera fixed above the system was used to record the grasps at several positions to compare the experimental results with the simulations. The tip of a force gauge

Table 5.1 Parameters of the simulation presented in Fig. 5.6 (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$)

l_1	l_2	a	b	c	d_2	d_3
1	1	0.5	0.7654	1.5811	0.5	0.25
d_4	w	θ_{01}	θ_{02}	d_1	r	n_{div}
0	0.04	45	46	0.15	0.75	20
r_{sphere}	k_i^f		k_{ext}^{obj}		k_{int}^{obj}	
0.01	174.714		$1.7327 \cdot E$		$0.2902 \cdot E$	

Table 5.2 Parameters of the simulation presented in Fig. 5.7 (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$)

l_1	l_2	a	b	c	d_2	d_3
1	1	0.757	0.412	1.581	0.5	0.5
d_4	w	θ_{01}	θ_{02}	d_1	r	n_{div}
0	0.1	45	45	0.15	0.75	40
d_5	k_i^f		k_{ext}^{obj}		k_{int}^{obj}	
0.25	6314.2		$0.0586 \cdot E$		$1.403 \cdot E$	

(Shimpo, FGV-100XY) was used to push the input lower link of the finger (actuation) and measures are taken regularly for different values of input force applied. Results are compiled by merging the recorded forces from the gauge with the corresponding positions of the finger.

Actual power and precision grasps of the molded soft cylinder are illustrated in Figs. 5.9 and 5.10. In Fig. 5.9, from the initial state (Fig. 5.9(a)) the finger gets closer to the object until a contact with its proximal phalanx is established (Fig. 5.9(b)). Then, thanks to the mechanical adaptation provided by the transmission mechanism, the second phalanx starts to rotate until it comes in contact with the object. During this phase, the penetration of the proximal phalanx in the soft object increases as the actuation torque gets greater. In the fingertip grasp pictured in Fig. 5.10(c), deformation was noticeably larger due to concentration of the contact to a smaller area of the object.

The comparisons of the deformations obtained from the PSM and the resulting experimental data are shown in Fig. 5.11 (parameters are presented in Tab. 5.3 and 5.4). These three

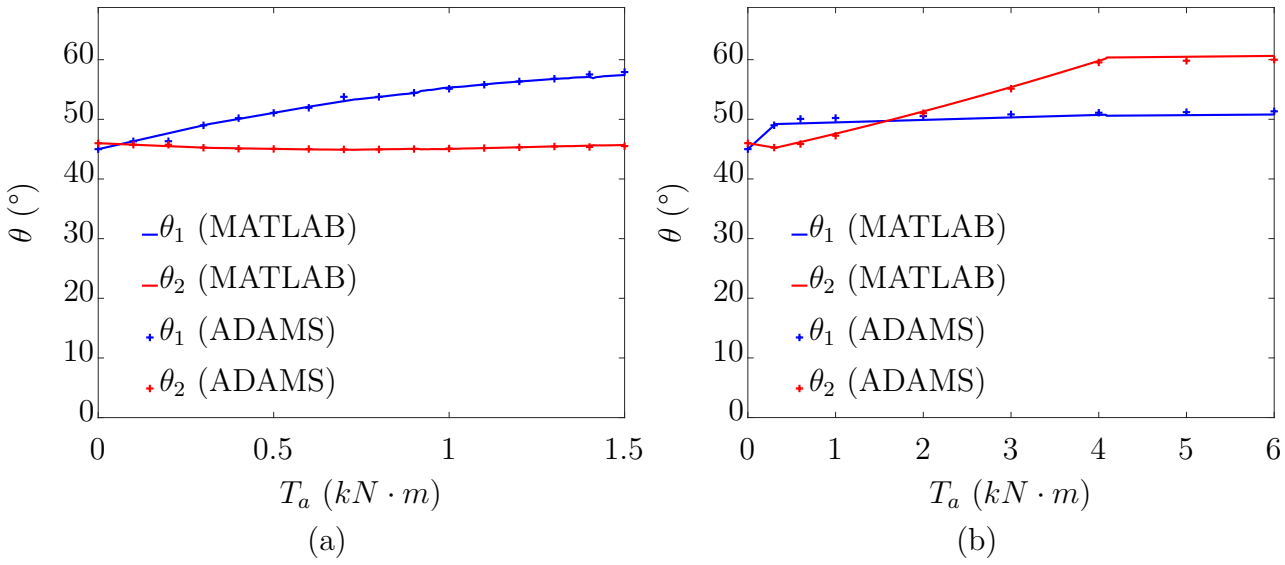


Figure 5.6 Variations of relative angles in the finger simulated with the numerical model and a commercial software ((a) $E_{obj} = 10^3 Pa$, (b) $E_{obj} = 10^5 Pa$)

figures correspond to the following cases: without any object (before first contact), during a power grasp, and during a precision grasp respectively. The small discrepancy between the simulated and the measured positions of the finger in Fig. 5.11(a) can be explained by unmodeled effects such as friction and fabrication tolerances, especially in the thickness of the compliant living hinges created by laser cutting. However, the curves in Figs. 5.11(b) and 5.11(c) are noticeably close despite these imperfections. In both figures the phase of the grasp before contact with the object is distinguishable at the beginning of the curves where variations of the angles are large. Then, during the power grasp the first phalanx is constrained by the object but can still increase its penetration of the latter, thus the curve of θ_1 has a smaller slope. In contrast, θ_2 is increasing faster as the second phalanx moves to envelop the object. During the precision grasp the distal phalanx is constrained by the object deformation while the proximal is also relatively not moving much due to the kinematic constrain imposed by the linkage.

5.4 Discussions

The proposed model is not restricted to the specific design of compliant underactuated finger based on a five-bar mechanism presented in the previous section and can be extended to a wide range of architectures. Whether the finger while still being underactuated relies on another transmission mechanisms (e.g. tendon/pulley as illustrated in Fig. 5.12(a)) or

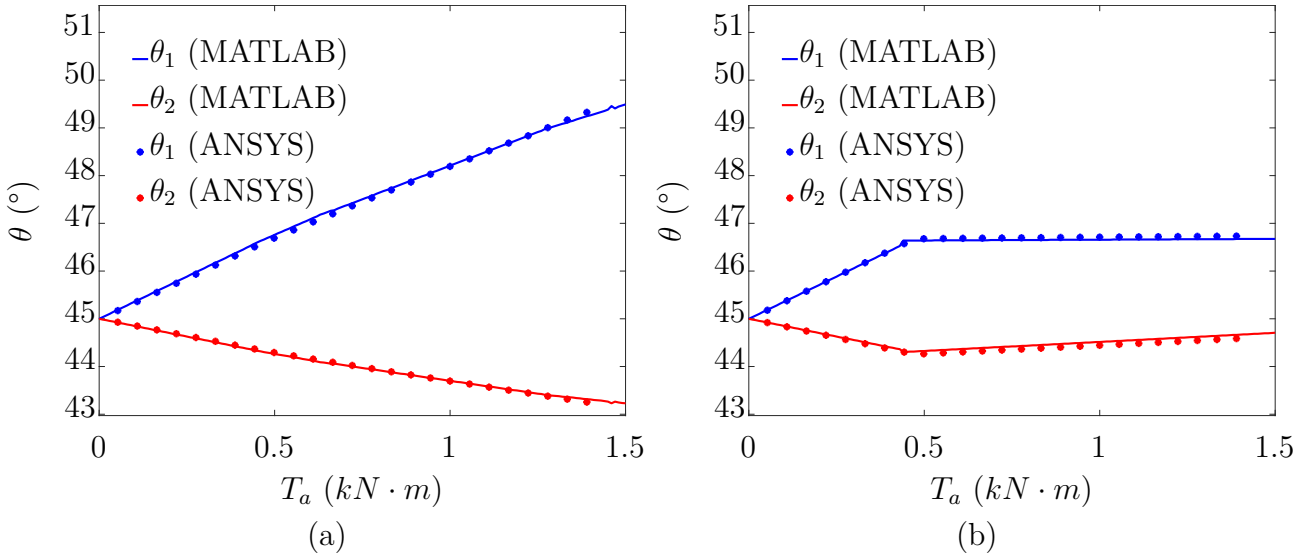


Figure 5.7 Variations of relative angles in the finger simulated with the numerical model and a finite element analysis software ((a) $E_{obj} = 10^5 Pa$, (b) $E_{obj} = 10^8 Pa$)

Table 5.3 Parameters of the comparison presented in Fig. 5.11(a) (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$)

l_1	l_2	a	b	c	d_2
0.1	0.1	0.0757	0.0412	0.1518	0.05
d_3	d_4	w	θ_{01}	θ_{02}	k_i^f
0.05	0	0.01	45	45	0.075

another actuation type completely, e.g. a fully-actuated force-controlled finger (Fig. 5.12(b)), or even with a dual-finger rotational gripper (Fig. 5.12(c)). Simulations of their interactions with a soft object modeled with a PSM can be obtained by adapting the input torque vector and the Transmission matrix presented in Section 5.2. For instance, with tendon/pulley underactuated fingers the input force vector is simply modified by including the pulling force instead of the actuation torque and updating the Transmission matrix coefficients. Namely, to simulate the illustrated design in Fig.5.12(a), a three-phalanx tendon-driven finger, if there

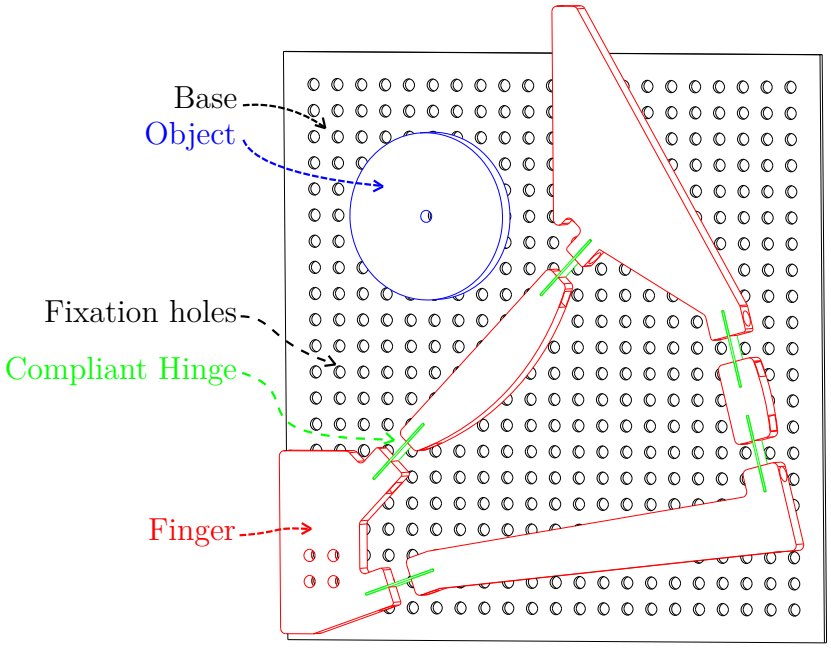
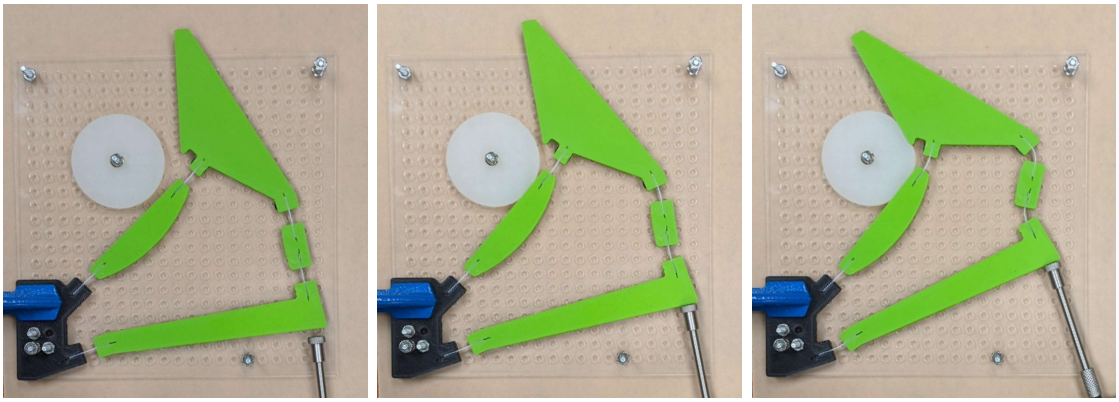


Figure 5.8 CAD model of the experimental setup

are p_j contacts on the j^{th} phalanx the input force vector can be expressed by:

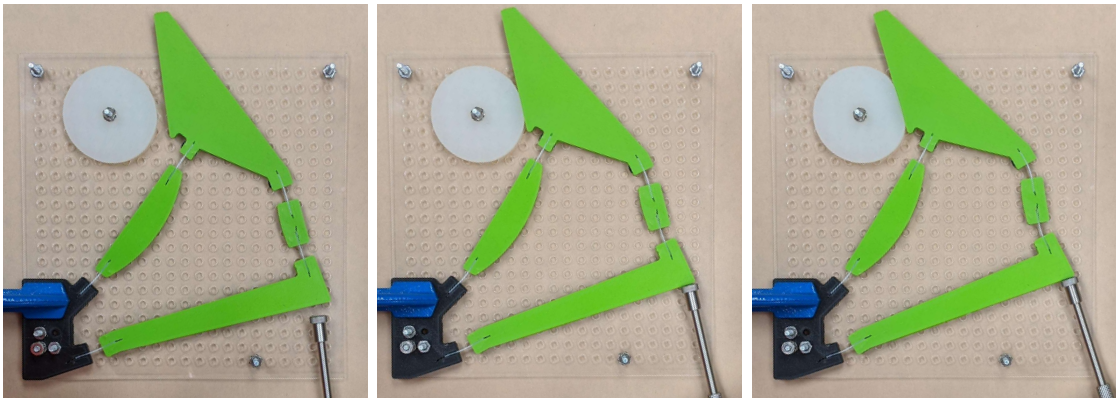
$$\boldsymbol{t}_{\text{tendon}} = \begin{bmatrix} F_a \\ -k_{1,1}\Delta\theta_{1,1} \\ \vdots \\ -k_{1,p_1}\Delta\theta_{1,p_1} \\ -k_{2,1}\Delta\theta_{2,1} \\ \vdots \\ -k_{2,p_2}\Delta\theta_{2,p_2} \\ -k_{3,1}\Delta\theta_{3,1} \\ \vdots \\ -k_{3,p_3}\Delta\theta_{3,p_3} \end{bmatrix}, \tag{5.34}$$

with F_a the magnitude of the pulling force exerted on the tendon. Subscripts i, j indicate stiffnesses or angles related to the j^{th} joints on the i^{th} phalanx.
Then, the Transmission matrix of this finger is calculated based on the relationships between



(a) (b) (c)

Figure 5.9 Power grasps of a soft object (contacts on both phalanges)



(a) (b) (c)

Figure 5.10 Precision grasps of a soft object (distal phalanx contact only)

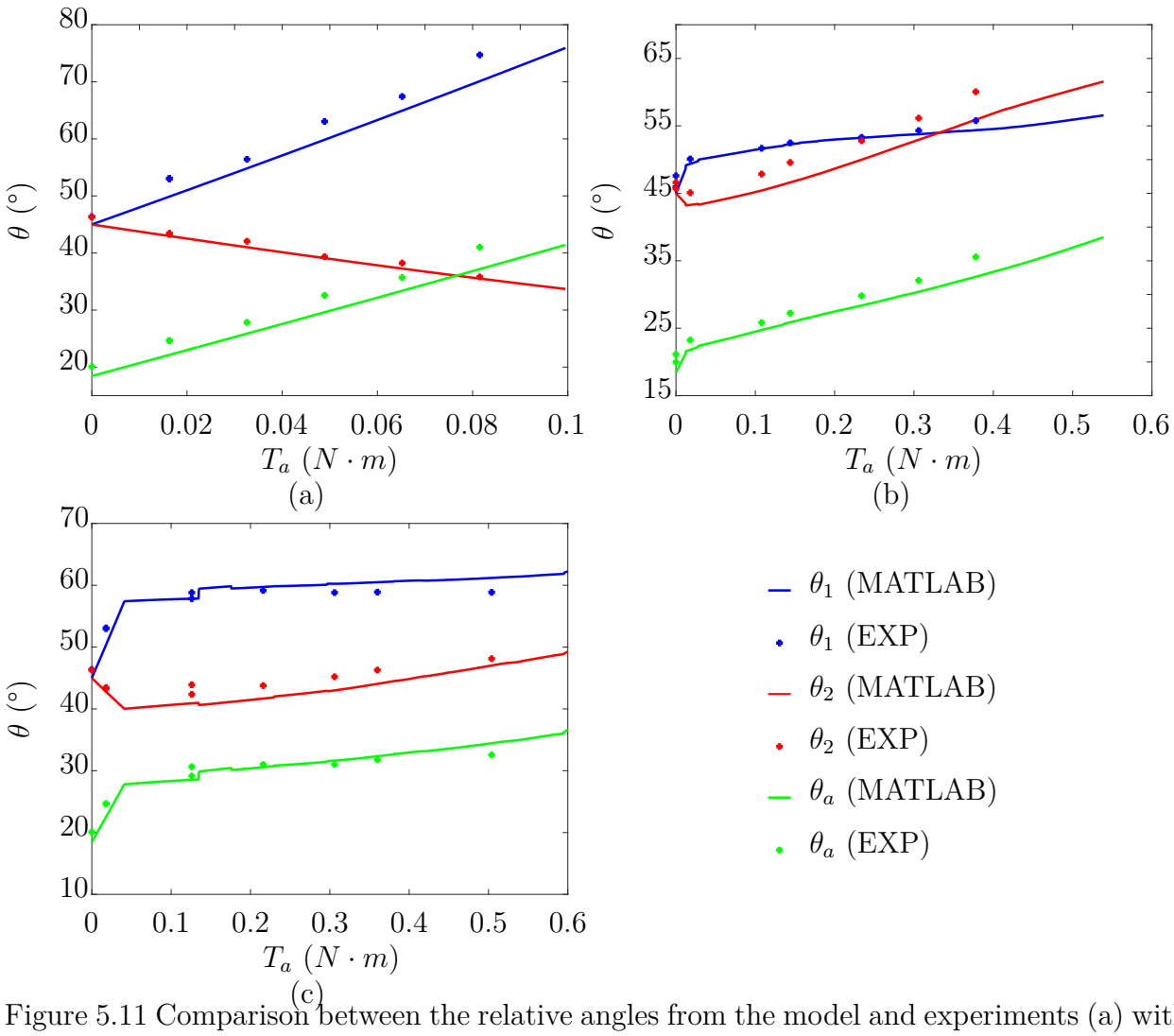


Figure 5.11 Comparison between the relative angles from the model and experiments (a) with the finger (no object), (b) pictured in Fig.5.9 (power grasp) and (c) pictured in Fig. 5.10 (pinch grasp)

the variation of l_t the tendon length and the interphalanx angles:

$$\boldsymbol{T}_{\text{tendon}} = \begin{bmatrix} \frac{\dot{l}_t}{\dot{\theta}_{1,1}} & \cdots & \frac{\dot{l}_t}{\dot{\theta}_{1,p_1}} & \frac{\dot{l}_t}{\dot{\theta}_{2,1}} & \cdots & \frac{\dot{l}_t}{\dot{\theta}_{2,p_2}} & \frac{\dot{l}_t}{\dot{\theta}_{3,1}} & \cdots & \frac{\dot{l}_t}{\dot{\theta}_{3,p_3}} \\ & & & \boldsymbol{I} & & & & & \end{bmatrix}. \tag{5.35}$$

Adapting the input torque vector and the Transmission matrix for fully-actuated finger/gripper as presented in Fig.5.12(b) and (c) is straightforward since the joint velocities corresponding

Table 5.4 Parameters of the object simulation presented in Fig. 5.11(b) and (c) (all lengths are in meters, angles in degrees, stiffnesses in $kN \cdot m$)

n_{div}	r	k_{ext}^{obj}	k_{int}^{obj}
30	0.032	80.3	1161.7
d_1	d_5	d_1	d_5
Power grasp		Pinch grasp	
0.0505	0.0725	0.0805	0.032

to the input torques are directly the variation of the interphalanx angles. Therefore, one has:

$$\boldsymbol{t}_{\text{fully-actuated}} = \begin{bmatrix} T_{a_p} \\ -k_{p_1}\Delta\theta_{p_1} \\ \vdots \\ -k_{p_n}\Delta\theta_{p_n} \\ T_{a_d} \\ -k_{d_1}\Delta\theta_{d_1} \\ \vdots \\ -k_{d_n}\Delta\theta_{d_n} \end{bmatrix} \tag{5.36}$$

$$\boldsymbol{T}_{\text{fully-actuated}} = \begin{bmatrix} 1 & 0 & \dots & 0 & 0 & \dots & 0 \\ & & \boldsymbol{I} & & & \boldsymbol{O} & \\ 0 & & \dots & 0 & 1 & 0 & \dots & 0 \\ & & \boldsymbol{O} & & & \boldsymbol{I} & \end{bmatrix} \tag{5.37}$$

$$\boldsymbol{t}_{\text{gripper}} = \begin{bmatrix} T_{a_p} \\ -k_{p_1}\Delta\theta_{p_1} \\ \vdots \\ -k_{p_n}\Delta\theta_{p_n} \end{bmatrix} \tag{5.38}$$

$$\boldsymbol{T}_{\text{gripper}} = \begin{bmatrix} 1 & 0 & \dots & 0 \\ & \boldsymbol{I} & & \end{bmatrix} \tag{5.39}$$

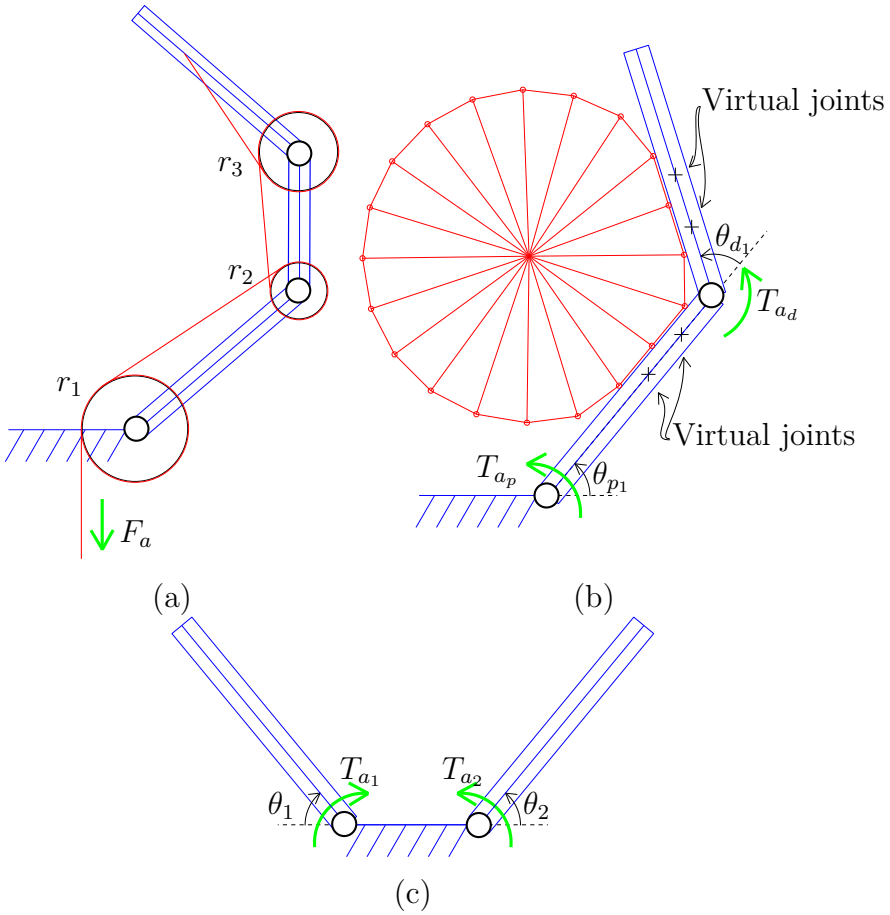


Figure 5.12 Model of (a) a tendon-driven finger, (b) a fully-actuated finger grasping a modeled object and (c) a gripper

where T_{a_x} $(x = p, d)$ are the two actuation torques of the fingers, θ_{p_i} is the relative angles of the i^{th} revolute joint and k_{p_i} the stiffness of the combined spring. The same notation is used for the second phalanx except the subscript is d is used to stand for distal. Subscripts p_1 and d_1 correspond to the proximal and distal interphalanx joints, while p_i $(i = 2, \ldots, n)$ and d_j $(j = 2, \ldots, m)$ correspond to additional virtual joints depending on the number of contact on the respective phalanges.

When two fingers are modeled in the case of the rotational gripper, the projection, Jacobian and Transmission matrices as well as the input vector have to be adapted to calculate Eqn. (5.30) and (5.31) for each finger.

A simulation of a fully-actuated finger using the equations above and compared with a dynamic simulation is illustrated in Fig. 5.13. Parameters of the simulation are presented in Tab. 5.5 including the actuation torques. First, the motor localized at proximal phalanx lowest joint is actuated until $0.02N \cdot m$ and then, the actuation between the proximal and the

distal phalanx is increased. Similarly, results of simulations with the model for the gripper and tendon-driven finger are illustrated in Fig. 5.14 and 5.15 respectively. The parameters of these simulations are presented in Tab. 5.6 and 5.7. In the case of the gripper, symmetrical actuation was assumed ($T_a = T_{a_r} = -T_{a_l}$) and the center of the object was closer to the left finger at rest. As can be plainly seen, the simulations using the presented model fit the dynamic simulation results very well even with these other finger designs.

To conclude and discuss potential applications of this model, it should be reminded that in order to design adaptive fingers, minimal penetrations are often preferred, especially when the objects are soft and/or fragile. Thus, for optimization purposes limits can be set for the magnitudes of these deformations and our model can be used to evaluate them efficiently. As an example, in Fig. 5.16 the maximal permissible actuation magnitudes are plotted depending on the Young's Modulus of the object for a set threshold on the deformation. The parameters of this simulations correspond to one of Section 5.3. For medical applications or in the agricultural sector this type of plot could be used for application guidelines and to ensure that the deformation of the object seized stays below a certain percentage. For instance, materials with a Young's Modulus between $10^2 Pa$ to $10^6 Pa$ closely approximate brain tissues [162] and golden delicious apples [163] respectively.

The methodology presented in this paper could be adapted to approximate the shapes and stiffnesses of objects in three-dimension which would allow to simulate more complex cases. This extension could be done with minimal adaptation of the model using the approach proposed in [164] for instance. From a practical perspective, this 3D update would include the addition of: one equation for the projection of the forces along the $\boldsymbol{z}$-axis, the evaluation of the internal forces in three-dimension in Equ. (5.10), and the extension of the projection matrices to handle 3D vectors. As for the fingers, abduction and adduction motions of underactuated fingers (if possible) are rarely performed simultaneously with their closing/opening since these fingers are seldom used for manipulation, especially with deforming objects. Thus, an adjustment of the Jacobian and Transmission matrices and the input vector from the planar to the spatial case is less likely to be necessary. Each finger would essentially still be "squeezing" the object in its own plane of action. Only the coupling between the fingers through the object deformation needs to be changed.

Then, even if frictionless contacts are a conservative scenario and gravity can often be neglected compared to contact forces, it is nevertheless possible to take into account static

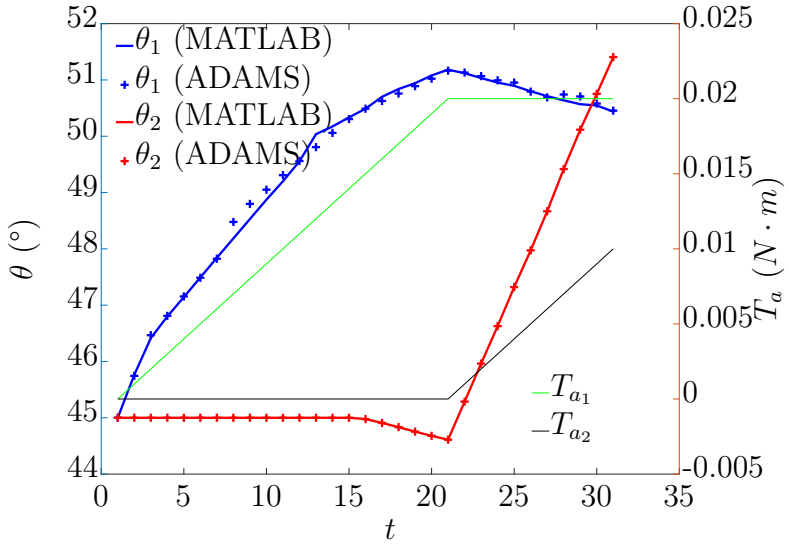


Figure 5.13 Comparison between the relative angles from the model of a fully-actuated finger and dynamic simulations

Table 5.5 Parameters of the simulation presented in Fig. 5.13 (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$)

l_1	l_2	θ_{01}	θ_{02}	d_3	w	d_1
0.1	0.1	45	45	0.05	0.005	0.0285
d_5	r	n_{div}	r_{sphere}	k_i^f	k_{ext}^{obj}	k_{int}^{obj}
0.0425	0.064	20	0.001	0.075	76.62	11.99

friction as well as the weight of the object in our model. These refinements will first require the determination of appropriate masses and friction coefficients at every discretized points of the object which was not shown in the original Van Gelder model in [118]. However, other works in the literature have addressed similar concerns, if not specifically for Van Gelder's models. For instance, in [164] the estimation of discrete points masses for a lumped model was proposed and this method is similar to our calculation of the spring stiffnesses, i.e. masses are estimated depending on the shapes of the adjacent triangles at the considered node as well as the density of the material. The gravitational force applied at a point i with m_i the

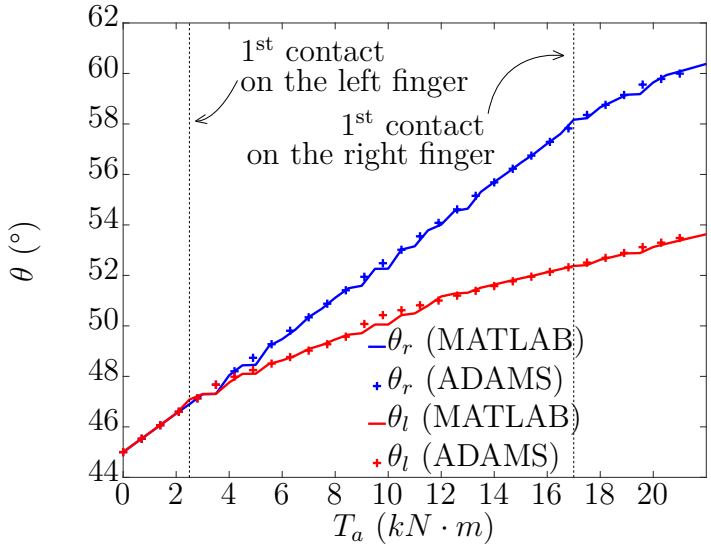


Figure 5.14 Comparison between the relative angles from the model of a gripper and dynamic simulations

Table 5.6 Parameters of the simulation presented in Fig. 5.14 (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$)

l_r	l_l	θ_{0r}	θ_{0l}	d_3	w	d_1
0.2	0.2	45	45	0.025	0.005	0.0225
d_5	r	n_{div}	r_{sphere}	$k^f_{r/l}$	k^{obj}_{ext}	k^{obj}_{int}
-0.0075	0.070	20	0.001	0.075	76.62	11.99

estimated local mass may then be included rather simply in Eqn. (5.8), i.e.:

$$\begin{bmatrix} f^i_x \\ f^i_y \end{bmatrix} = \begin{bmatrix} -\boldsymbol{S}_{i,i-1} \\ \boldsymbol{S}_{i,0} + \boldsymbol{S}_{i,i-1} + \boldsymbol{S}_{i,i+1} \\ \boldsymbol{S}_{i,i+1} \end{bmatrix}^T \begin{bmatrix} \boldsymbol{u}_{i-1} \\ \boldsymbol{u}_i \\ \boldsymbol{u}_{i+1} \end{bmatrix} + \boldsymbol{P}_i \begin{bmatrix} k_{i,0} b_{i,0} l^0_{i,0} \\ k_{i,i-1} b_{i,i-1} l^0_{i,i-1} \\ k_{i,i+1} b_{i,i+1} l^0_{i,i+1} \end{bmatrix} + m_i \boldsymbol{g} \tag{5.40}$$

with $\boldsymbol{g}$ the gravitational acceleration vector. Then, Eqn. (5.10) would become:

$$\boldsymbol{f}_o = \boldsymbol{S}\boldsymbol{u} - \boldsymbol{P}\boldsymbol{k} + \begin{bmatrix} m_1 \boldsymbol{I} & \dots & m_n \boldsymbol{I} \end{bmatrix}^T \boldsymbol{g} \tag{5.41}$$

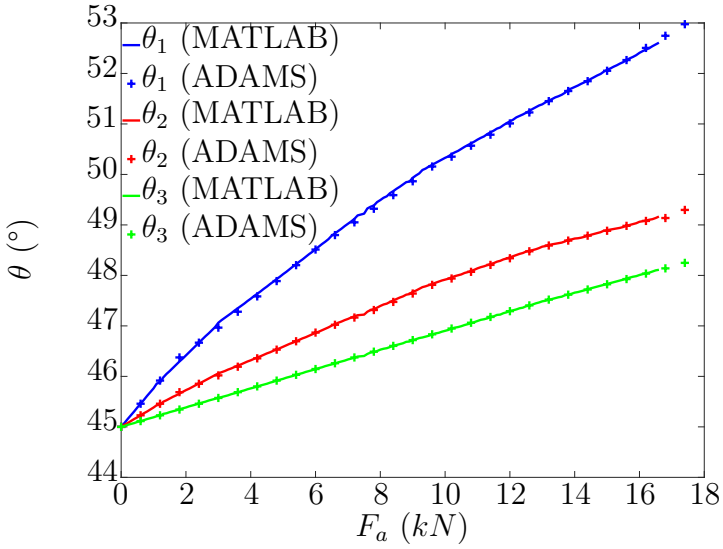


Figure 5.15 Comparison between the relative angles from the model of a tendon-driven finger and dynamic simulations

and the rest of the algorithm we proposed remain unchanged. Friction between the finger and the grasped objects on the other hand can be modeled by first updating the Jacobian matrix defined in Eqn.(5.19) to a extended form $\boldsymbol{J}^*$, as described in [37], namely:

$$\boldsymbol{J}^* = \boldsymbol{J} - \boldsymbol{\mu} \begin{bmatrix} \boldsymbol{r}_{11}^T \boldsymbol{y}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T \boldsymbol{y}_2 & \boldsymbol{r}_{22}^T \boldsymbol{y}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1p}^T \boldsymbol{y}_p & \boldsymbol{r}_{2p}^T \boldsymbol{y}_p & \boldsymbol{r}_{3p}^T \boldsymbol{y}_p & \dots & \boldsymbol{r}_{pp}^T \boldsymbol{y}_p \end{bmatrix} \tag{5.42}$$

where $\boldsymbol{y}_i$ the vector defined by $\boldsymbol{y}_i = \boldsymbol{z} \times \boldsymbol{x}_i$ and $\boldsymbol{\mu}$ is a diagonal matrix where each coefficient relates the normal to the tangential forces at the associated contact point. These coefficients are not necessarily the coefficient of static friction μ_s but their values range from $-\mu_s$ to $+\mu_s$.

Assuming materials are know, this static friction coefficient is also determined and for an input torque the proposed model can evaluate a set of stable positions where the contact forces are within the friction cone, around the stable position calculated without friction as in Section 5.2. The range of the resulting parameters (position of the finger and the points of the object) for a possible stable position will increase quickly as the input torque and/or the number of contact points increase. A possible solution to limit the burden of computing the range of all possible stable positions would be to evaluate these positions in a sequential algorithm using small increments of the input torque and determine if the contact locations on the phalanges in contact can remain constant or not. For example, at the next increment

Table 5.7 Parameters of the simulation presented in Fig. 5.15 (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$)

l_i	θ_{0i}	d_3	w	r_1
0.1	45	0.05	0.01	0.01
r_2	r_3	d_1	d_5	r
0.005	0.0025	0.0305	0.0305	0.082
n_{div}	r_{sphere}	k_i^f	k_{ext}^{obj}	k_{int}^{obj}
20	0.001	0.075	7.662	1.199

Table 5.8 Parameters of the simulation presented in Fig. 5.16 (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$)

l_1	l_2	a	b	c	d_2
0.1	0.1	0.0758	0.0412	0.1581	0.05
d_3	d_4	w	θ_{01}	θ_{02}	d_1
0.05	0	0.01	45	45	0.0505
r	n_{div}	d_5	k_i^f	k_{ext}^{obj}	k_{int}^{obj}
0.032	30	0.0725	0.075	$0.0803 \cdot E$	$1.1617 \cdot E$

of the input torque after the first contact occurred, when the contact force was still zero, determination of the new configuration can be done considering a contact location unchanged but an unknown friction force magnitude (namely one coefficient μ_i of $\boldsymbol{\mu}$). Thus, the overall number of unknowns remains the same as the number of equations, similarly to the original case described in Section 5.2 where that contact location was the unknown. If a configuration can then be found where the contact force is within the friction cone (i.e. $-\mu_s \leq \mu_i \leq +\mu_s$), the contact location can be assumed to stay the same and a set of stable positions can be calculated, limiting the range of the solutions. On the other hand, if no stable positions can be found, the algorithm must be started anew with another value of the contact location which is incremented by a slight step in the opposite direction of the friction force calculated at the previous step because then, sliding must occur on that phalanx. This sequence must then be repeated for each contact point and clearly can become very computationally demanding. However, it should be noted that friction does not appear to play a significant role during the deformation of the object in practice as the results from our original frictionless model

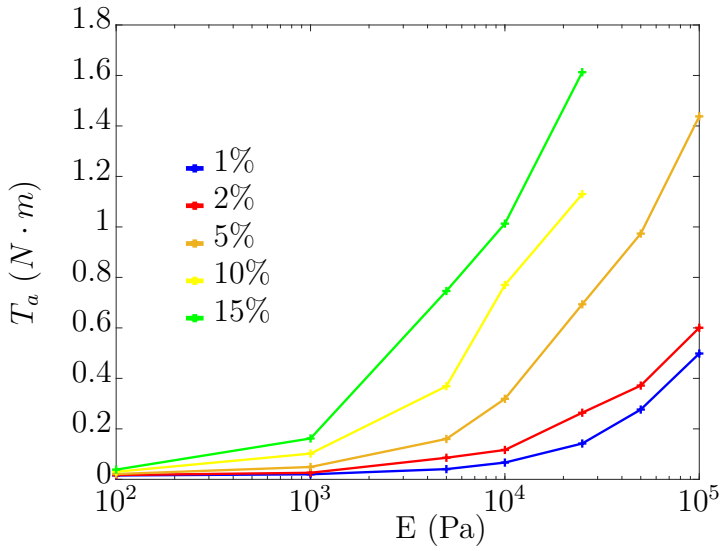


Figure 5.16 Maximal actuation magnitudes of a finger for a set penetration threshold in the grasped object

seem to closely match experimental results as shown in Section 5.3.2.

Lastly, an extension of the model could be considered to include the case of freely moving objects. The possible motions of a rigid circular object grasped by a pair of underactuated fingers was extensively studied in [48] using potential energy analyses and supported by matching experimental measurements. It was shown in that work that the whole object and finger system has only two stable equilibrium positions, one where the former is in a power grasp and one corresponding to a precision grasp, similarly to the cases studied in Section 5.3.2 and illustrated in Figs. 5.9-5.10. Indeed, after the application of a small disturbance at the stable position, the object was shown to move back to this original stable position. This phenomenon highlights that there exist a particular location for the object where the latter tends to stay. In that sense, one could consider this point as fixed to the ground, echoing the model we proposed here. Therefore, an extension of the proposed model with PSM may consider the object motions by first evaluating the stable equilibrium positions (one for a power grasp and one for a precision grasp) considering the object rigid. Then, the object deformations at one of these positions can be simulated with the model presented here. Providing that the overall shape of the object does not deviate significantly from a circle, it is expected that the results of our model would be close to the actual grasp. A more refined technique would be to update the object center point by finding the minimal potential energy of the system following the methodology from [48] but including the potential energy stored by the object in addition to the finger at each step of our simulation. This could

prove to be challenging due to the sheer number of variables. Considering the case where one center point of the object is fixed thus models both the compression of the soft object after it has reached an equilibrium position but also the case where this object is indeed and truly attached to the ground.

5.5 Conclusion

In this paper a numerical model for compliant robotic finger grasping a soft object was shown in order to efficiently simulate its penetration, and therefore, predict potential damages the objects can be subjected to. Moreover, the model can be applied for multiple contacts on the same phalanx of an underactuated finger using a virtual subdivision technique. After the presentation of the model, validations with dynamic simulations and FEA were the first step before final comparisons with results obtained with an experimental test bench. Deformations of a grasped object by an underactuated gripper in some case cannot be neglected and for specific applications it is a critical parameter to evaluate. Due to the increasing efforts in the field of compliant underactuated gripper, effective and simple models are necessary to improve practical designs. While most works in the literature focus on the simulations of object deformations (to accurately estimate stresses and distorted shapes) or on the deformation of the grippers adapting to rigid objects, the model presented here efficiently was shown to be capable of efficiently handling both at the same time.

CHAPITRE 6 ARTICLE 3 : BISTABLE COMPLIANT UNDERACTUATED GRIPPER FOR THE GENTLE GRASP OF SOFT OBJECTS

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Nicolas Mouazé et Lionel Birglen

This paper presents the design of an underactuated gripper with an embedded bistable mechanism. This gripper is specifically targeted at seizing soft objects while minimizing the deformation of the latter. Decreasing the magnitudes of these deformations is key in many practical applications to reduce potential damages due to excessive contact pressure. For instance, for both food processing and medical applications bruising the manipulated item is usually unacceptable. The gripper presented in this work aims at solving this issue by taking advantage of a recently published new method to accurately model the interactions between a compliant gripper and a deformable object. The method is also extended to include contact with the palm of the gripper. The main advantage of this technique compared to existing traditional approaches such as finite element analyses (FEA) is its computational efficiency which allows its use in demanding optimization algorithms, as will be shown. In the first part of this paper, the overall design of the gripper is proposed and the corresponding model is described. The optimization process is then presented and completed with the experimental testing of a prototype.

6.1 Introduction

Most robotic grippers in practical applications are used for a simple and unique tasks: grasping an object. However, the latter can significantly differ in sizes, materials, or masses and these uncertainties can be difficult to handle with the simplest gripper designs as one still found in abundance. For example, in surgical environments grippers can be used to hold and secure steel needles as well as delicate tissues and organs. The latter is of great importance because it has been observed that local bruising leads to microlesions which in turn correlate with longer recovery time [165]. This issue created the need for gentler, often compliant, graspers as proposed in the literature [80, 136, 137]. Similarly, when produces are picked and placed in packaging, indelicate grippers can leave unsightly bruises. Because of customer preferences, these damaged produces are often disregarded and end up as waste.

One way to decrease the excessive pressure created by simple gripper is to distribute the contact forces over a larger area. To this aim, underactuated fingers have been introduced in the literature. These fingers have more degrees of freedom (DOF) than actuators and use a dedicated transmission mechanism combined with passive elements (typically springs) to distribute the actuating torques to the driven DOF. At the heart of this technique, the transmission mechanisms are commonly based on tendon-pulleys [166], gears [167] or linkages [168]. Underactuated fingers are also referred to as self-adaptive [132, 169] due to the distribution effect of the transmission linkage which generate a closing motion of the finger where the stop of one phalanx does not prevent the other to continue moving. An enveloping motion is thereby created purely mechanically without the need for sensors or numerous actuators. Hence, underactuation is one solution for grasping delicate tissues and produces keeping deformation of the latter under control. In principle, underactuated fingers share many similarities with soft grippers as referred to as in the soft robotics field. These soft grippers are also a solution to avoid damaging contact pressures and can be seen as an extension of underactuated fingers where the number of phalanges (and thus, the number of DOF) goes to infinity while structural deformation of the finger material is simultaneously used. An intermediate between soft and rigid-link underactuated grippers are compliant underactuated grippers in which local deformation of the structural part is used to create kinematic bonds. Compliant mechanisms can significantly improve the design of mechanisms and use flexible materials in combination with appropriate geometries to create a linkage [132, 134, 136, 170, 171]. It is a common approach in the manufacturing of microelectromechanical systems. Because the deformation of a compliant mechanism is often restricted to a small part of the material, conversely to soft robotic systems, the most common modelling method used relies on finding an adequate pseudo-rigid-body model (PRBM) [82, 172]. A similar approach to simulate the deformation of soft objects has been introduced in [120] based on a point-spring model (PSM). It is a simple alternative to FEA that is both accurate even for large deformations and far less computationally intensive. In a PSM a set of points connected by springs discretizes the geometry of an object. The fundamental difference with FEA is that the geometric distribution of these points is not random and that they are very few in number. For instance, it was shown that twenty points is sufficient to accurately predict the deformation of a circular object [125]. Values of the springs' stiffnesses in a PSM depend on the material and the simulated geometry as shown in the literature [118].

In addition to compliance, underactuated grippers can also become bistable with the inclusion of a properly designed complementary linkage [173]. Compliant bistable mechanisms

are a class of compliant mechanisms which possess two configurations where they are in stable static equilibrium. The first one typically corresponds to the initial position, at rest, when no flexural part of the linkage is deformed. A second configuration is then found after some motion has occurred and in this second configuration, at least some elastic energy is stored by the mechanism. In a gripper, this second configuration can be set to correspond to a fully closed position and thus, when there is an object preventing the gripper to reach this second stable equilibrium, grasp forces can be created. The interesting aspect of these grasp forces is that they are generated by the elastic energy released by the mechanism and not by the actuator driving the gripper. Thus, grasp forces can be maintained at no energetic cost [174, 175]. This property was used for example in [176] where a bistable linkage was added to the palm of an underactuated gripper to improve robustness against external disturbances during high-speed manipulations.

In this paper, a process to design a compliant bistable underactuated gripper dedicated to the grasp of fragile objects is shown. The main constraint of this task is to mitigate the deformations of the grasped soft objects as much as possible. To the best of the authors' knowledge, the challenge of designing an underactuated gripper interacting with deformable objects has never been addressed in the literature. First, the PRBM of a candidate design is presented, followed by the definition of the objective function and the constraints to consider for an optimization. Then, the design process to achieve the desired goal is detailed step-by-step and results are shown.

6.2 Model

This section introduced the design and the corresponding numerical models to evaluate deformations of objects interacting with grippers. Assuming the gripper has to grasp fragile objects, cautious manipulations of the tools in particular during interactions are expected. Therefore, the motions and deformations in the system are assumed slow and kinetic energies are neglected compared to the potential energies. Frictionless interaction and neglecting gravity are common hypotheses in analysis of underactuated grippers, as the first one is conservative (worst-case scenario for stable grasps) and usually gravity can be neglected compared to grasp forces. Adaptations of the model discussed here to cases with non negligible friction and masses are discussed in [121]. Finally, to reduce the number of variables and computing time, the left and right fingers will have mirrored geometrical parameters.

6.2.1 Pseudo-Rigid Body Design

The geometry of the gripper proposed in this work is illustrated in Fig. 6.1 and since it is assumed to be symmetrical, only half the gripper is shown. The kinematic structure of this design can be decomposed into three parts:

1. first, a two-phalanx underactuated finger (joints A_1 and A_2, in blue) driven by a three joint transmission mechanism (joints B_1, B_2, and B_3),
2. a bistable four-bar linkage (B_1, C_2, C_3, C_4, in purple),
3. an optional input handle (B_1, D_2, D_3, D_4, in yellow).

A deformable object is also shown in Fig. 6.1. This object is initially circular (orange disc) but is deformed by the contact with the finger. The palm of the gripper (namely the line passing through A_1 and normal to axis $\boldsymbol{y}$) is assumed fixed and points B_1 and D_4 are attached to this palm. In addition to the phalanges, the horizontal line (OA_1) of the palm is the only part contacting the object. For ease of usability in case the design is actuated manually, the optional four-bar input handle is shaped to have a motion similar to a scissor. The input of the finger in that case is the rotation of link D_3D_4 (cf. green arrow). Because the gripper will be made compliant, every revolute joint of this model is completed with a rotational spring consistently with the PRBM method.

To illustrate how bistability can be obtained with the four-bar linkage $B_1C_2C_3C_4$, the general motion of a common bistable four-bar is presented Fig. 6.2(a). In this figure, θ_1 is the actuated angle ranging in the illustrated example from 50 to 100°. If one uses the following geometric parameters: $l_1 = 0.1m$, $l_2 = 0.2m$, $l_3 = 0.5m$, $l_4 = 0.5m$ and the following stiffnesses in the joints: $K_1 = 1N{\cdot}m$, $K_2 = 1.25N{\cdot}m$, $K_3 = 1.5N{\cdot}m$ and $K_4 = 5000N{\cdot}m$, the resulting motion of the mechanism has three equilibrium positions illustrated in Fig. 6.2(a). Amongst these equilibrium positions, two are stable and one is unstable. The associated potential energies as functions of the input angle are illustrated in Fig. 6.2(b). In the previous figure, V_{tot} and V_i are the potential energies of the complete system and of the i^{th} spring (corresponding to the angle θ_i) respectively. Three configurations where the derivative of the total potential energy is zero and thus, the mechanism is in static equilibrium, can be found: one at the initial position ($V = 0J$ in the illustrated case) and two others indicated by crosses in Fig. 6.2(b). The two minima correspond to stable equilibrium positions where the mechanism can stay without any input force/torque and will return if disturbed by small perturbations. The mechanism can also stay in the position of maximum total energy without input but in this case it cannot resist to any disturbance, corresponding to an unstable equilibrium. Evaluations of the first and second derivative of the energy are particularly useful for computational optimizations to find the remarkable positions (first derivative equal to zero) and their stability (sign of the

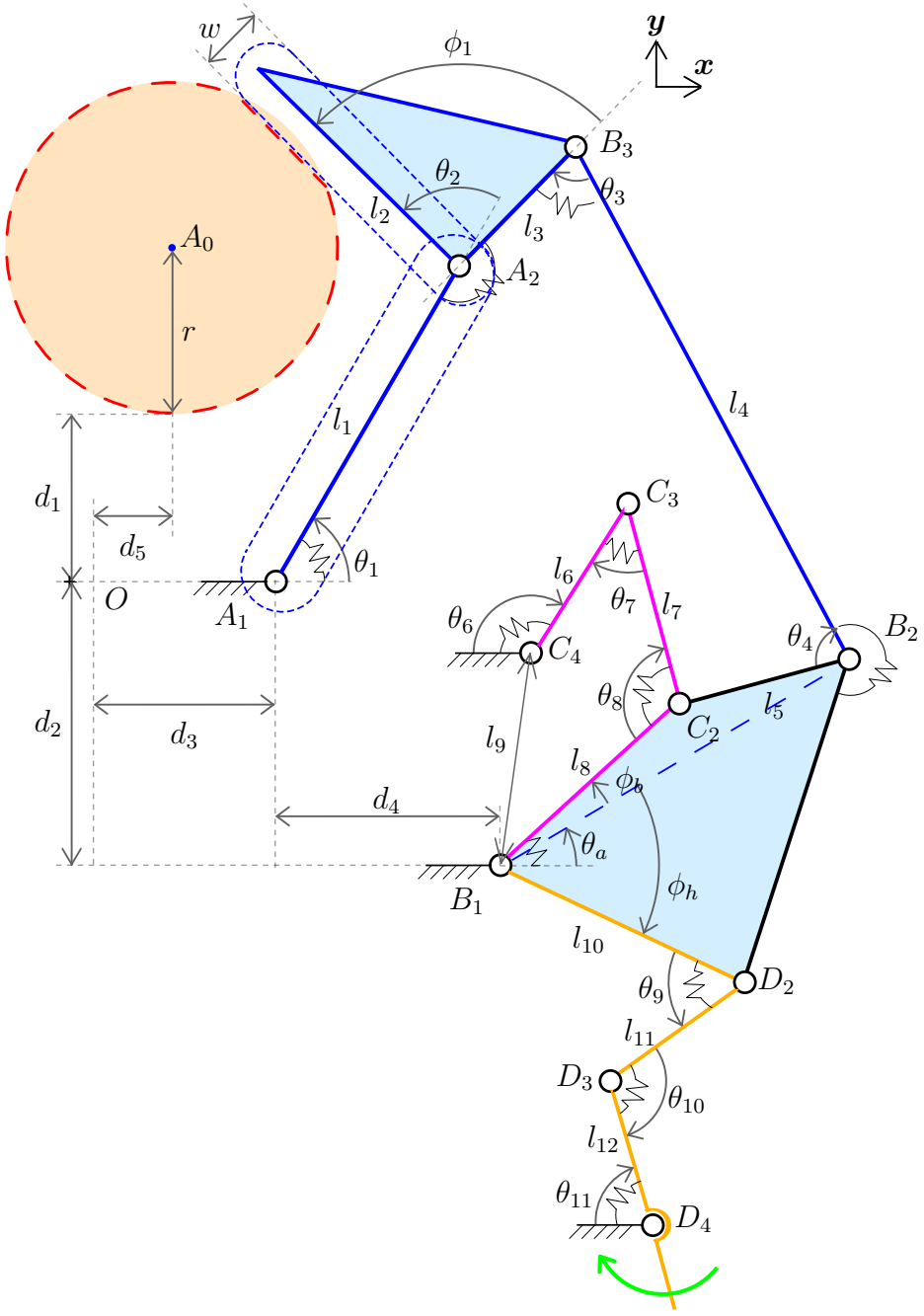


Figure 6.1 Kinematic model of the compliant bistable underactuated finger with a deformable object.

second derivative). The evolution of the first and second derivatives of the potential energies are also illustrated in Fig. 6.2(c) and (d).

The total potential energy being the sum of the elastic energies in each springs, it is clear that the stiffer spring, and especially its retrograde motion, is the main reason for the bistable behavior. Only this spring energy will increase and then, return to almost zero during the motion whereas all the others continually increase. The example illustrated here show how a bistable mechanism work and it can be easily adapted to a finger as used in this work to keep it stable in both the open and closed positions. Indeed, a configuration of four-bar linkage could be calculated to match the potential energies of the finger and create extreme positions of energy. A delicate balance must however be found in practice: if the grasping force created by the bistable linkage is too weak a small perturbation will move back the whole mechanism in the open position while a strong snapping action of this bistable can damage the object. It could be noted that we chose to embed the bistable linkage inside the underactuated finger but the handle could also have been used to this aim if present. The finger was been chosen to maximize compactness as the void inside of an underactuated finger as used in this work is geometrically required to generate good grasp forces but typically left unused.

6.2.2 Deformation of the Soft Object

In order to simulate the interaction between the gripper and an object, a point-spring model is used to account for the shape and elasticity of that object when deformed. For example, in Fig. 6.3, there are n points discretizing the contour of an object and completed by one central point. Springs connect all the points and divide the geometry in triangles similarly to a planar FEA. As shown in [118], semi empirical formulae for the stiffnesses of these springs can be used to match the results of a FEA software even for large deformations as is typically the case for soft objects.

From [121], a general relationship between $\boldsymbol{f}_o$ the vector of internal forces in the object and $\boldsymbol{u}$ the displacements of these points can be developed and takes the form of:

$$\boldsymbol{f}_o = \boldsymbol{S}\boldsymbol{u} - \boldsymbol{P}\boldsymbol{k} + \boldsymbol{M}\boldsymbol{g} \tag{6.1}$$

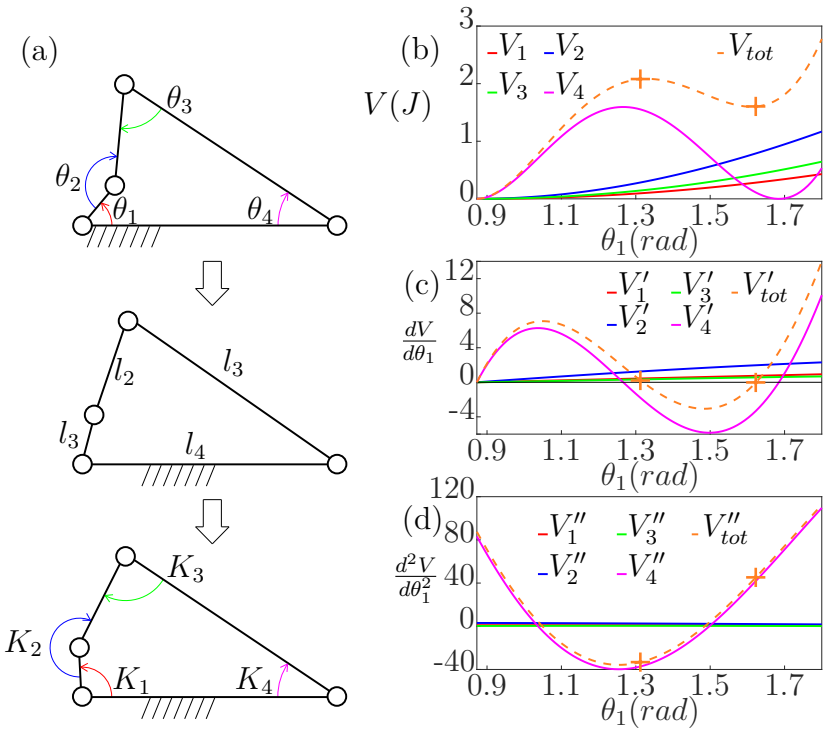


Figure 6.2 Motion of a bistable mechanism. (b) Evolution of the potential energies, their (c) first and (d) second derivatives.

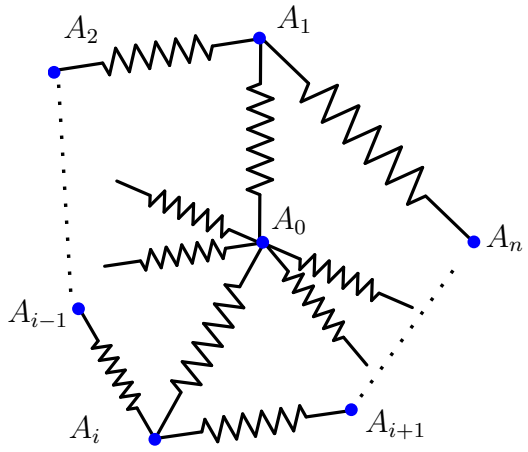


Figure 6.3 Point-Spring model with n points modeling the contour of the object and a central point connecting all contour points.

with:

$$\boldsymbol{f}_o = \begin{bmatrix} \boldsymbol{f}_0^T & \dots & \boldsymbol{f}_n^T \end{bmatrix}^T, \tag{6.2}$$

$$\boldsymbol{S} = \begin{bmatrix} \sum_{j=0}^{n} \boldsymbol{S}_{0,j} & \vdots & -\boldsymbol{S}_{0,i} & \vdots & -\boldsymbol{S}_{0,n} \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ -\boldsymbol{S}_{i,0} & \vdots & \sum_{j=0}^{n} \boldsymbol{S}_{i,j} & \vdots & -\boldsymbol{S}_{i,n} \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ -\boldsymbol{S}_{n,0} & \vdots & -\boldsymbol{S}_{n,i} & \vdots & \sum_{j=0}^{n} \boldsymbol{S}_{n,j} \end{bmatrix}, \tag{6.3}$$

$$\boldsymbol{u} = \begin{bmatrix} \boldsymbol{u}_0^T & \dots & \boldsymbol{u}_n^T \end{bmatrix}^T, \tag{6.4}$$

$$\boldsymbol{P} = \begin{bmatrix} \boldsymbol{P}_0 & \dots & \boldsymbol{0} & \dots & \boldsymbol{0} \\ \vdots & \ddots & \vdots & & \vdots \\ \boldsymbol{0} & \dots & \boldsymbol{P}_i & \dots & \boldsymbol{0} \\ \vdots & & \vdots & \ddots & \vdots \\ \boldsymbol{0} & \dots & \boldsymbol{0} & \dots & \boldsymbol{P}_n \end{bmatrix}, \tag{6.5}$$

$$\boldsymbol{k} = \begin{bmatrix} \boldsymbol{k}_0^T & \dots & \boldsymbol{k}_n^T \end{bmatrix}^T, \tag{6.6}$$

$$\boldsymbol{M} = \begin{bmatrix} m_0\boldsymbol{I} & \dots & m_i\boldsymbol{I} & \dots & m_n\boldsymbol{I} \end{bmatrix}^T. \tag{6.7}$$

$\boldsymbol{S}$ is the stiffness matrix as typically defined in finite element models and is here augmented with the product $\boldsymbol{Pk}$ accounting for large deformations. The details of both elements can be found in [121]. Vectors $\boldsymbol{f}_i$ and $\boldsymbol{u}_i$ are modelling the force applied on the i^{th} point and its displacement respectively. Finally, $\boldsymbol{M}$ is a mass matrix, $\boldsymbol{g}$ the gravitational acceleration vector and m_i the mass of point i.

The matrices $\boldsymbol{P}_i$ can be detailed as:

$$\boldsymbol{P}_i = \begin{bmatrix} g_{1,i,0} & \dots & g_{1,i,n} \\ g_{2,i,0} & \dots & g_{2,i,n} \end{bmatrix} \tag{6.8}$$

with $g_{1,j,k} = g_{2,j,k} = b_{j,k} = l_{j,k}^0 = 0$ if the j^{th} and k^{th} points are not connected by a spring or else:

$$\begin{cases} g_{1,j,k} &= \cos(\alpha_{j,k}) \\ g_{2,j,k} &= \sin(\alpha_{j,k}) \\ b_{j,k} &= 1 - \cos(\alpha_{j,k} - \alpha_{0,j,k}) \end{cases}. \tag{6.9}$$

Vectors $\boldsymbol{k}_i$ are defined by:

$$\boldsymbol{k}_i = \begin{bmatrix} K_{i,0} b_{i,0} l^0_{i,0} \\ K_{i,1} b_{i,1} l^0_{i,i} \\ \vdots \\ K_{i,n} b_{i,n} l^0_{i,n} \end{bmatrix} \tag{6.10}$$

where $K_{j,k}$, $\alpha_{j,k}$, $\alpha_{0,j,k}$ and $l_{j,k}$ are the stiffness, angle with the $\boldsymbol{x}$-axis in the considered configuration and at rest, and the length (also at rest) of the spring connecting the j^{th} and k^{th} points.

Estimations of the accuracy of that discretization method have been performed in [125] to measure its precision and the required calculation time depending on the division of the object. Compared to the formulation introduced in the previous reference, in this paper one point can be connected to as many points as required and therefore, multiple discretizations are possible (not only triangles).

6.2.3 Gripper-Object Interaction

To evaluate the positions of the different points of the gripper-object system, conditions for static equilibrium of the fingers can be first expressed, namely:

$$\boldsymbol{f}_o^T \boldsymbol{D}_{x,s}^T \boldsymbol{J}_s - \boldsymbol{t}_s^T \boldsymbol{T}_s = \boldsymbol{0}^T, \tag{6.11}$$

$$\boldsymbol{f}_o^T \boldsymbol{D}_{y,s}^T \boldsymbol{J}_s - \boldsymbol{t}_s^T \boldsymbol{T}_s = \boldsymbol{0}^T, \tag{6.12}$$

which relate the contact forces and their opposing internal forces in the object projected along the $\boldsymbol{x}$ and $\boldsymbol{y}$ axes respectively. The subscripts s refer to the s^{th} finger (left or right) and these equations have to be evaluated for both fingers. In addition, the equilibrium between the palm of the gripper and the object points in contact is expressed by:

$$\boldsymbol{D}_p \boldsymbol{f}_o = \boldsymbol{0}, \tag{6.13}$$

while

$$\boldsymbol{D}_w \boldsymbol{f}_o = \boldsymbol{0} \tag{6.14}$$

is the relationship between the position and the internal forces for points of the object not in contact with the gripper. In the previous equations, $\boldsymbol{T}_s$ and $\boldsymbol{J}_s$ are the Transmission and

the Jacobian matrices of the finger [37]. Vectors $\boldsymbol{t}_s$ are the input torques. Finally, $\boldsymbol{D}_{x,s}$ and $\boldsymbol{D}_{y,s}$ are projection matrices and $\boldsymbol{D}_w$ and $\boldsymbol{D}_p$ are diagonal matrices. Matrices $\boldsymbol{D}_{x,s}$, $\boldsymbol{D}_{y,s}$, and $\boldsymbol{D}_w$ are defined in [121]. The elements of $\boldsymbol{D}_p$ are an extension of the model in the previous reference taking into account the potential contact with the palm. The elements of matrix $\boldsymbol{D}_p$ are denoted d_{ij}^w ($i = 1 \dots n$ and $j = 1 \dots 2n$) with:

$$d_{ij}^w = \begin{cases} 1 & \text{if } 2i = j \text{ and the } i^{th} \text{ object point touches the palm,} \\ 0 & \text{otherwise.} \end{cases} \tag{6.15}$$

6.2.4 Grasp Simulation

The simulation of a soft grasp by the gripper starts with the fingers at rest and the object is assumed circular with its center along the axis of symmetry. The input angles of the fingers are θ_a (left and right) in Fig. 6.1 and the corresponding input torques are incremented step-by-step simultaneously on each finger symmetrically moving the fingers closer to the object until a contact occurs. Then, the fingers apply forces on the object, pushing it, deforming it while also adapting to its shape to a degree varying with the relative stiffness between the object and the gripper. The object is free to move in the plane and will deform significantly if it cannot escape the grasp or find a position where it can maintain its integrity.

Considering the object could move freely, when the gripper is closing it will push the object until the latter reaches a position where it will be squeezed under non-zero contact forces (after the object cannot be pushed any further to avoid deformations). To speed up the computing time in this phase of the simulation, the first configuration where the object will cease to move and be subjected to squeezing forces is found: a simulation of the finger closing without object is performed and at every actuation step the area between the fingers is evaluated and compared to the object shape. When the object cannot fit inside anymore, the procedure ends and the last position gives the first full contact configuration.

6.3 Validation and Calibration

To validate the approach presented in this paper, experimental data with grippers (or just fingers) and soft objects needs to be compared to the proposed model. To this aim, a first unoptimized prototype, whose CAD model is shown in Figure 6.4, with a soft object were both built. The links of the gripper are 3D printed with plastic filaments and they are connected by compliant living hinges. These living hinges are either laser cut in a $0.7mm$ thin Delrin plate (in red) or also 3D printed (in green). The gripper lays flat parallel to a

table in order to minimize the effect of gravity and to reduce friction, the gripper is offset from the table by a few millimeters. Soft circular objects are made of a silicon compound (Smooth-On, Ecoflex^{TM} 00-30) and its initial position is randomly set inside the gripper, not necessarily with its center along the axis of symmetry. Positions are recorded with a camera at different input torques, measured with a force sensor.

It should be noted that the compliant hinges in the finger are short beams made from Delrin while circular hinges are used in the bistable and the handle mechanisms. This is due to early testing of a prototype: because of the important return torque in the joints of the bistable, short beams were shown to be poor approximation of perfect revolute joints and prone to buckling. Therefore, these joints were changed to circular hinges which performed much better. With specimens built for two-points flexure characterization of the flexible hinges, this prototype allows for a fine calibration of the model, mostly adjusting the stiffnesses of the compliant hinges and the PSM. Secondly, with the parameters obtained from this calibration an optimization is performed to look for the best geometries of the transmission mechanism (positions of points B_2 and B_3 at rest).

Figure 6.5 shows an example of an enveloping grasp with the arbitrarily designed gripper. The geometric parameters of this gripper are listed in Tab. 6.1 where parameter n_{div} is the number of contour points (equal to the number of triangles) used to approximate the object. For this prototype, the bending test of the compliant hinges yields their equivalent stiffness which is found to be approximately 0.22 $N \cdot m \cdot rad^{-1}$, a value close to the 0.18 $N \cdot m \cdot rad^{-1}$ from the literature [177]. With this value and a similar characterization of the springs in the PSM, see [120], the closing motion of the gripper in simulation matches that of the prototype to an excellent degree as illustrated in Fig. 6.6. In the latter figure, subscripts r, l and m refer to the right, left, and mean value of the finger angles. The points "EXP" are

Table 6.1 Geometric parameters of the gripper used for calibration purpose (all lengths are in centimeters, angles are in degrees and stiffnesses in $N \cdot m \cdot rad^{-1}$).

$\theta_{a,0}$	$\theta_{1,0}$	$\theta_{2,0}$	l_1	l_2	l_3	l_4	l_5
18.435	45	45	5	5	4	3.5	7.9
d_2	d_3	d_4	r	k_f	n_{div}	k_{int}	k_{ext}
2.5	2.5	0	9	0.22	30	389.5	5244.9

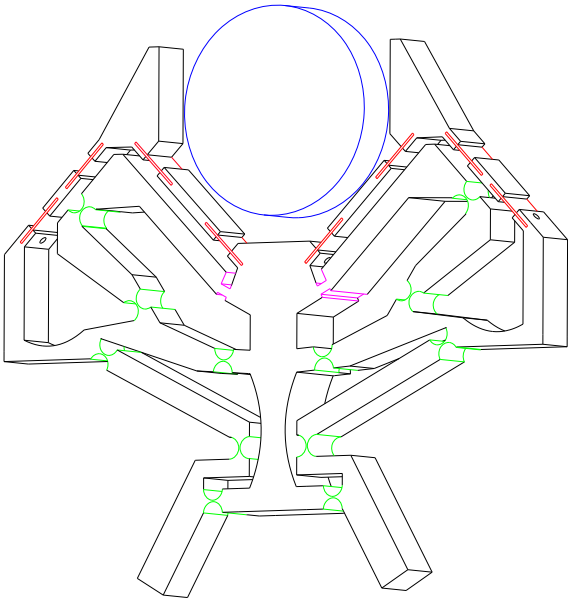


Figure 6.4 CAD model of a prototype.

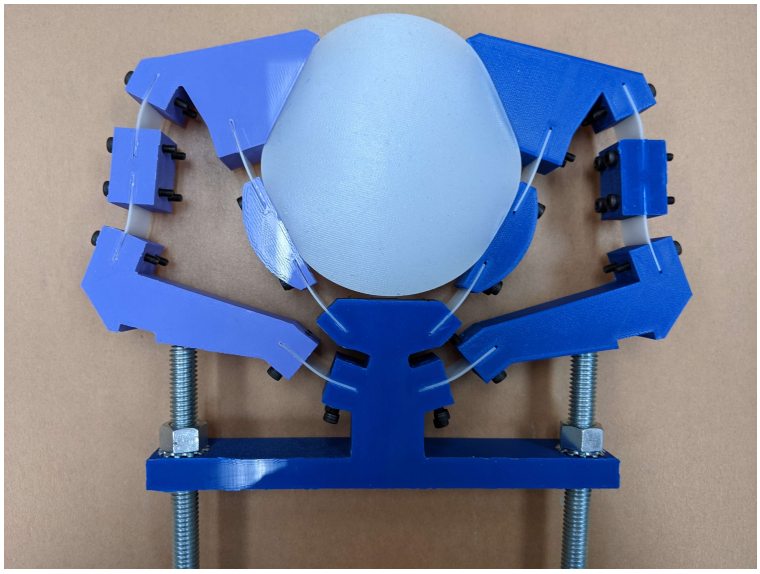


Figure 6.5 Enveloping grasps by a gripper for calibration.

from experimental measurements while the solid lines are from the model described in the previous section. As can be seen, the hypotheses proposed in Sec. 6.2 and the algorithm to find the initial squeezing configuration are validated. In these experiments, the object was first pushed by the fingers to the center without significant deformation until all phalanges touched the object. Then, when enveloped by the fingers the deformations remain identical on the right and left sides. The two phases can be distinguished by the major slope changes observed in the plots.

6.4 Gripper Design

6.4.1 Procedure

The selection of the best geometric parameters for the gripper is realized by a procedure consisting in three main steps: first, an optimization of the fingers transmission mechanism by simulating several object grasps; then, design of the handle if required; finally, inclusion of the bistable mechanism. The difficulty of finding a suitable position for the bistable mechanism in a compact finger may require to look for a trade-off between feasibility and effectiveness. Therefore, a Pareto set has to be evaluated and will lead to back and forth refinement between the different steps of this procedure.

As introduced in Sec. 6.1 mechanical adaptation of the finger to an object is one of the main distinctive asset of underactuated fingers. With the specific objective to reduce deforming objects discussed here, a performance index μ is proposed here to measure the maximum deformation of a soft object. This index is established as the maximal difference of the length of the spring in the associated PSM during the motion of the gripper, namely:

$$\mu = \max_{Ta} \Big(\max_{i \neq j} |l_{i,j} - l^0_{i,j}| \Big). \tag{6.16}$$

This performance index needs to be minimized. However, the ultimate objective of a gripper is to actually grasp an object and minimizing the deformation of that object may lead to useless cases, e.g. fingers not touching the object at all. To take care of this issue, the performance index is computed for an actuation torque ranging from 0 N to a set final actuation torque $T_{a,f}$. This final torque is chosen to be one quarter of the torque required to make the fingers touch each other at their fingertips when there is no object. Setting this final torque to a specific value guarantees that the optimal design is not one where there is no contact. The simulated objects are 9cm silicon cylinders.

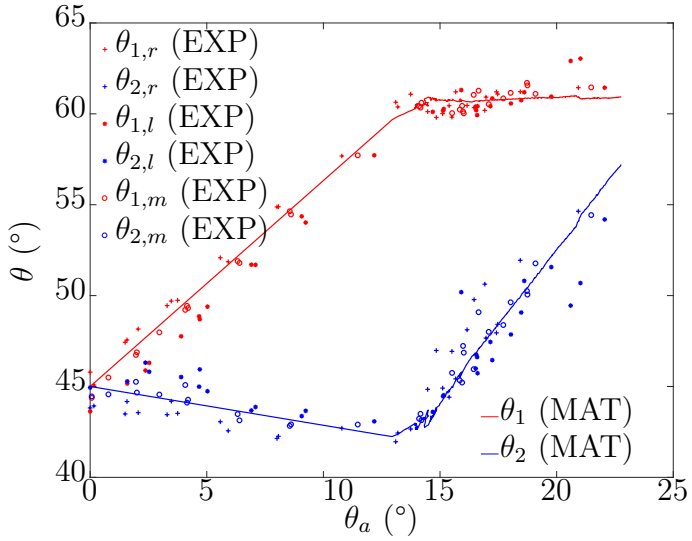


Figure 6.6 Comparison between simulation and experimental tests.

For the second step of the design procedure, knowing the initial and final positions of the actuation angle θ_a, a handle can be adapted for a user with a simple four-bar mechanism to approximate the motion of scissors, as previously introduced. Thus, the established variation of θ_a corresponds to a variation of θ_{11} typical of handheld instruments. A desired final handle position $\theta_{11,obj}$ can be fixed and simulations of the four-bar performed to match this objective with the already calculated final position of θ_a. The performance of a configuration can be evaluated by computing the difference $(\theta_{11,f} - \theta_{11,obj})^2$. To avoid possible bistable behaviour of this part of the mechanism angle θ_{10} needs to be kept over 180°during the motion.

6.4.2 Optimization of the Finger

With the calibrated parameters of the object and gripper, an optimization of the transmission mechanism (defined by points B_2 and B_3) is performed for minimal deformations of the object during grasping. This optimization was achieved by a genetic algorithm (run with Matlab) and Tab. 6.2 presents two configurations, one of which obtained from the software. The design $C1$ corresponds to the optimal design from the optimization whereas $C2$ is a configuration with average performances relatively to the defined performance metric. The latter will serve as a control design, i.e. for comparison purposes. Tab. 6.2 also shows their respective performance metrics and the calculated final torque required. These grippers were built to verify if the calibration values still fit the simulations. Figs. 6.7 and 6.8 illustrate comparisons between the interphalanx angles calculated by simulations and measured with

both prototypes. Fig. 6.9 also shows the optimized gripper enveloping a soft silicone cylinder when actuated by a torque equal to $0.95N \cdot m$, while a simulated model at the same torque is superimposed on the picture for comparison.

As can be clearly seen from the figures, the simulations still match the experiments very well. It can also be observed that the deformations are indeed greater with the control gripper. The optimized design evenly deforms the cylinder and therefore, provide a gentler shape adaptation. From the performances of the design candidates during the numerical optimization, it appears that the farthest and the highest point B_2 is, the smoother the adaptation of the phalanges is. However, this property implies that a gentler gripper, as illustrated in Fig. 6.9, is also quite bulky. For this reason, a Pareto front needs to be established in order to observe possible trade-off between attenuating the deformations and improving compactness. Fig. 6.10 shows this Pareto front: the effect of the position of B_2 on the adaptability is evaluated, a is the projection of B_2 on the $\boldsymbol{x}$-axis and b is the projection of B_2 on the $\boldsymbol{y}$-axis. The optimized value of b and the performance metric μ_1 are then shown for each a.

The Pareto front shows three distinct zones: first, when B_2 is closer to the axis of symmetry than B_3 (small value of a), when B_2 is below B_3, and when B_2 is further to the axis of symmetry than B_3 (large value of a). The design in the last case present the best overall performance. However, they are also the most massive ones whereas the compact configurations in the first zone have relatively lower performances. The designs in-between these two zones seem an acceptable compromise where relatively constant performances ($\mu \approx 4\%$) are obtained, a value that is quite close to the best performance ($\mu = 3\%$) but producing a more compact geometry.

Table 6.2 Geometric parameters of the grippers (all lengths are in meters, angles are in degrees, and stiffnesses in $N \cdot m \cdot rad^{-1}$ and torque in $N \cdot mm$).

	$\theta_{a,0}$	d_2	l_3	l_4	l_5	k_f	μ_1	$T_{a,f}$
C1	16.8	5	3.8	6.4	12.8	0.11	3.8	364
C2	20.8	5	3.3	7	6.6	0.11	5.3	358

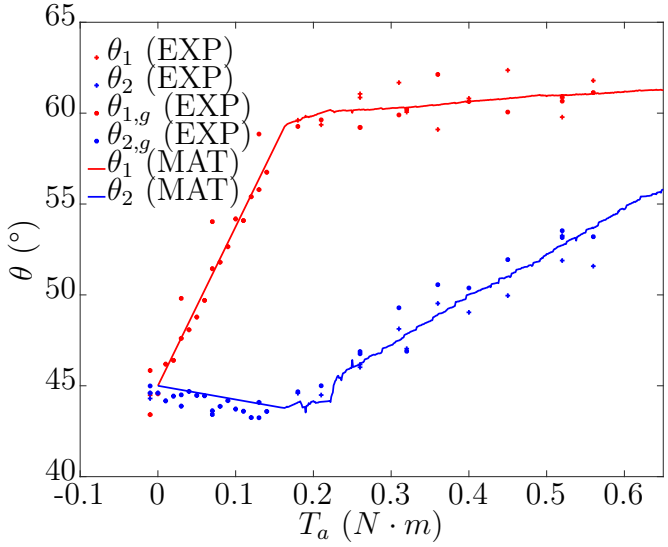


Figure 6.7 Comparison between the relative angles from the model and experiments for the control gripper.

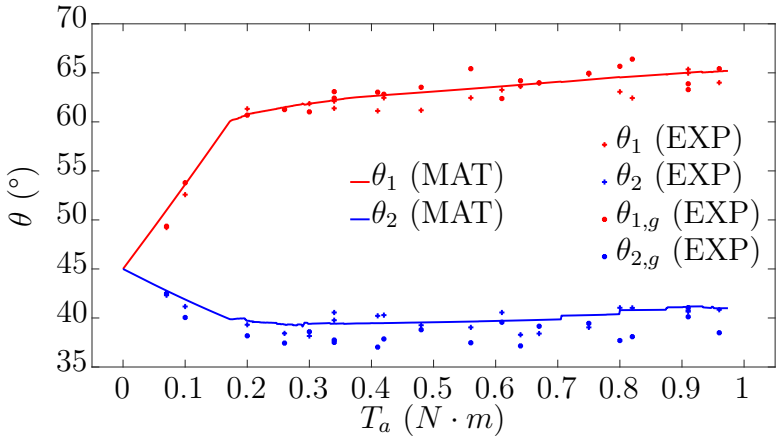


Figure 6.8 Comparison between the relative angles from the model and experiments for the optimal gripper.

Figure 6.9 Optimal gripper grasping a silicone puck.

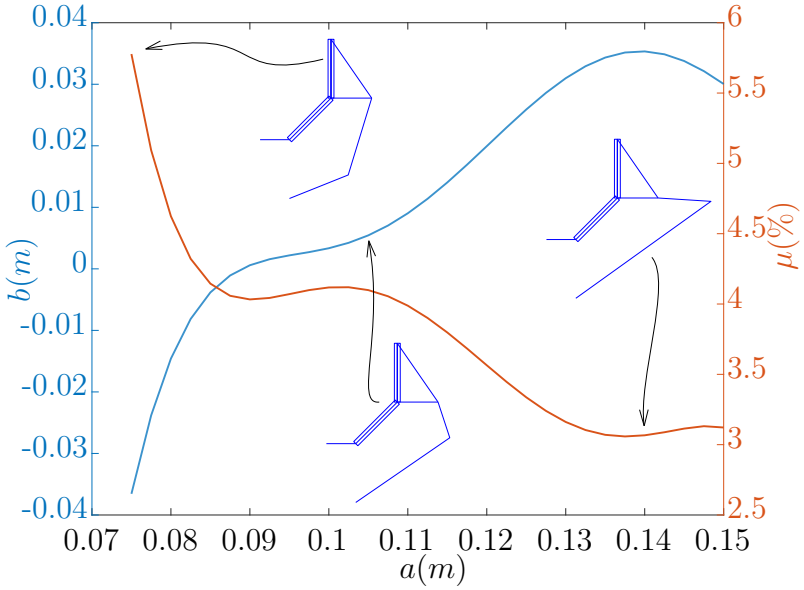


Figure 6.10 Pareto set minimizing object deformations during a grasp.

6.4.3 Handle, Bistable, and Complete Gripper

Finally, the handle and bistable mechanisms are added to the optimal design. If scaled down, a user will be able to actuate the gripper with a natural motion, like using a scissor and the gripper will close at a chosen configuration set by the bistable. The desired design objective for the handle is to have a variation of the position of the handle from rest to the configuration where the left and right fingertips are touching each other is equal to 25°. Knowing the geometrical parameters of the optimal finger, its motion can be simulated and the variation of its input angle θ_a evaluated. From θ_a, the geometry of the handle mechanism can be deduced with the simple geometric equations describing the kinematics of four-bar linkages [178]. Fig. 6.11 shows the input angle of the optimal finger θ_a as a function of θ_{11}, the angle of the handle of the simulated model, compared to experimental data. The angle variation of the handle corresponds to the set objective.

Then, after the handle is designed, the potential energy of the whole gripper (both fingers) and handle can be calculated. For each increment of the handle input torque, an equivalent stiffness k_{equ} can be estimated (where $k_{equ} = T_a/\Delta\theta_a$) corresponding to an equivalent joint in a four-bar linkage as illustrated in Fig. 6.2. The stiffer spring in the bistable mechanism will have to counter the energy accumulated from k_{equ} which ranges from 3.3 to $3.1 N \cdot m \cdot rad^{-1}$ for the optimal parameters at hand. Because of this limited range of variation, the maximum value of the equivalent spring is used to compute the bistable configuration. Figs. 6.13 and 6.12 illustrate the final prototype with all three parts (fingers, handle, and bistable four-bar) grasping 9 cm and a 6 cm deformable cylinders. Complete geometrical parameters of the design can be found in Tab. 6.3.

Table 6.3 Geometric parameters of the complete gripper (all lengths are in millimeters, angles are in degrees, and stiffnesses in $N \cdot m \cdot rad^{-1}$)

$\theta_{a,0}$	l_3	l_4	l_5	l_6	l_7	l_8
34	35	32	112	80	50	50
l_9	l_{10}	l_{11}	l_{12}	k_D	k_C	k_B
30	60	80	26	0.11	0.55	93

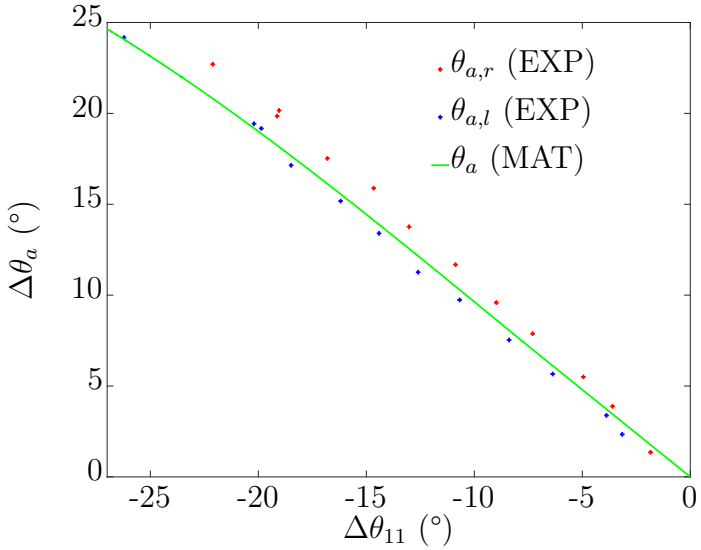


Figure 6.11 Comparison between simulations and experimental data of the variations of the actuated angle of the finger depending on the handle position.

Figure 6.12 Complete optimal gripper grasping a 9 cm soft cylinder.

Figure 6.13 Complete optimal gripper grasping a 6 cm soft cylinder.

During the grasp of the larger cylinder the bistable effect is not seen as the the object is too large for the gripper to reach the second basin of attraction created by the resulting potential energy. With a smaller cylinder, for instance the 6 cm diameter object, the mechanism is indeed bistable and the gripper stays closed on the object without any actuation effort. The prototype also demonstrates the accuracy of the simulation: see Figs. 6.14 and 6.15 where both are compared.

6.5 Application

As mentioned in the introduction, during the handling of fruits and vegetables special attention is needed to limit possible damages. To this aim, finite elements have been used to model a tomato for example and evaluate its deformation under compression as micro-fractures are expected during a rough grasp [179]. In [180] a specific control system has been implemented in a robot designed to harvest apples. With a mechanical adaptive gripper such as the one discussed in this work the requirement for sensors and the complexity of the controller can be reduced even if the orientation and shape differ from one produce to another. In this last section, the prototype previously designed is experimented in a more realistic setting by grasping three fruits/vegetables with various stiffness: an empire apple ($E = 3.33MPa$) [163]), a bell pepper, and a tomato ($E = 0.66MPa$) [179]). This last produce is an important candidate

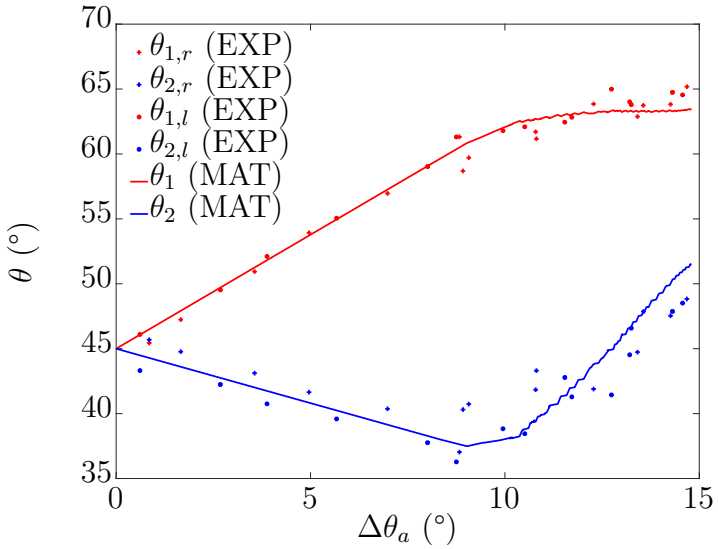


Figure 6.14 Comparison between the relative angles from the model and experiments with a 9 cm cylinder (pictured in Fig. 6.12).

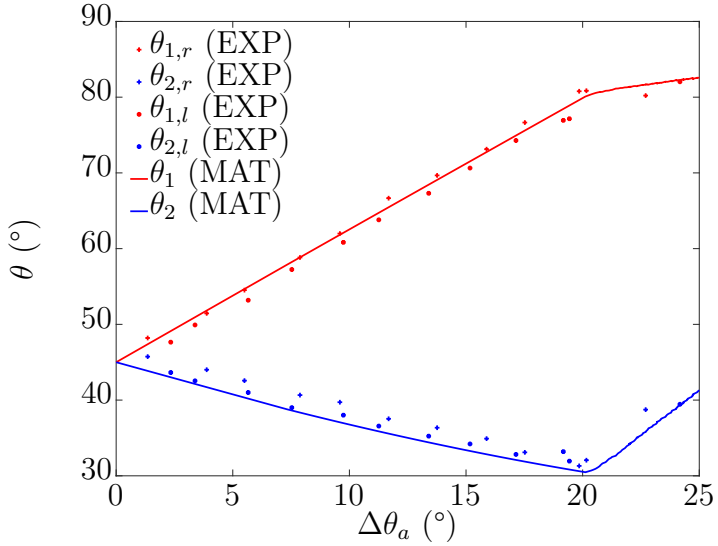


Figure 6.15 Comparison between the relative angles from the model and experiments with a 6 cm cylinder (pictured in Fig. 6.13).

to mitigate deformation as they are commonly afflicted by bruises leaving dark marks on the fruit a few days after the grasp. A DC motor with an incremental encoder (Maxon 355561) is attached to the optimized and non-optimized prototypes to rotate a threaded rod with a nut pushing a 3D printed beam and actuate the fingers. Fruits and vegetable were randomly placed in front of the gripper to be grasped. Fig. 6.16 illustrated these experiments. The difference of adaptation between the grippers is greatly observed with the tomato that is indeed a softer "object".

Experiments demonstrate that the grippers can grasp the bell pepper and the apple easily and lift them without major deformation. Comparatively, with the tomato the gripper has to produce a greater squeezing force to avoid slippage. However, the non-optimized gripper clearly deforms the fruit to a greater degree as can be observed in Fig. 6.17(a). The first phalanx (in blue) obviously flattened the tomato unlike in Fig. 6.17(b) with the optimized gripper. With the final design, the softness of the shape adaptation capability is also clearly demonstrated (see Fig. 6.17(c)). During the experiment, the non-optimised gripper applies a noticeably greater force with its first phalanx, contrarily to the two other grippers.

Data from simulations can be used to adapt a gripper for an automatised production line. For example for a food production chain working with tomatoes, knowing the range of diameter of the fruits usually on the line and the limit of deformation they can withstand without bruise, one actuation torque can be fixed for all tomatoes. Figure 6.18 illustrates the input torque to apply to deform a soft cylinder ($E = 0.66MPa$) by 1 to 7% (and the minimal torque for contact) for different diameters. With these data, a rectangle of acceptable torques to assure a grasp can be drawn for multiple size of object. For example, for cylinders with radii ranging from 2.5 to 5.5 cm (assuming tomatoes are close to cylinders) and if the acceptable deformations threshold is 5%, torques localized in the red rectangles in 6.18 are adequate to actuate the corresponding gripper. A torque of 0.9 $N \cdot m$ is set as upper limit and above this value the gripper cause deformations leading to damages with 3 cm radius cylinders. Furthermore, with torques below the lower threshold (0.4 $N \cdot m$), the contact forces may be too weak to create a safe grasp of smaller objects. Therefore, 0.5 $N \cdot m$ was selected as the lower torque magnitude to ensure grasping without damaging the targeted objects within the targeted size range.

The Young Modulus of the objects can also be a varying parameter in the design process as a tomato softens when ripening, before and after harvesting. Results from Fig. 6.18 can be combined with data from Fig. 6.19 and 6.20 to analyze this effect. The deformations can be

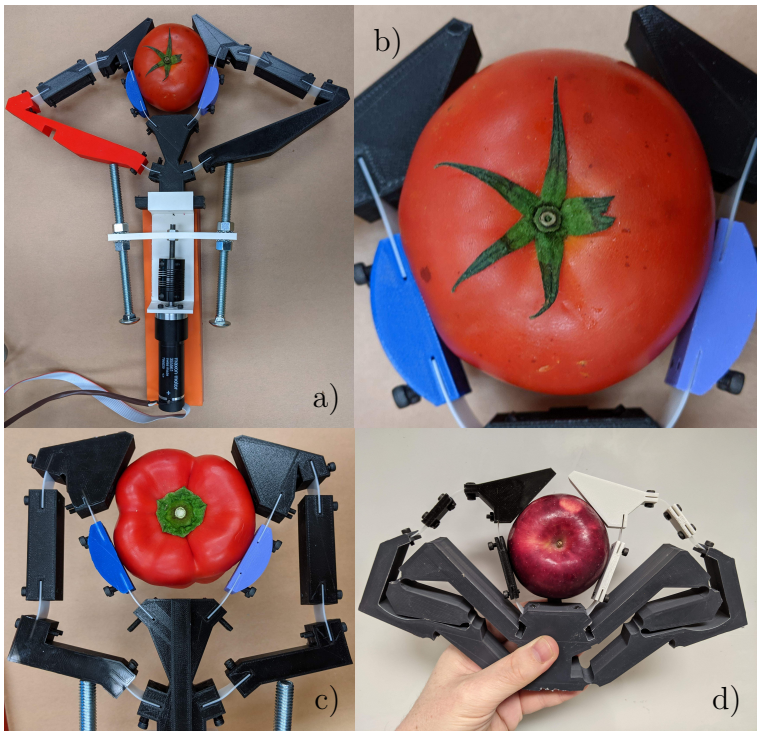


Figure 6.16 Grasp of a tomato by the optimized finger (a)-(b), a bell pepper by the non-optimized finger (c), and an apple by the final prototype (d).

evaluated depending on the Young Modulus, which is $0.66MPa$ on average when picked but expected to decrease by 30% [181] after a few days, with the minimal and maximal diameters of the cylinders. Red rectangles in Figs. 6.19 and 6.20 (dashed lines) also indicate the range of torques to grasp the object while maintaining the deformation under a chosen magnitude. In this case, 0.5 $N \cdot m$ seems again to be an acceptable lower torque for the application.

From a practical point of view, the optimized gripper with the bistable mechanism can grasp cylinders ranging from 4.6 cm to 11 cm, as well as randomly shaped objects close to these diameters. The bistable mechanism is effective for diameters ranging from 4.6 cm to 6.4 cm (bistability can be observed) while objects between 6.4 and 11 cm can still be grasped but the second stable position cannot be reached so the power saving feature of the bistable is not in action. Fig. 6.21 illustrates the bistable gripper grasping a tomato with a diameter close to 6 cm without any input force applied on the input links (the rod are not pushing the bottom linkages for illustration) but the grasp is both gentle and secure.

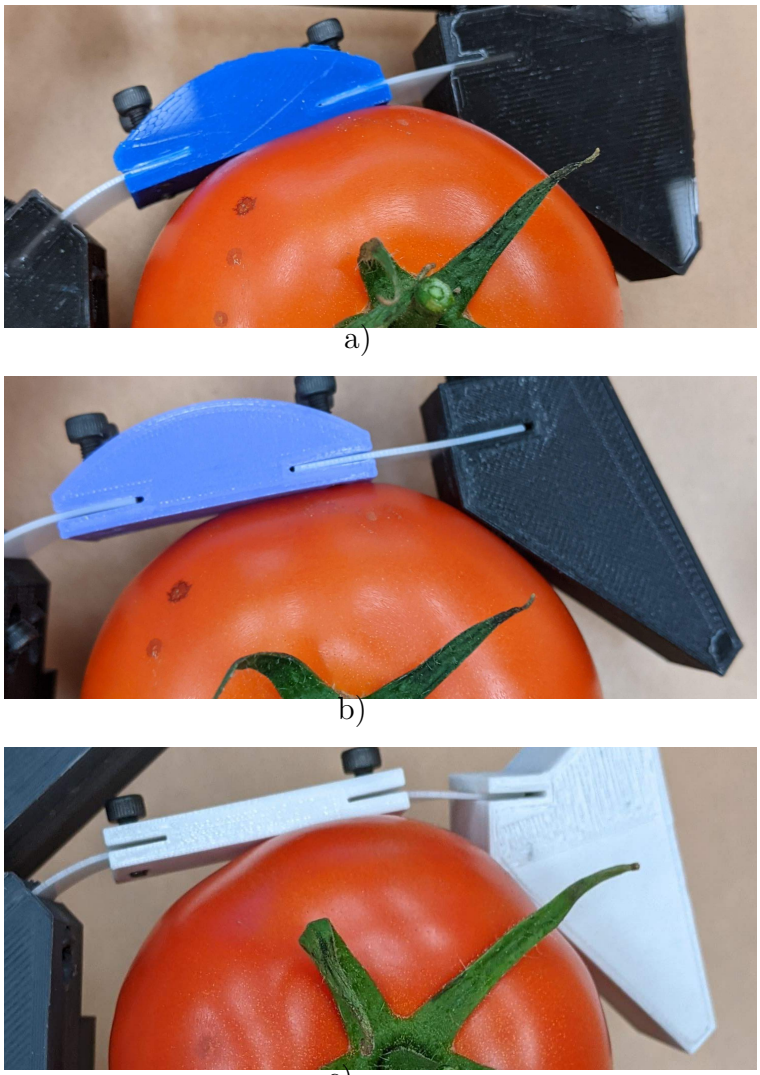


Figure 6.17 Grasp of a tomato by the non-optimized finger (a), by the optimized finger (b), and by the final prototype (c).

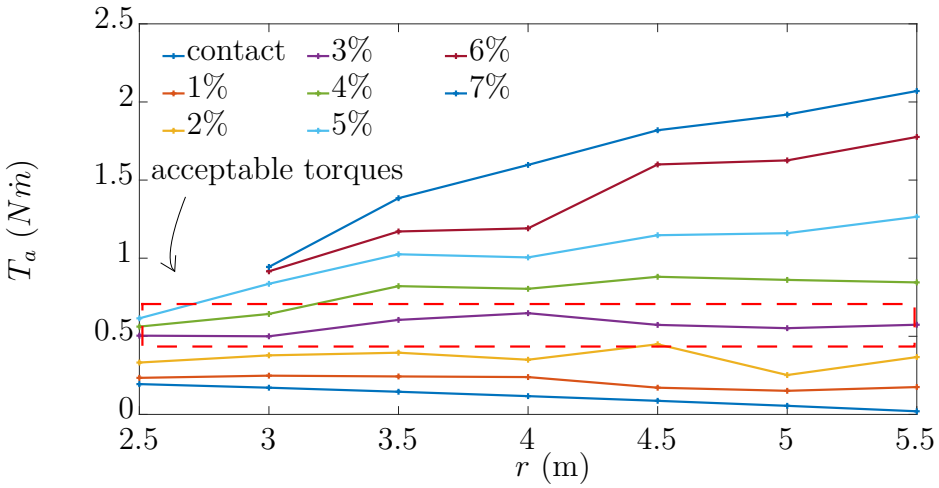


Figure 6.18 Input torque needed to deform a cylinder by 1 to 7 % depending on the cylinder radius ($E = 0.66MPa$).

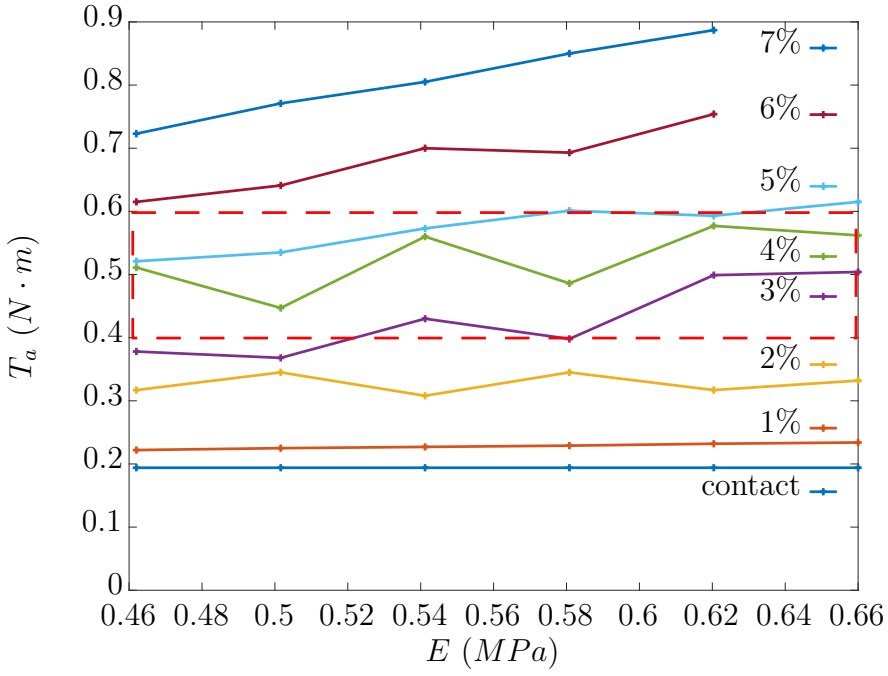


Figure 6.19 Input torque needed to deform a cylinder by 1 to 7 % depending on the Young Modulus ($r = 2.5\ cm$).

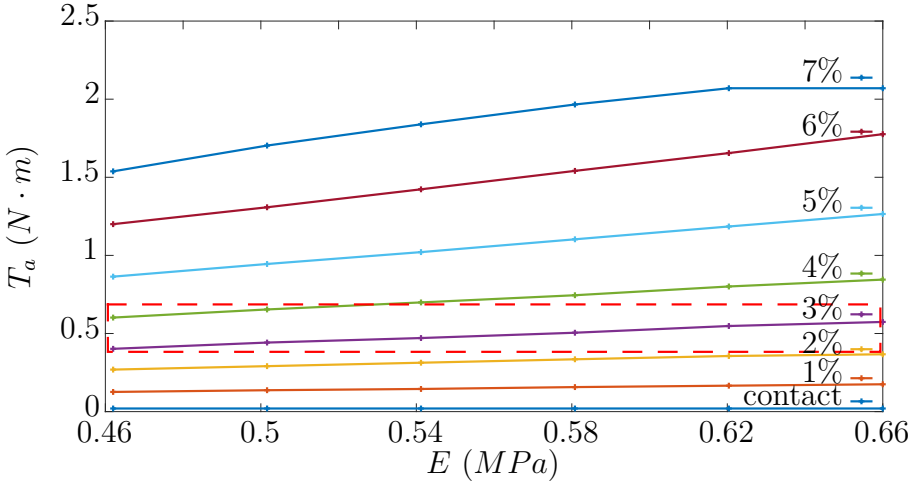


Figure 6.20 Input torque needed to deform a cylinder by 1 to 7 % depending on the Young Modulus ($r = 5.5\ cm$).

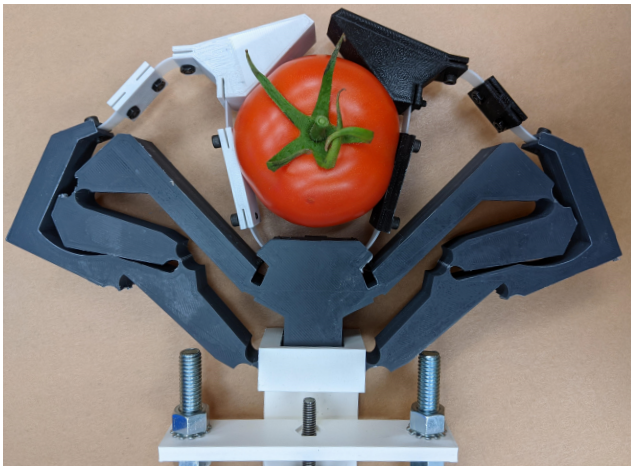


Figure 6.21 Grasp of a tomato by the bistable prototype.

6.6 Conclusion

In this paper, an optimal design of compliant and bistable underactuated gripper is proposed. To the best of the authors' knowledge it is the first time this type of gripper is specifically designed for the undamaging grasp of soft object. The final design produced by the methodology is a bistable inexpensive gripper operated like a scissor with few assembly steps. A general model of the interactions between a free moving deformable object and compliant grippers was first presented and adapted to an optimization. The object was approximated with a point-spring model and potential contacts with the palm are taken into account. In the second part, an optimization of the gripper was presented and a prototype was built to demonstrate the effectiveness of the methodology presented in the paper.

6.7 Résultats complémentaires

Dans la partie précédente, plusieurs prototypes ont été fabriqués et testés : un de géométrie arbitraire pour la calibration, un optimisé (appelé C1), un second de géométrie empirique (appelé C2) pour comparer son efficacité à celui optimisé et au final, une pince complète. Les paramètres de leurs architectures sont listés dans le tableau 6.4 (seulement la partie doigt pour la pince finale). Le prototype pour la calibration est noté C0 et le final F. Les paramètres optimisés sont les longueurs des membrures du mécanisme de transmission l_3, l_4 et l_5. Il est à noter que si les pièces des pinces sont de couleurs différentes, cela est seulement dû au fait que les filaments de PLA utilisés étaient ceux présents dans le laboratoire lors de la fabrication. Le matériau ainsi que les paramètres de fabrication utilisés sont les mêmes pour toutes les pièces imprimées en trois dimensions. Pour chacun de ses prototypes, des comparaisons de leurs comportements avec des objets calculés tels que prédits par des simulations numériques et ceux observés par expérience ont été réalisées et sont présentées dans l'article soumis. Dans cette partie, des résultats complémentaires absents lors de la soumission de l'article sont présentés portant sur les écarts observés lors des comparaisons.

6.7.1 Incertitudes de mesure

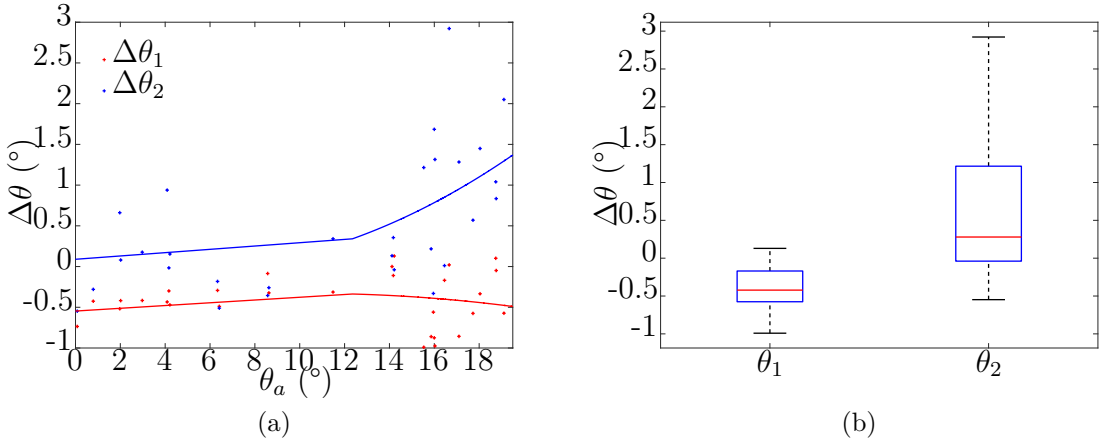
Sur la figure 6.22(a), on peut observer les écarts des positions des phalanges calculées avec le modèle et celles mesurées à l'aide du prototype utilisé pour les calibrations en fonction de l'angle d'actionnement du mécanisme (positions illustrées sur la figure 6.6). Les écarts $\Delta\theta_i$ calculés sont les différences entre les angles $\theta_{i,m}$ obtenus par expérience et θ_i obtenus à l'aide du modèle (pour $i = 1$ ou 2). La figure 6.22(b) est un diagramme en boîte représentant les répartitions de ces écarts. Celui-ci a été tracé pour la phalange proximale (indiqués par $\Delta\theta_1$) et pour la phalange distale (indiqués par $\Delta\theta_2$). La médiane pour $\Delta\theta_1$ est de $-0,4°$, tandis que la moyenne est de $-0,4°$ et l'écart-type de $0,31°$. Pour la seconde phalange, la médiane de l'écart entre les résultats est de $+0,3°$, la moyenne de $-0,5°$ et l'écart-type de $0,8°$. Ces écarts sont aussi ceux que l'on retrouve dans les autres comparaisons.

On peut observer sur ces figures que les écarts les plus grands se trouvent dans la zone où l'on observe des contacts entre l'objet et le doigt. Dans le cas de la figure 6.22(a) pour des valeurs de θ_a supérieur à 12°. De plus, dans cette situation les écarts sont plus importants pour la seconde phalange que pour la première et ceci se retrouve aussi dans d'autres résultats (voir les figures 6.8, 6.12 et 6.13). Le mouvement non parfaitement symétrique des deux doigts lors d'expérience est l'une des causes des écarts observés en particulier lors du contact.

En plus de cela et de la non-prise en compte de certains phénomènes physiques dans le modèle

Tableau 6.4 Paramètres des prototypes fabriqués dans ce chapitre (toutes les longueurs sont en centimétres, les angles en degrés et la raideur en $N \cdot m \cdot rad^{-}1$).

	C0	C1	C2	F
$\theta_{a,0}$	18,435	16,8	20,8	34
$\theta_{1,0}$	45	45	45	45
$\theta_{2,0}$	45	45	45	45
l_1	5	5	5	5
l_2	5	5	5	5
l_3	4	3,8	3,3	3,5
l_4	3,5	6,4	7	3,2
l_5	7,9	12,8	6,6	11,2
d_2	2,5	2,5	2,5	2,5
d_3	2,5	2,5	2,5	2,5
d_4	0	0	0	0
k_f	0,22	0,11	0,11	0,11



(a) (b)

Figure 6.22 Écarts entre les positions des phalanges calculées par simulation et par l'expérimentation.

comme le frottement et la gravité, les erreurs que l'on peut observer avec ou sans interaction peuvent être dû à la méthode de mesure. Afin d'estimer un ordre de grandeur des incertitudes dues à celle-ci, des cales étalons d'angle ont été utilisés afin de comparer les valeurs réelles de leurs angles à celles mesurées. L'une de ses mesures ainsi que les valeurs des écarts mesurés sont représentés sur la figure 6.23. Comme pour la mesure de la position des phalanges, les angles ont été mesurés à partir des images obtenues d'un appareil photo. La moyenne des écarts entre les angles des gabarits réels et ceux mesurés est de $-0,02°$, avec des valeurs comprises entre $-2,1°$ et $+1,17°$. Les écarts mesurés avec le prototype sont compris entre $-1°$ et $+1°$ pour la première phalange et $-0,5°$ et $+2,9°$ du même ordre de grandeur que les erreurs estimées avec les gabarits. Ces incertitudes de mesures correspondent à celle que l'on retrouve dans les autres comparaisons présentées précédemment.

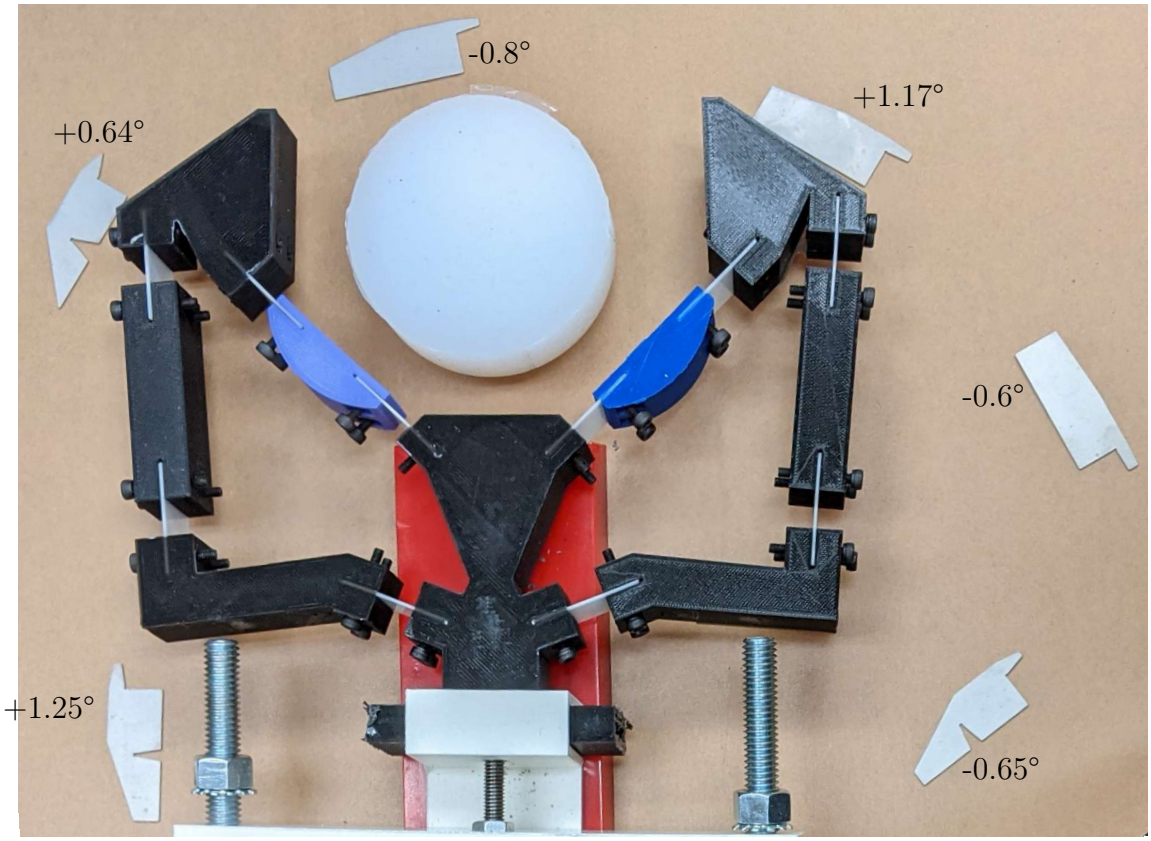


Figure 6.23 Mesure de gabarit

CHAPITRE 7 DISCUSSION GÉNÉRALE

La présentation des travaux sur les doigts sous-actionnés dans la revue de littérature a mis en avant le manque de travaux théoriques sur la conception de pince sous-actionnés pour la saisie d'objet déformable. Les conceptions comprenant généralement des présentations de résultats empiriques de saisie par des pinces non-optimisées. Les travaux présentés dans ce manuscrit visent à répondre à ce manque, mais il reste encore plusieurs défis à relever et certains aspects n'ont pas encore été discutés. Plusieurs limites du modèle doivent être par exemple encore étudiées. Dans la suite, des pistes pour les recherches futures sur le sujet sont présentées. Dans cette partie nous allons nous concentrer sur le second modèle qui est présenté dans le chapitre 5, car montrant de meilleur résultat. Il serait quand même intéressant de se replonger dans le premier modèle afin de comprendre, avec plus de recul, pourquoi un tel écart existent entre les deux approches.

7.1 Développement du modèle de l'objet en trois dimensions

Les modèles présentés précédemment ont été étudiés pour des cas en deux dimensions avec des hypothèses sur le frottement et la gravité ce qui limite leurs plages d'application. Comme évoqué dans le chapitre 5 des modèles masse-ressort 3D incluant la masse ont été présentés récemment dans la littérature. Par exemple, dans [164] le modèle présenté peut être utilisé afin de calculer les valeurs des raideurs en fonction du matériau utilisé et ainsi simuler la rigidité de l'objet. Dans cet article, une formule afin de distribuer la masse de l'objet à chaque point est aussi proposée. De la même manière que les valeurs des raideurs, les masses dépendent des volumes adjacents aux points, ainsi qu'à la masse volumique. On va donc pouvoir utiliser les résultats de ces études pour simuler l'interaction de pince avec des objets plus complexes.

Dans cette section un modèle complet en trois dimensions de l'objet va être détaillé avec la prise en compte du poids de l'objet. Celui-ci n'a pas encore été validé par comparaison avec des résultats d'expériences ou avec des analyses éléments finis, ni même vérifié. Ce pourquoi il est inclus dans la section Discussion. Dans un premier temps les équations reliant les déplacements des points du MMR et les forces qui leur sont appliquées (sans contact avec un objet extérieur tel qu'une pince) vont être détaillées.

7.1.1 Pour un ressort :

La figure 7.1 représente le déplacement d'un ressort (en trait gras) entre sa position initiale 0 (entre A_0 et B_0) non contrainte et la position i (entre A_i et B_i). La longueur initiale sera notée l_0, la variation de longueur du ressort Δl. De plus, le vecteur de norme Δl et de même direction que le ressort à l'état i sera noté $\boldsymbol{\Delta l}$ dans la suite du texte, c'est-à-dire :

$$\boldsymbol{\Delta l} = \Delta l \begin{bmatrix} \sin(\beta_i)\cos(\alpha_i) \\ \sin(\beta_i)\sin(\alpha_i) \\ \cos(\beta_i) \end{bmatrix}. \tag{7.1}$$

Les angles β_0 et β_i représentent les angles entre l'axe $\boldsymbol{z}$ et les vecteurs $\boldsymbol{A_0B_0}$ et $\boldsymbol{A_iB_i}$, respectivement. Les angles α_0 et α_i représentent les angles entre l'axe $\boldsymbol{x}$ et les vecteurs $\boldsymbol{A_0B_0}$ et $\boldsymbol{A_iB_i}$ projetés dans le plan P normal à $\boldsymbol{z}$ passant pas $\boldsymbol{A}_0$, respectivement. $\boldsymbol{u}_A$ et $\boldsymbol{u}_B$ sont les vecteurs déplacements de l'état 0 à l'état i comme représenté sur la figure 7.1 des points A et B, respectivement.

La première étape est de définir la variation de longueur du ressort en fonction des déplacements des extrémités du ressort. C'est-à-dire entre Δl et $\boldsymbol{u}_A$, $\boldsymbol{u}_B$.

On suppose A_0 l'origine du repère. On note $\boldsymbol{p}_C$ le vecteur allant de A_0 vers C. On pose $\boldsymbol{R}(\boldsymbol{z}, \alpha)$ la matrice de rotation d'angle α autour de l'axe $\boldsymbol{z}$ telle que :

$$\boldsymbol{R}(\boldsymbol{z}, \alpha) = \begin{bmatrix} \cos(\alpha) & -\sin(\alpha) & 0 \\ \sin(\alpha) & \cos(\alpha) & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{7.2}$$

et $\boldsymbol{R}(\boldsymbol{n}, \beta)$ la matrice de rotation d'un angle β autour d'un axe $\boldsymbol{n}$ (à calculer à l'état 0 et à l'état i) telle que :

$$\boldsymbol{R}(\boldsymbol{n}, \beta) = \begin{bmatrix} n_x^2(1-c_\beta)+c_\beta & n_xn_y(1-c_\beta)-n_zs_\beta & n_xn_z(1-c_\beta)-n_ys_\beta \\ n_xn_y(1-c_\beta)+n_zs_\beta & n_y^2(1-c_\beta)+c_\beta & n_yn_z(1-c_\beta)-n_xs_\beta \\ n_xn_z(1-c_\beta)-n_ys_\beta & n_yn_z(1-c_\beta)+n_xs_\beta & n_z^2(1-c_\beta)+c_\beta \end{bmatrix}, \tag{7.3}$$

avec :

$$\boldsymbol{n} = \begin{bmatrix} n_x & n_y & n_z \end{bmatrix}^T = \frac{(\boldsymbol{p}_B - \boldsymbol{p}_A) \times \boldsymbol{z}}{\|(\boldsymbol{p}_B - \boldsymbol{p}_A) \times \boldsymbol{z}\|}, \tag{7.4}$$

ainsi que $c_\beta = \cos(\beta)$ et $s_\beta = \sin(\beta)$.

Dans la suite, on posera :

$$\boldsymbol{R}_0 = \boldsymbol{R}(\boldsymbol{z}, -\alpha_0)\boldsymbol{R}(\boldsymbol{n}_0, -\beta_0), \tag{7.5}$$

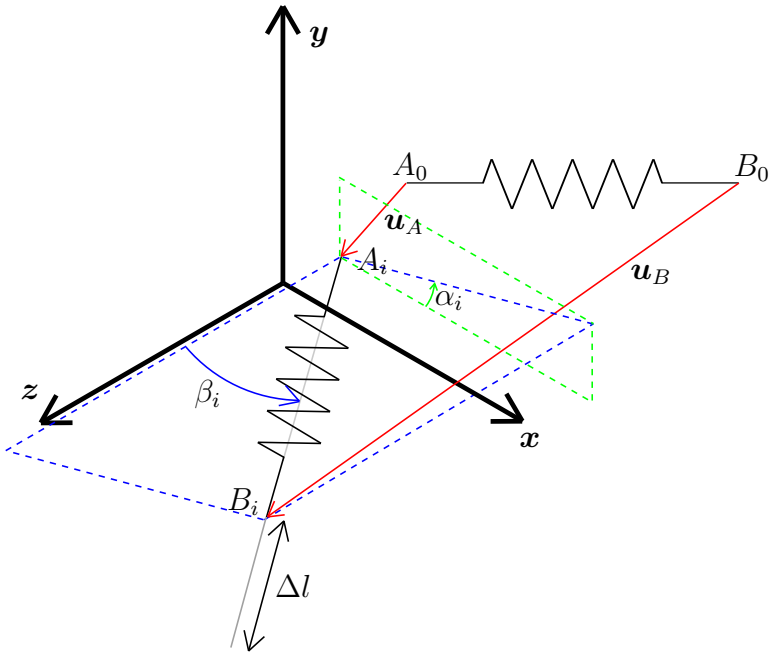


Figure 7.1 Déplacement d'un ressort d'une position 0 à une position i

$$\boldsymbol{R}_i = \boldsymbol{R}(\boldsymbol{z}, -\alpha_i)\boldsymbol{R}(\boldsymbol{n}_i, -\beta_i), \tag{7.6}$$

$$\boldsymbol{R}_0 - \boldsymbol{R}_i = \begin{bmatrix} g_1 & g_2 & g_3 \\ . & . & . \\ . & . & . \end{bmatrix}, \tag{7.7}$$

et :

$$\boldsymbol{R}_0 = \begin{bmatrix} h_1 & h_2 & h_3 \\ . & . & . \\ . & . & . \end{bmatrix}. \tag{7.8}$$

En utilisant une boucle géométrique, on obtient :

$$\Delta l = \boldsymbol{a}^T \boldsymbol{u} + l_0 b, \tag{7.9}$$

avec :

$$\boldsymbol{a} = \begin{bmatrix} h_1 - g_1 \\ h_2 - g_2 \\ h_3 - g_3 \\ g_1 - h_1 \\ g_2 - h_2 \\ g_3 - h_3 \end{bmatrix}, \tag{7.10}$$

$$b = g_1 l_0 \sin(\beta_0)\cos(\alpha_0) + g_2 l_0 \sin(\beta_0)\sin(\alpha_0) + g_3 l_0 \cos(\beta_0), \tag{7.11}$$

et

$$\boldsymbol{u} = \begin{bmatrix} \boldsymbol{u}_A \\ \boldsymbol{u}_B \end{bmatrix}. \tag{7.12}$$

À partir de l'équation 7.9 et de la loi de Hooke, on obtient en posant K_{AB} la valeur de raideur du ressort :

$$\begin{bmatrix} \boldsymbol{f}^A \\ \boldsymbol{f}^B \end{bmatrix} = \begin{bmatrix} \boldsymbol{S}_{AB} & -\boldsymbol{S}_{AB} \\ -\boldsymbol{S}_{AB} & \boldsymbol{S}_{AB} \end{bmatrix} \begin{bmatrix} \boldsymbol{u}_A \\ \boldsymbol{u}_B \end{bmatrix} - K_{AB}\boldsymbol{a} l_0 b, \tag{7.13}$$

tel que :

$$\boldsymbol{S}_{AB} = -K_{AB} \begin{bmatrix} (h_1 - g_1)^2 & (h_1 - g_1)(h_2 - g_2) & (h_1 - g_1)(h_3 - g_3) \\ (h_1 - g_1)(h_2 - g_2) & (h_2 - g_2)^2 & (h_2 - g_2)(h_3 - g_3) \\ (h_1 - g_1)(h_3 - g_3) & (h_2 - g_2)(h_3 - g_3) & (h_3 - g_3)^2 \end{bmatrix} = \boldsymbol{S}_{BA}. \tag{7.14}$$

7.1.2 Pour n points :

Dans le cas général où un objet est discrétisé avec n points sur son contour et un point central fixe (noté ici avec l'indice 0), l'équation précédente peut se généraliser. Cette équation va dépendre de la forme de la discrétisation de l'objet, en particulier des positions des ressorts reliant les masses. Deux points peuvent par exemple être reliés à un nombre différent d'autres points. Dans le cas où un point i est relié à q autres points $(i_1, \ldots, i_q)$ et en prenant en compte sa masse m_i et le vecteur accélération de la pesanteur $\boldsymbol{g}$, l'équation reliant les déplacements des points à la force appliquée sur le point i devient :

$$\boldsymbol{f}^i = \begin{bmatrix} \boldsymbol{S}_{0,i} + \sum_{j=1}^{q} \boldsymbol{S}_{i_j,i} & -\boldsymbol{S}_{i_1,i} & \ldots & -\boldsymbol{S}_{i_q,i} \end{bmatrix} \begin{bmatrix} \boldsymbol{u}_i \\ \boldsymbol{u}_{i_1} \\ \vdots \\ \boldsymbol{u}_{i_q} \end{bmatrix} - \boldsymbol{P}_i \begin{bmatrix} K_{i,0} b_{i,0} l^0_{i,0} \\ K_{i,i_1} b_{i,i_1} l^0_{i,i_1} \\ \vdots \\ K_{i,i_q} b_{i,i_q} l^0_{i,i_q} \end{bmatrix} + m_i \boldsymbol{g}, \tag{7.15}$$

avec $\boldsymbol{P}$ la matrice telle que :

$$\boldsymbol{P} = \begin{bmatrix} h_{1,i,0} - g_{1,i,0} & h_{1,i,i_1} - g_{1,i,i_1} & \dots & h_{1,i,i_q} - g_{1,i,i_q} \\ h_{2,i,0} - g_{2,i,0} & h_{2,i,i_1} - g_{2,i,i_1} & \dots & h_{2,i,i_q} - g_{2,i,i_q} \\ h_{3,i,0} - g_{3,i,0} & h_{3,i,i_1} - g_{3,i,i_1} & \dots & h_{3,i,i_q} - g_{3,i,i_q} \end{bmatrix} . \tag{7.16}$$

Cette équation sera à calculer pour tous les points de l'objet.

7.1.3 Modèle des forces de contact d'une pince sous-actionnée en trois dimensions :

Si un modèle en deux dimensions de l'objet ne peut dans tous les cas être satisfaisant, il en va de même du modèle de la pince. Une pince peut par exemple avoir plus que deux doigts, dans des plans différents. De plus des systèmes pour faire varier les plans des doigts peuvent exister [56, 182] afin de faire tourner les doigts autour de la paume et ainsi créer des systèmes avec des mouvements d'abduction/adduction. Dans la suite, l'adaptation du modèle des doigts en trois dimensions va être discutée, suivi de l'interaction avec l'objet. On considérera chaque doigt dans un plan orthogonale à la paume.

Comme dans le cas en deux dimensions présenté à la section 5 on ne peut pas résoudre le problème avec la méthode actuelle tirée de [37] et il faudra donc aussi dans ce cas séparer virtuellement les phalanges. De plus, l'étude des doigts sous-actionnés a généralement été réalisée dans un plan car les pinces ne contenait que deux doigts face à face, pour un cas en trois dimensions, on va donc chercher à étendre le modèle pour d doigts. L'étude de plusieurs doigts en trois dimensions peut se faire de la même manière qu'un seul en considérant d'abord chaque plan individuel correspondant à chaque doigt, puis en utilisant des matrices de projections adaptées à chaque doigt, avant de finalement relier les forces de contact des phalanges aux points objet en contact. Les modèles présentés pour un doigt sont donc toujours valables.

On suppose que la paume est dans le plan $\mathcal{D}_p$ comprenant les axes $\boldsymbol{x}$ et $\boldsymbol{z}$. On explicite les équations pour un doigt h dans un plan $\mathcal{D}_{f_h}$ orthogonal à $\mathcal{D}_p$ comprenant les axes $\boldsymbol{x}_{f_h}$ et $\boldsymbol{y}$. $\boldsymbol{x}_{f_h}$ est construit à partir d'une rotation de l'axe $\boldsymbol{x}$ de γ_{f_h} autour de l'axe $\boldsymbol{y}$. La figure 7.2 illustre les repères pour une phalange i d'un doigt h.

L'équation reliant les forces de contacts sur le doigt à l'actionnement est la même que dans le cas en deux dimensions, mais sera à calculer dans différents plans suivant le doigt, c'est-à-dire :

$$\boldsymbol{f_p}^T \boldsymbol{J} = \boldsymbol{t}^T \boldsymbol{T}, \tag{7.17}$$

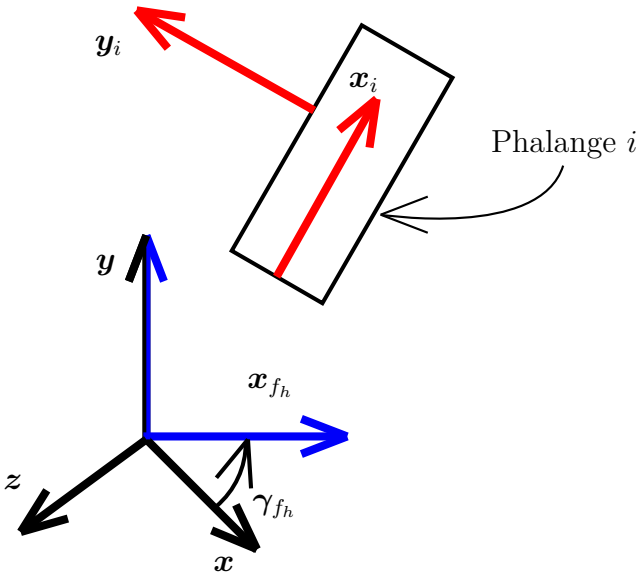


Figure 7.2 Modèle de la phalange i du doigt j.

avec $\boldsymbol{f_p}$ ($\in \mathbb{R}^{p\times 1}$) le vecteur des forces de contacts du doigt h tel que :

$$\boldsymbol{f_p} = \begin{bmatrix} f_1 \\ \vdots \\ f_p \end{bmatrix}. \tag{7.18}$$

$\boldsymbol{J}$ ($\in \mathbb{R}^{p\times p}$) la matrice jacobienne est elle à adapter en prenant en compte les frottements :

$$\boldsymbol{J} = \begin{bmatrix} \boldsymbol{r}_{11}^T\boldsymbol{x}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T\boldsymbol{x}_2 & \boldsymbol{r}_{22}^T\boldsymbol{x}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1n}^T\boldsymbol{x}_n & \boldsymbol{r}_{2n}^T\boldsymbol{x}_n & \boldsymbol{r}_{3n}^T\boldsymbol{x}_n & \dots & \boldsymbol{r}_{nn}^T\boldsymbol{x}_n \end{bmatrix} - \boldsymbol{\mu}_y \begin{bmatrix} \boldsymbol{r}_{11}^T\boldsymbol{y}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T\boldsymbol{y}_2 & \boldsymbol{r}_{22}^T\boldsymbol{y}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1n}^T\boldsymbol{y}_n & \boldsymbol{r}_{2n}^T\boldsymbol{y}_n & \boldsymbol{r}_{3n}^T\boldsymbol{y}_n & \dots & \boldsymbol{r}_{nn}^T\boldsymbol{y}_n \end{bmatrix} - \boldsymbol{\mu}_z \begin{bmatrix} \boldsymbol{r}_{11}^T\boldsymbol{z}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T\boldsymbol{z}_2 & \boldsymbol{r}_{22}^T\boldsymbol{z}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1n}^T\boldsymbol{z}_n & \boldsymbol{r}_{2n}^T\boldsymbol{z}_n & \boldsymbol{r}_{3n}^T\boldsymbol{z}_n & \dots & \boldsymbol{r}_{nn}^T\boldsymbol{z}_n \end{bmatrix}. \tag{7.19}$$

$\boldsymbol{r}_{ki}$ est le vecteur allant de du point A_k au $i^{ième}$ point de contact et $\boldsymbol{x}_i$ le vecteur unitaire de même axe que la phalange i dans la direction de la phalange suivante (en s'éloignant de la paume). $\boldsymbol{y}_i$ est le vecteur normal à la phalange i et $\boldsymbol{z}_i$ construit à partir de $\boldsymbol{x}_i$ et $\boldsymbol{y}_i$. Le vecteur $\boldsymbol{t}$ et la matrice $\boldsymbol{T}$ sont comme dans les sections précédentes à adapter en fonction de l'architecture du doigt concerné.

7.1.4 Interactions objet-pince en trois dimensions

Pour la suite, on pose n le nombre de points pour discrétiser l'objet, n_c le nombre de points en contact avec le doigt, n_{nc} celui des points sans contact ($n = n_c + n_{nc}$), ainsi que p le nombre de phalanges virtuelles, p_c le nombre de phalanges virtuelles en contact ($p_c = n_c$), p_{nc} le nombre de phalanges sans contact ($p = p_c + p_{nc}$). À partir de l'équation 7.17, nous allons devoir faire le lien entre les forces de contact des phalanges et les forces dans l'objet. En fonction des contacts entre les phalanges et les points de l'objet, nous pouvons écrire :

$$\boldsymbol{f}_p = \boldsymbol{D}_x\boldsymbol{f}_o, \tag{7.20}$$

$$\boldsymbol{f}_p = \boldsymbol{D}_y\boldsymbol{f}_o, \tag{7.21}$$

$$\boldsymbol{f}_p = \boldsymbol{D}_z\boldsymbol{f}_o. \tag{7.22}$$

On pose d_{ij}^x, d_{ij}^y et d_{ij}^z les éléments des matrices de projection $\boldsymbol{D}_x$, $\boldsymbol{D}_y$ et $\boldsymbol{D}_z$ respectivement (avec $i \in \mathbb{R}^p$ et $j \in \mathbb{R}^{3n}$), tel que :

$$d_{ij}^x = \begin{cases} \frac{1}{\cos\gamma_{f_h}\cos(\sum_{j=1}^{p}\theta_i)} & \text{si } j \equiv 2(3),\ \cos\gamma_{f_h}\cos(\sum_{j=1}^{p}\theta_i) \neq 0, \\ & \text{et que la } i^{\text{ème}} \text{ phalange touche le } \frac{j+2}{3}^{\text{ème}} \text{ point}, \\ 0 & \text{sinon}, \end{cases} \tag{7.23}$$

$$d_{ij}^y = \begin{cases} \frac{1}{\sin\sum_{j=1}^{p}\theta_i} & \text{si } j \equiv 1(3),\ \sin\sum_{j=1}^{p}\theta_i \neq 0, \\ & \text{et que la } i^{\text{ème}} \text{ phalange touche le } \frac{j+1}{3} \text{ point}, \\ 0 & \text{sinon}, \end{cases} \tag{7.24}$$

$$d_{ij}^z = \begin{cases} \frac{1}{\sin\gamma_{f_h}\cos(\sum_{j=1}^{p}\theta_i)} & \text{si } j \equiv 0(3),\ \sin\gamma_{f_h}\cos(\sum_{j=1}^{p}\theta_i) \neq 0, \\ & \text{et que la } i^{\text{ème}} \text{ phalange touche le } \frac{j}{3} \text{ point}, \\ 0 & \text{sinon}. \end{cases} \tag{7.25}$$

Ce qui nous donne finalement en combinant 7.22 et 7.17 :

$$\boldsymbol{f}_o^T \boldsymbol{D}_x^T \boldsymbol{J} = \boldsymbol{t}^T \boldsymbol{T}, \tag{7.26}$$

$$\boldsymbol{f}_o^T \boldsymbol{D}_y^T \boldsymbol{J} = \boldsymbol{t}^T \boldsymbol{T}, \tag{7.27}$$

$$\boldsymbol{f}_o^T \boldsymbol{D}_z^T \boldsymbol{J} = \boldsymbol{t}^T \boldsymbol{T}. \tag{7.28}$$

Dans le cas où il y a des valeurs inconnues dans la matrice $\boldsymbol{J}$, c'est-à-dire liée à une phalange sans contact, celles ci n'interviennent pas dans le calcul car multipliées par un élément nul des matrices $\boldsymbol{D}_x$, $\boldsymbol{D}_y$ et $\boldsymbol{D}_z$.

Le système nous donne $3p_c$ équations venant des lignes où les forces de contact sont non nulles projetées suivant l'axe $\boldsymbol{x}$, $\boldsymbol{u}$ et l'axe $\boldsymbol{z}$. Tandis qu'il y a trois fois les mêmes équations pour les forces de contact aux phalanges sans contact, donc on n'a seulement p_{nc} équations en plus. Pour simuler l'interaction entre la pince et l'objet, il faudra calculer les équations 7.26, 7.27 et 7.28 pour les d doigts. Ensuite, à ces équations il faut rajouter les équations des forces au niveau des points de l'objet qui ne sont pas en contact. pour cela, on pose :

$$\boldsymbol{0} = \boldsymbol{D_w} \boldsymbol{f}_o, \tag{7.29}$$

avec d_{ij}^w ($i \in \mathbb{R}^{3n}$ et $j \in \mathbb{R}^{3n}$) les éléments de la matrice $\boldsymbol{D}_w$ définis tel que :

$$d_{ij}^w = \begin{cases} 1 & \text{si le point } i + 2/3 \text{ n'est pas en contact avec un doigt et } i \equiv 2(3), \\ 1 & \text{si le point } i + 1/3 \text{ n'est pas en contact avec un doigt et } i \equiv 1(3), \\ 1 & \text{si le point } i/3 \text{ n'est pas en contact avec un doigt et } i \equiv 0(3), \\ 0 & \text{sinon.} \end{cases} \tag{7.30}$$

L'équation 7.29 ajoute $3n_{nc}$ équations.

Finalement, on obtient

- $3n_{nc}$ équations pour les points non en contact avec un doigt grâce à 7.29.
- $\sum_{i=1}^{d} 3p_{c,i}$ équations pour les points en contact avec une phalange.
- $\sum_{i=1}^{d} p_{nc,i}$ équations pour les phalanges sans contact.

tandis que l'on a :

- $3n_{nc}$ inconnues : coordonnées des points non en contact avec un doigt.
- $\sum_{i=1}^{d} p_i$: angles des phalanges
- $\sum_{i=1}^{d} 2p_{c,i}$: distances de contact suivant la longueur et suivant la largeur sur la phalange

On a donc :

- $3n_{nc} + \sum_{i=1}^{d} 3p_c + \sum_{i=1}^{d} p_{nc}$ équations
- $3n_{nc} + \sum_{i=1}^{d} p + \sum_{i=1}^{d} 2p_c$ inconnues

Or $p = p_{nc} + p_c$, donc :

$$3n_{nc} + \sum_{i=1}^{d} 3p_c + \sum_{i=1}^{d} p_{nc}(\text{équations}) = 3n_{nc} + \sum_{i=1}^{d} p + \sum_{i=1}^{d} 2p_c(\text{inconnues}). \tag{7.31}$$

Finalement, le système d'équation devrait pouvoir être résolu afin de simuler les interactions en trois dimension. L'effet de la paume pourrait aussi être pris en compte. Il restera ensuite à valider le modèle par comparaison de simulations avec des expériences.

7.1.5 Prise en compte du frottement

Pour la prise en compte du frottement, bien que l'on observe déjà celui-ci entre le silicone et le PLA, il serait intéressant d'observer d'autres cas présentant plus ou moins de frottement entre la pince et l'objet afin d'effectuer d'autres comparaisons avec le modèle numérique. Dans le chapitre 5 une méthode est proposée afin de prendre en compte le frottement. Celle-ci peut se mettre sous la forme de l'algorithme présenté dans la figure 7.3. Si l'on fait l'hypothèse que l'on a un seul point de contact, pour le calcul d'une position i pour un couple d'actionnement $T_{a,i}$, on fait d'abord l'hypothèse qu'une distance de contact ne change pas, puis on évalue la

position du système, ainsi que la valeur du coefficient de frottement μ_i. Dans ce cas-là, k_i n'est plus une inconnue, mais remplacée par μ_i. Si la valeur μ_i est dans le cône de frottement, on pourra considérer la position calculée comme possible, dans le cas contraire il faudra faire évoluer k_i et refaire le calcul jusqu'à trouver une position possible. La méthode n'a pas encore été testée. Elle paraît difficilement applicable pour plusieurs points de contact simultanés, car si pour une position toutes les forces ne sont pas dans le cône de frottement, pour quelles contacts va t-on considérer qu'il y a eu du glissement : tous les contacts avec un coefficient trop grand, certains parmi ceux-là ou faut-il en considérer d'autres [38] ?

7.2 Application à d'autres architectures de doigts

7.2.1 Applications aux *soft grippers*

Il reste encore plusieurs aspects à travailler en particulier sur le modèle des doigts. Les doigts présentés et étudiés avaient par exemple un nombre limité de DDL, qu'ils soient pleinement ou sous-actionnés. Et dans les cas où l'on souhaite étudier des pinces plus flexibles qui se rapprochent plus des tentacules que des doigts humains, les *Soft-Gripper*, leur infinité de DDL ne permet pas encore d'adapter les équations. Leurs modèles sont complexes à mettre en place et peu présents dans la littérature. De plus, il va être difficile de travailler avec une infinité de liaison rotoïdes pour approcher la souplesse de tentacule en prenant en compte aussi les interactions avec l'objet. Des approximations avec un nombre raisonnable de liaisons pourraient être possibles. Finalement, s'il est possible de déterminer une matrice de transmission approchée pour une architecture de *Soft-Gripper*, il faudra une attention toute particulière lors de la séparation des phalanges présentée dans le chapitre 5. Cette amélioration en plus de l'adaptation du modèle en trois dimensions pourrait permettre dans le futur d'optimiser des pinces telles que celle présentée dans [183] qui est utilisée pour saisir une framboise.

7.2.2 Tailles des phalanges

En effet, le problème de la taille des phalanges des *Soft-Grippers* soulève un problème que l'on retrouve aussi avec les architectures sous-actionnées. Lors de l'interaction entre les points du MMR et des phalanges, si plusieurs points de contact sont proches, cela peut entraîner des phalanges virtuelles petites relativement à la largeur du doigt. Si c'est le cas, une hypothèse est que lors de la recherche d'une position d'équilibre, cela peut être compliqué pour le programme de trouver une bonne configuration car une grande variation des angles relatifs entre les phalanges sera parfois trouvée lors de la résolution des équations. Ce qui ne correspondra

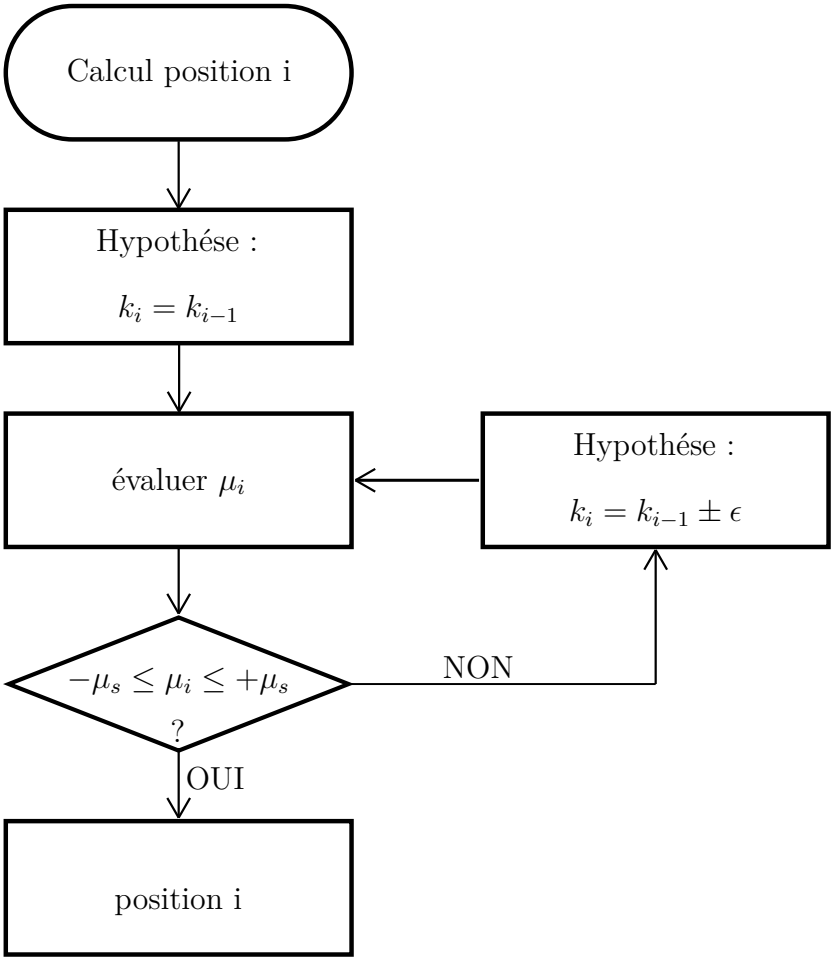


Figure 7.3 Algorithme pour prendre en compte le frottement dans le modèle numérique.

pas à la réalité où les phalanges réelles ne se plient pas. De plus ce problème est accentué plus la largeur des phalanges est grande. Il faudra donc étudier les longueurs minimales que peuvent prendre les phalanges, virtuelles ou non, afin de ne pas rencontrer ce problème ou changer le procédé de séparation des phalanges.

En effet, afin de séparer une phalange virtuellement et de se limiter à un contact par phalange les axes des liaisons additionnelles sont placées à mi-distance entre deux points de contact. Cette découpe a été choisie arbitrairement, mais d'autres choix sont possibles. Par exemple, choisir de placer des liaisons afin d'avoir les points de contact à l'extrémité des phalanges correspondantes. Encore une fois, ce procédé est à tester à l'aide de comparaison entre simulations et expériences. Il y a potentiellement des emplacements optimaux pour ces liaisons, mais aussi pour les valeurs de raideurs des ressorts virtuels qui leurs sont associés. En effet on souhaite que ces ressorts soient le plus rigides possibles, mais une trop grande valeur peut entraîner des problèmes de conditionnement de matrice. Dans les simulations présentées dans cette thèse les valeurs de raideur des liaisons compliantes réelles étaient chacune même ordre et après calibrations des raideurs virtuelles, les simulations correspondaient aux mesures expérimentales. L'établissement d'une procédure afin de déterminer des raideurs fictives applicables sans problème de conditionnement à tous les cas permettrait de ne pas avoir à refaire cette étape de calibration à chaque changement de liaisons compliantes.

7.2.3 Évaluation de plusieurs mécanismes de transmission

Pour les architectures classiques utilisant des phalanges rigides l'application du modèle ne nécessite que le développement des matrices de transmission qui relie l'actionnement aux DDL. Il est donc possible de mener des optimisations avec différentes architectures possibles de transmission afin d'évaluer leurs performances avec des objets déformables et fragiles. Les tests réalisés dans cette thèse ont été limités à un seul mécanisme de transmission principalement, même si plusieurs validations ont été réalisées grâce à ADAMS avec des mécanismes à tendons, tout en travaillant avec deux, voire plus rarement trois, phalanges. Évaluer l'augmentation du nombre de phalanges et les autres transmissions sur les forces de saisies, l'adaptation et la déformation de l'objet donneraient des indications sur les directions à prendre lors de la conception d'une pince sous-actionnée. Pour des applications spécifiques, une pince comprenant des doigts avec des architectures différentes pourrait être envisageable.

7.2.4 Optimisation des liaisons compliantes

Enfin un autre point sur lequel la thèse ne s'est pas penché, peut-être plus négligeable par rapport aux objectifs est l'optimisation de la géométrie des liaisons compliantes utilisées dans

le doigt. À la fois en terme de raideur, car pour facilité la fabrication les mêmes géométries pour chaque articulation ont été utilisées, ce qui limitait les défauts de fabrication, mais aussi la forme car les liaisons entre les phalanges peuvent pincer les objets potentiellement déformés.

7.3 Limite, amélioration et évaluation du modèle de l'objet

Au niveau du modèle de l'objet, plusieurs cas doivent encore être expérimentés et comparés à des simulations afin d'observer les limites dans lesquelles on pourra considérer le MMR suffisamment précis relativement à son application.

7.3.1 Homogénéité de la raideur de l'objet

Dans la littérature, le modèle points-ressorts a toujours été testé pour simuler des cas où l'objet est constitué d'un matériau isotrope et donc à une rigidité homogène. C'est aussi ce qui a été considéré dans cette thèse et ce n'est bien sûr que rarement le cas en pratique. Pourtant les formules de raideur développées dans la littérature comme l'équation 4.1 devraient déjà permettre de considérer un assemblage de différents matériaux isotropes avec des modules de Young distincts. Dans la formule proposée, les raideurs sont calculées en fonction de la géométrie des surfaces adjacentes et du module de Young du matériau qui les constitue. C'est pourquoi, il est possible de calculer, dans les cas en deux dimensions, des ressorts à la jonction entre deux matériaux de rigidités différentes. Ces formules pourraient donc être appliquées pour des solides anisotropes que l'on peut séparer en plusieurs sous-solides isotropes. On peut ainsi imaginer simuler la déformation d'une tomate en considérant deux divisions de celle-ci. D'abord la partie charnue avec un premier module de Young, puis les parties centrales avec un second module de Young, tout en positionnant des points du modèle aux limites de chaque solide. La figure 7.4(a) illustre un exemple d'une séparation, avec une partie extérieur d'une couleur avec un module de Young E_1 et une partie intérieure d'une autre couleur avec E_2. Par contre, cette approche n'est pas adaptée pour les matériaux anisotropes où une telle séparation n'est pas réalisable, comme le présente la figure 7.4(b). Dans ce cas-là, d'autres approximations peuvent être envisagées pour corriger cette limite, par exemple en calculant des modules de Young moyens ou encore en distribuant la rigidité selon l'emplacement des ressorts si cela est possible. D'autres cas plus complexes : non-linéarité et viscosité en fonction du temps, applicable dans le cas de tissus naturel par exemple, seront aussi des paramètres que l'on pourra essayer d'intégrer dans le modèle. Ils seront aussi à prendre en compte si l'on fait une analyse dynamique du système. Dans tous les cas, des mesures expérimentales devront confirmer la précision de ces approches.

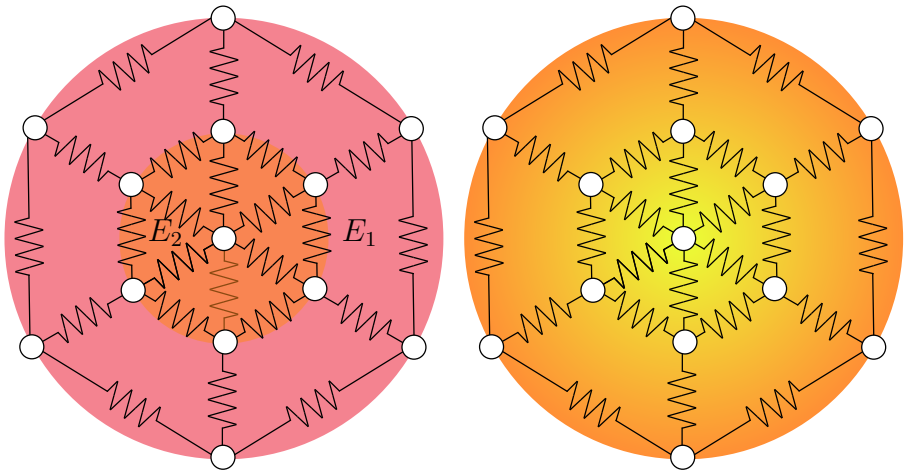


Figure 7.4 Application du modèle point-ressort à un objet (a) constitué de deux matériaux isotropes, (b) anisotrope.

7.3.2 Déformations importantes et précision

Les modèles présentés sont des approximations de la réalité à simuler et leur utilisation s'est fait pour des cas où il n'ait pas souhaitable d'avoir beaucoup de déformation. C'est pourquoi le modèle n'a pas été testé pour de grandes déformations et dans ce cas là la précision du modèle n'est pas connu. Ce scénario va à l'encontre de ce qui est l'avantage principal des doigts sous-actionnés, mais on peut imaginer la conception d'une pince permettant à la fois d'imposer beaucoup de force à un endroit localisé d'un objet avec une saisie par pincée, tandis qu'une prise enveloppante va, elle s'adapter et ainsi réduire les déformations. Il serait donc intéressant dans le futur de tester par simulation avec ANSYS ou par expérimentations, quelle est la précision du MMR en fonction du taux de déformation de l'objet. Ceci est valable aussi pour les modèles du doigt compliant.

7.3.3 Expérimentations avec des objets de formes différentes

Dans le chapitre 4, le modèle présenté à été comparé à des simulations éléments finis pour des objets carrés. Le modèle présentant de moins bons résultats que celui présenté dans le chapitre 5, les travaux n'ont pas continué sur celui-ci, pourtant le second modèle lui n'a été utilisé que pour simuler des objets cylindriques. La comparaison avec des objets de formes différentes sera intéressante pour valider le modèle, mais aussi pour étudier certains problèmes du modèle de l'objet. En effet, premièrement avec le modèle de Van Gelder [118] si un triangle qui discrétise l'objet a un angle intérieur supérieur à 90°alors l'approximation des raideurs

des ressorts peut amener à une valeur négative, ce qui ne fait pas de sens physiquement. Aussi dans la suite de ce projet, d'autres équations ou discrétisations que celles de Van Gelder pourraient être testées. De plus, pendant une simulation il se peut que des ressorts se chevauchent, c'est un cas qui n'a pas encore été rencontré et qui devrait poser problème. En particulier pour utiliser la technique des phalanges virtuelles qui n'est pas applicable s'il n'y a pas de distance entre deux points de contact. Si ce cas se présente, cela peut amener à revoir la discrétisation en cours de simulation.

7.3.4 Simulation de cas particuliers

Dans les discussions, il a été évoqué que le modèle devait être testé avec d'autres scénarios (3D, autre transmission, objet quelconque, etc.). Ceci illustre ce que l'on souhaite finalement : un modèle qui fonctionne pour le plus de situations possibles. Le modèle devra donc être éprouver dans un maximum de cas, en particulier ceux où l'on pourrait avoir des problèmes, comme des problèmes de convergence de calcul par exemple. Dans l'annexe C, des problèmes de convergence liés à la taille des phalanges virtuelles sont ainsi mis en avant. Ceux-ci pourraient s'expliquer par des problèmes de conditionnement de la matrice jacobienne J dans laquelle apparaissent les longueurs des phalanges et les distances de contact. Si l'on prend un exemple de situation où l'on a des points de contact très proches sur une phalange proximal et un seul point sur une phalange distale, la matrice jacobienne fera donc intervenir des longueurs très petites par rapport aux autres. Il faudra donc trouver une solution afin d'éviter ou de corriger ce problème de convergence.

7.3.5 Optimisation du modèle points-ressorts

Enfin des évaluations du MMR pourraient permettre dans le futur d'établir un processus de discrétisation de l'objet. En effet, suivant si le compromis que l'on cherche entre la précision et la rapidité, il est possible de placer les points du modèle différemment. Dans le travail présenté, il a été choisi d'utiliser un point central et des points sur le contour arbitrairement, mais on peut se demander si l'on peut se passer du point central ou s'il faut à l'inverse ajouter plus de point. Dans *Point-Spring Modeling of the Grasp of a Deformable Object by an Angular Gripper* un article soumis pour le Symposium du *Canadian Committee for the Theory of Machines and Mechanisms* de 2021 la précision du modèle a été évaluée en faisant varier le nombre de triangles discrétisant l'objet afin de trouver le nombre optimal de point sur le contour. Il peut être possible d'aller encore plus loin et de placer des points sur le contour seulement dans la zone de contact et de calculer les raideurs des ressorts en fonction. Aussi, des triangles ont été utilisés, il est peut-être aussi possible d'utiliser d'autres polygones

toujours avec les mêmes équations.

Enfin, il serait intéressant de comparer le modèle points-ressorts optimisé à plusieurs modèles éléments-finis (simplifié, non simplifié) afin d'observer les avantages et les inconvénients de chacun, en particulier en terme de temps de calcul. Avoir des ordres de grandeur de la réduction (voir l'augmentation) des temps de calcul permettrait ensuite de faire des choix sur les stratégies à adopter pour étudier des mécanismes. Ainsi d'autres applications que des optimisations pourraient être imaginées faisant intervenir potentiellement d'autres résultats calculés à partir de ce modèle. Le calcul des forces de contact pourrait par exemple être un résultat exploitable.

7.4 Interaction objet-doigts

Les limites et améliorations présentées précédemment portaient sur le modèle développé à partir d'une étude quasi-statique car cela était le point central de ce doctorat, il y a pourtant d'autres aspects non évoqués qui sont encore à noter. En particulier, la prise en compte de force extérieure ou d'effet de la dynamique, en plus de la friction.

En effet, lors de la simulation de la saisie d'un objet par une pince, la stabilité va être aussi évaluée par la capacité de la pince à contrer d'éventuelles perturbations. D'autant plus qu'une pince sert généralement à manipuler des objets. Cet aspect n'a pas encore été étudié et il est encore tôt avant de pouvoir prétendre l'inclure dans cette étude, mais de futurs études pourront se servir de ce modèle pour tester des architectures en appliquant des forces extérieures aux systèmes qui pourraient potentiellement éjecter l'objet. Ceci permettra de se passer de la fabrication de prototype pour expérimenter. Il serait aussi intéressant d'étudier les interactions en prenant en compte la dynamique et de ne plus supposer des mouvements lents [184]. Dans ce cas, il pourrait être intéressant d'inclure dans le modèle masse-ressort de l'objet des amortisseurs.

De plus, le modèle a été développé pour des optimisations, donc il est attendu que sa compilation soit le plus rapide possible. C'est pourquoi un travail est encore possible afin d'accélérer le calcul du programme.

CHAPITRE 8 CONCLUSION ET RECOMMANDATION

Les travaux présentés dans cette thèse, ainsi que l'objectif fixé au début de ce doctorat portent sur le développement d'un modèle permettant l'estimation rapide de l'interaction entre des objets mous et des pinces mécaniques, en particulier les systèmes sous-actionnés compliants. Ce dernier chapitre consiste en une synthèse des travaux, puis des perspectives qui pourront être réalisées en fonction des limitations actuelles.

8.1 Synthèse des travaux

L'étude de la saisie d'objet par des pinces mécaniques en ne considérant que des objets indéformables ne constitue qu'un cas très particulier de la saisie. En effet, on ne peut considérer un tissu humain, par exemple, rigide afin d'optimiser une pince chirurgicale. De plus, la méthode éléments finis, processus classique pour le calcul précis de déformations lors de l'interaction de solides déformables, est une méthode qui nécessite de la puissance informatique et de long temps de calcul. C'est pourquoi, le développement de modèles simples et rapides afin d'étudier des cas plus généraux, avec des objets déformables, est indispensable pour la conception de préhenseurs adaptés à la saisie d'objets fragiles. Ceux-ci permettront l'optimisation des architectures, en particulier celle de doigts mécaniques sous-actionnés qui ont l'avantage d'adapter la position de leurs phalanges à la forme des objets, afin de minimiser des déformations. Cette thèse s'intéresse donc au développement de méthodes pour l'estimation des interactions entre pince et objet déformable et leurs applications. Le choix s'est porté sur un modèle masse-ressort pour modéliser l'objet. Utilisé habituellement dans le domaine des effets spéciaux pour son rendu réaliste mais imprécis, c'est un modèle simple et rapide. Comme dit précédemment, les pinces qui ont été considérées sont des pinces dites sous-actionnées, car lorsque l'on souhaite saisir un objet déformable, il est généralement souhaitable de garder l'intégrité physique de ce dernier afin de ne pas l'endommager. Le sous-actionnement est donc un candidat adapté à cet usage.

Dans un premier temps, deux méthodes sont présentées afin de simuler l'interaction entre le modèle d'une pince et celui d'un objet. Le premier fait intervenir les énergies potentielles, tandis que le second utilise directement les forces de contact. Pour ce dernier, les modèles des forces de contact développés dans de précédent travaux ne pouvaient être directement appliqués, car limités à un point de contact par phalange. Ce qui n'est pas envisageable dans le cas où un objet va se déformer sur les surfaces d'un doigt. Il a donc fallu développer une technique pour n'avoir qu'un contact par doigt. Chaque modèle a ensuite été évalué par

comparaison avec des résultats d'analyse éléments finis et avec des mesures expérimentales. Les deux ont amené à la publication d'article.

Finalement, suivant les résultats obtenus et la comparaison de la précision des modèles, le meilleur est choisi afin d'optimiser une architecture de pince, et plus spécifiquement le mécanisme de transmission car c'est la partie qui distribue l'actionnement aux phalanges. L'objectif est de minimiser la déformation d'un objet que la pince doit saisir. Certaines contraintes ont été intégrées, comme le fait de concevoir une pince qui ne nécessitera aucune étape de montage et donc être réalisable en une seul élément. Cette optimisation a aussi amené à la soumission d'un article.

L'apport concret de cette thèse est donc finalement un modèle numérique pour simuler rapidement les interactions objets déformables/pinces mécaniques. Ceci appuyé par la fabrication d'un prototype de pince compliante optimisé afin de saisir un objet en le déformant au minimum. On peut imaginer utiliser ces travaux dans le futur pour la fabrication de pinces chirurgicales sans dent ou celle de pinces industrielles montées sur un robot où utilisées comme outil à main, voire même pour d'autres systèmes que des pinces, telles que des rétracteurs.

8.2 Limitations de la solution proposée

Les modèles d'interaction pince-objet présentés dans cette thèse ont plusieurs limitations. Premièrement, leurs utilisations impliquent au préalable la connaissance d'un modèle pour l'architecture des pinces considérées. Pour les mécanismes pleinement et sous-actionnés, ceux-ci sont relativement aisés à développer, cependant pour les architectures de types *Soft-Robotic*, ceux-ci sont complexes et peu présentes dans la littérature. En outre, si l'on considère qu'un *Soft-Gripper* a une infinité de DDL (sans approximation), le modèle actuel ne permet pas de simuler les interactions. Deuxièmement, le modèle actuel de l'objet se base sur une approximation. Il a été choisi, car simple et rapide, ce qui permet un compromis entre la précision et la rapidité par rapport à la méthode des éléments finis communes. Lorsque l'on recherche une précision parfaite, cette dernière méthode sera à privilégier,

8.3 Améliorations futures

Principalement, des validations du modèle dans plus de scénario font parties des améliorations futures à ce travail. D'abord, des validations du modèle en trois dimensions pourraient ainsi permettre plus d'applications de celui-ci. Par ailleurs, le système a pour l'instant été testé avec des formes simples en deux dimensions (des cercles pour des cylindres, des rectangles pour des pavés dans le cas du premier modèle), dans des cas où l'on souhaite minimiser la

déformation. Il sera donc intéressant dans le futur d'éprouver le modèle dans d'autres cas, comprenant d'autres géométries, avec un objet subissant de grandes déformations, etc.

La programmation du modèle doit aussi être améliorée afin d'être plus robuste pour limiter les problèmes de convergence. De plus, la construction du programme n'a pas été optimisée à son maximum. Malgré sa rapidité actuelle, celle-ci peut donc encore être améliorée.

Finalement, les travaux présentés dans cette thèse portent seulement sur la saisie d'objet, la suite logique de ce travail va donc être sur la manipulation de ses objets. Une étude dynamique du système serait intéressante à prendre en compte.

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ANNEXE A ARTICLE 4 : PRELIMINARY DESIGN OF A REVOLUTE TO PRISMATIC MORPHING COMPLIANT JOINT

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Abstract

The reduction of weight and size of mechanisms are important and difficult challenges considering portability, energy efficiency, and simplicity of fabrication. One of the solutions to address these issues consists of mechanisms with variable topology for which the mobility of the output is a succession of several simpler elementary motions. This change of mobility allows for achieving complex motions without necessitating a complicated design where many actuators or types of mechanical transmissions are required. Indeed, these variable topology mechanisms, also referred to as morphing mechanisms, have the ability to change their output motion throughout their workspace. Hand tools, medical devices and aerospace robotic end-effectors are potential applications of this technology. In this paper, conceptual designs of such a revolute to prismatic morphing joint and its implementation using compliant hinges are proposed. Additionally, performance indexes pertaining to the desired output motion are proposed. First, a pseudo-rigid body model of a design candidate is presented and simulations of this model are compared with finite element analyses to ensure accuracy. Then, several design features are quantitatively evaluated in order to propose improvements for future versions of the design. Finally, an early prototype illustrates the potential and feasibility of the proposed design as well as a possible application.

La réduction de l'encombrement et du poids des mécanismes sont des problèmatiques majeures et difficiles en tenant compte des questions de portabilité, de minimisation de la consommation d'énergie et de facilité de fabrication. Une solution apportée consiste à utiliser des mécanismes à topologie variable. La mobilité en sortie de ces mécanismes est une succession de mouvements élémentaires simples. Cette caractéristique permet d'accomplir des mouvements complexes sans avoir besoin d'une architecture compliquée avec plusieurs actionneurs ou mécanismes de transmission. En effet, ces mécanismes à topologie variable ont la capacité de changer leur mouvement de sortie dans leur espace de travail. Dans cet article, un concept préliminaire de mécanisme passant d'une articulation rotoïde à une articulation prismatique,

ainsi que sa réalisation avec des liaisons compliantes sont proposés. De plus, des indices de performance sont présentés. En premier lieu, des simulations d'un modèle pseudo-rigide du design sont présentées et comparées avec des analyses par éléments finis afin de s'assurer de la validité du modèle. Plusieurs élements du design sont évalués ensuite quantitativement afin de proposer des améliorations pour de futurs versions. Finalement, une version préliminaire d'un prototype illustre le potentiel du design proposé, ainsi qu'une application possible.

Introduction

Mechanisms with variable topology (MVT) have the ability to change their mobility during operation by switching their topologies [1]. These switches or variations occur through modifications of the number, type and/or configuration of their joints. This explains why they are sometimes referred to as *reconfigurable* mechanisms with variable joints in contrast to invariant topological structures. They are also referred to as morphing systems by biomimetism inspiration [2]. Similarly, morphing wings can be designed that change their aerodynamic properties. In that case the focus is often placed on material properties, e.g. in [3], or in their topologies [4, 5].
One of the main interests of MVT is to be able to achieve complex and shifting motions with the simplest mechanical structure possible. Indeed, motions usually realized by several mechanisms can be achieved with a single MVT solution. Similarly to underactuated mechanisms, MVT have the advantage of decreasing the number of actuators required for complicated tasks [37]. Cost, weight and size are thereby reduced and thus, MVT are a promising way to design mechanisms in particular in medical or aerospace fields where footprints and payloads are significant issues. In [7] a leg mechanism is shown that switches from one topology, which is advantageous when crossing an obstacle, to another stabler and faster when there is no obstacle, a feature closely related to the design presented in [9, 23]. Another example is the adjustable plier with a Revolute (R) to Prismatic (P) variable joint presented in [10]. Finally, in [11] a 3-URU parallel manipulator was shown to have multi-operational mode throughout its workspace, e.g. spatial translations, orientational wrist, planar motion.

MVT are mostly created with two topologies states [12] and can transition from one to the other. A simple and typical example is a four-bar linkage switching from an RRRR mechanisms to an RRRP [13–15] structure as illustrated in Fig. A.1. As can be seen in this figure, one of the revolute joints can morph into a prismatic joint with a continuous motion at the input and it allows for a smooth transition between a rotation and a translation of the output link. This morphing joint is also called a variable kinematic joint. From the MVT

classification proposed in [16] the mechanism illustrated in the previous figure is a type-II MVT because the variable topology is only due to a variation of the joint geometry. A direct switching between both configurations is an ideal case, but in practice there is always a transition and the latter poses a major design challenge. The variable joint morphs from a one degree-of-freedom (DOF) topology to a combination of two topologies with one DOF each in the transitional position. From this state, the system could finally become a RRRR or a RRRP mechanism without preference posing a potential issue as to how to control the motion. Other morphing techniques are presented in [2] while in [17] planar five-bar mechanisms are presented. Spatial systems with variable joints also exist, see for instance [18, 19].

To simplify manufacturing and assembly of linkages, mechanisms can be built relying on structural deformation to approximate lower-pair kinematic pairs. This technique is commonly referred to as compliant mechanism design [81, 82]. Two main approaches to design and build a compliant mechanism can be highlighted based on the literature. First, designs with flexure hinges where a localized thinning of the material is used to approximate revolute joints. Since the resulting deformation of the mechanisms is mainly concentrated in these thinned out parts of the mechanism, this is designated as localized compliance. Conversely, relying on distributed compliance is a second approach used when the mobility is created with an elongated section [22]. The latter technique allows typically for a wider range of motion because the stress undergone by the material due to the deformation is distributed to a greater volume of material. However, its accurate modeling is more challenging than with localized compliance. Both of these approaches are well known solutions to create compliant linkages [23]. To model the motions of compliant mechanisms using classical machine theory with added lumped compliances, Pseudo-Rigid Body Models (PRBMs) have been developed and make it easier to simulate and optimize compliant designs. Compliant revolute joints for instance can be very accurately modeled using PRBM and are most commonly found in

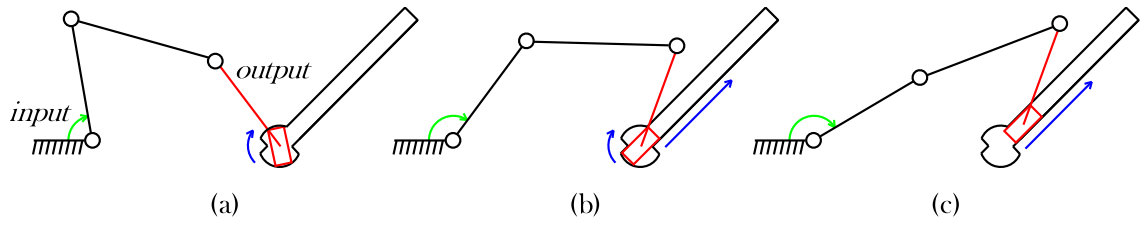


Figure A.1 Motions of a RRRR/RRRP morphing mechanism : (a) RRRR topology phase, (b) transition between topologies and (c) RRRP phase.

practice. Prismatic joints on the other hand do not yet have a simple compliant equivalent although a series of thin parallel beams is often used in Micro-Electro-Mechanical Systems (MEMS) to constrain a seismic mass to a linear path [24, 25]. In [26] for example, a bistable micromechanism is introduced where a block of material is guided with two pairs of parallel thin beams of material that allow for that block to translate only. Another technique to transition from one mobility to another relies on constraint singularities as shown in [27] were a compliant gripper was presented which can operate in two different modes, namely its two jaws can either move along a circular translation (pinch mode) or rotate with respect to the base.

While underactuated compliant mechanisms are widespread for grasping applications [28, 80, 132], to the best of the authors' knowledge there is no report of a compliant MVT design in the literature. The purpose of this paper is to present a proposition for the latter with a morphing compliant linkage transitioning from a revolute to a prismatic output motion while minimizing the intermediate phase. First, a general design is proposed in Section A followed by the introduction of performance indexes in Section A. Then, results obtained with a numerical model are compared to dynamic simulations of that model (using MSC ADAMS) and also to results from a finite element analysis software (ANSYS) in Section A. Later, improvements of the design are proposed and their performances are evaluated in Section A by the presentation of a very first prototype illustrating the potential of this mechanism in Section A and concluding with the presentation of a potential application in Section A.

Modelling

The conceptual design proposed in this work is presented in Fig. A.2. The output is the motion at the right part of the mechanism (in red). The left part is assumed to be fixed to the ground. This mechanism is a compliant implementation of the linkage illustrated in the previous section. Between the ground link at the left and the output at the right, there are two kinematic chains. The first (upper) one is constituted by a thin beam (in green) shaped as a "V" and the lower one (in blue) includes two living flexural hinges. The objective is that the right output part of the mechanism, when actuated by a force at the point P, has first an almost pure rotational motion and then, switches to a linear translation. It is expected that during the first part of this motion, movement of the output is mostly due to the flexibility of the thin beam and the right flexural hinge, the former spreading out as the input force is not of sufficient magnitude yet to bend the left flexural hinge. Then, when the thin beam

is completely unfolded and stretched, this beam and the bottom link act as two guides to create the linear motion again similarly to the linkage presented in Section A.

Figure A.3 illustrates the pseudo-rigid body model (PRBM) of the proposed compliant design at rest as well as the two ideal motions to reach when actuated. From the initial state (in Fig. A.3(a)), when the mechanism is actuated by a force as illustrated in Fig. A.3(b), the output of the MVT (namely the link from A_2 to A_5 as illustrated) is rotating around a fixed point. Then, as the force is increasing the trajectory of the output changes from a circular to a linear path (Fig. A.3(c)). In each of these phases a virtual four-bar linkage can be used to approximate the motion of the mechanism. The transition between both phases is controlled by the stiffnesses of the lumped springs and the geometry of the design. In the first phase, A_2 is almost fixed since the stiffness K_1 is sufficient to prevent the rotation around A_1 due to the actuation force. Links from A_1 to A_2 and from A_1 to A_3 are thus motionless. The vertices of the equivalent four-bar linkage in that phase are A_2, A_3, A_4 and A_5 as illustrated in dashed red lines in Fig. A.3(b). Then, as the actuation force needed for a rotation around A_4 increases, the distance between A_3 and A_5 remains almost constant, which then defines the vertices of the second equivalent four-bar linkage for the translation phase, namely : A_1, A_2, A_5 and A_3. The last case is equivalent to designs with two pairs of parallel beams as in [31]. In the following, A_1 is assumed to be the origin of the coordinate system and the force applied at point P has a constant orientation of $\alpha = -45°$ with respect to the $\boldsymbol{x}$-axis. This angle value is chosen in order for that force to be in the direction of first the rotation and then, the translation. The magnitude of the input force is assumed to be continuously incremented.

Positions and orientations of the output can be calculated from the parameters of the model and the input force increment by using the virtual work principle. To follow this approach, variations of the elastic potential energy stored in each of the springs modeling the compliant mechanism (with stiffness K_s with $s = 1, \ldots, 5$ in the PRBM) is first calculated as :

$$\Delta E_{K_s,i} = \frac{K_s}{2} \Delta(\theta_{s,i} - \theta_{s,0})^2 \tag{A.1}$$

where the 0 and i subscripts indicate positions at the initial state and at the i^{th} increment of the actuation respectively. The stiffness K_s of the s^{th} spring is estimated with an approximated model from the literature [81,82]. The thin beam is modelled as two flexible segments with forces and moments at their ends. Thus, the model with a combined force-moment end

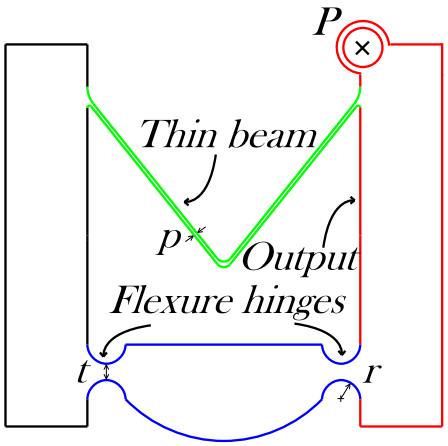


Figure A.2 Conceptual design of a morphing compliant R→P mechanism.

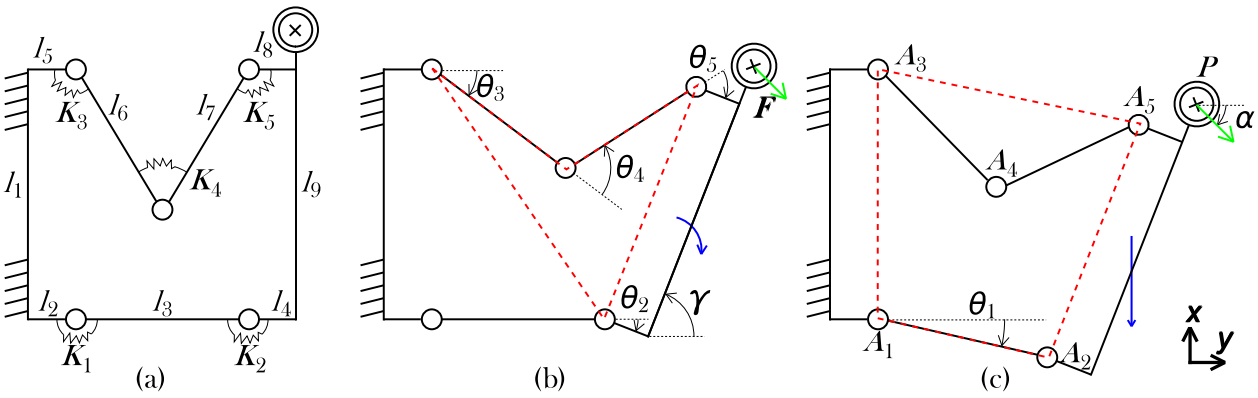


Figure A.3 PRBM of the MVT a) at rest, and desired motion during b) the rotation and c) the translation phase.

loading from [81] is used for this kinematic chain. It is worth mentioning that the accurate modelling of an initially curved compliant beam does not have a simple solution and the model chosen here is only a rough approximation.

Then, the virtual work of the actuation force $\boldsymbol{F}$ is :

$$\delta W_{F_i} = -F_i \delta \sqrt{(x_i - x_0)^2 + (y_i - y_0)^2} \cos(\boldsymbol{F}^T \boldsymbol{u}), \tag{A.2}$$

where F_i is the magnitude of the input force at the i^{th} increment, (x_i, y_i) the position of P the point of application of the force at the i^{th} increment and $\boldsymbol{u} = [x_i - x_{i-1}, y_i - y_{i-1}]^T$ the displacement vector between the position at the i^{th} and $(i-1)^{th}$ increments. At each step i and for each value of the input force F_i, the total value δW_i of the work from the actuation force δW_{F_i} and the variation of the potential energy of the internal springs is evaluated for several positions around the previous configuration. Then, the actual position of the mechanism corresponding to the minimum of W_i previously calculated is found, i.e. :

$$\delta W_i = \delta W_{F_i} + \sum_{s=1}^{5} \Delta E_{K_s,i}. \tag{A.3}$$

This allows to compute the values of the joint angles for each of the input force magnitude.

Performance Metrics

The motion of the compliant MVT can be decomposed in three phases and thus, performance indices are also established according to this decomposition in order to evaluate the efficiency of the proposed design. In the first phase, the performance of the system can be measured according to the capability of the output to produce a pure rotation around a fixed point. This phase is assumed to take place between a zero input force $F_0 = 0$ at rest and a threshold magnitude F_a at the a^{th} increment. Thus, based on a proposed set of design parameters (geometry and material), the first step is to search for the best coordinates (X_0, Y_0) of the center of rotation of the output link and the distance R between this point and point P of the linkage. These parameters are found with the simplex search method implemented in MATLAB to establish the best circle segment that fits the path of the evaluated design in the selected range of force. Then, μ_1 the first performance metric can be computed as :

$$\mu_1 = \sum_{i=1}^{a} \left(\sqrt{(\boldsymbol{r}_{P_i}^T \boldsymbol{x} - X_0)^2 + (\boldsymbol{r}_{P_i}^T \boldsymbol{y} - Y_0)^2} - R \right)^2 \tag{A.4}$$

where $\boldsymbol{r}_{P_i}$ is the position of the point P relative to the origin A_1 for the input force magnitude F_i. The previous index evaluates the discrepancy between the ideal calculated circle path and the real trajectory of a selected point of the output link, namely P in the previous equation. In an ideal rotational motion of the output link, all points of the latter will move on a circle around the same fixed point. Therefore, another performance metric to quantify the motion of the output link can be similarly defined characterizing how close to a circle around the same point with coordinates (X_0, Y_0) the point A_6, arbitrarily chosen as the middle of A_2 and A_5, is :

$$\mu_2 = \sum_{i=1}^{a} \left(\sqrt{(\boldsymbol{r}^T_{A_{6,F_i}} \boldsymbol{x} - X_0)^2 + (\boldsymbol{r}^T_{A_{6,F_i}} \boldsymbol{y} - Y_0)^2} - \sqrt{(\boldsymbol{r}^T_{A_{6,F_0}} \boldsymbol{x} - X_0)^2 + (\boldsymbol{r}^T_{A_{6,F_0}} \boldsymbol{y} - Y_0)^2} \right)^2 \tag{A.5}$$

where $\boldsymbol{r}_{A_{6,F_k}}$ is the position vector of point A_6 for the force magnitude F_i. Both of the latter two metrics are defined as sums of squared values to stay positive in all cases.

The second phase, between the input force magnitudes F_a and F_b $(b > a)$ is defined as the transition between the revolute and prismatic compliant joint behaviors. During this phase, the actual motion of the output is a mix between a pure rotation and a pure translation. A perfect mechanism will have an infinitely small transition between both motions and thus, a simple metric to quantify this aspect is :

$$\mu_3 = F_b - F_a \tag{A.6}$$

which is ideally zero.

Finally, during the prismatic joint phase between F_b and F_c the force maximal magnitude $(c > b)$, the output link should have a linear motion and thus, the variation of its orientation (γ indicated in Fig. A.3) must be as close to zero as possible. Therefore, an appropriate performance metric for this phase is :

$$\mu_4 = \sum_{i=b}^{c} (\gamma_{F_i} - \gamma_{F_b})^2 \tag{A.7}$$

where γ_{F_i} is the value of γ for the magnitude F_i. Again, minimal values of this metric are ideal. All the previously defined performances indexes can be used to measure the performance of a particular design and also optimize the compliant mechanism in order to approach the desired motion as much as possible. However, it should be noticed that these metrics are not independent of the selected scale and have all different units (square length, force, and

square rad). Finally, the range of motion of the any evaluated design are measured to ensure that motionless or almost motionless designs are rejected since the magnitude of this range of motion is not explicitly taken into account by the metrics.

Validation

In Figs. A.4 and A.5, a simulation of the numerical model proposed in Section A is compared to results from commercial softwares : a dynamic package (MSC ADAMS) and a finite element solver (ANSYS). The simulated material properties are those of Delrin (Young's modulus $E = 3.1 \cdot 10^9 Pa$ and Poisson's ratio $\mu = 0.35$) and Tab. A.1 lists the PRBM stiffnesses and geometric parameters used as illustrated in Figs. A.2 and A.3. The variable e is the overall thickness of the mechanism in the out-of-the-plane direction (**z**-axis). Figure A.6 illustrates six successive positions along the range of the motion of the MVT and the associated deformations as evaluated with the FEA software. In the numerical model, spring stiffnesses modeling the distributed flexural beam are assumed equal ($K_3 = K_4 = K_5$). In Fig. A.4 the motion of P goes from the top ($F = 0$ N) and moves along the curve toward the lower part of the figure. To compare the evolution of the orientation and the transition between the revolute and prismatic behavior of the mechanism, the figure is annotated with the value of the force at different increment values. A circle centered at (X_0,Y_0) with a R radius (noted Ideal Circle Path) is added as well to illustrate the ideal revolute path at the beginning of the motion, where X_0, Y_0 and R are calculated as presented in Section A. Numerical values obtained are $X_0 = 49.6mm$, $Y_0 = 2.6mm$ and $R = 124.6mm$.

One can see from these results that the simulations from the numerical model and the dynamic software are indeed close, validating the numerical simulation based on the virtual work principle. However, a marked discrepancy can be observed between these curves and the results from the FEA. The difference between these results may be explained by the approximated model of the compliant thin beam. Changing the value of the springs in the PRBM has little effect on the discrepancy which can therefore be traced mainly to the ap-

l_1	l_2	l_3	l_4	l_5	l_6	l_7	l_8	l_9	r	t	p	e	K_1	K_2	K_3	K_4	K_5
95	7	86	7	3.75	68	68	3.7	95	7	6	1.25	1	51.2	51.2	0.2	0.2	0.2

Tableau A.1 Parameters of the simulations presented in Figs. A.4 and A.5 (all lengths are in mm, stiffnesses in N/m).

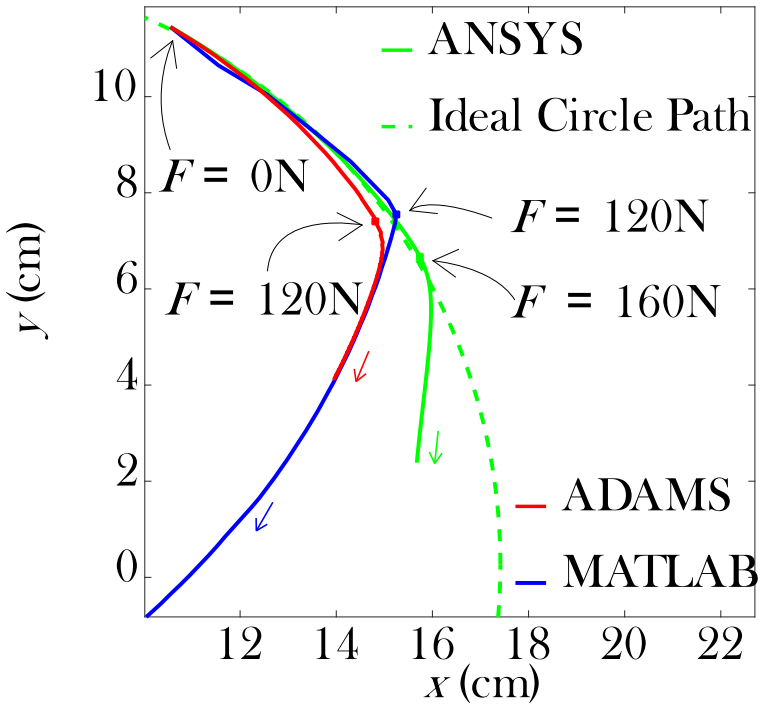


Figure A.4 Simulated motion of P.

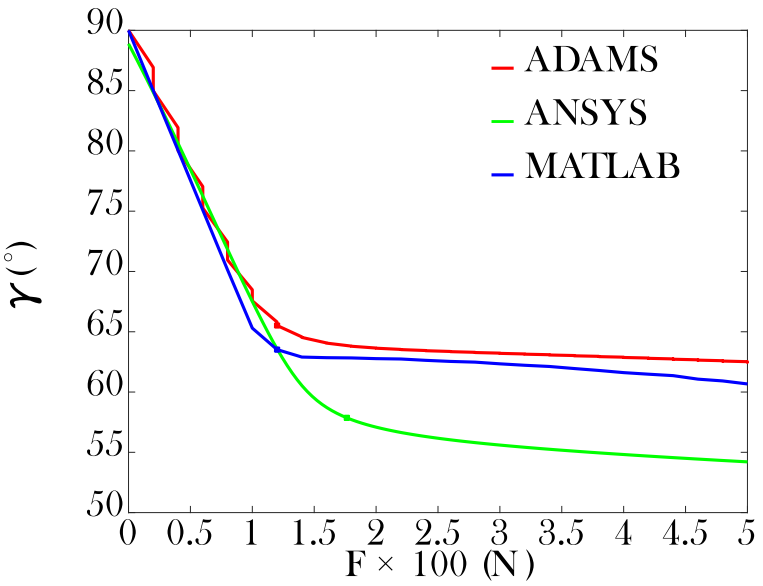


Figure A.5 Orientation of the output.

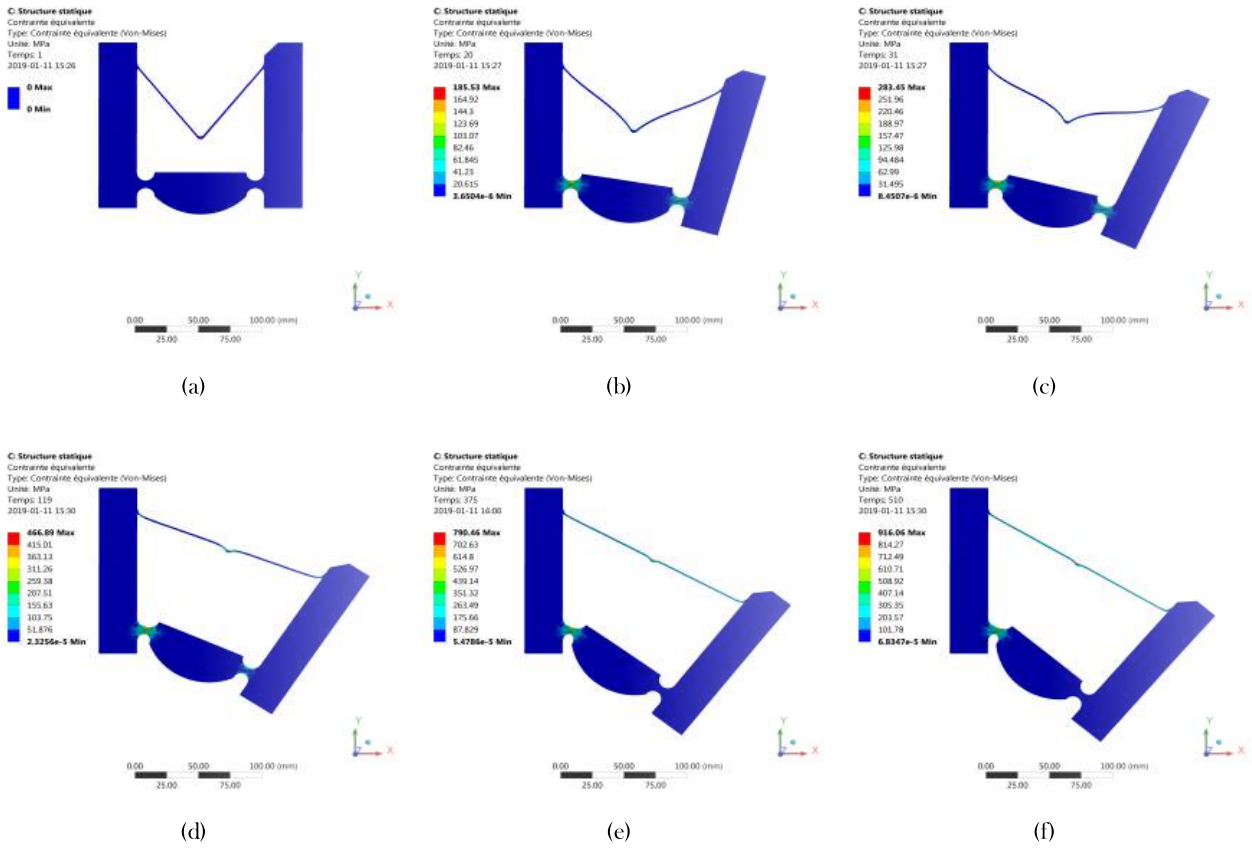


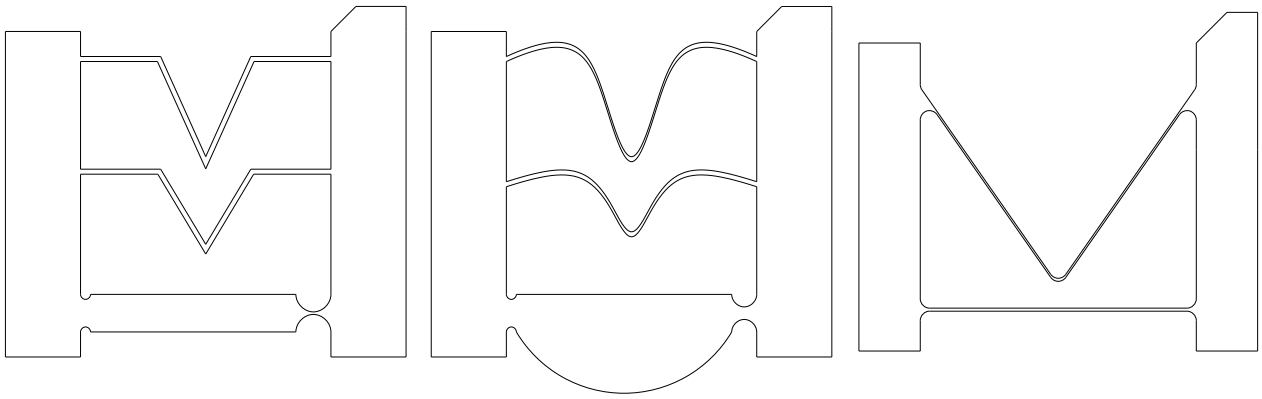
Figure A.6 Deformation of the MVT at (a) $F = 0N$, (b) $76N$, (c) $120N$, (d) $473N$, (e) $1500N$ and (f) $2000N$.

proximation of the geometry. The FEA and the model results in Fig. A.5 show two parts with linear variations of the orientation of the output. The value of the force at the end of the first linear part is F_a while F_b is at the starting point of the second phase. These forces are selected by inspection of the plots. At the beginning, between $F = 0N$ and $F = 120N$ (see Figs. A.6(a), A.6(b) and A.6(c)) the variations are important as expected for a rotational motion and then, between $F = 473N$ and $F = 2000N$ (see Figs. A.6(d), A.6(e) and A.6(f)), these variations becomes quite negligible suggesting that indeed, the movement is close to a linear translation. Also the motion fits a segment of the ideal circle path, even though the radius of the latter is quite large. For design evaluation, a prediction of the transition forces from the numerical model and considering the simulated variation of the orientation is a potentially interesting future development. It should be noticed however that an important issue has been left unaddressed and is quite apparent from the values of the input force magnitudes : the deformations computed by the FEA indicate that the Von Mises stresses in the mechanism are actually higher than the yield limit of Delrin. This issue will be discussed in the next Section.

Improvements

Figure A.7 illustrates other potential design candidates for the MVT. Figure A.8 shows the motions of the point P (noted "Output") and Fig. A.9 the orientations of the output link respectively for these new design candidates. In Fig. A.8, ideal circular paths for each design are illustrated and noted "Circle". These circles are centered at the coordinates (X_0,Y_0) with a radius R whose values were found with the similar methodology as discussed before and used to calculate the performance metric μ_1, see Section A. Table A.2 presents the respective performance indexes. Variation of the orientation is constant at the beginning and at the end, thus transitions force values are computed from the orientation plots to calculate the associated indexes. The end of the revolute joint phase is observed at $F_a = 120N$, $32N$, $128N$ and $0.09N$ for the initial design (ID), design 1 (D1), 2 (D2) and 3 (D3) respectively. Then, the transitions end at $F_b = 473N$, $721N$, $1030N$ and $2.02N$ for these same designs.

In design D1, two distributed compliance beams with identical geometry are used instead of a single one and the left flexure hinge was made stiffer in order to ensure that the bottom link starts to rotate only when the upper link is completely unfolded taking into account the results seen in Fig. A.6(b). The performance indexes table illustrates that the initial design has the weakest performance during the revolute phase (large value of μ_2) and also the prismatic one (highest μ_4 but the difference is less significant than with μ_2). However, it



(a) Design #1 (D1) (b) Design #2 (D2) (c) Design #3 (D3)

Figure A.7 Conceptual designs.

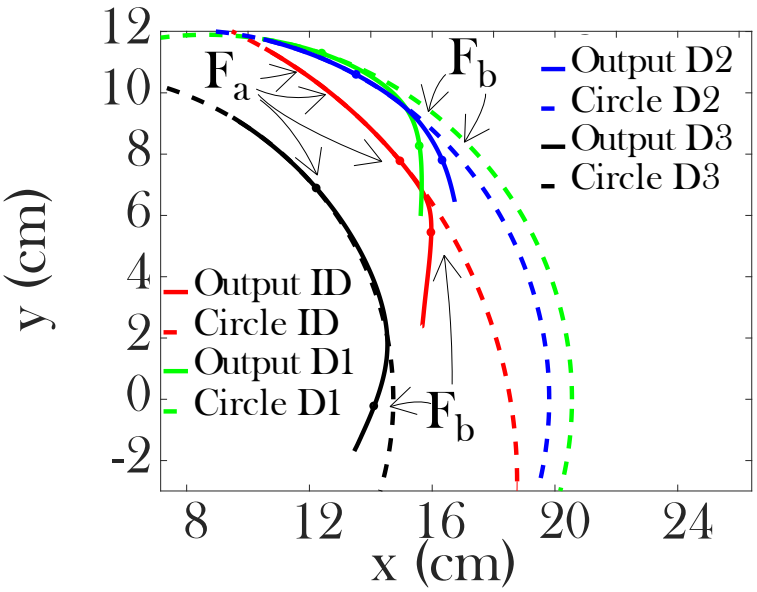


Figure A.8 Motion of the force contact point.

	Initial Design	Design 1	Design 2	Design 3
μ_1	0.0026	0.0030	0.0285	0.0756
μ_2	628	0.01	4	3.4
μ_3	353	689	901	1.93
μ_4	$5.4 \cdot 10^5$	$2.9 \cdot 10^5$	$2.8 \cdot 10^5$	$15 \cdot 10^5$

Tableau A.2 Performance metrics of the designs presented in Fig. A.7.

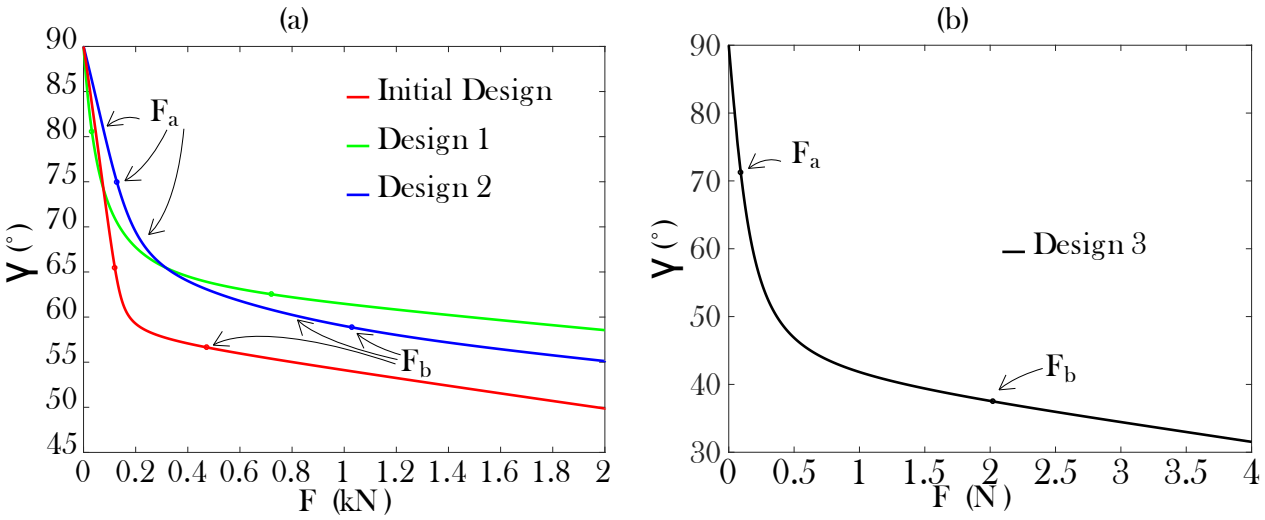


Figure A.9 Orientation of the output.

is the design with the smallest transition from one topology to the other with design D3 and also with the largest range of motion during the first phase. The second design D2 has a mix of several features and its geometry is adapted in order to maintain the Von Mises stress as low as possible. It presents good overall performances during the rotation and translation, but its transition phase is the longest. Finally, the last design (D3) uses a thin compliant lower beam instead of two flexure hinges in order to also decrease the Von Mises stress. The performance of this last design are actually the worst during the prismatic phase in terms of the proposed metric but on the other hand, it is the only one that keeps the stress under a level compatible with manufacturing at least for early experiments, see next Section. The output orientation of this design changes the most when the top beam is fully extended and the force applied at the output pushes that beam closer to the base, which contributed to the overall disappointing values of the metrics. Other potential solutions to deal with excessive stresses during the operation of the linkage is to use another material with a larger range of elastic deformation such as rubber, or to use instead of a thin beam cut from the same block of material an insert with low compliance (a NiTinol sheet or even fabric wires for instance) added to that block of material. This would make manufacturing slightly more complicated but it is a possible solution.

Prototype

Figure A.10 shows different steps during the motion of an early prototype based on the proposed design D3 and also on the Von Mises stresses calculated with the FEA constraining

manufacturing. Figures A.10(a) and (b) respectively show the rest and one of the transition positions. Figure A.10(c) illustrates one position during the translational phase when the top beam cannot extend anymore. The base (the green block on the left of the figures) is clamped to a support and the input force is manually applied (as shown in Fig. A.3) to the green block on the right, i.e. the output. The base and the output of the mechanism (in green) were 3D printed in PLA while the compliant parts are made from laser cut acrylic beams of 0.5 and $1mm$ thickness for the top and the bottom beams respectively. During the beginning of the motion, the V-shaped beam extends while the bottom beam bends only marginally. A large variation of the orientation of the output is observed as expected during the rotational phase. Then, after the transition between motion types, the fully extended V-shaped beam and the compliant beam at the bottom guide the output to keep the orientation as constant as possible and thus, the motion is close to a pure translation. See the multimedia attachment to this paper for a video of the prototype demonstrating the motion. Figure A.11 illustrates the comparison between the trajectory measured with the prototype and results from a FEA of the same geometry. The system being static at several positions because of friction and gravity cannot be neglected with this first prototype and hampers direct comparison between the force applied on the device and the force from the FEA. Thus, the actual orientation of the output link is plotted with respect to the positions of point P relative to $\boldsymbol{y}$-axis in Fig. A.11(b) while the abscissa of the plot was the input force in previous figures. As one can readily tell from the plots in the figure, the fit between measurements and the FEA is excellent.

Application

Finally, a potential application of the mechanism is discussed here where the latter can simplify the design while creating complex motions, namely underactuation in grasping fingers. In [32], unusual kinematic joints are shown to be of interest in the transmission of underactuated fingers. The latter rely on a transmission linkage similar in principle to a differential to drive their many output joints from a single torque source. Hence, underactuated fingers have many degrees of freedom but typically only a single actuator. A desirable feature of this technique is that if one phalanx is physically prevented from moving after having touched an object, the other phalanges will still continue their closing motions. Therefore, a natural enveloping grasp is achieved mechanically without sensors or control. Furthermore, underactuated fingers are commonly implemented as compliant mechanisms due to the ease of manufacturing [132] and modelled using a pseudo-rigid body model with only revolute joints. In Fig. A.12, two examples showing the closing motion of an underactuated finger is presen-

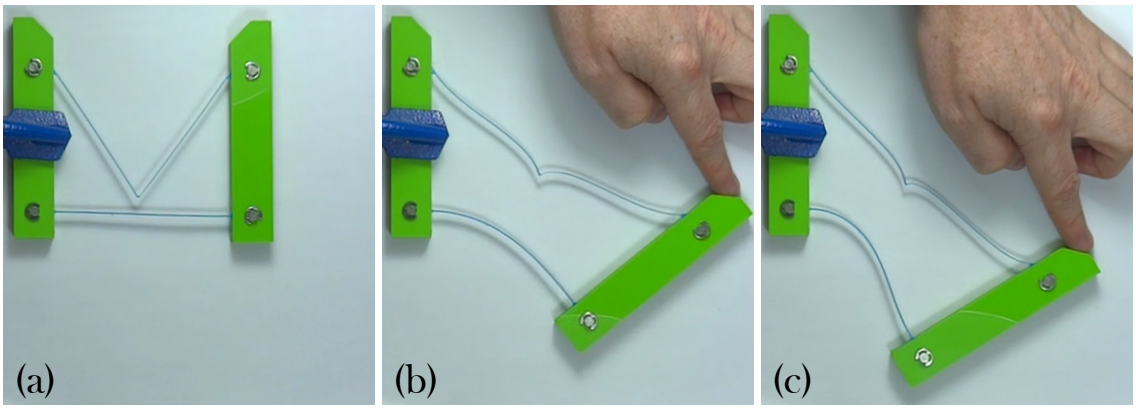


Figure A.10 Prototype of the MVT a) at rest, at the end of b) the rotation and of c) the translation phase.

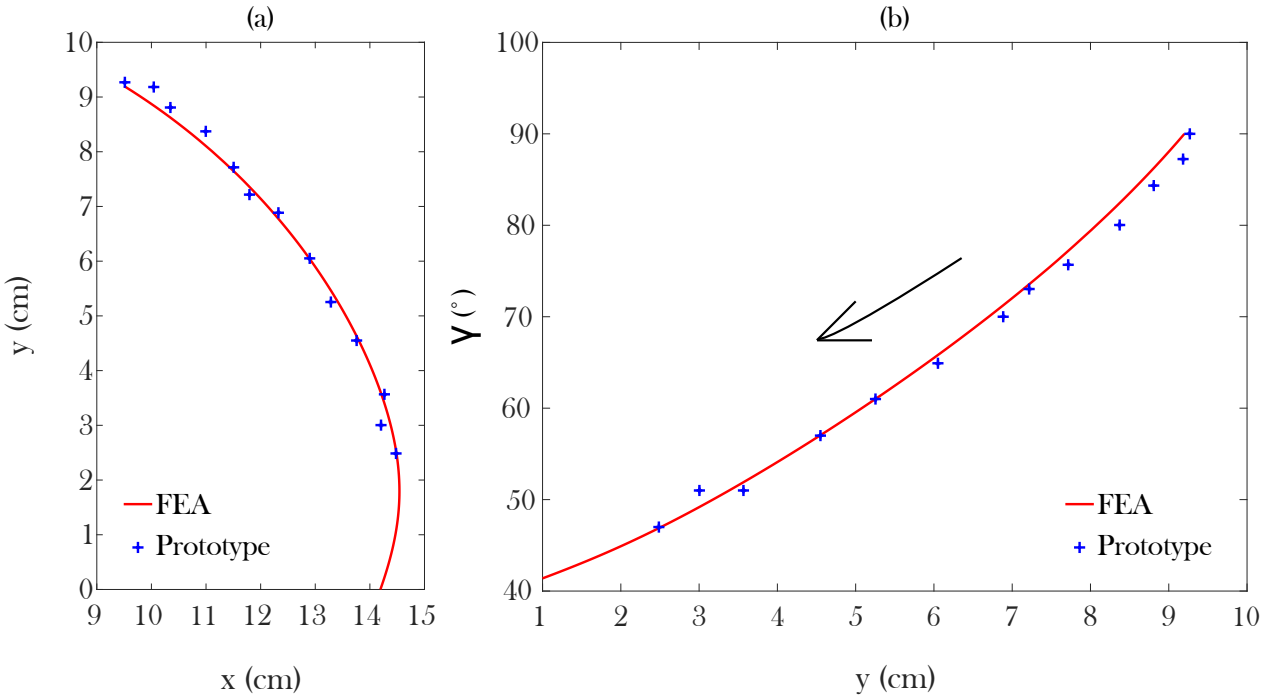


Figure A.11 Comparison between measurements and FEA simulations. a) Motion of the application point of the force, b) orientation of the output.

ted. Both fingers rely on a 5-bar transmission linkage but each with a different topology. In the first case (left column), the actuator is located at the lowest revolute joint whereas in the second case (right column), that joint is replaced with a morphing joint. The resulting closing motions can then be compared and the transition between topologies of the second finger with the morphing joint shows that its distal phalanx does not reach out as much as the initial design. This property can have significant advantages such as avoiding collisions between a pair of opposing finger, or as illustrated in the figure, grasping objects close to obstacles (walls, tables, etc.).

Conclusion

In this paper, the preliminary design of a compliant mechanism which is able to transition the motion of its output link from a rotation to a translation has been presented. This work constitutes a first step towards a more practical implementation and introduced a PRBM modeling the proposed system. This model was validated using a dynamical simulation package but the FEA results seem to point out that the selected PRBM is too coarse to accurately predict the required wide range of deformation. Performance metrics were then defined to help quantify how well the design candidate achieves its task and measure not only how accurately it can produce the desired output motions but also how "fast" it can transition from one to the other. Evaluation of these metrics with an accurate model can lead to optimization for specific desired applications with a chosen motion. Finally, design alternatives are proposed and compared to the initial iteration to highlight the compromise that seems to be required between the accuracy of the generated motion and the size of the transition phase. Finally, a prototype of one of these design alternatives is presented in order to illustrate the motion of the mechanism.

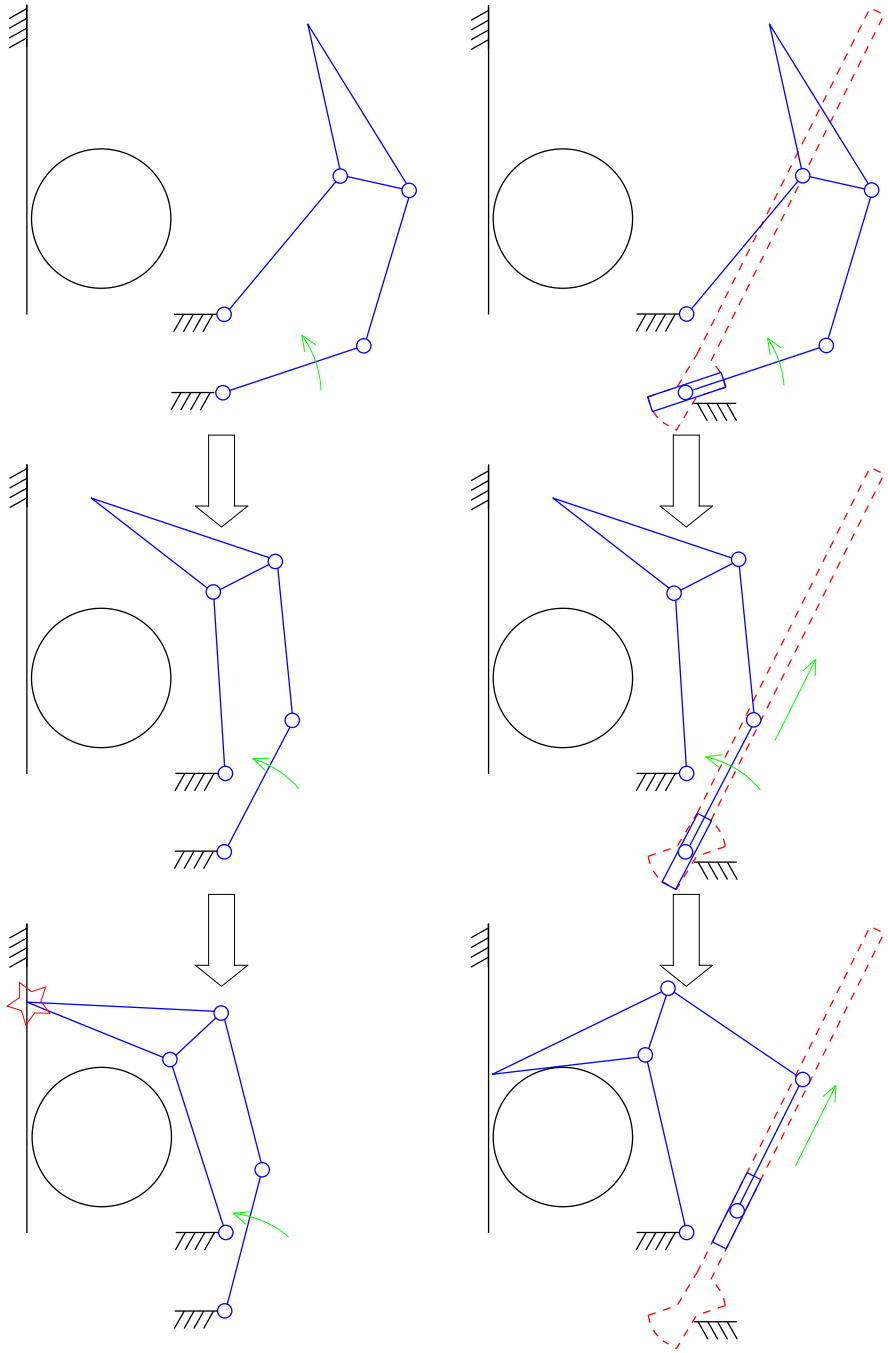


Figure A.12 Underactuated fingers with a) revolute joints only and b) a revolute to prismatic revolute joint.

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ANNEXE B ARTICLE 5 : NON-CYLINDRICAL WRAPPING RODS AND COMPLIANT ADAPTERS FOR MULTI-MODE TWISTING STRING ACTUATION

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Abstract

In this paper, improvements of the basic twisting string actuation commonly found in the literature are demonstrated based on the addition of two types of passive elements in order to optimize mechanical efficiency. First, the motion speed is increased with noncylindrical rigid wrapping. Secondly, passive compliant load adjusters are added to improve the overall mechanical advantage of the transmission in reaction to varying loads. A mathematical model of the new system is first detailed in the paper and provides numerical simulations subsequently validated with experimental results. The impact on the important characteristics of the transmission of the design parameters are shown in details and the simulations highlight several improvements of our proposed design compared to existing twisting wire systems. Finally, this novel transmission with a compliant adjuster is used to drive an underactuated robotic finger to demonstrate its effectiveness.

Introduction

Achieving a compact mechanical transmission with both light weight and high mechanical advantage has always been a difficult challenge in the design of mechatronic systems due to the often large number of degrees of freedom (DOF). This is even more critical in the closely related fields of prosthetic and assistive devices, see [22]. One of the techniques to reduce the weight of a mechatronic product while keeping numerous DOF is for instance to rely on underactuated mechanisms. This technique has been popularized in recent years especially for artificial hands as shown in [37]. Using an underactuated mechanism, one can design novel architectures of robotic fingers constituted by a serial arrangement of phalanges connected by revolute joints and combined with a transmission which distributes the actuation torque (usually a single motor is used) to every phalanx. This transmission mechanism can take

the form of a series of pulleys around which a tendon is routed as in [42]. The design thusly obtained partially mimics the human finger structure, an often desirable target as the latter is an "end-effector" with unparalleled versatility as argued in [4]. Specific tendon and pulley transmission mechanisms have also been shown to produce outstanding performances in terms of contact forces and grasp stability for underactuated fingers, e.g. by [40].

The design of a transmission mechanism is usually concomitant with the choice of an actuator. Pneumatic and hydraulic actuation systems are possible in many applications but not all of them as shown in [6]. For instance, with prosthetic devices, electric motors are generally preferred for many practical reasons such as noise level, environmental constraints, availability of power sources, etc. To connect such an electric actuator to a tendon-driven underactuated finger for instance, the simplest and most popular solution is to rely on a winch. In that case, when the motor shaft turns the tendon/string wraps around that winch thereby providing a pulling force which actuates the finger. This solution was used in many underactuated hands, e.g. by [8, 9, 35]. However, is not the only solution possible. Another mean to provide a pulling force from a rotating axis is to use twisting string actuators (TSAs) shown by [10]. A TSA typically consists in multiple wires attached to a motor output shaft on one end and to the load on the other. Initially, the wires are typically setup to be parallel to each other and to the shaft but when the latter rotates, these wires twist around each other. Since the shape of the wires resulting from this motion is a helix the distance between the endpoints becomes shorter than at the initial configuration when the wires are straight. This twisting action provides a force pulling these endpoints toward each other. Thus, the torque produced by the motor is transformed into a pulling force. This solution is notable for its simplicity, very high mechanical advantage, and low weight. However, it is not without disadvantages. For instance, its input-output relationship in terms of position is nonlinear and an undesired effect of achieving a high mechanical advantage is that the output motion is very slow. Furthermore, when dealing with artificial fingers, if a large force and small velocity is desired when these fingers are grasping an object, the opposite is true when moving through the air.

Despite these disadvantages, TSAs were shown to be an attractive design alternative in a variety of applications. For instance, they were used to drive not only robotic hands by [11] but also artificial eyes by [12] and elbow exoskeletons by [13]. Although mathematical models of TSA and their designs have been proposed in the literature, e.g. by [14] and [15], reported results typically focus on the contracted length of the strings as a function of the position of the motor. Transmission ratios on the other hand are only infrequently reported or even sim-

ply plotted. To solve the disadvantages of TSAs previously mentioned, improvements have also been proposed to design more mechanically efficient actuators. Optimal is here to be understood as having a wide range of transmission ratio values, thus such optimal design should be able to mitigate the degradation of mechanical efficiency due to a variation of the payload by adjusting the trade-off between output force and velocity. For example, if the payload is light the output motion should be accelerated while when stronger forces are required to drive the output, the actuation should have a higher transmission ratio. The adaptability of a transmission also ensures that no oversizing of the motor are needed during the design. For instance, [16] showed a cylinder (wrapping rod) placed inside of two twisting wires in order to increase the output velocity while the TSA is in "speed" mode and disengages in the "force" mode when the load magnitude increases. This actuation is therefore load-adaptive similarly to the one proposed in this paper but its implementation is less compact however because the twisting part of the mechanism of [16] is separated in two stages : with and without the cylinder and twisting is needed in both. For the same purpose, another solution was presented by [17] where the distance between the attachment points of the wires are also variable, i.e. depends on the load magnitude, based on a compliant linkage. This distance is generally referred to as offsets by [18] and has been shown to play an important role in the characteristics of the transmission. Both these solutions however also change the twisting length conversely to the one presented here. In the solution proposed in this paper, a novel TSA is shown inspired by these previous works and improving on the solutions proposed in the literature by not only combining both a novel shape of wrapping rod inside the strings to speed up the action of the actuator but also a variable and adaptive offset.

First, a mathematical model of our TSA is presented to formally establish the impact of the design parameters (offset lengths, twisting length, etc.) on the contraction profile and the transmission ratio. Then, a experimental test bench will be shown and used to support the proposed model. This test bench will be used to validate the behaviors of classical TSA as reported in the literature and then, with our new design.

Mathematical Model

The actuation sequence of a TSA can be separated in three phases. At the beginning, as illustrated in Fig. B.1(a), wires are not in contact between each other or with the rod, if there is one. This constitutes the first pre-twisting phase during which, as the motor rotates, wires are moving closer to each other. Then, in the second phase, actual twisting occurs and the wires are wrapping around each other in the area of fixed length L (as shown in

Fig. B.1(b), B.2(a) and B.2(b)). This twisting phase corresponds to the usual and desired actuated motion of the TSA. Finally, a last phase can occur if the motor shaft is exceedingly rotating in which the twisted bundle of wires starts forming knots as pictured in Fig. B.2(c). This constitutes overtwisting and should be avoided since the motion of the TSA output is unpredictable and these knots can prevent complete unwinding of the system. One should note that the more wires are used, the sooner overtwisting can happen. To prevent this issue the angle α (cf. Fig. B.3) between the wire and an arbitrary reference line perpendicular to the motor axis should be kept above a fixed threshold as proposed by [19].

In the model used here, it is assumed that there is no elongation of the wire during the motion and contact are frictionless. Also, if a rod is inserted between the wires it is assumed to be of finite length and contact with the wires to occur between two limit heights h_1 and h_2, see Fig. B.3 (vector $\mathbf{z}$ is parallel to the rotating motor axis). During the first phase of the movement, i.e. pre-twisting, wires are not touching each other or the rod. Thus, $\forall z \in [0; L]$ one has $R(z) < R_w(z, \theta)$ where $R(z)$ is the radius of the rod at the height z along the axis $\mathbf{z}$, θ is the motor angle, and R_w the distance between the wire and the motor rotating axis at height z. The latter length depends on the geometric parameters of the TSA, namely :

$$R_w(z, \theta) = \frac{1}{2}\sqrt{d_1^2 + (d_2^2 + d_1^2)\frac{z^2}{L^2} + \frac{d_2^2}{L^2}\sin^2(\theta)z^2} - r \tag{B.1}$$

where d_1 and d_2 are the offsets, lengths between the two wires contact points with the upper and middle bar respectively, while r is the string radius, see Fig. B.4. One can also denote c the contracted length of the wire, defined as the difference between the length of wire at the motor position θ and the minimal length (in the initial state $\theta = 0°$), i.e. :

$$c = \sqrt{L^2 + \frac{d_1^2}{4} + \frac{d_2^2}{4} + \frac{d_1 d_2}{2}\cos\theta} - \sqrt{L^2 + \frac{(d_1 - d_2)^2}{4}}. \tag{B.2}$$

This length is also equal to the variation of distance in the area without rotation as illustrated in Fig. B.1. In these equations, the wires are assumed always tight, that implies they are at the initial position. This geometrical model is based on the assumption that the wires are inextensible. Thus, the contracted length does not depend on the payload. After the first contact between the wires or with the rod, h_1, h_2, and α can be obtained by solving :

$$\tan\alpha = \frac{h_1}{A} = \frac{h_2}{B} = \frac{L - (h_1 + h_2)}{C} \tag{B.3}$$

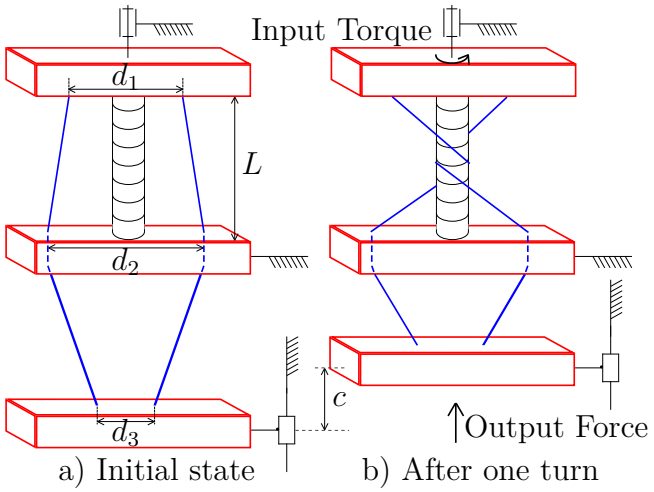
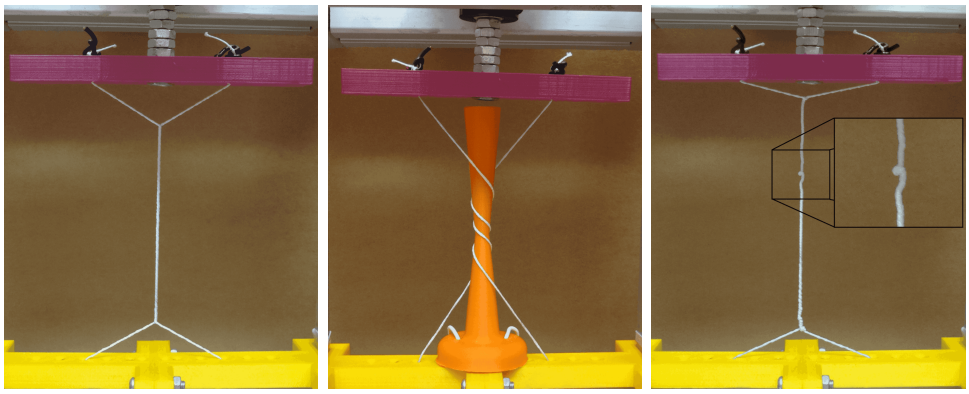


Figure B.1 Schematic TSA.



(a) Twisting phase. (b) Twisting around a rod. (c) Overtwisting.

Figure B.2 Different phases of winding.

where A, B and C are respectively the lengths of the projection in a plane perpendicular to $\mathbf{z}$ of the wires between the bottom fixation point and the first contact point, between the first and last contact points on the rod, and between the latter final contact point and the upper attachment point. These lengths can be expressed as :

$$A = \sqrt{\frac{d_2^2}{4} - (R(h_1) + r)^2}, \tag{B.4}$$

$$B = \sqrt{\frac{d_1^2}{4} - (R(h_2) + r)^2}, \tag{B.5}$$

$$C = \int_{\theta_1}^{\theta_2} (R(\gamma) + r)\, \mathrm{d}\gamma, \tag{B.6}$$

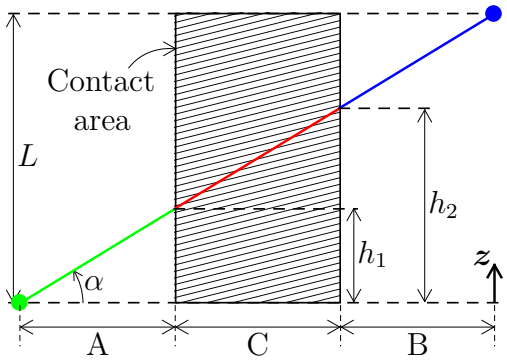


Figure B.3 Side view of the unfolded twisting area.

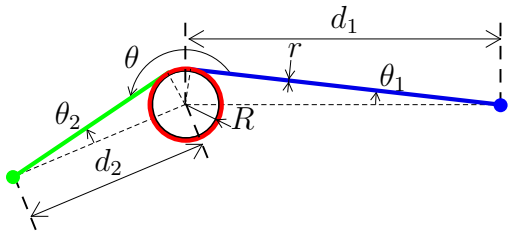


Figure B.4 Top view of the twisting area around a cylinder.

with :

$$\theta_1 = \frac{2R(h_2)}{d_1},\ \theta_2 = \frac{2R(h_1)}{d_2}. \tag{B.7}$$

Finally, the total contracted length during the twisting phase can be calculated as :

$$c = \sqrt{L^2 + (A+B+C)^2} - \sqrt{L^2 + \frac{(d_1 - d_2)^2}{4}} \tag{B.8}$$

Test Bench

A test bench was designed in order to study the relative impact of different configurations of a TSA and validate the model developed in the previous section. The CAD model of this test bench is illustrated in Fig. B.5. A DC motor equipped with an incremental encoder (Maxon 355561) is fixed at the top of an aluminum frame made from 45x45 mm extrusions. Plastic 3D printed parts are also used both for the upper (solid red lines) plate attached to the motor shaft and for the two other lower plates where several holes exist in order to fix wires (dashed blue lines) with known offsets. Thin kite commercial strings (honeywell Spectra Fibers) with a radius from 0.45 (unloaded) to 0.25 mm (loaded) and fishing lines (Caperlan braid 8X Dyneema) with a radius between 0.25 (unloaded) and 0.15mm (loaded) were used. These two types of wires are quite readily available and offer a good resistance

and durability. Please note however that both wires exhibits a notable decrease in diameter when subjected to a load. This reduction in diameter depending on the load cannot be taken into account directly in a geometrical model to ensure its accuracy. Thus, to ensure the accuracy of the simulation minimal and maximal diameters are measured. The wires are routed through the holes of the middle 3D printed plate attached to the structure. Thus, the translation of the lowest plate is the desired output of the TSA. In the following experiments, two wires are used and fixed symmetrically to the motor axis so the plates are bars actually. The offset at the bottom bar is denoted d_3 (see Fig. B.5). Finally, the initial state corresponds to the configuration illustrated in Fig. B.5 where the magnitude of the contraction length l is maximal.

One should note that the length l between the middle and the bottom bars changes as the wires are twisting or untwisting and thus, c is also the difference between $l(\theta)$ and l_0, the length of wire between the bottom and middle bar at the motor angle θ and the length at the initial state respectively. This distance at different angles of the motor is manually measured with a caliper. To measure the force created by the TSA a force sensor (Mecmesin Compact Force Gauge CFG+) is attached to the base and connected to the bottom bar by a spring. A positioning controller (Maxon EPOS2P) is used to control the position of the motor shaft and measure the current flowing between the terminals. Measuring both this current and the force developed at the output of the TSA allows to get an experimental estimate of the transmission ratio. As the upper bar rotates, wires start twisting and either wrap around each other (Fig. B.5) or around a rod inserted in the middle of the wires (Fig. B.6). This latter design is similar to the one presented by [20] where a cylindrical rod is used to accelerate the decrease of the contracted length but in our case a more general shape of this rod, i.e. with a variable radius, is considered.

Experiments

Parameters Evaluation

This section presents the experimental results obtained measuring the contraction lengths and transmission ratios depending on the motor angular positions for several choices of design parameters in order to validate the mathematical model and compare the effects of changing these parameters. Fig. B.7 for instance shows experimental measurements of c (curves labelled “exp” in the legends) using Spectra wires and the predicted numerical values using our model (labelled “mod” in the legends) both depending on L and d_1 respectively. Due to the nonlinear variation of the string radii, contraction lengths are numerically evaluated

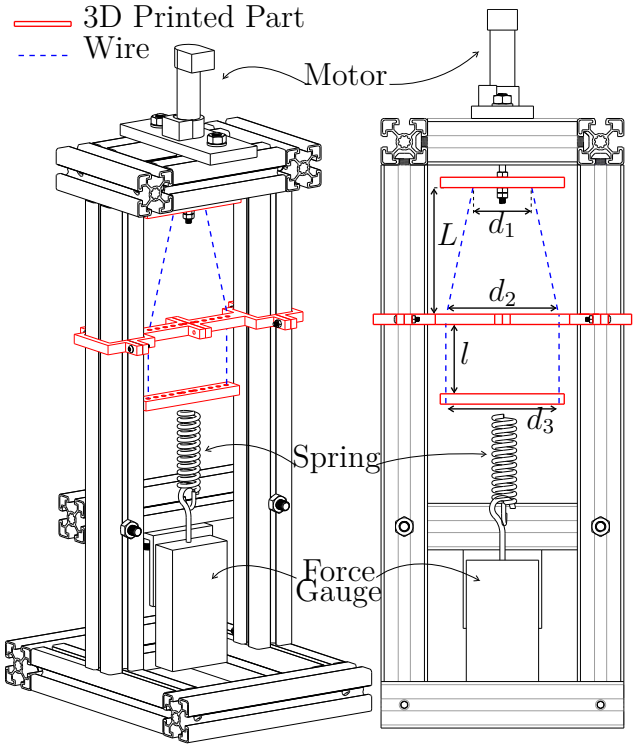


Figure B.5 The test bench.

twice : with $r_1 = 0.25$mm and $r_2 = 0.45$mm, i.e. the loaded and unloaded values respectively. In Fig. B.8, measurements with Spectra Fibers (labelled with "S" in the figure) are compared to the ones with Dyneema lines (labelled "D").

As is clear from these figures, actual measurements are close to the predicted curves and therefore, validates the mathematical model. Even if the model is only geometrical similar experiments with d_2 also confirms its accuracy. Furthermore, the shape of the curves can be separated in two phases : the pre-twisting phase where a drastic increase of the contraction length is observed (leftmost initial jump) and the twisting phase where the slope is gentler. Observing the curves also highlight the effect of varying the design parameters. For example, the smaller d_1 or d_2 are, the bigger the first jump is (smaller L has an opposite effect) but these variables do not have any significant impact on the second phase. On the other hand, varying the string radii change the curve gradient in the twisting part while no significant influence was apparent in the initial phase. Transmission ratios T can be estimated from a kinematic model such as the one presented in Section B using the conservation of energy if friction is assumed negligible as in [21], see Eq. (B.9). Actual transmission ratios can be measured experimentally by estimating the input actuator torque from current sensing (knowing the

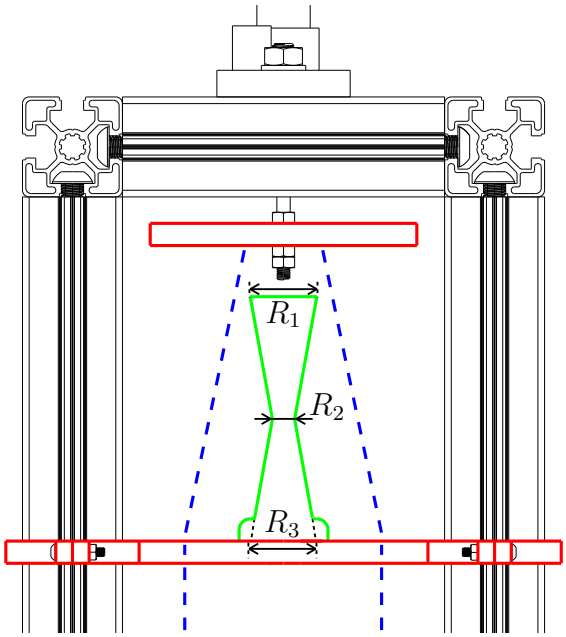


Figure B.6 Twisting area with an inserted rod.

motor torque constant) and computing the ratio of the latter with the measured force, i.e. :

$$T = \frac{F}{\tau} = \frac{\Delta\theta}{\Delta l} \tag{B.9}$$

where F is the measurement from the force gauge, τ the estimated actuator torque, $\Delta\theta \approx \theta_i - \theta_{i-1}$ is the variation at time step i of the motor angle measured from the encoder attached to its axis, and $\Delta l \approx c_i - c_{i-1}$ is the variation of contracted length computed from the model. Comparison between the experimental and predicted transmission ratios are plotted in Fig. B.9. During the pre-twisting phase, transmission ratios go from zero at rest to a peak value, then smoothly decrease during the twisting. As is obvious from this figure, the string radii has a great impact on the transmission ratios. Further analysis of the impact of the design parameters on the values of the ratio brings out that increasing L or decreasing one offset increases the values of T. The most significant change depends on the value of r, a thinner string maximize the transmission ratio. As illustrated, the latter is almost divided by two as the strings radii increase during the twisting.

Interfering rods

Using the developed model, the shape and influence of an interfering object placed in the middle of the wire can be studied. Three cylinders are first used with constant radii $R_1 =$

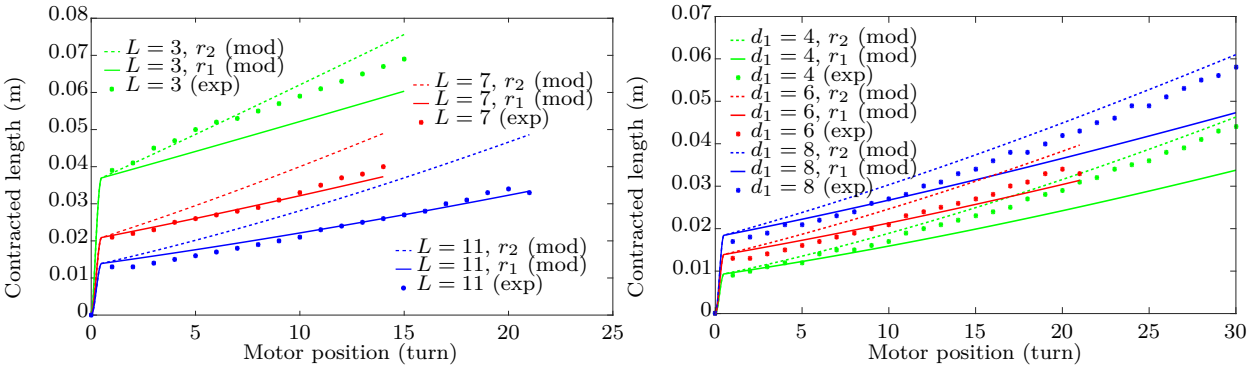


Figure B.7 Contracted length with Spectra fibers as a function of the motor position and L in cm for $d_1 = 6$ cm and $d_2 = 6$ cm (left) and as a function of d_1 in cm for $L = 11$ cm and $d_2 = 6$ cm (right).

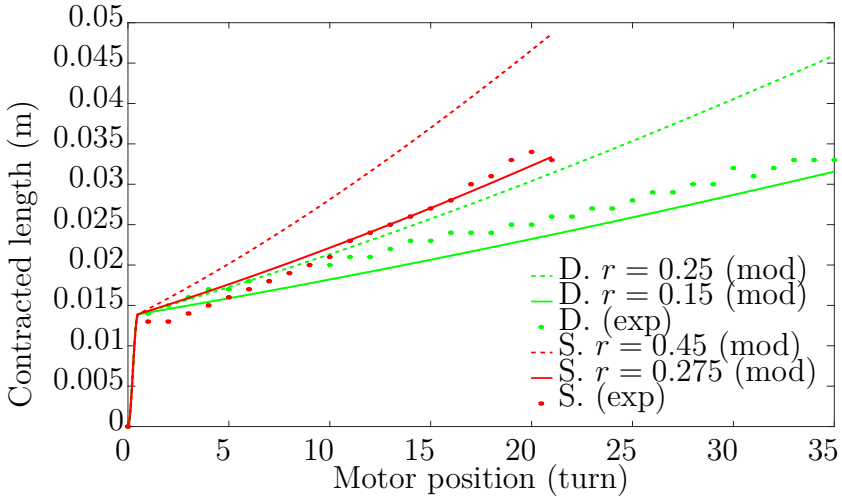


Figure B.8 Contracted length as a function of motor position and r (in mm). $L = 11$ cm and $d_1 = d_2 = 6$ cm.

$R_2 = R_3$ equal to 3, 5 and 8 mm respectively (see Fig. B.6). Then, non-uniform rods can be studied and in our experiments one with a larger middle section ($R_2 = 8$ mm) than its ends ($R_1 = R_3 = 3$ mm) and one with the inverse curvature ($R_1 = R_3 = 8$ mm and $R_2 = 3$ mm) were used. These non-uniform radius rods will be referred to as NUR 1 and NUR 2 respectively. Non-uniform radii motivated the development of Eq. (B.1) and (B.8) to calculate the moment of contact depending on the values of R along the **z**-axis. The objective is to achieve having the same effect as the small cylinder at the beginning of the twisting phase and then, an increasingly larger cylinder or conversely, that the twisting occurs at the larger middle part first and then, extends to both ends. The chosen radii of the wires to simulate the change in the contraction length is $r = 0.275mm$. With these rods, first experiments show that the motor is not strong enough to produce even a single complete turn when the force sensor is

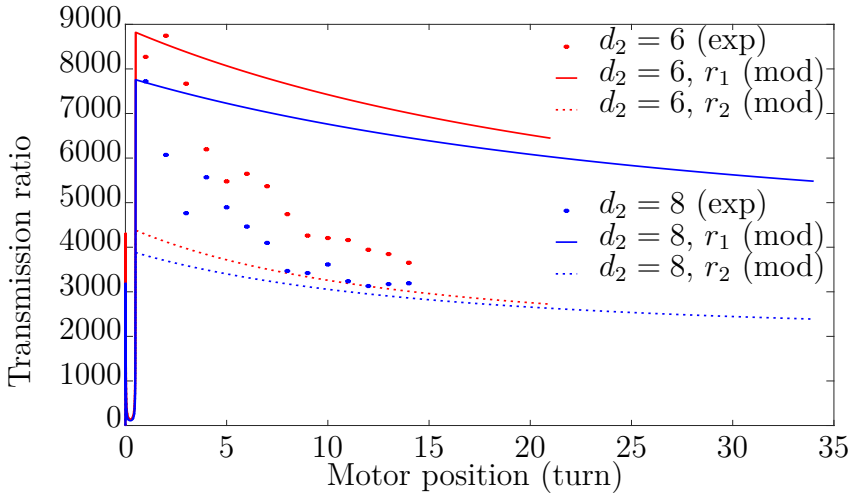


Figure B.9 Transmission ratio as a function of the motor position and d_2 (in cm). Wires are Spectra fibers, $L = 11$ cm and $d_2 = 6$ cm.

assembled and thus, the latter was substituted with a 0.5kg payload mass.

Comparisons between the model and experimental plots are presented in Fig. B.10. Experimental and simulated contracted lengths are close to each other and a jump of similar magnitude can be observed from the initial state to the first point of contact as was the case without any interfering rod (in blue). Then, as expected, during the twisting phase an increase in the radius of the cylinder leads to an increase of the rate of change of the contracted length, i.e. the velocity output. In the case of the first non-uniform rod, the contracted length curve is close to the one of the cylinder with the largest radius at the beginning of the twisting. The more the motor twists the wires, the more the curves go apart as the twisting occurs on smaller sections of NUR 1. Similarly, at first the contraction length of the second non-uniform rod is close to the smaller cylinder and then quickly increases as the wires twisted around an increasing section. A nonuniform rods like NUR 1 used in a TSA with an underactuated gripper has the same function as the tendon sheaths in human fingers illustrated by [4] during a grasping movement. At the beginning of the bending of an initially straight and extended finger they cause the finger to quickly approach the targeted object with a small amount of output force, then conversely at the end of the motion, the latter is maximized as the motion is slowed and the object is grasped.

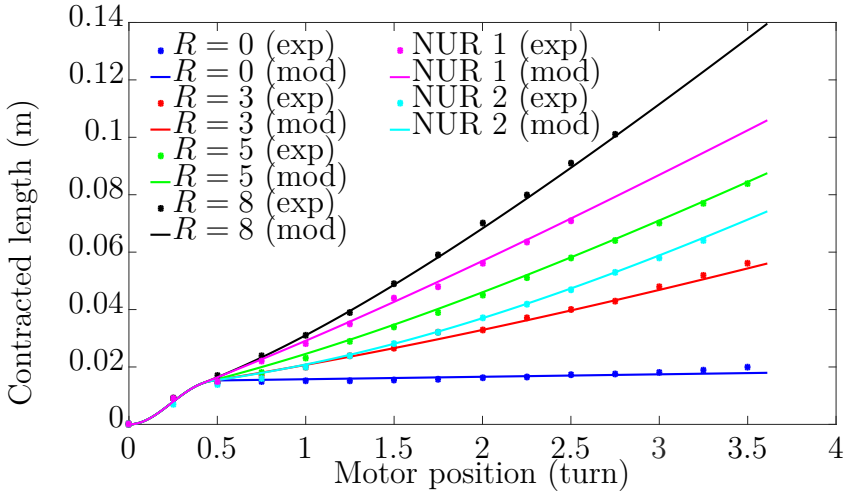


Figure B.10 Evolution of the contracted length as a function of the motor position and type of interfering object (R are in cm). Wires are Spectra fibers, $L = 11$ cm, $d_1 = 6$ mm and $d_2 = 6$ mm.

Interfering compliant rod

The shapes of the rods used in previous section influence the contraction speeds depending on the motor position. Thus, the design of one rod relies on the assumption that the position where the TSA should be fast and where it should be strong is known beforehand. For example, if between zero and three motor turns the load is light and after three turns the load is heavier, a rod can be designed to be quick at the beginning of the motion and then, stronger after three turns accordingly. Additionally, the transition from wires wrapping around a particular diameter to another needs to be smooth since an abrupt transition of diameters can lead to unpredictable wrapping and therefore, an immediate transition from a speed mode to a force mode is difficult. Finally, the previously discussed rods cannot adapt to a sudden change arising from unmodeled perturbations. For all these inconstant cases, fixing the transmission ratio curve by design of the wrapping rod might not be optimal and possibly, leads to unacceptable inefficiency. These issues limit the use of a specific geometry of rod to particular cases where the load is well defined. Adapting the advantages of the interfering rigid rods but with a shape defined depending on the situation is proposed here as a solution. Namely, the shape of the rod can be made to change from a thin shape for speed mode to a thicker one for force mode when an additional load is applied and inversely. This shape adaptive rod would increase the overall efficiency of the system and its range of applications.

The CAD model in Fig. B.11 presents a design solution to achieve such a variable shape rod based on a compliant structure : two parallel 3D printed rigid disks (in blue) are fixed

together with a structural stiff beam (in black) and wires (in green) are threaded from the top to the bottom disk to create a cylindrical envelope. When the twisting string of the actuation wrap around the envelope created by these wires, it will act as a large diameter cylinder keeping its shape when the load is light but then, it could shrink and become thinner when heavier loads are applied since the actuation string of the transmission will be subjected to a greater tension. Furthermore, using wires again to achieve compliance still preserves one of the best advantages of TSA namely lightness. Different ways of threading the wires connecting the two disks are possible as illustrated in Fig. B.11(b) and (c) and this threading pattern will affect the initial shape of the rod as well as its interactions with the twisted strings. At rest, the diameter of a cylinder made by a straight threading from the top to the bottom disc is a linear function of the height (a cone) or constant if the diameters of the end disks are equal. This is probably the simpler shape but others are possible using an oblique threading where the attachment points of the wires are shifted. The resulting compliance of the cylinder depends on the tightness of the wires creating the cylindrical shape and it will affect its adaptability to the load. The more the wires are stretched with a preloading the more resistant to deformation the cylindrical envelope created by these wires becomes. A rough idea of the loads to be actuated and possible perturbations compared to the motor power can determine the trade-off to choose between speed and force and how to stretch the cylinder accordingly. In our prototype of this compliant rod, The bottom disk is attached to the middle plate of the test bench and we refer to the design as a "compliant rod," consisting of the rigid central rod, two disks and the compliant envelope created by the wires.

Figure B.12 shows experimental results from a TSA equipped with this compliant rod and lifting different payload masses (0.1kg, 0.2kg and 0.5kg). In this example, the compliant rod has a straight threading and the overall distance between its top and bottom disks is 12cm (11cm wires and two 5mm identical rigid disks). The diameter at rest of the arrangement is 32mm. During experiments, higher deformations of the compliant cylinder are observed with the heaviest load and if a load is added the cylinder indeed rapidly changes its shape. It should be noted that when deformed the device cannot truly be considered a cylinder with various diameters anymore as parts of its sides may be more strained than others. For example, after half a spin of the motor, it is typical that the TSA strings are compressing the cylinder on two opposite area whereas some areas are not yet in contact. Thus, a proper numerical model is more complex to develop than for rigid noncylindrical rod. Nevertheless, the illustrated curves in Fig. B.12(a) show that the speed of contraction decrease after a couple of turns of the motor to go from the speed mode to force mode for bigger loads and the passive compliant element passively adapts its shape. Figs. B.12(b) and B.12(b) illustrate the adaptation after one motor turn. In the first case, no load is attached to the TSA and the

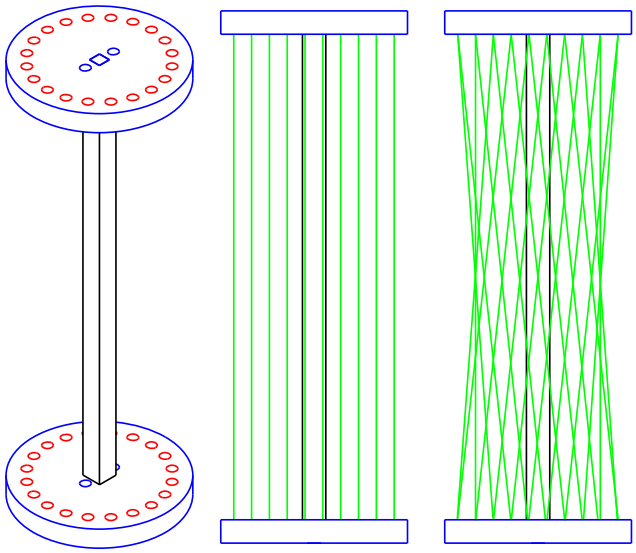


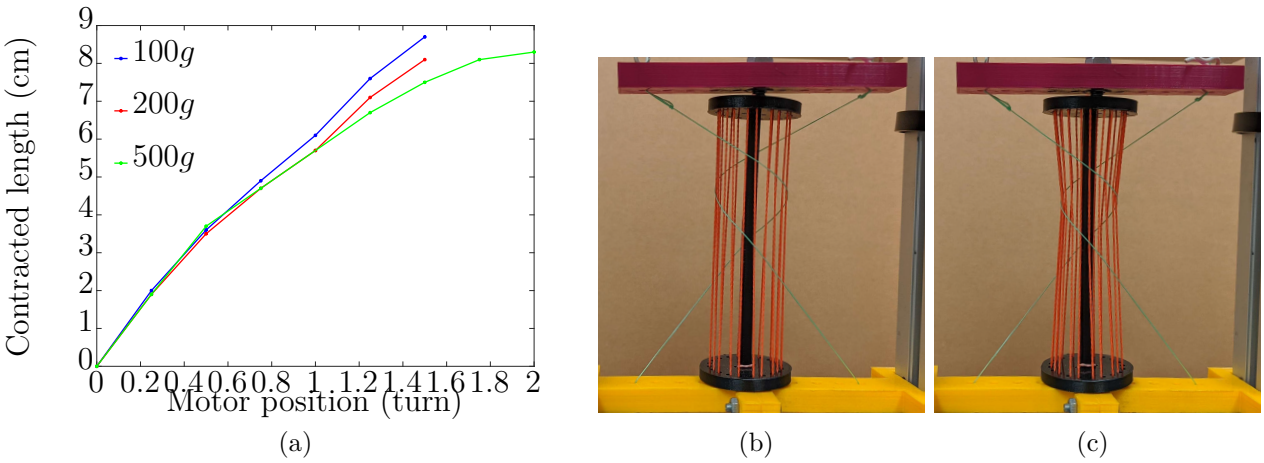
Figure B.11 (a) Structure of the compliant rod without wire and two possible threading patterns : straight (b) or oblique (c).

stiffness of the threading is high enough to keep a cylindrical shape whereas in the second case, the TSA is attached to a 0.5kg load and the cylinder becomes noticeably thinner.

Compliant offset adapter

A third and final improvement over classic TSA, namely an adaptive load adjuster based on a compliant mechanism (see Fig. B.13) is also added replacing the fixed middle bar. This adjuster allows to change d_2 for different load situations : the wires pass through the part and are guided by two slotted plates (one above and one below the latter) to keep them aligned. When the wires twist, if the load exceeds a certain threshold, internal reactions force the wires to move closer by elastically deforming the compliant part and thus, d_2 decreases. Consequently, the speed of the system also decreases and the transmission ratio increases which slows down the motion but increases the output force.

Comparisons for different values of the offset d_2 are illustrated in Fig. B.14. These variations are obtained with three different load masses and three initial values of d_2, indicated by subscripts 1, 2 and 3. Initially, wires are placed through the adjuster at $d_{2,n}$ with $n = 1, 2, 3$ before the load is added and the first measurement is taken, this explains the slight differences with $d_{2,3}$ at rest for different values of the load. The compliant adapter bends even if the wires are not twisting yet. With $d_{2,1}$ no difference can be observed as the wires are not in



(a) (b) (c)

Figure B.12 Experiments with the compliant cylinder : measurements of the contracted length with different payloads (a). Compliant rod with no payload (b) and a 0.5kg payload (c).

contact with the adjuster. For all nine curves, two phases can be observed as in the previous section. First, the magnitude of d_2 decreases from its initial value and a significant drop of d_2 during the pre-twisting occurs compared to a slight decrease in the twisting phase. As expected, the heavier the load is, the greater the decrease of d_2. Furthermore, depending on the initial position of the wire the magnitude of the decrease with the same load is amplified, as such the drop is double from $d_{2,3}$ to $d_{2,1}$ for a 0.2kg payload mass. The initial values of d_2 are limited by estimation of the deformations the compliant adjuster can resist. Observations of deformations during initial experiments showed that the specific geometry of compliant adjuster used here cannot withstand larger offsets as plastic deformation occur indicating that failure could occur after some cycles. A pseudo-rigid body model could be developed in the future to evaluate the usable range of deformation and accurately predict the adaptation range of d_2.

Application and Discussion

Finally, two underactuated fingers inspired by the designs shown by [37] and driven with pulleys were attached to the TSA (see Fig. B.15). As discussed in the introduction, underactuated fingers have a limited number of motors and adapt mechanically their shapes to the object they grasp, similar to computer-controlled compliant grasps. Thus, combining them with a TSA transmission seems a natural extension to create light and affordable mechatronic devices which can change their mechanical properties in reaction to disturbances without requiring any electronic control or even sensing. The kinematic architecture of the underac-

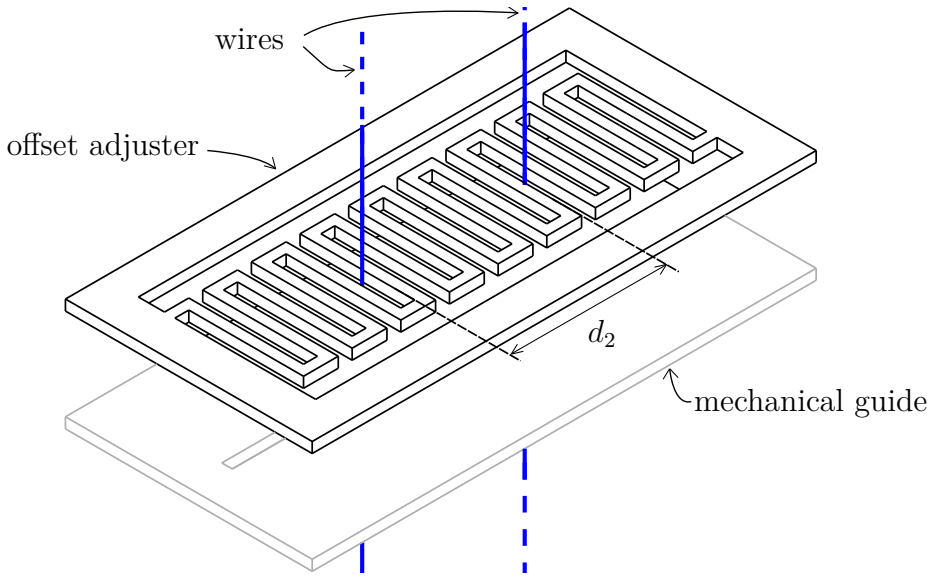


Figure B.13 Compliant offset adjuster.

tuated fingers used in this paper is the same of the one presented by [40] and each of their driving tendons is a wire of the TSA. Hence, both wires of the TSA are used as outputs. A noncylindrical wrapping rod and a compliant adjuster are added to the TSA one at a time. For the specific design shown here and without interfering elements, the fingers close in 12 motor revolutions, while only 1.2 turns are needed when NUR 1 is added. With the compliant adjuster, in the initial state the wires in the fingers are not taut and the adjuster is undeformed but when the wires are twisting, the wires in the fingers tighten. When the latter wires are fully stretched, the fingers close. The offset d_2 is changing as the motor position evolves, dropping from $5.5cm$ at the initial state to $3.4cm$ when the fingers are fully closed.

The compliant offset adjuster increases the adaptability of the complete system. More generally, if an underactuated finger reaches a configuration where its adaptation is maximal and an increment of the input force may lead to deformations or even damage to the object, an additional compliant mechanism such as the compliant offset adapter or an interfering compliant rod can adapt to the increase of the payload and therefore, serves as a secondary safety feature for little cost. The rigid wrapping rod is less versatile but is a controlled (by design) solution to adapt the transmission ratio. This solution is more adapted to scenarios with expected change of payload and still does not require the addition of another motor or additional sensors.

To the best author's knowledge issue with friction and mechanical properties of the wires

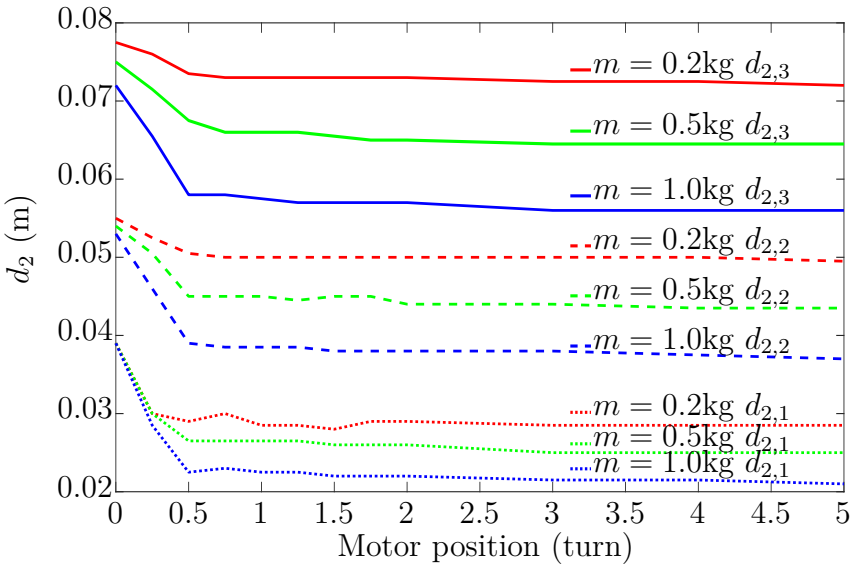


Figure B.14 Measurements of the offset d_2 with the compliant offset adjuster and different payloads.

are often observed but rarely included in theoretical models. As mentioned in Section B the presented comparison between experimental and simulation results took possible change of geometry of the wire into account by considering both extreme radii as measured experimentally. Additionally, the stiffest available strings were chosen to limit deformations. Friction is expected between a wire and 3D printed parts, e.g. interfering rods, and also with other wires. This physical phenomenon is absent from the geometric model and thus, a complete dynamic of quasi static model is still necessary for accurate simulations and future designs of TSA. Considering zero deformation of wires for example with an heavy payload will lead to inaccurate model. Moreover, a dynamic analysis is the next step to model the adaptation of the compliant adjusters and its potential resonance.

The main challenge to model the radius shrinking of a wire and its elongation is that its stiffness changes with the twisting. An empirical method is used by [22] to estimate the relationship relating the stiffness to the contraction. Two coefficients are evaluated by measuring the stiffness compared to the contraction with a TSA composed of one, two or three wires and then, by approximating the results. [23] models friction in the motor and between the payload and its environment by kinematic and static equations while modeling the elongation of the wires with virtual springs. Stiffness of two springs simulates the elastic deformation of the cross-section of the wire (its radius) and its length to consider variations of these geometric parameters. Then, based on static equations involving the stiffnesses of the springs and assuming small deformations the simulations of the motor torque during the motion are fitted with the experimental results.

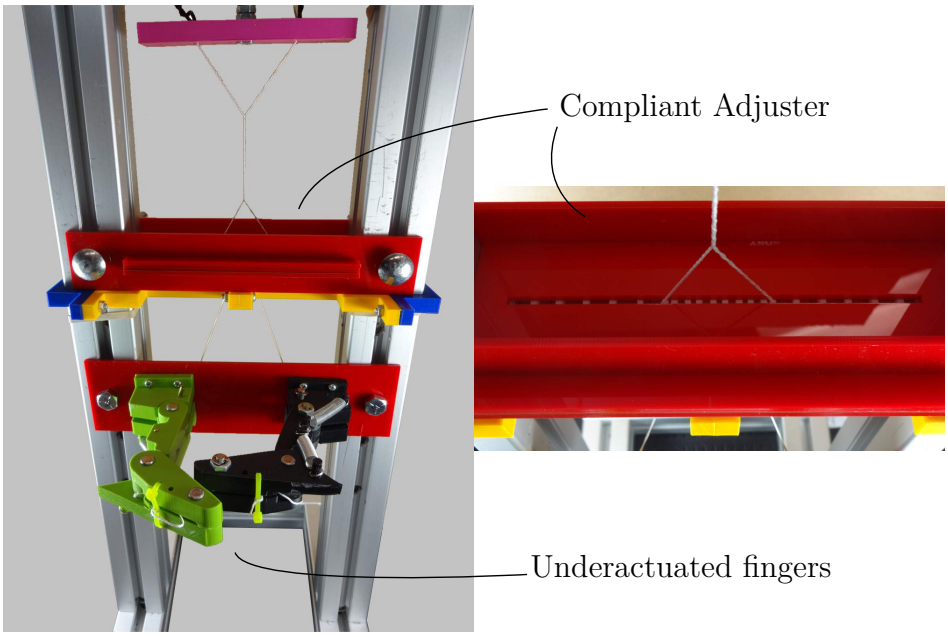


Figure B.15 Twisting strings actuating underactuated fingers.

Conclusion

This paper presents a TSA enhanced by three additional features inspired by the literature. First, noncylindrical rigid wrapping rods are used to accelerate the actuation depending on the evolution of the twisting and then, a compliant cylinder made of a stretched threading pattern is shown to passively adapts its shape to payloads to optimize wrapping speed. Finally, a compliant adjuster adapts the mechanical advantage based on the output load. Experimental measurements were shown to match the proposed mathematical model for a wide range of parameters and finally, an application of this TSA where it drives underactuated fingers was briefly illustrated as a potential application.

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ANNEXE C ARTICLE 6 : POINT-SPRING MODELING OF THE GRASP OF A DEFORMABLE OBJECT BY AN ANGULAR GRIPPER

Cet article a été présenté lors de la conférence avec comité de lecture du *Canadian Committee for the Theory of Machines and Mechanisms 2021* et disponible en ligne depuis le 3 juin 2021.

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Abstract

The accurate evaluation of the deformation undergone by a soft object when it is seized by a gripper has many important applications but remains a difficult challenge due to the computational burden associated with this task. For instance, during surgery excessive pressure has been associated with local trauma (micro-lesions) leading to indesirable bleeding. For produce pick and place, the same type of excessive pressure can cause unsightly bruises and unnecessary waste as these damaged produces generally cannot be sold to the customer. In this paper, an alternative to the usual Finite-Element Analysis (FEA) method is used to accelerate the computation of an effective model, simpler but keeping excellent accuracy. While introduced in a previous work by the authors, this model is here refined by analyzing the impact on the simulation accuracy of the number of elements used to approximate the soft object. The latter being critical for computation efficiency. Simulations are discussed here with the objective of confirming what is the minimal viable number of elements one can use in the case of an angular gripper grasping deformable circles of various radii. After the introduction of the numerical model, comparisons with FEA of the results obtained from the model depending on the number of elements are shown. Finally, experiments with a surgical forceps and deformable cylinders confirm the effectiveness of the model in producing accurate results.

L'évaluation précise de la déformation subie par un objet mou lorsqu'il est saisi par un préhenseur a de nombreuses applications importantes mais reste un défi difficile à relever en raison de la charge de calcul associée à cette tâche. Par exemple, lors d'une opération chirurgicale, une pression locale excessive peut être associée à un traumatisme local (micro-lésions) entraînant des saignements indésirables. Pour la manipulation de produits agricoles, ce même type de pression excessive peut causer des traces disgracieuses menant à des pertes inutiles, car ces produits endommagés ne peuvent habituellement pas être vendus au client. Dans ce document, une alternative à la méthode habituelle d'analyse par éléments finis est utilisée pour accélérer le calcul d'un modèle efficace tout en conservant une excellente précision. Bien qu'introduit

dans un travail précédent par les auteurs, ce modèle est ici affiné en analysant l'impact sur la précision de la simulation du nombre d'éléments utilisés pour approcher la forme de l'objet mou. Ce nombre étant essentiel pour l'efficacité du calcul. Des simulations sont discutées dans le but de confirmer quel est le nombre minimal viable d'éléments que l'on peut utiliser dans le cas d'une pince angulaire saisissant des cercles déformables avec différents rayons. Après l'introduction du modèle numérique, des comparaisons avec l'analyse par éléments finis des résultats obtenus à partir du modèle en fonction du nombre d'éléments sont présentées. Enfin, des expériences avec une pince chirurgicale et des cylindres déformables confirment l'efficacité du modèle à produire des résultats précis.

Introduction

Design optimization of grippers intended for the grasp or interaction with deformable objects usually involves the comparison and therefore, simulations, of a significant number of architectures. These simulations are typically relying on a Finite Element Method (FEM) which is the most widely software approach used for structural analysis. This method is accurate, but also requires significant computation time [92,93] which makes certain optimization techniques difficult to use such as genetic algorithms, especially when large deformations are required. Another method referred to as mass-spring model (MSM) is commonly used for real-time application because of its simplicity and speed [117]. It consists of using mass points to discretize the geometries of the deformable objects to be simulated. These mass points are connected by springs approximating the elasticity of the material. In the available literature, values for the spring stiffnesses in a MSM are estimated with different techniques usually taking into account the geometry of the discretization (area of contiguous polygons, lengths, etc.) and material properties (Young's modulus, Poisson's ratio) [4, 119, 142]. Although stiffness evaluation processes are usually for two-dimensions cases, an adaptation for three-dimensional scenarios including masses localized at the discretized points has also been recently proposed [164].

The current most common application of MSM is visual effect for animation, where the goal is to quickly visualize realistic motions but not necessary accurate ones [140]. This trade-off is not acceptable for all applications especially when it can have important practical consequences, such as with surgery simulators where inaccurate results can have negative impact on training [9]. However, MSM can be made significantly more accurate, even for large deformation, when enough points are considered. In [120] a point-spring model (PSM), a MSM with no mass, was used to evaluate the deformation of a soft object grasped by compliant grippers with multiple architectures. The possibility to quickly go from one design

to another and obtaining quick results from the simulation was the rationale of the choice of method and the results were shown to comparable to FEA at a much lighter computational burden. In this method, the points of the PSM only discretize the object contour shape and its fixed center, keeping complexity at a minimum. When forces are applied on the contour points, the internal springs of the model deform to mimic the flexibility of the soft material constituting the object. Thereby, deformations undergone by the object can be evaluated to optimize gripper design architectures. An example of application of this type of simulation is when one needs to design a medical tool minimizing the deformation, or at least mitigate damage, of fragile tissues or organs while keeping a maximum number of contact points for a secure grasp [134, 135].

One challenge of the aforementioned optimization is to determine the parameters of a PSM discretization for the soft object which will be both accurate and quick. As FEM, a simulation with a coarse meshing may not result in a reasonably accurate value while better results with a finer mesh are expected but may be very long to compute. The work presented in [13] focused on the validation of the numerical equations modeling grippers with deformable objects in order to fit with experimental measurements but without concerns about the required computational time. For instance, the number of divisions of the objects was not a parameter analyzed in the latter reference despite its critical importance for computation time and accuracy). In this article, different numbers of elastic elements are studied to evaluate a PSM simulating the grasp of a deformable circle by an angular (rotational) gripper. First, the numerical model of the object, the jaws of the gripper, and their interaction will be detailed. Then, a comparison of the accuracy of a discretization depending on its number of elements will be presented to determine the best distribution of contour points of the PSM for efficiency (accuracy and speed). Finally, PSM simulations are compared to experiments with a commercial surgical forceps and soft cylinders.

Model

Figure C.1 illustrates the grasping scenario discussed in this paper : an angular gripper is squeezing a round soft object. The fingers of the gripper are symmetrical and the center of the object is assumed to lie along that axis of symmetry. For legibility purpose, the parameters in Fig. C.1 are shown only for one finger. When an actuation torque T_a is applied on both fingers simultaneously (assumed identical and shown as a green arrow in the figure), the gripper closes to squeeze the object. The object shown at rest in orange then deforms along the dotted red line due to the contact pressure from the fingers. The angle θ_a is the actuation angle and its value at rest is θ_{a_0} in the following sections. A torsion spring with stiffness k_f

is attached to both revolute joints to statically constrain the gripper when $T_a > 0$ but before it makes contact with object. Thus, when the actuation torques vanish the fingers go back to their initial positions with this model.

Figure C.2 presents a potential discretization of the object with a PSM. Points and springs define triangles and it should be noted that all points except the center belong to the contour of the object. The center point is used in this paper to fix the center of the object to the ground. As mentioned in the introduction, this discretization was selected for its simplicity. Appropriate spring stiffnesses are calculated depending on the Young's modulus and Poisson's ratio of the simulated material and contiguous triangles geometry as proposed in [118]. For example, the stiffness estimate of the spring between the green and blue triangle in Fig. C.2 depends not only on the material but on the areas of these triangles and their lengths (r and h). The contour points are assumed to be uniformly distributed along the circle. Their number are defined as the number of triangles and all contour springs have the same stiffness coefficients (noted k_{ext}) as well as the internal spring (noted k_{int}). When decomposing a round object in that manner, care must be taken to avoid a discretization leading to triangles with internal angles over 90°as it leads to negative stiffness with the Van Gelder model [118].

The numerical model proposed in [13] and laying the foundation of the present paper requires three steps as well as a few hypotheses that need to be recalled here. First, the motions are assumed slow and gravity is negligible compared to the grasp forces. Then, frictionless contact forces are modelled and thus, these contact forces are perpendicular to the finger surface. As for the methodology steps, the equation of the generated contact forces $\boldsymbol{f}_p$ on the phalanges of an underactuated finger on rigid objects introduced in [37] is first used. This equation relates these contact forces to an input torque vector $\boldsymbol{t}$ and also introduces matrices depending on the transmission mechanism (the Transmission matrix $\boldsymbol{T}$), the geometry of the finger as well as the contact locations (the Jacobian matrix $\boldsymbol{J}$), i.e. :

$$\boldsymbol{f}_p^T \boldsymbol{J} = \boldsymbol{t}^T \boldsymbol{T}, \tag{C.1}$$

with :

$$\boldsymbol{f_p} = \begin{bmatrix} f_1 & \dots & f_p \end{bmatrix}^T \tag{C.2}$$

$$\boldsymbol{t}_{\text{gripper}} = \begin{bmatrix} T_a \\ -k_f \Delta\theta_a \\ -k_2 \Delta\theta_2 \\ \vdots \\ -k_p \Delta\theta_p \end{bmatrix} \tag{C.3}$$

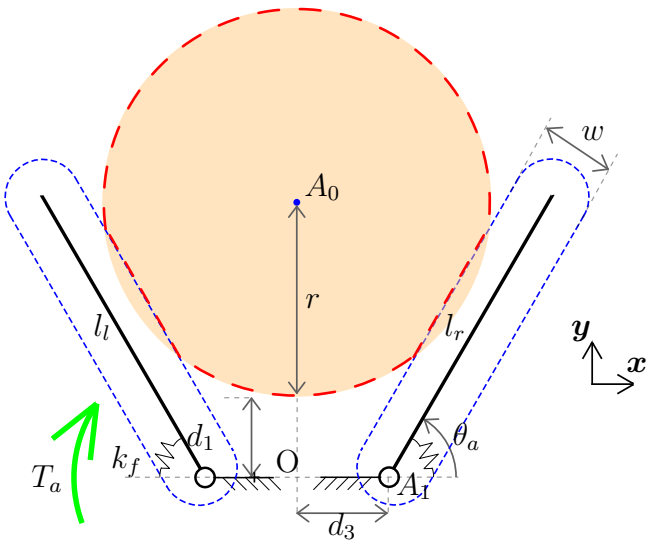


Figure C.1 Model of the gripper with a deformable soft object in red dotted line.

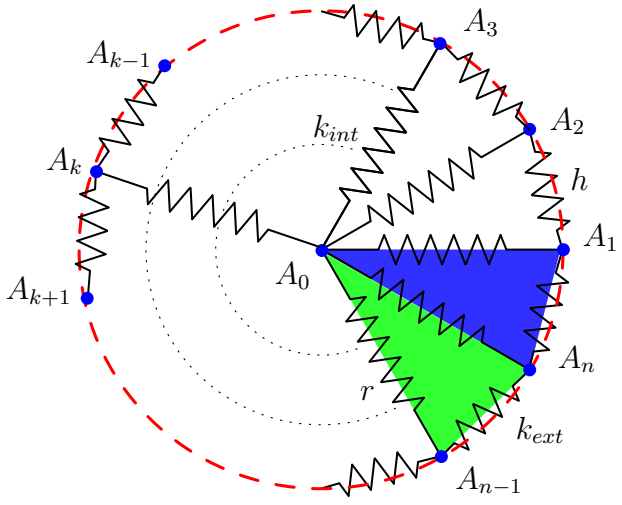


Figure C.2 Point-Spring Model of a Circular Object.

where θ_i and k_i ($i \in [2, p]$) are the interphalanx angles and spring stiffnesses at the joints between them. The Transmission matrix corresponding to the angular gripper is simply :

$$\boldsymbol{T}_{\text{gripper}} = \begin{bmatrix} 1 & 0 & \dots & 0 \\ & & \boldsymbol{I} & \end{bmatrix} \tag{C.4}$$

and the grasp Jacobian matrix is :

$$\boldsymbol{J} = \begin{bmatrix} \boldsymbol{r}_{11}^T \boldsymbol{x}_1 & 0 & 0 & \dots & 0 \\ \boldsymbol{r}_{12}^T \boldsymbol{x}_2 & \boldsymbol{r}_{22}^T \boldsymbol{x}_2 & 0 & \dots & 0 \\ \dots & \dots & \dots & \dots & \dots \\ \boldsymbol{r}_{1p}^T \boldsymbol{x}_p & \boldsymbol{r}_{2p}^T \boldsymbol{x}_p & \boldsymbol{r}_{3p}^T \boldsymbol{x}_p & \dots & \boldsymbol{r}_{pp}^T \boldsymbol{x}_p \end{bmatrix} \tag{C.5}$$

where $\boldsymbol{r}_{ki}$ is the vector from point A_k to the i^{th} contact point and $\boldsymbol{x}_i$ the unit vector along the i^{th} phalanx in the direction of the end of the finger.

While Eq. (C.1) has been extensively used for underactuated hands, it can also be used for a fully-actuated gripper as considered here but needs to be adapted. Indeed, if the number of contact per phalanx is greater than one, the model of [37] is invalid and this situation is clearly to be expected for the interaction of the angular gripper with a PSM based soft object, especially when it starts to be squeezed. In [13], a methodology is presented to overcome this limitation of the original model. In order to use the equation for $p > 0$ contacts on one finger, the latter is virtually divided into p links, or phalanges, connected by revolute joints. Each virtual joint is also combined with a stiff rotational spring (k_2 to k_p). Therefore, Eq. (C.1) becomes valid again since there is again only one contact per (virtual) phalanx and can be used to model the interaction between the finger and the points of the PSM. In the modeled scenario of the rotational finger, the relationship between the input torques (due to the actuation and the springs) has to be calculated twice, one for each finger, since the finger positions are not expected to always be equal when the discretization is non-symmetrical.

The second step of the algorithm is the computation of the relationship between $\boldsymbol{f}_o$ the vector of internal forces in the PSM and $\boldsymbol{u}$ the vector of the displacements of all the object points as presented in [13]. This relationship can be expressed as :

$$\boldsymbol{f}_o = \boldsymbol{S}\boldsymbol{u} - \boldsymbol{P}\boldsymbol{k} \tag{C.6}$$

where $\boldsymbol{f}_o$ and $\boldsymbol{u}$ are vectors defined as :

$$\boldsymbol{f}_o = \begin{bmatrix} f_x^1 & f_y^1 & \dots & f_x^i & f_y^i & \dots & f_x^n & f_y^n \end{bmatrix}^T \tag{C.7}$$

$$\boldsymbol{u} = \begin{bmatrix} \boldsymbol{u}_1^T & \dots & \boldsymbol{u}_i^T & \dots & \boldsymbol{u}_n^T \end{bmatrix}^T \tag{C.8}$$

with $\boldsymbol{u}_i$ the displacement vectors of points A_i (see Fig. C.2). The scalars f_x^i and f_y^i are the components of the forces at the endpoints of the i^{th} spring projected along the $\boldsymbol{x}$ and $\boldsymbol{y}$ axes of the reference frame (using subscripts x or y to indicate which axis). The matrices $\boldsymbol{S}$, $\boldsymbol{P}$ and vector $\boldsymbol{k}$ depend on the stiffnesses and the positions of the points. Details on how to compute these matrices can be found in [13].

Finally, the last step is to establish a connection between Eq. (C.1) and (C.6). This can be done by using the the projection matrices defined in [13] and the resulting equations are :

$$\boldsymbol{f}_p = \boldsymbol{D}_x \boldsymbol{f}_o, \tag{C.9}$$

$$\boldsymbol{f}_p = \boldsymbol{D}_y \boldsymbol{f}_o, \tag{C.10}$$

$$\boldsymbol{0} = \boldsymbol{D}_z \boldsymbol{f}_o. \tag{C.11}$$

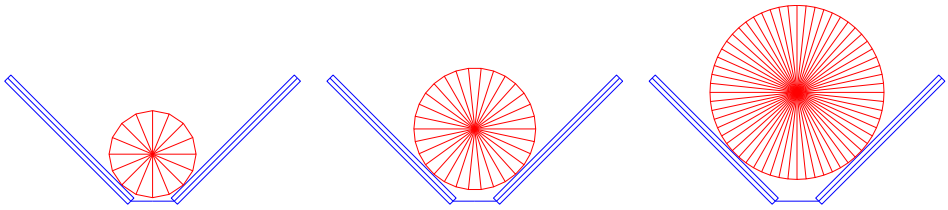
Matrices $\boldsymbol{D}_x$ and $\boldsymbol{D}_y$ associate each internal force applied on a point to the corresponding phalanx force. The forces are also projected to match the direction of the fixed frame. For phalanges not touching the PSM, their contact forces are equal to zero. Matrix $\boldsymbol{D}_z$ completes the static equilibrium equations for the contour points not in contact with the phalanges.

From the initial position of the gripper and object pair, by increasing the input torques incrementally and solving Eqs. (C.9), (C.10) and (C.11) the motion of the finger and deformations of the PSM can be simulated efficiently. In the numerical examples of this paper, the model is implemented in Matlab and the difference between the left- and the righthand side of these equations are numerically minimized with a Nelder-Mead simplex algorithm.

Number of Triangles

In this section, the accuracy of simulations with a PSM having different number of triangles are evaluated by simulating the grasp of a small ($r = 5cm$), a medium ($r = 7cm$) and a large ($r = 10cm$) circle while varying the number of contour points. The simulated configurations are presented in Fig. C.3 and the associated parameters are listed in Tab. C.1 and C.2. These results are obtained following the methodology of the previous section.

Starting from the approximation calculated with the Van Gelder model as an initial estimate,



a) $r = 0.05m$ b) $r = 0.07m$ c) $r = 0.1m$

Figure C.3 Models of circular object grasped by a jaw gripper.

Tableau C.1 Parameters of the finger in the simulation presented in Figs. C.3, C.4 and C.5 (all lengths are in meters, angles in degrees, stiffnesses in $N \cdot m$).

l_r	l_l	θ_{a_0}	d_3	w	k^f
0.2	0.2	45	0.025	0.01	0.073

the spring stiffness values are calibrated to fit the results of a single corresponding FEA where a complete circle is grasped. A large number of elements is first used in the simulation, i.e. when a greater number does not affect the results significantly. Figure C.4 illustrates an example comparing the angle θ between the fingers for a specific input torque as calculated with an FEA software (ANSYS) and a PSM dividing the circle in 4, 12 and 20 triangles. Figure C.5 shows for the three different sizes of circles the differences of position of the finger after deformations with four to sixty triangles. The required computation time with the medium size object is also illustrated in Fig. C.6. For comparison, the FEA needs around $20,000s$ to compute one scenario.

From Fig. C.4 and C.5, one can observe that as expected the accuracy with a coarse mesh of the PSM is poor especially with the largest object. With a significant increase in the number of triangles, the results converge to a consistent difference. This was to be expected considering the same phenomenon appears with FEA. However, it is noticeable that accurate results can be obtained with a rather limited number of elements, especially again comparing with the FEA software. There is, however, a minimal number of elements to retain : a PSM with four divisions for example, namely a square, is clearly too far from the original geometry of the circle. During an optimization of a gripper with such a coarse discretization several architectures may be poorly evaluated. Obviously, the finer the discretization is, the closer the results fit with the FEA. With twelve points the evolution of the angle seems to converge to good results with most deformations. The object side in contact with the gripper is typically

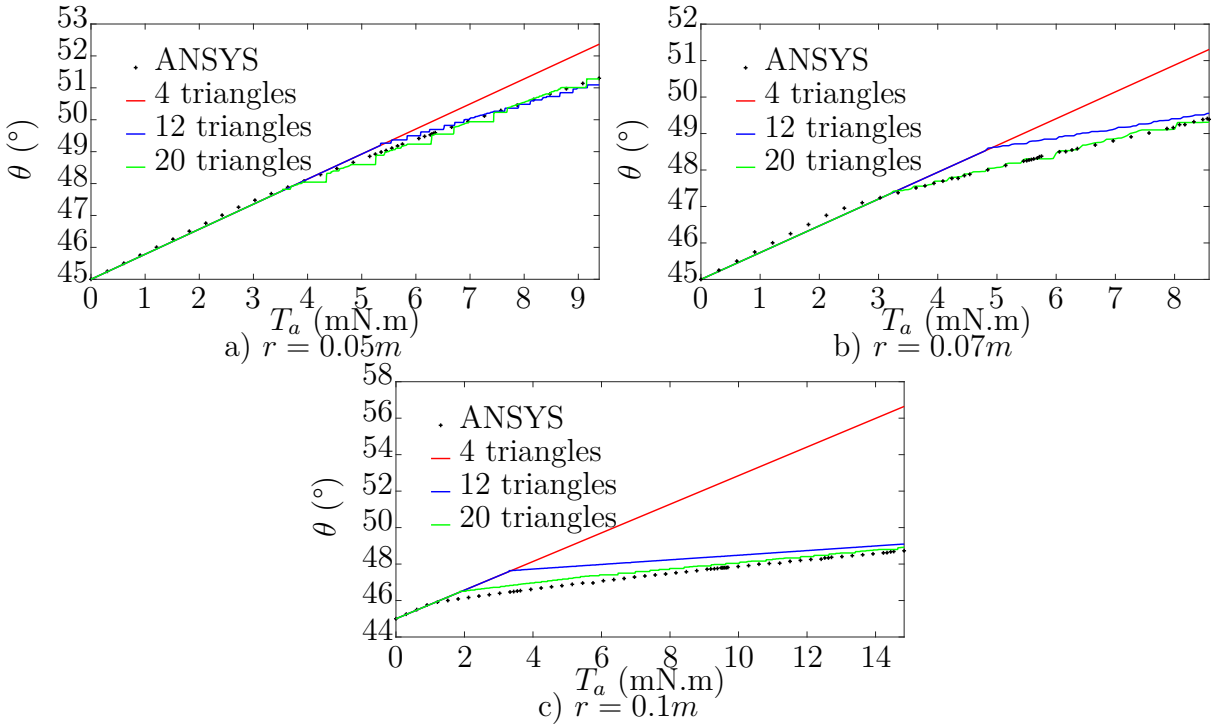


Figure C.4 Simulations with the numerical model compared to results with a FEA.

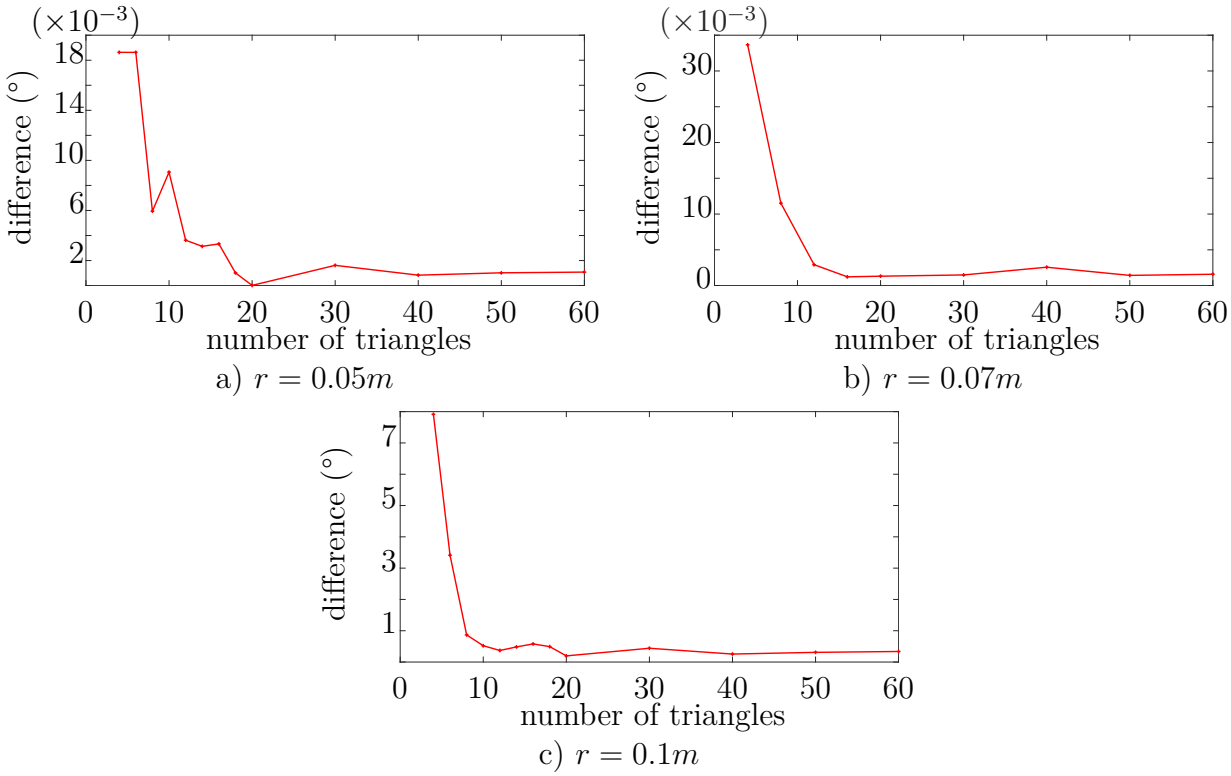


Figure C.5 Precision of the simulation depending on the number of triangles and circles with three different radii.

Tableau C.2 Parameters of the object in the simulation presented in Fig. C.3, C.4 and C.5 (all lengths are in meters)

r	d_1
0.05	0.004
0.07	0.013
0.1	0.025

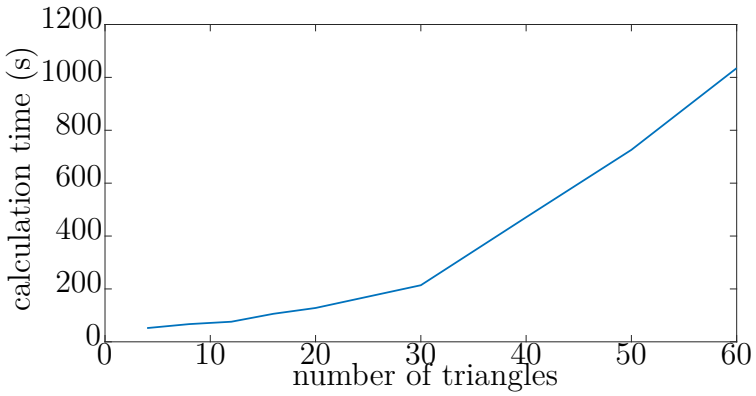


Figure C.6 Calculation times for different divisions of the object ($r = 0.07$).

flattened and thus, positions of points aligned on the surface may not be as important as the number of those not in contact.

One can notice that the differences observed in Fig. C.5 at sixty divisions (or more) are not zero. This is due to an imperfect calibration of the spring stiffnesses but this remaining error is small. Depending on the size of the object, the number of triangles to reach a prescribed range of accuracy varies. Indeed, the convergence depending on this number of elements is slower with a small object ($r = 5cm$) but speeds up with larger radii. In this example, with twenty triangles or more, all results seem to match the FEA quite well, as the remaining gap with the FEA is not significantly improved if a finer discretization is tried. However, calculation time also increases exponentially (see Fig. C.6) with more points to compute and therefore, this value (20) may be a good trade-off between computation time and accuracy. An identical evolution of calculation times was observed with the other objects. It should be noted that when the meshing is too fine, the use of virtual phalanx leads to convergence issue for the numerical model. Short distances between contour points imply short virtual phalanges in case of multiple contacts and thus, the relative angles between these phalanges become difficult to calculate.

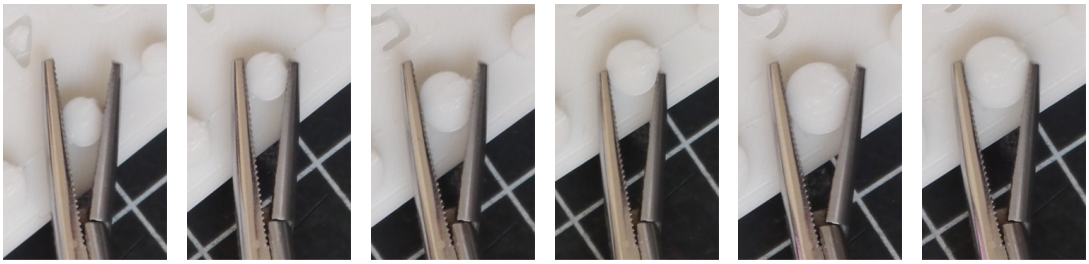
Experiments

To compare the proposed numerical model with experimental results, a surgical Hartmann alligator forceps (illustrated in Fig. C.7) is used to grasp deformable circles. This forceps is driven close by the action of extension springs at the handle to provide a stable and consistent actuation torque at its jaws as a function of their openings. Figure C.8 shows the forceps grasping many different soft circles. These circular objects were 3D printed with 100% infill using a thermoplastic polyurethane (PolyFlex TPU95). The finger spring of the model is in our case zero and the input torque T_a is created by the extension of the springs at the handle. To measure the actuation torque at the jaw of the forceps, a force sensor (Marveldex Force Sensing Resistor) was inserted during the grasp of several steel dowel pins with different diameters (instead of the soft object) to measure the generated forces, and in all experiments the angles between the jaws and the distance of contact were measured on pictures. The actuation torque T_a was then estimated for different openings of the forceps based on a linear interpolation. Figure C.10 presents the torque from the measurements with the forceps and from the numerical model. After a calibration of the PSM to match the applied torque in one scenario by comparison between the experiment and a simulation (see Fig. C.8(f) and C.9(f)), the other grasps are simulated. The results corresponding to Fig. C.8 are presented in Fig. C.9. Based on the Young Modulus of the thermoplastic polyurethane used ($E = 29MPa$) and the approximation of the Van Gelder model, a coefficient of 0.033 was applied to the spring stiffnesses to fit the simulation.

Tabs. C.3 and C.4 present the parameters of the finger and the objects, respectively. For the simulations, the spring value of k_f and the initial angles of the jaw are chosen arbitrarily. A spring value equal to zero would have led to an infinite number of possible position of the jaw. The torques due to this spring are therefore subtracted from the actuation torque in order to calculate the real torque applied on the object. The resulting torques correspond to the results shown in Fig. C.10. These torques resulting from the model corresponding to the angle observed in the experiment are similar to the measured ones. The gap between the values may be due to the lack of precision of the force sensor from which consistent readings were sometimes difficult to obtain.

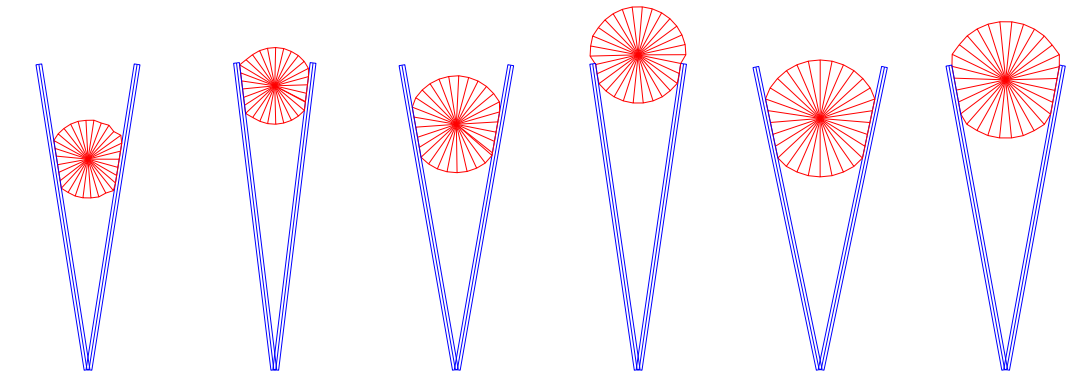
Shapes of the deformed PSM also match the observed ones with the forceps. One example where both shapes from Figs. C.8(d) and C.9(d) are superposed is illustrated in Fig. C.11. With the real and simulated jaws at the same position, the computed deformations are very close in all cases, event at the edge of the fingers where the curvature of the cylinder changed drastically. The small teeth on the forceps jaws also appear to have little influence of the deformation of the objects.

Figure C.7 Surgical forceps.



(a) (b) (c) (d) (e) (f)

Figure C.8 Grasping by the forceps of circles with diameters of $4mm$ (a and b), $5mm$ (c and d) and $6mm$ (e and f).



(a) (b) (c) (d) (e) (f)

Figure C.9 Simulation of grasping by a forceps of cylinders with diameters of $4mm$ (a and b), $5mm$ (c and d) and $6mm$ (e and f).

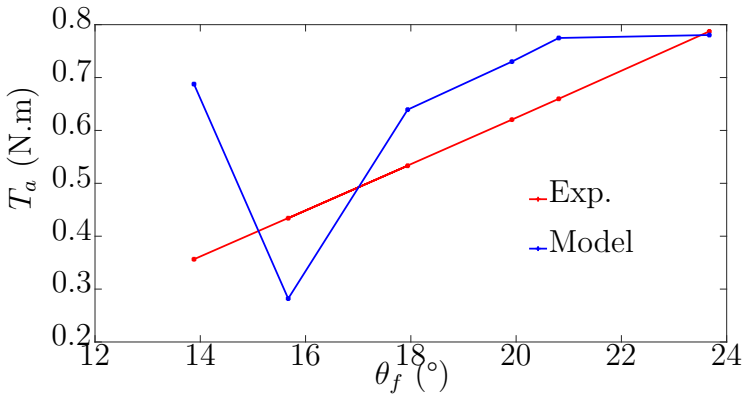


Figure C.10 Comparison between the torque from the model and experimental results.

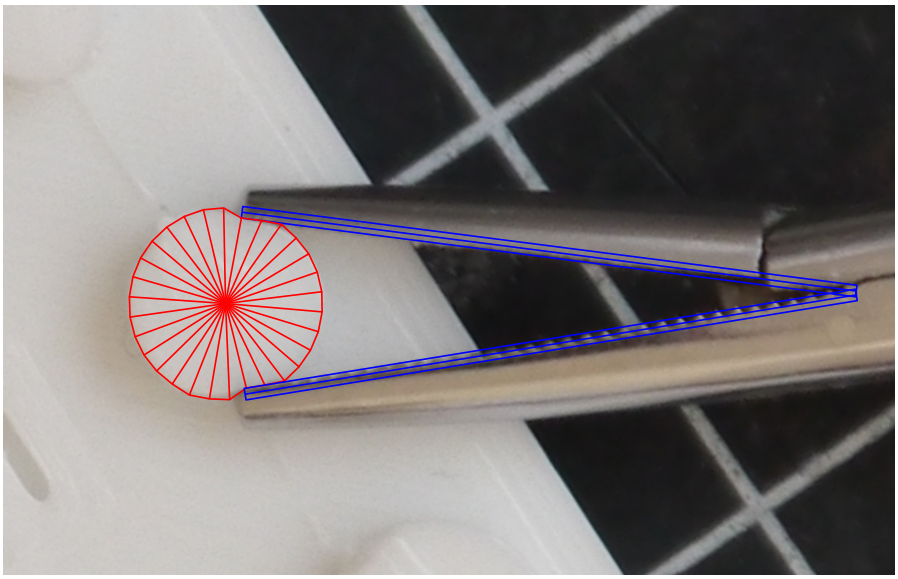


Figure C.11 Superposition of the deformed shape predicted by the model and an experimental picture ($r = 2.5mm$, $d_1 = 16mm$).

Tableau C.3 Parameters of the finger in the simulation presented in Fig. C.9 (all lengths are in millimeters)

l_r	l_l	d_3	w
16	16	0.1	0.3

Tableau C.4 Parameters of the object in the simulations presented in Fig. C.9 (all lengths are in millimeters).

	r	d_1
(a)	2	11
(b)	2	15
(c)	2.5	13
(d)	2.5	16
(e)	3	13
(f)	3	15

Conclusions

While a FEA produces accurate simulations, the calculation time and complexity of using this approach is a major drawback for certain applications, such as design optimizations or real-time rendering. Because of its simplicity, a PSM is shown here to be an alternative to achieve accurate simulations within a reasonable time. In this paper, a PSM was tested to simulate the interaction between an angular gripper and deformable circles. The goal was to find the trade-off between accuracy and calculation time in terms of number of elements used to approximate the circles with the ultimate objective to optimize future design of grippers. After a brief introduction of the model developed in a previous work, this model is used to evaluate the deformations of different sizes of objects depending on the number of triangles for different squeezing pressures and the calculation times are recorded. Twenty elements seem to be sufficient in most cases. However, to obtain the same level of accuracy, the results indicate that fewer elements are necessary with larger size objects (compared to the gripper). Finally, experimental observations with a surgical forceps grasping cylinders and the equivalent results from the numerical model are compared. This comparison yielded that both the experimental and simulated deformations are close to each other and thus, the

model seems accurate.

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