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A Design Framework that Employs a Classification Scheme and Library for Compliant Mechanism Design

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Library for Compliant Mechanism Design

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ABSTRACT

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Limited resources are currently available to assist engineers in implementing compliant members into mechanical designs. As a result, engineers often have little to no direction incorporating compliant mechanisms. This thesis develops a conceptual design framework and process that utilizes a proposed classification scheme and a library of mechanisms to help engineers incorporate compliant mechanisms into their applications.

As the knowledge related to the synthesis and analysis of compliant mechanisms continues to grow and mature, and through the classification scheme established in this thesis, compliant mechanisms may become more extensively used in commercial mechanical designs. This thesis also demonstrates a design approach engineers can use to convert an existing rigid-body mechanism into a compliant mechanism by using the established classification scheme and a library of compliant mechanisms. This approach proposes two possible techniques that use rigid-body replacement synthesis in conjunction with a compliant mechanism classification scheme. One technique replaces rigid-body elements with a respective compliant element. The other technique replaces a complex rigid-body mechanism by decomposing the mechanism into simpler functions and then replacing a respective rigid-body mechanism with a compliant mechanism that has a similar functionality. These techniques are then demonstrated by developing and designing a competitive and feasible compliant road bicycle brake system.

Keywords: compliant mechanisms, design methodology, lamina emergent mechanisms, classification scheme

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CHAPTER 1. INTRODUCTION

1.1 Problem Statement

To succeed in an increasingly competitive and global market, mechanical designs need to reduce cost and spatial requirements. Pahl et al. [1] discuss the importance of identifying cost factors in the design process, and proposes the following methods: reduce part-count, reduce assembly and manufacturing time, and simplify manufacturing processes. Compliant mechanisms potentially provide these cost advantages compared to rigid-body mechanisms [2].

Though compliant mechanisms are advantageous for many products, engineers often have difficulty incorporating them into their designs [2, 3]. This difficulty is often caused by limited industry experience in implementing compliant mechanisms into devices when extensive analysis and synthesis methods are devoted toward more traditional rigid-body mechanisms. Consequently, rigid-body mechanisms are often implemented into a product where compliant mechanisms could have proved more advantageous [3]. One way to make compliant mechanisms more accessible to engineers is to have a resource of compliant mechanisms they can use in their designs.

The purpose of this thesis is to introduce a classification scheme for compliant mechanisms, provide a framework were the classification scheme can be used to compile a library of designs, illustrate an example of a reference source by compiling a library of lamina emergent designs, and illustrate design examples using this classification scheme with a library of designs.

1.2 Research Approach

1.2.1 Classification Scheme

The first phase of this research is the development of a classification scheme for compliant mechanisms. This classification scheme will be one where it can be implemented into a resource for engineers. The first step in compiling this classification scheme is to determine current classification schemes used with rigid-body mechanisms. These schemes will provide the basic groundwork for categorizing compliant mechanisms. Once this groundwork is established, the main characteristics of compliant mechanisms will be divided and categorized. With this classification scheme established a framework for a reference library of designs will be provided.

1.2.2 Lamina Emergent Mechanism Library

In recent years a wide variety of compliant mechanisms have been developed and proposed. A new class of compliant mechanisms, called Lamina Emergent Mechanisms (LEMs), has been developed where it has spatial and fabrication advantages. LEMs are fabricated in a plane (e.g. sheet metal), though their motion is out of the fabrication plane [4]. Because LEMs are fabricated in a planar form, they utilize less expensive manufacturing processes, and their initial configuration is spatially compact though they can perform complex motions. The second research phase involves demonstrating how a library of existing design may look like by illustrating a library of lamina emergent designs.

1.2.3 Product Design with Lamina Emergent Mechanism Library

The final phase in the research will illustrate how the classification scheme and a library of compliant designs could be used in conjunction with the design methodology rigid-body replacement synthesis to design a device.

1.3 Background

1.3.1 Compliant Mechanism Classification Scheme

Numerous mechanism classification schemes have been developed and refined. Frank Reuleaux, known as the “Father of Kinematics,” was one of the first engineers to use symbols to represent machines and kinematic pairs [5]. Using his symbol notation he classified mechanisms into three categories: class or name of body or line, form of the body, and relation of one element to its pair [5–7].

Artobolevsky saw the need to compile and classify a variety of mechanisms for a reference book for designers and inventors. To do this, Artobolevsky classified machines into two categories, first according to structural and constructional features (e.g. Elements of Mechanisms, Simple Lever Mechanism), and then subdivided them into their service function. The indices for these groups are indicated with one to three letter abbreviations of the category name (e.g. Simple Lever Mechanisms index is SL) [8]. Numerous other resource handbooks have been created for the purpose of being a sourcebook of mechanisms for designers and inventors [9–12]. Though these books contain numerous machines, none include compliant mechanisms into the machine designs. Thus the proposed classification scheme would assist future reference books to incorporate compliant mechanisms.

An important factor to consider while classifying compliant mechanisms is that they often behave similar to rigid-body mechanisms [13]. Part of the classification scheme will overlap with classification strategies used for rigid-body mechanisms. However, many other attributes of compliant mechanisms will require additional classification such as the basic compliant mechanism classifications from Midha et al. [13].

Existing classification schemes may be gathered and a reference source specifically for compliant mechanisms can be compiled using a method similar to Artobolevsky's approach. This scheme will be demonstrated by developing a database of LEMs (consistent with the scheme), and demonstrate the use of the resulting database by designing a mechanism.

1.3.2 Lamina Emergent Mechanisms

LEMs incorporate the characteristics of compliant mechanisms, ortho-planar mechanisms, metamorphic mechanisms, and change point mechanisms [4]. Compliant mechanisms are devices that achieve mobility through deflection of flexible members [14]. Ortho-planar mechanisms are defined by Praise et al. as "mechanisms with links that can be simultaneously located in a plane with motion out of that plane" [15]. Dai and Jones define metamorphic mechanisms as a "mechanisms whose number of effective links changes as it moves from one configuration to another" [16]. Change point mechanisms are mechanisms where the Grashofs equation is equal (i.e. $s+l=p+q$), meaning that all links are collinear [17]. LEMs are advantageous in a number of design applications due to their characteristics: compact in highly constrained space, compact in transportation but deployed during operation, compact packaging and shipping, and manufacturable where limited processes are desirable [4].

With understanding of the mechanics of LEMs and how they are a branch of compliant mechanisms, along with some basic classifications already established [4, 18, 19], an classification method will be established.

CHAPTER 2. PROPOSED COMPLIANT MECHANISM CLASSIFICATION SCHEME

2.1 Introduction

[1]The purpose of this chapter is to propose a classification scheme for categorizing compliant elements and mechanisms, with the goal of providing a foundation for resources that would help engineers rapidly sort through the body of existing compliant elements and mechanisms, and find the one that is most suited for implementation in their design. Such a classification scheme would also increase engineers' awareness of compliant mechanisms already in existence as well as increase their ability to develop new mechanisms that use compliant mechanisms.

Although compliant mechanisms generally possess similar functionality to rigid-body mechanisms, they can be advantageous compared to traditional rigid-body mechanisms in that they (1) require fewer parts, (2) are easier to fabricate and assemble, (3) have more repeatable motion, (4) may cost and weigh less, and (5) are easily miniaturized [2, 14]. Engineers may be reluctant to incorporate compliant mechanisms into their designs [2, 3], since they may be more familiar with the synthesis and analysis of rigid-body mechanisms than compliant mechanisms. For example, the motions and forces associated with rigid-body mechanisms can be decoupled to predict and design for a specific task; whereas compliant mechanisms have highly coupled kinematics and kinetics. Nonlinearities associated with large-deflection motion also complicate their design and analysis. As a result of the inherent difficulty associated with compliant mechanisms and shortage of examples, many engineers have pursued designs that utilize rigid-body mechanisms and have over-looked, or not had the knowledge or capability to design compliant mechanisms. With the growing demands for the advantages that compliant mechanisms offer, more extensive work has been performed to incorporate synthesis techniques [14, 21–23] and to simplify the analysis [14, 24–27] for designing compliant mechanisms.

[1]In Proceedings of the ASME International Design Engineering Technical Conferences [20]

This chapter proposes a scheme to classify compliant mechanisms and explains how a possible library of compliant designs, based on this scheme, could be used to help engineers implement compliant mechanisms into their applications. This classification scheme incorporates similar classification techniques used to categorize traditional rigid-body mechanisms as well and is consistent with existing nomenclature and classification schemes [13]. With a classification scheme selected, it will be possible for new and old compliant mechanisms to be compiled into a library of compliant designs.

2.2 Background

Mechanism classification schemes for rigid-body mechanisms have been developed and refined. Frank Reuleaux was one of the first engineers to use symbols to represent machines and kinematic pairs [5]. Using a symbol notation he classified mechanisms into three categories: class or name of body or line, form of the body, and relation of one element to its pair [5–7].

Following the work of Frank Reuleaux, two major schemes for classifying rigid-body mechanisms have been developed. The first scheme subdivides mechanisms into linkage modules and classifies them according to the behavior of their kinematic chains. The second scheme classifies mechanisms according to both their constructional features and function. These schemes are discussed below.

2.2.1 Linkage Classification Schemes

Extensive work has been done to understand and classify linkages. Two common linkages are the four-bar and six-bar linkages.

Four-bar mechanisms are perhaps the most common mechanisms used by engineers. Grashof was one of the first who analyzed and classified four-bar mechanisms; as a result he was a forerunner to four-bar mechanism research [28]. He classified mechanisms according to the Grashof criterion, which describes four-bar mechanisms by the mathematical equation

$$s + l \leq p + q \tag{2.1}$$

where *s*, *l*, *p*, and *q* are the lengths of the shortest, longest, and the two intermediate links, respectively. If equation (2.1) is satisfied, the mechanism is classified as a Grashof mechanism, and Grashof mechanisms can be categorized further depending on which link is the shortest [29]. Barker refined Grashof's classification scheme by separating four-bar mechanisms into three major categories, Grashof, non-Grashof, and change point [17].

Watt and Stephensen mechanisms are typical classifications of six-bar mechanisms. A Watt mechanism is characterized by having its two ternary links connected. A Stephensen mechanism is characterized by having its two ternary links separated by a binary link [30].

2.2.2 Feature and Function Classification

A variety of machine source books have been written which classify mechanisms according to their function and construction features. Additional approaches have been proposed for classifying mechanisms according to their function [31, 32].

Artobolevsky saw the need to compile and classify a variety of mechanisms for a reference book for engineers [8]. He classified machines into two categories: first according to structural and constructional features (e.g. Elements of Mechanisms, Simple Lever Mechanism), and then subdivided them according to the service function. The reference index for these groups are indicated with two to three letter abbreviations of the category name (e.g. Simple Lever Mechanisms index is SL) [8]. Numerous other resource handbooks have been created for the purpose of being a sourcebook of mechanisms for engineers [9–12]. These books contain few compliant mechanisms; thus, the proposed classification scheme could assist future reference books incorporate compliant mechanisms.

2.3 Compliant Mechanism Characteristics

Identifying characteristics that define a mechanism as compliant is useful for classification. Compliant mechanisms transfer motion, force, or energy by using the deformation of flexible members [14]. This is unlike rigid-body mechanisms that achieve their motion from discrete parts connected by moving joints. Although compliant mechanisms share similar characteristics with rigid-body mechanisms, additional classification is required. For example, there is one basic configuration for a rigid-link parallel-guided mechanism, but 28 compliant mechanisms that correspond to this motion [22, 33].

Compliant mechanisms store elastic energy through the deformation of their flexible members, while a rigid-body mechanism cannot store elastic energy without the addition of other components such as springs. Furthermore, the behavior of compliant mechanisms (their range of motion, stiffness characteristics, load capacity, etc.) are strongly dependent on the choice of material, whereas the material used to make a rigid-body mechanism does not affect the mechanisms behavior as long as the material is sufficiently rigid. Also, because compliant mechanisms may have fewer parts, they may require less assembly time than rigid-body mechanisms. This is because their motion and energy storage elements are integrated into fewer components. This results in some advantages of compliant mechanisms such as they do not require lubrication, and they achieve their motion without friction or backlash. However, because compliant mechanisms deflect through flexible members, fatigue of flexible members should be considered.

Midha et al. [13] provided the nomenclature and classification of link and segment identification. The proposed classification scheme is intended to extend this to provide a foundation for a possible future library of compliant designs.

2.4 Classification Scheme

To achieve the objective of assisting engineers in more easily identifying compliant mechanisms for their designs, the classification should be organized in a simple and intuitive manner.

The proposed scheme is broken into two classification approaches: categorization (which embodies the designs), and index (which references the designs in the categorization classification approach). The first approach categorizes these mechanisms using a method similar to that employed by Artobolevsky [8], which is classifying mechanisms according to their function. This approach includes the mechanism depiction and a concise description of its behavior. The second approach indexes the mechanisms that are categorized. This index will help engineers find multiple mechanism design possibilities that are suited for a desired application and/or satisfy constraints imposed in the fabrication method (e.g. if the mechanism is a biomedical mechanism, the mechanism's material needs to be biocompatible).

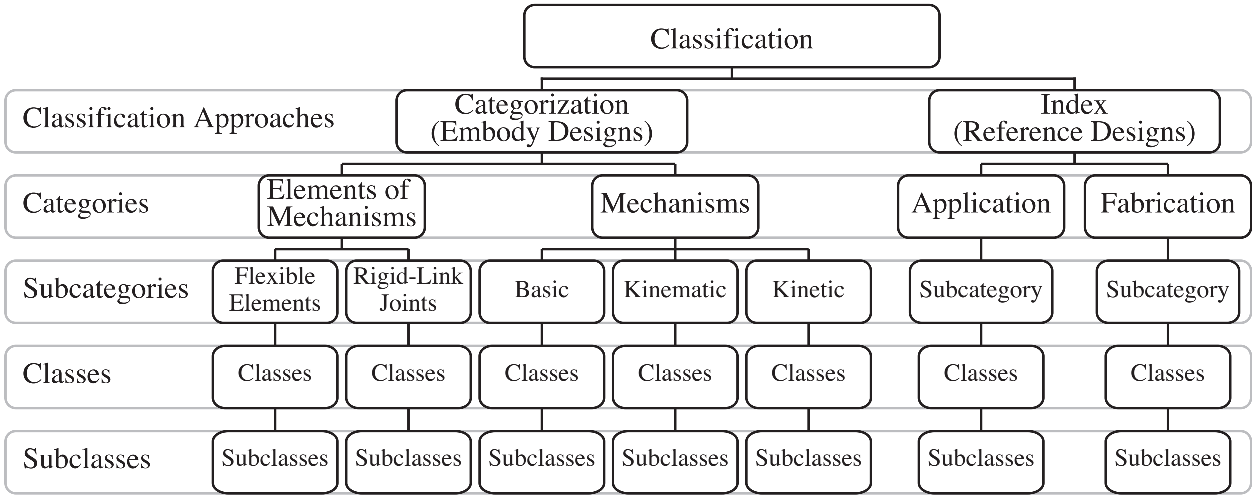


Figure 2.1: Classification scheme hierarchy

The complete hierarchy of the classification scheme is found in Figure 2.1. Where each classification approach is subdivided into categories, subcategories, classes, then subclasses to appropriately categorize a compliant design.

2.4.1 Categorization

Two systems of categorizing compliant mechanisms were determined to be convenient for engineers: categorizing according to *Elements of Mechanisms* and *Mechanisms*.

Elements of Mechanisms

Compliant *Elements of Mechanisms* are defined as a system of compliant and/or rigid segments that achieve a distinct motion. Understanding the elements used in compliant mechanisms can help engineers understand how a compliant mechanism operates and the advantages and disadvantages of these elements. Also, techniques have been established where compliant elements may be used to replace rigid joints [3]. Some examples of elements of compliant mechanisms are the large-displacement elements by Trease et al. [34], the compliant rolling-contact element (CORE) by Cannon and Howell [35], the lamina emergent torsion (LET) joints by Jacobsen et al. [4], and the split-tube flexures by Goldfarb and Speich [36].

Table 2.1: *Elements of Mechanisms*' subcategories and classes

Flexible Elements (FE)	• Beam	(FB)
	• Revolute	(FR)
	– Hinge	(FRH)
	– Scissor	(FRS)
	– Torsion	(FRT)
	– Lamina Emergent	(FRL)
	• Translate	(FT)
	– Lamina Emergent	(FTL)
	• Universal	(FU)
	– Lamina Emergent	(FUL)
	• Flexible Elements: Other	(FO)
Rigid-Link Joints (RLJ)	• Revolute	(RR)
	• Prismatic	(RP)
	• Universal	(RU)
	• Rigid-Link Joints: Other	(RO)

The *Elements of Mechanisms* category will be subdivided into two different subcategories, then into different classes where existing designs can be categorized. The two subcategories are: *Flexible Elements* and *Rigid-Link Joints*. It was deemed necessary that the *Rigid-Link Joints* subcategory should be included in this classification because compliant mechanisms utilize both flexible and rigid elements to achieve their kinematic and kinetic behavior. The specific class characterizes the functional operation of the element. In some cases, additional subclasses are appended to a class where there are unique characteristics of elements that needed to be further classified. The subcategories and their subsequent classes for the *Elements of Mechanisms* category are listed in Table 2.1, these classes are defined in section 3.2.1.

Mechanisms

Compliant *Mechanisms* are defined as a system of rigid bodies connected by elements to achieve a desired motion and/or force transmission. The *Mechanism* category is subdivided into three subcategories: *Kinematic*, *Kinetic*, and *Basic*. Mechanisms with the primary purpose of obtaining a specified motion, path, orientation, or other positioning relationship, are classified under the *Kinematic* subcategory. Those mechanisms with the primary purpose associated with their force-defection relationship, energy storage, or other force or energy related function, are classified under the *Kinetic* subcategory. *Basic* mechanisms are those where the kinematics or kinetics of the mechanism is not defined. The motion (kinematics) and force-deflection behavior (kinetics) of compliant mechanisms are highly coupled; however most compliant mechanism applications are designed with a primary function related either to their intended motion or their force-deflection behavior. These subcategories are then subdivided into classes for categorization of existing compliant mechanism designs. Additional subclasses may be appended to a class that will define a unique characteristic of a mechanism, where further classification was required. The subcategories and their subsequent classes for the *Mechanisms* category are listed in Table 2.2, these classes are defined in section 3.2.2.

Limitations of the Classification Scheme

The proposed classification scheme for compliant mechanisms is based upon existing schemes that classify rigid-body mechanisms. As a result, it is difficult to classify compliant elements and mechanisms in a distinct class. This is because (1) mechanisms may by classified in the *Elements of Mechanisms* category because their behavior is similar to the function of a rigid-link element, (2) the mechanisms are not classified by all their kinematic and kinetic characteristics but by their dominating characteristics, and (3) the classification is ever expanding to accommodate new elements of mechanisms or mechanisms that require a new class in order to be classified.

2.4.2 Index

The purpose of the index is to assist the engineer in finding compliant mechanisms by using different approaches. To accomplish these different approaches, two separate indexing methods have been developed. The first method helps engineers find compliant mechanisms that already exist for certain *Applications*. The second method helps engineers find compliant mechanisms according to issues relating to their *Fabrication*. This second section was deemed necessary because the characteristics and behavior of a compliant mechanism is largely determined by how they are

Table 2.2: *Mechanisms*' subcategories and classes

Basic Mechanism (BA)	• Four-Bar Mechanism	(BF)
	• Six-Bar Mechanism	(BS)
Kinematics (KN)	• Translational	(TS)
	– Precision	(TSP)
	– Large Motion Path	(TSL)
	– Orthogonal	(TSO)
	• Rotational	(RT)
	– Precision	(RTP)
	– Large Motion Path	(RTL)
	– Orthogonal	(RTO)
	• Translation—Rotation	(TR)
	– Precision	(TRP)
	– Large Motion Path	(TRL)
	– Orthogonal	(TRO)
	• Parallel Motion	(PM)
	– Precision	(PMP)
	– Large Motion Path	(PML)
	• Straight Line	(SL)
	• Stroke Amplification	(SA)
	• Spatial Positioning	(SP)
	– Precision	(SPP)
	• Metamorphic	(MM)
	• Ratchet	(RC)
	• Kinematic: Other	(KMO)
Kinetics (KN)	• Energy Storage	(ES)
	– Clamp	(ESC)
	• Stability	(SB)
	– Bistable	(SBB)
	– Multistable	(SBM)
	• Constant Force	(CF)
	• Force Amplification	(FA)
	• Dampening	(DP)
	• Mode	(MD)
	– Buckle	(MDB)
	– Vibration	(MDV)
	• Kinetic: Other	(KNO)

fabricated. As more compliant mechanisms are designed, these designs will be appended to the Index.

2.5 Library of Compliant Designs

One purpose for the classification scheme is to provide a foundation for a possible future library of compliant designs. A proposed organization of this library is described in this section, and several examples are provided.

2.5.1 Categorization

The library's categorization section contains the compliant designs. Associated with each design is a reference number and reference categorization that indicates the subcategory, and class of the design. The reference number specifies the category of the design, followed by an number assigned to the design (i.e EM-# represents an element and M-# represents a mechanism). The first reference categorization specifies the subcategory followed by the second reference categorization that specifies the class. The reference categorization subcategory and class are indicated by indices that can be found in Tables 2.1 and 2.2. The first two letters of the indices indicate the specific class, and if there is a third letter it indicates the specific subclass (e.g. FR is a flexible *revolute* class and FRH is the subclass *hinge* in the flexible *revolute* class).

2.5.2 Index

The index contains two separate indexing methods where compliant mechanisms will be classified according to their application and/or fabrication constraint. These categories will reference the mechanisms defined in the categorization section described above. The mechanism will be referenced by the mechanism's reference number and the reference index.

2.5.3 Mechanism Depiction

Each design is to be shown on its own chart to conveniently convey its pertinent information, and to help engineers quickly identify the element or mechanism and its characteristics. Each

<table>
<tr><td>Element or Mechanism Reference #</td><td>Name</td><td>SUBCATEGORY CLASS</td></tr>
<tr><td colspan="2">Figure 1
(1)

Figure 2
(2)</td><td>General description of element or mechanism, and if more information is available a reference will be included.

(1) References the element's or mechanism's segments, and if fabrication is important the figure will be displayed in the manufacturing layout.

(2) Displays the element's or mechanism's deformation.</td></tr>
</table>

Figure 2.2: Library of compliant designs template

chart consists of a reference number (indicated in the upper left-hand corner of the chart), name (upper center), reference categorization (upper right-hand corner), drawing (lower left-hand side), description along with any references where more information can be found (right-hand side), and a description of the drawings in an enumerated format (lower right-hand side) for each design, as shown in Figure 2.2.

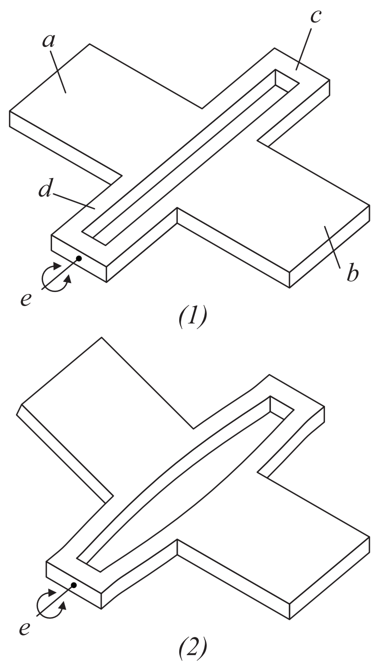
2.5.4 Examples

Three examples are provided to demonstrate the proposed classification scheme. The three examples are a LET joint, a HexFlex™, and a parallel-guided mechanism.

LET Joint

A LET joint is a single-layer, flexible element that provides rotational motion out of the plane of fabrication; this flexible element can be used for both macro and micro applications that require large angular deflections [37]. The LET joint would be classified under *Elements of Mecha-*

EM-1	Outside LET Joint	FE FRL



This element is a lamina emergent torsion (LET) joint. It is suited for applications where large angular rotation is desired, but high off-axis stiffness is not critical. It can be fabricated in a single plane. [37]

(1) Rigid segments *a* and *b* are attached to a mechanism. The flexible segments *c* and *d* are in bending and torsion, respectively, causing a rotation about the *e* axis.

(2) Deformed configuration of rotation about the *e* axis.

Figure 2.3: Outside LET Joint example

nism category, in the *Flexible Elements* subcategory, *revolute* class, and *lamina emergent* subclass. Figure 2.3 demonstrates how this mechanism would appear.

An engineer may approach a design problem with the desire to replace an existing rigid joint with a compliant element to simplify the assembly process. The design may be constrained to be made using a laser cutter, capable of cutting only planar devices, while maintaining the capability of out-of-plane motion. With these functional requirements in mind, the engineer would find the Outside LET Joint as a possible candidate.

An alternative approach is to consult the fabrication index under design layout. Under this index there is a lamina emergent mechanism (LEM) [4] class where the engineer can find a variety of elements of mechanisms and mechanisms that can be manufactured in a plane, but operate out of the plane of fabrication.

HexFlex

The HexFlex™ is a single-layer, multi-axis spatial positioning control mechanism, which can be used for both macro and micro applications that require precision positioning [38]. This mechanism would be classified under the *Mechanism* category, in the *Kinematic* subcategory, *spatial positioning* class, and *precision* subclass. Figure 2.4 demonstrates how this mechanism would appear.

Engineers have been designing multi-axis micromanipulators for MEMS application [40, 41]. However, an engineer may use the classification scheme to find a multi-axis micromanipulator that utilizes the advantages of compliant mechanisms (i.e. simplicity of design and ease of manufacturing/integration into MEMS devices [38, 42], and low cost). An engineer seeking this type of mechanism would look under the *Mechanism* category in the *spatial positioning* class under the *Kinematic* subcategory. In this class he/she would find the mechanism indicated in Figure 2.4 and similar mechanisms.

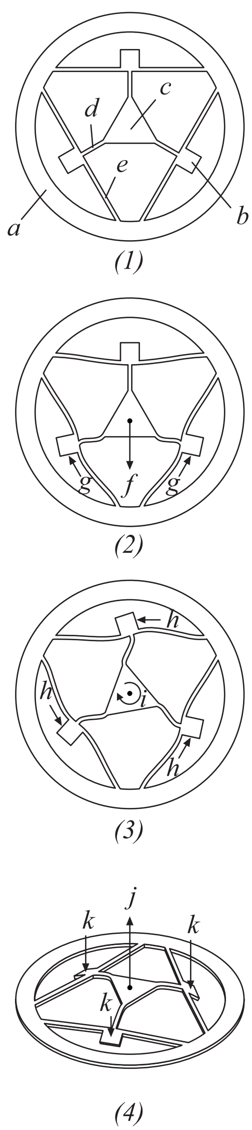
The engineer could also approach the design problem by finding a replacement mechanism through the application index. In the application index the engineer would look in the manipulators class under the MEMS category. The index would indicate the design shown in Figure 2.4 and similar mechanisms.

Parallel-Guided Mechanism

A parallel-guided mechanism is a mechanism whose two opposite links remain parallel throughout the mechanism's motion. This mechanism would be classified under the *Mechanism* category, in the *Kinematic* subcategory and *parallel motion* class, and also in the *Kinetic* subcategory and *energy storage* class. Figure 2.5 demonstrates how this mechanism would appear.

Engineers in the bicycle industry may want to design components that increase performance, decrease weight, and decrease cost [43]. A bicycle derailleur is one candidate for replacement by a compliant mechanism. A typical derailleur is a rigid-link parallel-guided mechanism with springs attached to provide the desired force-deflection relationship. An engineer could look in the *Kinematic* subcategory in the *parallel motion* class, and in the *Kinetic* subcategory under the *energy storage* class. In this classification the engineer would find multiple designs to choose from,

M-1	HexFlex	KM SPP



The HexFlex™ is a single-layer, multi-axis spatial positioning control mechanism, which can be used for both macro and micro applications that require precision positioning. [39]

(1) Rigid body *a* is fixed. Rigid bodies *b* are the actuator action tabs. Rigid body *c* is the motion stage. Flexible elements, *d* and *e*, allow infinitesimal motions.

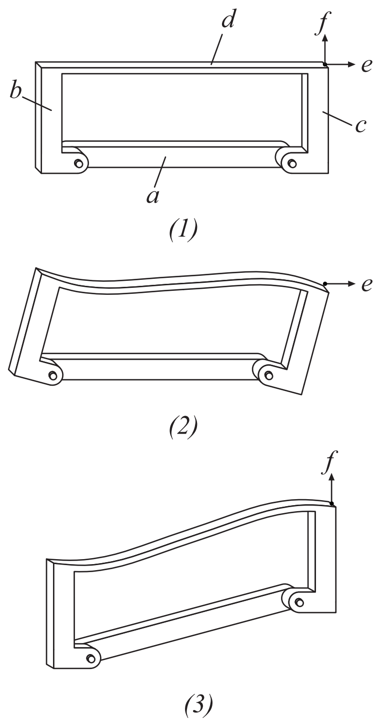
(2) Deformed configuration by planar displacement of the actuator tabs in the *g* direction, which causes the motion stage to displace in the *f* direction.

(3) Deformed configuration by planar displacement of the actuator tabs in the *h* direction, which causes the motion stage to rotate about the *i* axis.

(4) Deformed configuration by orthogonal displacement of actuator tabs in the *k* direction, which causes the motion stage to translate in the *j* direction.

Figure 2.4: HexFlex example

M-2	Parallel Guided	KM/KN PML/ES



This mechanism's links remain parallel throughout the mechanism's motion and is capable of large deflections with energy storage. [43]

(1) Rigid body *a* is a rigid link. Rigid bodies *b* and *c* are rigid segments. Segment *d* is a fixed-guided beam.

(2) When rigid body *a* is fixed, the mechanism deforms in the *e* direction.

(3) When rigid body *b* is fixed, the mechanism deforms in the *f* direction.

Figure 2.5: Parallel-Guided Mechanism example

including the design indicated in Figure 2.5. The engineer may also find this design by looking under the application index.

2.6 Conclusion

Many engineers are not familiar with compliant mechanisms—their function, application, implementation or their advantages. Currently, no library of compliant mechanisms exists with a classification scheme for helping engineers identify potential compliant mechanisms for a design. Such a resource may serve to increase engineers' awareness of compliant mechanisms and help them identify mechanisms appropriate for their applications.

The approach proposed in this thesis serves as a foundation for creating such a resource. The scheme would allow engineers to achieve compliant mechanism designs through multiple approaches: (1) their function and configuration, (2) their applications, and (3) according to fabrication constraints.

CHAPTER 3. LIBRARY OF LAMINA EMERGENT DESIGNS

3.1 Introduction

When creating new devices, it is often helpful to get inspiration from existing designs; this is particularly valuable when designing sophisticated devices that use compliance. One contribution to this thesis is the implementation of lamina emergent designs into a design reference, thus this chapter contains lamina emergent designs (see Tables 3.1 and 3.2). A more extensive reference resource for compliant designs are provided in Appendix A.

3.2 Class Definitions

For designs to be classified in their most ideal class, a greater understanding of the classes needs to be established. This section contains the class definitions that can help the reader understand the type of design that should be classified in a certain class.

3.2.1 Elements of Mechanisms Class Definitions

Flexible Elements (FE)

- **Beam (FB)** represents flexible elements that have a large length relative to the size of its width and thickness, and where the thickness is small compared to its width. These flexible elements usually have well-defined end loads, and their motion can be characterized by a rigid-link segment with a rotation about a characteristic pivot.

- **Revolute (FR)** represents flexible elements that have one degree of rotation; these elements have characteristics in which they behave similar to rigid-link pivots.

  – **Hinge (FRH)** represents unique flexible revolutes where the pivot is placed at the end of a rigid segment, such that these elements behave like a rigid-link hinge.

Table 3.1: LIbrary of LEMS (*Elements of Mechanisms*)

Name	*Reference Index*	*Categorization Index*	
		Subcategory	Class
Switch Back	EM-7	FE	FB
Reduced Inside	EM-22	FE	FRL
Reduced Outside Area Joint	EM-23	FE	FRL
Outside LET Joint	EM-24	FE	FRL
Inside LET Joint	EM-25	FE	FRL
Notch Joint	EM-26	FE	FRL
Groove Joint	EM-27	FE	FRL
LEM Translator	EM-29	FE	FTL
Reduced Outside Area Joint	EM-32	FE	FUL
Outside LET Joint	EM-33	FE	FUL
Inside LET Joint	EM-34	FE	FUL

Table 3.2: LIbrary of LEMS (*Mechanisms*)

Name	*Reference Index*	*Categorization Index*	
		Subcategory	Class
Rotational LEM	M-9	KM	RT
Rotational LEM	M-13	KM	RTL
Bricard 6R (LEM)	M-14	KM	RTL
Parallel-Guided LEM	M-25	KM	PML
Parallel-Guided LEM	M-26	KM	PML
Multi-Layer Parallel-Guided LEM	M-27	KM	PML
Hoeken (LEM)	M-28	KM	SL
Pantograph (LEM)	M-29	KM/KN	SA/FA
Multiple Stage Platform	M-30	KM/KN	SP/ES
HexFlex	M-31	KM	SPP
Lamina Emergent 4 Bar	M-32	KM	MM
Bistable Locking COPMM	M-33	KM/KN	MM/SBB
COPMM Bistable Switch	M-34	KM/KN	MM/SBB
Bistable COPMM	M-35	KM/KN	MM/SBB
Ortho–Planar Spring	M-37	KN	ES
Bistable Button	M-40	KN	SBB

  - **Scissor (FRS)** represents unique flexible revolutes where the pivot is not placed at the end of a rigid segment, such that these elements have a scissor action.
  - **Torsion (FRT)** represents unique flexible revolutes where the pivot is characterized at the center of these elements, and these elements behave such that they are in torsion.
  - **Lamina Emergent (FRL)** represents elements that are fabricated in a plane, but act like a pivot that operates out of the plane of fabrication.

- **Translate (FT)** represents flexible elements that allow one degree of translation, these elements have characteristics in which they behave like a prismatic joint.
  - **Lamina Emergent (FTL)** represents unique flexible translational elements where the elements are fabricated in a plane, but translate out of the plane of fabrication.

- **Universal (FU)** represents flexible elements that have two or more orthogonal, rotational degrees of freedom.
  - **Lamina Emergent (FUL)** represents unique flexible universal elements that can be fabricated in a plane, but have at least one degree of rotation that is out of the plane of fabrication.

- **Other (FO)** represents flexible elements that have unique characteristics, and are not easily classified in a specific class.

Rigid-Link Joints (RLJ)

- **Revolute (RR)** represents rigid-link elements that provide a single rotational degree of freedom between connecting links.
- **Prismatic (RP)** represents rigid-link elements, also known as sliding joints, that provide a single translational degree of freedom between connecting links.
- **Universal (RU)** represents rigid-link elements with two orthogonal rotational degrees of freedom.

- **Other (RO)** represents rigid-link elements that have unique characteristics, and are not easily classified in a specific class.

3.2.2 Mechanisms Class Definitions

Basic Mechanisms (BA)

- **Four-Bar Mechanisms (BF)** can be characterized to have similar configurations as a rigid 4-bar mechanism and also have general kinematic and kinetic characteristics.

- **Six-Bar Mechanisms (BS)** can be characterized to have similar configurations as a rigid 6-bar mechanism and also have general kinematic and kinetic characteristics.

Kinematic Mechanisms (KM)

- **Translational (TS)** represents mechanisms that have one or more translational degree of freedom.
  - **Precision (TSP)** represents unique translational mechanisms that have highly repeatable motion.
  - **Large Motion Path (TSL)** are unique translational mechanisms that have a large degree of translation relative to the size of the mechanism.
  - **Orthogonal (TSO)** are unique translational mechanisms where the input and output actions are in orthogonal directions.

- **Rotational (RT)** represents mechanisms that have one or more rotational degree of freedom.
  - **Precision (RTP)** represents unique rotational mechanisms that have highly repeatable motion.
  - **Large Motion Path (RTL)** are unique rotational mechanism that have a large degree of rotation relative to the size of the mechanism.
  - **Orthogonal (RTO)** represents unique rotational mechanisms where the input and output actions are in orthogonal directions.

- **Translation—Rotation (TR)** represents mechanisms that have one or more rotational and translational degrees of freedom.
  - **Precision (TRP)** represents unique translation—rotation mechanisms that have highly repeatable motion.
  - **Large Motion Path (TRL)** represents unique translation—rotation mechanisms that have a large degree of motion relative to the size of the mechanism.
  - **Orthogonal (TRO)** represents unique translation—rotation mechanism where the input and output actions are in orthogonal directions.
- **Parallel Motion (PM)** represents mechanisms whose opposite links remain parallel throughout the mechanisms motion path.
  - **Precision (PMP)** are unique parallel-motion mechanisms that have highly repeatable motion.
  - **Large Motion Path (PML)** represents a unique parallel-motion mechanisms that have a large degree of translation relative to the size of the mechanism.
- **Straight Line (SL)** represents mechanisms that have a specific point on the mechanism which produces a straight line though part of its motion.
- **Stroke Amplification (SA)** represents mechanisms that have a ratio of output motion to input motion, such that the output motion is greater than the input motion.
- **Spatial Positioning (SP)** represents mechanisms that control the position of a platform.
  - **Precision (SPP)** represents unique spatial positioning mechanisms that have highly repeatable motion.
- **Metamorphic (MM)** represents mechanisms whose number of effective links changes as they move from one configuration to another.
- **Ratchet (RC)** are mechanisms that allow a translational or rotational degree of motion in only one direction while preventing motion in the opposite direction.

- **Other (KMO)** represents mechanisms that have unique motion characteristics and are not easily classified in a specific kinematic class.

Kinetic Mechanisms (KN)

- **Energy Storage (ES)** represents mechanisms that store mechanical energy while it is being deformed.
  - **Clamp (ESC)** represents unique energy storage mechanisms that use the energy stored to grasp or hold an item.
- **Stability (SB)** represents mechanisms that have one or more stable position, where the mechanism is in equilibrium.
  - **Bistable (SBB)** represents mechanisms that have two stable equilibrium position.
  - **Multistable (SBM)** represents mechanisms that have more than two stable equilibrium positions.
- **Constant Force (CF)** represents mechanisms that produce a constant output force for a range of input displacements.
- **Force Amplification (FA)** represents mechanisms that have a ratio of output force to input force, such that the output force is greater than the input force.
- **Dampening (DP)** represents mechanisms designed for dampening purposes, such that the mechanisms reduce the force/motion amplitude of an oscillating system.
- **Mode (MD)** represents mechanisms that achieve a distinct pattern.
  - **Buckle (MDB)** represents mechanisms that achieve different modes through buckling.
  - **Vibration (MDV)** represents mechanisms that achieve different modes through vibration of an oscillating system.
- **Other (KNO)** represents mechanisms that have unique energy or force characteristics and are not easily classified in a specific kinetic class.

CHAPTER 4. UTILIZING A CLASSIFICATION SCHEME TO FACILITATE DESIGN

4.1 Introduction

[1]As the field of compliant mechanisms continues to mature and improve, and simplified analysis techniques become available, there is a growing need to compile compliant element and mechanism designs into a reference library [45] so engineers can embody compliance by using synthesis techniques. A compliant mechanism classification scheme was developed in Chapter 2 for the purpose of helping engineers find existing compliant mechanisms that they can incorporate into their products. This classification scheme also provided a framework for a possible library of compliant designs.

The purpose of this chapter is to demonstrate how to use this classification scheme in the design process and to illustrate examples from a possible library of compliant designs. To do this, the chapter indicates how type synthesis can be used in conjunction with the classification scheme. The chapter also demonstrates how the classification scheme could be used to redesign a rigid-body mechanism with the goal to illustrate the usefulness of the classification scheme and a library of compliant designs.

4.2 Background

To better understand the chapter, this section will provide a review of information related to: (1) a classification scheme for compliant elements and mechanisms, and (2) design methodologies for compliant mechanisms.

[1]In Proceedings of the ASME International Design Engineering Technical Conferences [44]

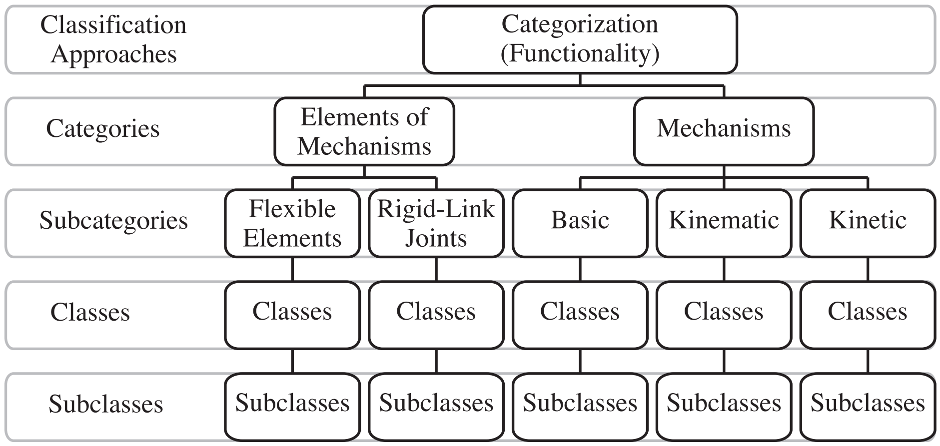


Figure 4.1: Functionality classification approach hierarchy

4.2.1 Classification Scheme

The classification scheme proposed in chapter 2 was developed to categorize compliant elements and mechanisms to help engineers rapidly sort through a body of compliant designs and find one suited for their application. To accomplish this objective, the classification scheme used three categorization approaches:

- Functionality
- Application
- Fabrication Constraints

This chapter focuses on using the functionality design approach. The functionality approach (referred to as *Categorization*) separates compliant designs into *Elements of Mechanisms* and *Mechanisms*. These categories are then subdivided into subcategories, classes, and subclasses (see Figure 4.1). The classes and subclasses of these categories are shown in Tables 2.1 and 2.2 with their definitions in section 3.2.

Elements of Mechanisms

Elements of Mechanisms are defined as a system of compliant and/or rigid segments that achieves a distinct motion. The *Elements of Mechanisms* category is divided into two subcategories: *Flexible Elements* and *Rigid-Link Joints*; because compliant mechanisms can use both flexible and rigid elements to achieve their function. The *Flexible Elements* subcategory contains more elements than the *Rigid-Link Joints* subcategory because the classification scheme is intended for compliance.

Mechanisms

Mechanisms are defined as a system of rigid bodies connected by elements to achieve a desired motion and/or force transmission. The *Mechanisms* category is subdivided into three subcategories: *Kinematics*, *Kinetics*, and *Basic*. This category was organized in this fashion because most compliant mechanism applications are designed with a primary function relating either to their intended motion or their force deflection behavior. Mechanisms with the primary purpose of obtaining a specified motion, path, orientation, or other positioning relationship are classified under the subcategory *Kinematics*. Those with the primary purpose associated with their force-defection relationship, energy storage, or other force or energy related function are classified under the subcategory *Kinetics*. Mechanisms that do not have a defined kinematic or kinetic characteristic (such as four-bar and six-bar mechanisms) are placed in the *Basic* subcategory.

4.2.2 Design Methodologies

Berglund et al. [3] indicated that methodologies are needed to help engineers design mechanical devices with flexible members. This chapter focuses on rigid-body replacement synthesis, but other designs methodologies have also been developed. Gallego and Herder [46] gave a comprehensive and conceptual overview of the different design methods for compliant mechanisms, which are: (1) rigid body replacement synthesis [3, 14, 22], (2) freedom and constraint topologies [47,48], (3) building blocks [23,49], and (4) topology optimization [50,51]. This chapter will focus on the rigid-body replacement method, because this technique can be beneficial to engineers who are more familiar with the synthesis and analysis of rigid-body mechanisms than compliant mechanisms. Then they can use the classification scheme to convert the rigid-body mechanism into a compliant mechanism.

Rigid-body replacement synthesis involves designing a rigid-body mechanism that accomplishes the desired function and then converting the design into a compliant mechanism [21]. The process is outlined in Figure 4.2. The classification scheme is helpful in the "Convert Rigid-body to Compliant Mechanism" step (see Figure 4.3); for instance, the conversion process replaces the rigid-link joints with flexible elements that can be found using the classification scheme. Furthermore, when the kinematic geometry of the rigid-body mechanism is determined, these individual elements can be chosen based upon the allowable stress, loading requirements and additional functional performance, and multiple design configurations can be found by using methods such as type synthesis. Once the conversion process has taken place, the kinematic and kinetic functionality can be analyzed [14].

4.3 Type Synthesis

Through type synthesis (specifically topological synthesis), permutations of alternate designs can be formed by considering the elemental level of compliant mechanisms [22, 33, 52, 53]. This method is helpful because the designs in the *Mechanisms* category are devices that are embodied for a specific function, though they do not contain all the possible configurations. Through type synthesis numerous other configurations can be deployed that may also have additional functionality because of the characteristics that are associated with compliance.

4.3.1 Topological Synthesis

Type synthesis is a process that "predicts which combination of linkage topology and types of joints may be best suited to solve a particular task" [52]. Type synthesis can be subdivided into (1) topological analysis, (2) topological synthesis, and (3) number synthesis. Topological synthesis is the process that enumerates mechanism structures based upon motion, degrees of freedom, and the number of links; this method is based upon graph theory [22, 54]. Murphy et al. [22] provides a systematic technique for enumerating non-isomorphic compliant mechanisms using type synthesis. Derderian et al. [33] and Brooks et al. [55] used Murphy's technique to find all the possible configurations for a compliant parallel-guided mechanism and 5-bar mechanical disc brake mechanism, respectively.

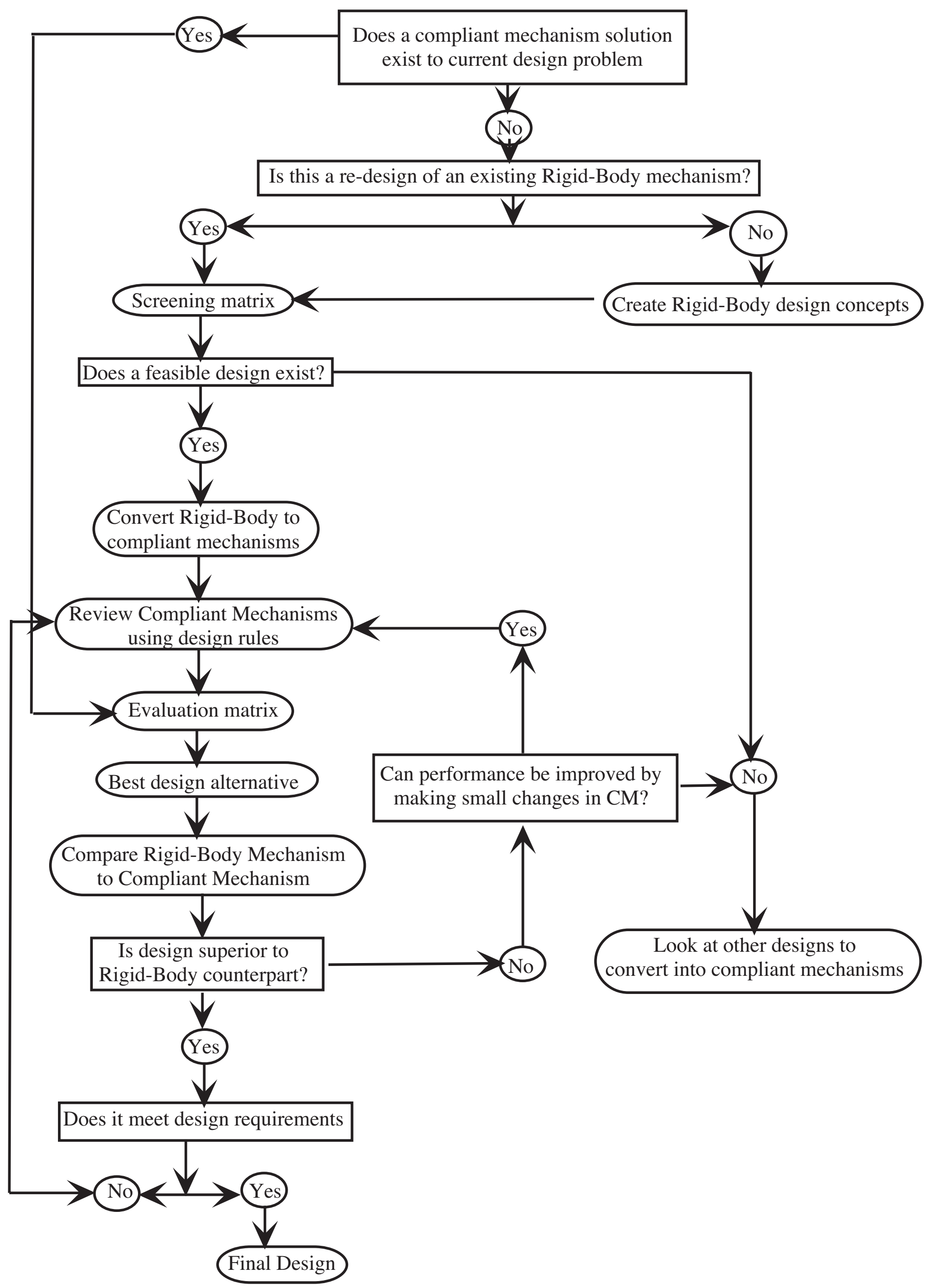


Figure 4.2: Rigid-body replacement synthesis design approach [3]

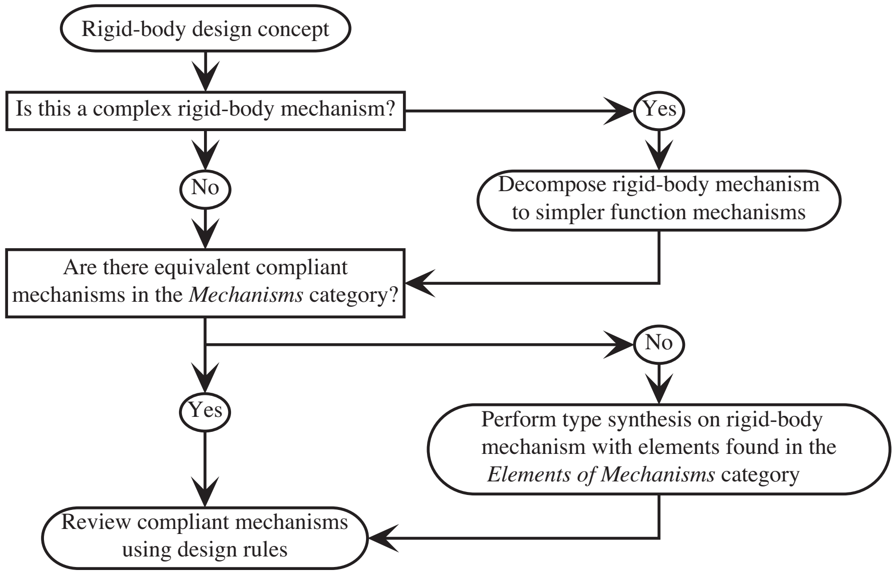


Figure 4.3: Rigid-body replacement synthesis conversion using a classification scheme

A key objective of this chapter is to demonstrate that type synthesis can be performed on a rigid-body mechanism to determine numerous configurations, and can use compliant elements for additional functionality. The elements chosen can be found in the classification scheme's *Element of Mechanism* category. Thus, this chapter indicates how to use a library of compliant designs with type synthesis, rather than a mathematical approach to discover all possible configurations of a specific mechanism for the described elements. This method can also be performed on existing mechanisms in the *Mechanism* category.

4.3.2 Parallel-Guided Mechanism Example

A compliant parallel-guided mechanism was chosen to demonstrate the possible benefits of using type synthesis. Derderian et al. [33] performed type synthesis on a compliant parallel-guiding mechanism with pin joints, flexible beams, and small-length flexural pivots that resulted in 28 possible configurations. By using the classification scheme more elements can be found in a library of compliant designs. Some possible revolute elements that can be included in this process are: living hinges [14], cross-axis flexural pivot [14, 56], CORE [35], isolated and inversion based high-compression elements [57], split tube flexures [36], large-displacement rotational element [34], LET joints [4], etc. Figure 4.4 illustrates possible configurations, where the mechanism is

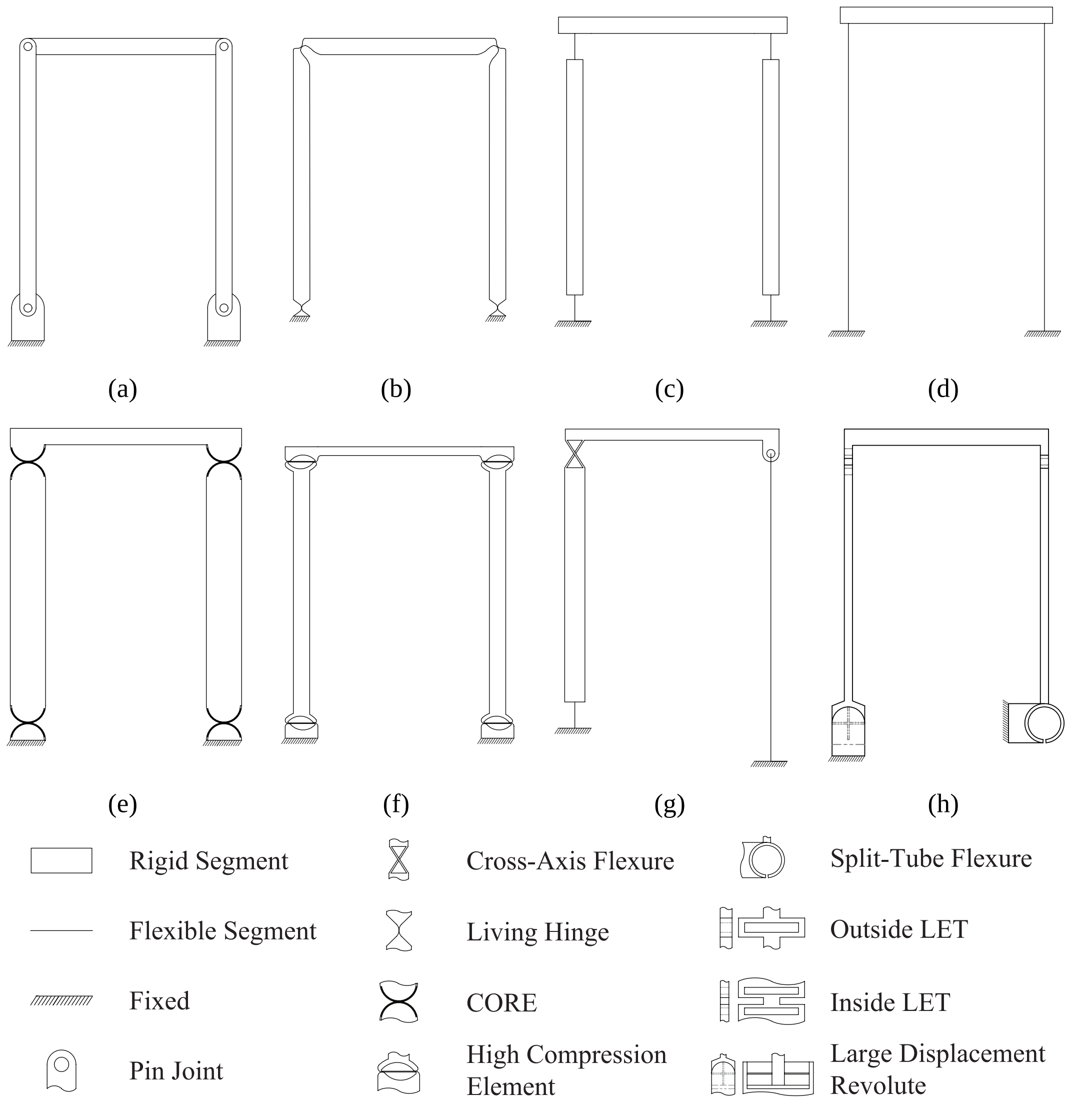


Figure 4.4: Parallel-guided mechanisms with the revoluting elements as (a) pin joints, (b) living hinges, (c) small-length flexural pivots, (d) flexible beams, (e) CORE and a (f) high-compression compliant element, and a combinations of elements ((g) and (h))

composed of the same revolute element (see Figures 4.4(a)-4.4(f)), or a combination of elements (see Figures 4.4(g) and 4.4(h)). Graph theory can be used for a more complete representation of all the possible configurations for the elements listed above.

Compliant elements can still be used for the general function of a parallel-guided mechanism, but some can also be used for some specific or unique functions as well, such as:

- Energy Storage

- Precision
- Large Motion Path
- High Compressive Loads

The elements chosen in the type synthesis process will determine additional functionality of the mechanism, while maintaining the original function. For example, if the design of a parallel-guided mechanism is also required to operate in a high-compressive-load environment; the types of elements that should be used are compression based elements, such as: pin joints, CORE, and isolated and inversion based high-compressive elements [57]. Also, the engineer should be aware of the characteristics of certain elements. For example, the CORE provides a rotational degree of freedom, but the axis of rotation changes location through its operation. Thus, if the CORE was chosen for a parallel-guided mechanism it would be advantageous for all of the four rotational elements to be a CORE.

4.4 Design Example

There are two design approaches based upon the classification scheme that can be used to redesign a rigid-body mechanism as a compliant mechanism. The first approach is to find an existing compliant mechanism in the *Mechanism* category of the classification scheme, which would have a similar function as a rigid-body mechanism. The second approach is to use rigid-body replacement synthesis to replace the elements or decomposed mechanisms with a compliant counterpart.

This chapter will illustrate both design approaches by redesigning an Audi A4 cup holder (see Figure 4.5). The first step is to determine which approach to consider. Since there is currently no compliant design that has a similar functionality as the cup holder in the *Mechanism* category, the best approach is to use rigid-body replacement synthesis.

Rigid-body replacement synthesis is a design process that begins with designing a rigid-body mechanism and converting it into a compliant mechanism. This conversion process has two approaches (see Figure 4.3). The first approach is to decompose a complex mechanism into mechanisms that have a relatively simple functionality, then replace respective rigid-body mechanisms

Figure 4.5: Audi A4 cup holder

with a compliant counterpart. The second approach is replacing the rigid-body elements with compliant elements that have a similar functionality. A common approach used in conjunction with this form is type synthesis, as discussed in section 4.3.

4.4.1 Design Rules

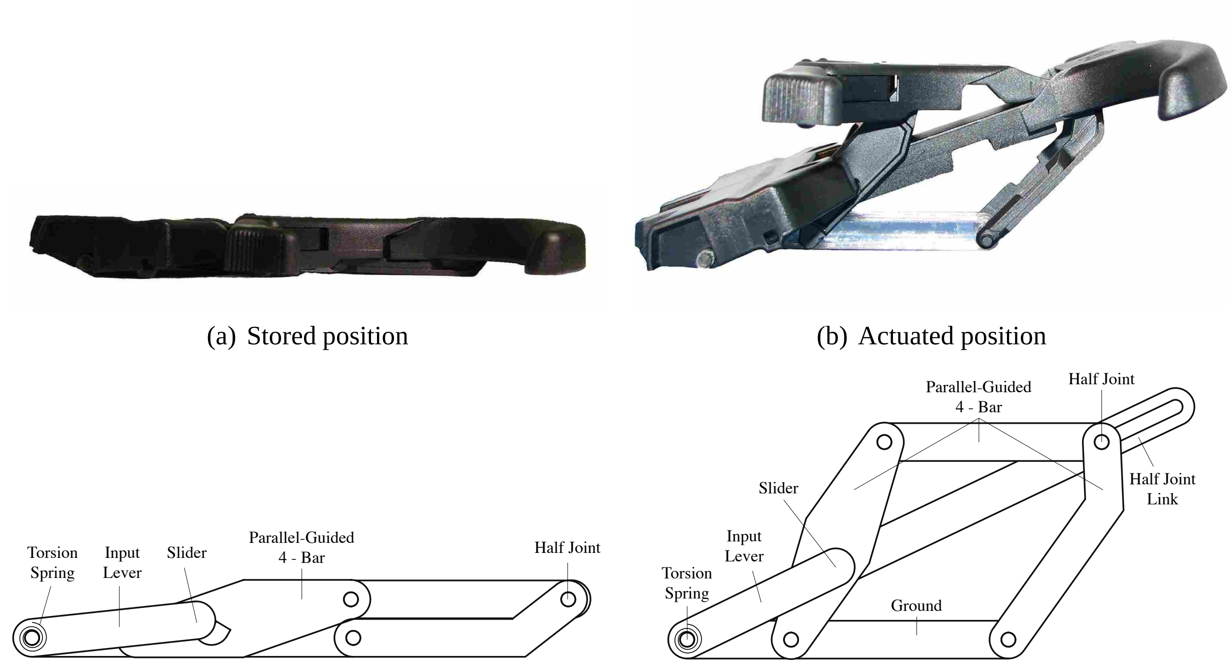
The design rules for selecting and designing compliant mechanisms from rigid-body replacement synthesis is provided by Berglund et al. [3], as shown in Figure 4.2. For the case where the design is based upon an existing rigid-body mechanism they gave six major steps. These steps are outlined below:

1. Rigid-body design alternatives are screened to identify the most viable alternative to be converted to a compliant mechanism

2. The conversion process yields alternative compliant designs

3. Compliant alternatives are reviewed using the general design rules to assure good designs

4. Design alternatives are evaluated to select the best design



(a) Stored position

(b) Actuated position

(c) Schematic of stored position

(d) Schematic of actuated position

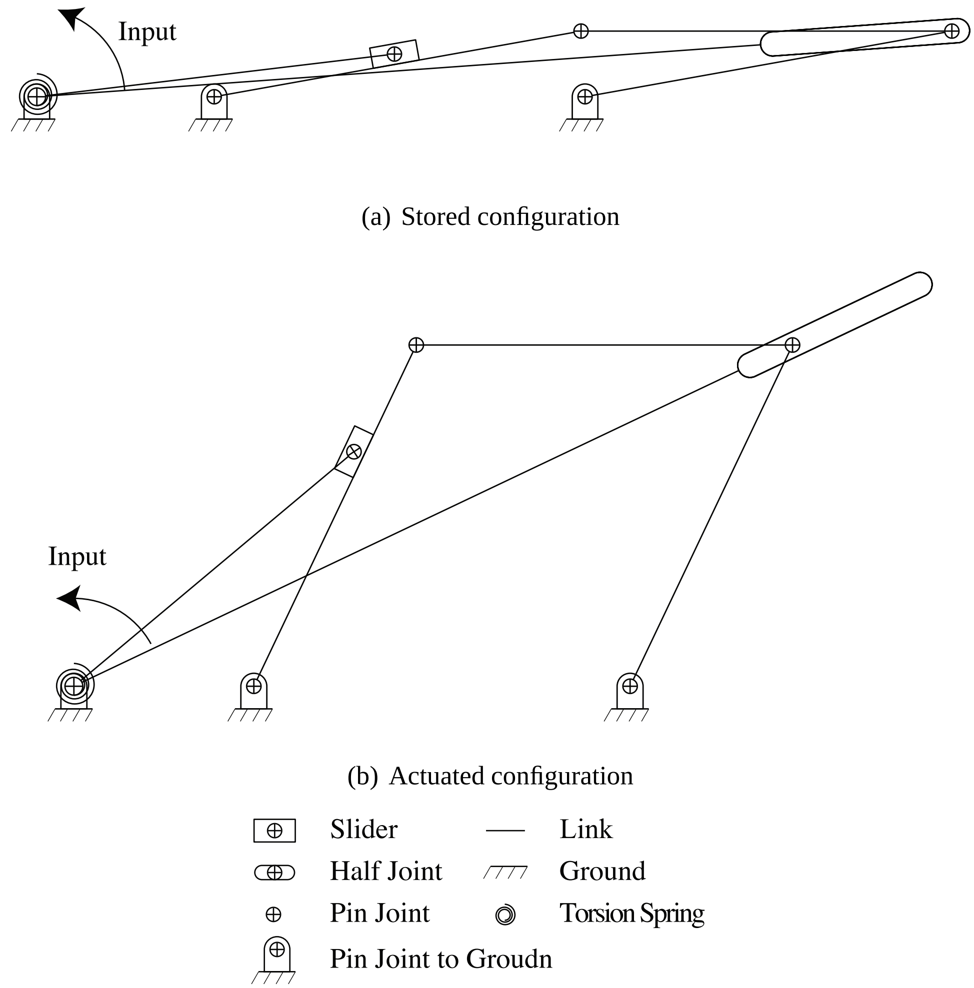
Figure 4.6: A ((a) and (b)) photo and ((c) and (d)) schematic of an existing cup holder in the ((a) and (c)) stored and ((b) and (d)) actuated positions

5. The best design is compared to the best rigid-body alternative

6. Design rules can again be used to refine the performance of the design, or help select another alternative if it is not superior to the rigid-body design.

According to the steps outlined above, a preliminary process is to accumulate rigid-body design alternatives. These design alternatives may come from reverse engineering the existing device and by using concept generation techniques. By reverse engineering the cup holder it is found that it consists of a parallel-guided 4-bar mechanism, an input lever that is actuated by a preloaded torsional spring and is connected to the 4-bar mechanism by a unique slider, and an additional link that is connected to the 4-bar mechanism by a half joint (see Figure 4.6).

Using the cup holder design as a benchmark multiple design concepts were generated. These designs are then screened to identify the most viable design alternatives that can be converted into a compliant mechanism. The design alternatives that were considered for this example are named rigid-body design 1 (RBD1) and rigid-body design 2 (RBD2). RBD1 is actuated by a



(a) Stored configuration

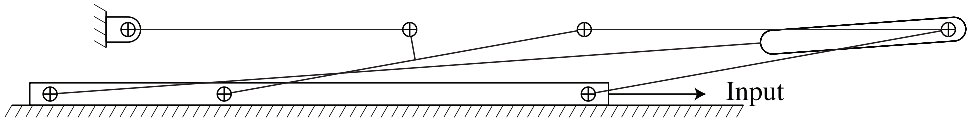
(b) Actuated configuration

Figure 4.7: RBD1 in its (a) stored and (b) actuated configuration

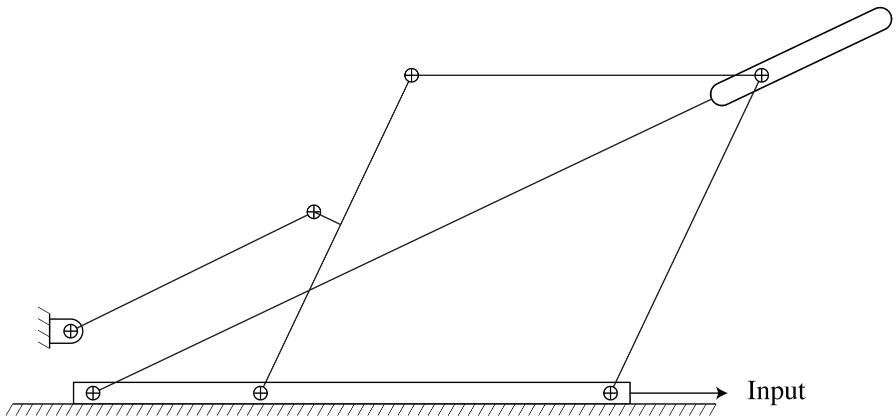
preloaded torsional spring and is based upon the original cup holder design (see Figure 4.7). RBD2 is displacement actuated (see Figure 4.8); this design uses the a similar configuration as the original cup holder design, though it is actuated by the displacement it undergoes when being removed from its stored position.

4.4.2 Conversion Process

Using rigid-body replacement synthesis to convert rigid-body mechanisms into compliant mechanisms is a technique that can be beneficial to engineers because they may be more familiar with the analysis and synthesis of rigid-body mechanisms. However, while designing rigid-body



(a) Stored configuration



(b) Actuated configuration

Figure 4.8: RBD2 in its (a) stored and (b) actuated configuration

mechanisms to convert into a compliant mechanism, one must understand that the motions and forces associated with rigid-body mechanisms can be decoupled, whereas compliant mechanisms have highly coupled kinematics and kinetics. For example, the RBD1 mechanism has a torsional spring for actuation. The compliant mechanism design counterpart will inherently have energy storage through the mechanism's deflection. Therefore, the manufactured configuration should be designed in the actuated rigid-body position (see Figure 4.7(b)), which will remove the need for the torsion spring and input lever for actuation. In the case of the RBD2, the manufactured configuration of the compliant mechanism design counterpart should be in the stored rigid-body position (see Figure 4.8(a)). This is attributed to the mechanism being actuated by a displacement, thus the energy storage from the mechanism's deflection will assist the mechanism to return into its stored configuration.

RBD1

Element Replacement - The classes of elements that would be beneficial for this type of design are those that have relatively large displacement and sufficient amount of energy storage. The design alternatives can be fewer because there is no advantage to having multiple element types for the parallel-guided mechanism for this design. A parallel-guided mechanism has four single rotational degree-of-freedom elements. By using designs from the *Elements of Mechanisms* category, the *Flexible Elements* subcategory, the *revolute* class, and the *hinge* subclass, a list of possible element types can be found. Some of these elements are: beams, small length flexural pivot, cross-axis flexural pivot, large displacement rotation element, split-tube flexure, and CORE (see Appendix A.1). Because the compliant elements inherently have energy storage through deflection, the actuating lever with the torsional spring are no longer necessary for these designs. The designs will still include the rigid-body half-joint element type for its prescribed function.

Mechanism Replacement - The classes of mechanisms that can be used in this type of design are those that can be designed in the actuated position. This is because the mechanism will store energy when it is deflected into the stored position, causing the mechanism to actuate. Also, the main operating mechanism is a parallel-guided mechanism; thus the mechanisms that should be considered are those that have a kinematic function of parallel motion. These mechanisms can be found in the *Mechanism* category, the *Kinematic* subcategory, and *parallel guided* class. More designs can be found in the *large motion path* subclass because these mechanism have relatively large deflections. Some possible mechanism designs that can be found in these classes are mechanisms #M-25 and #M-26 in Appendix A.2.

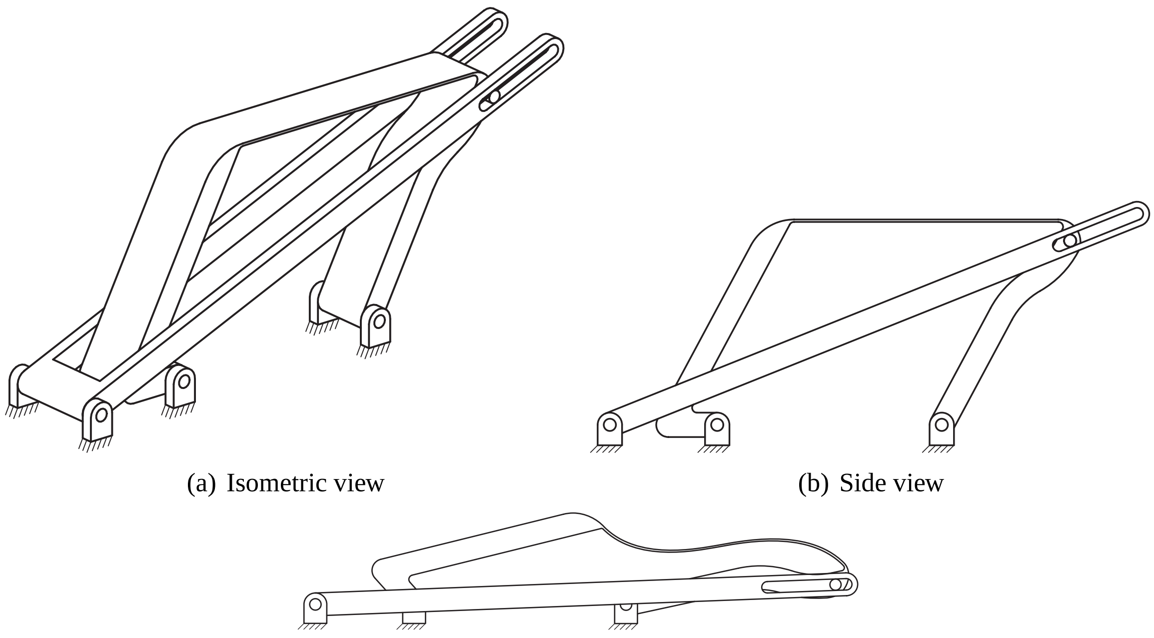
RBD2

Element Replacement - The classes of elements that would be beneficial for this type of design are similar to those indicated in section 4.4.2. The difference, however, are the elements for this type of design are ones that can be implemented into the parallel-guided mechanism that is planar or near planar in its manufactured form. Similar element designs can be found in the same subclass indicated in section 4.4.2. Other possible designs can be found in the *lamina emergent* subclass. This subclass would contain beneficial element designs that have a single degree of rotation and can be fabricated in a planar position [4]. Some possible element types for this design are: beams, living hinges, a large displacement rotation element, a split-tube flexure, CORE, or LET joints (see Appendix A.1).

Mechanism Replacement - The classes of mechanisms that can be used in this type of design are ones that can be designed in the stored position. This is because the mechanism has a displacement actuation, causing the mechanism to actuate into the deformed position as the mechanism is removed from its stored position. The energy stored through deflection causes the mechanism to reposition into the stored form when the mechanism is closed. The mechanisms for this type of function can be found in the same class as indicated in section 4.4.2. Some of the specific mechanisms that can be used are mechanism designs #M-25, #M-26, and #M-27 in Appendix A.2.

4.4.3 Compliant Design Discussion

After the rigid-body mechanism are converted into compliant mechanism, the designs are to be reviewed by general design rules. This is to ensure that the mechanism design concepts are beneficial for the specific application. An engineer may evaluate concepts that are based upon a selection criteria [58]. The criteria should be based upon the application of the mechanism and its relationship to its fabrication. This mechanism is for a cup holder, thus the mechanism should be fairly robust to hold liquid containers, have a relatively large degree of motion for a compliant mechanism, and fabricated using less expensive manufacturing and assembly processes. Under this criteria, numerous types of elements would not be feasible, thus eliminating any designs that use them. Of the elements listed above (and shown in Appendix A.1), those that meet this criteria are: beams, living hinges, and LET joints. The next step is to evaluate the designs to determine which concepts to pursue in more depth. The top results for the design concepts RBD1 and RBD2 are the compliant beam mechanism design and the compliant lamina emergent mechanism design, respectively. These concepts could be improved by repeating the design process and optimizing their performance.



(a) Isometric view (b) Side view

(c) Deflection

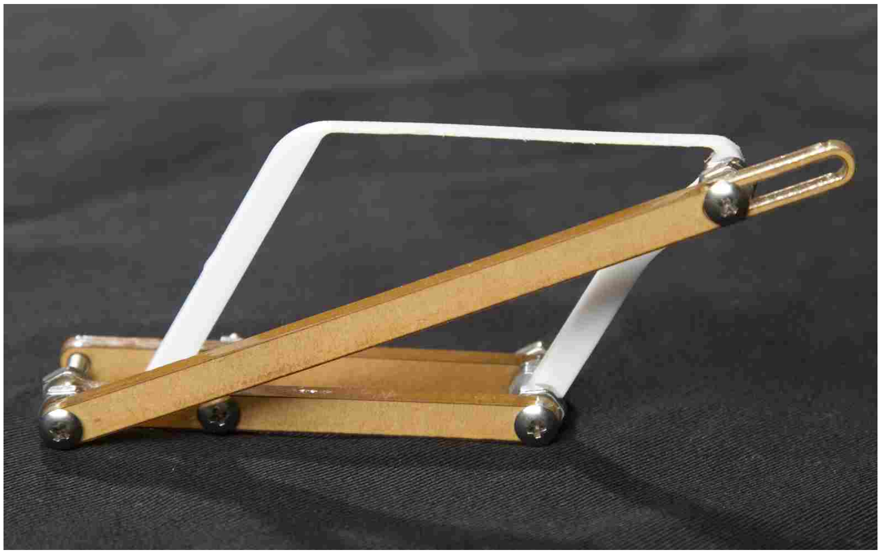
Figure 4.9: Compliant beam mechanism concept

Compliant Beam Mechanism Design

The compliant beam design (see Figure 4.9) is based upon RBD1. One way the conversion process could be achieved for this design is by replacing the coupler on the rigid parallel-guided mechanism with a fix-guided beam element (see *Fixed Guided* element, #EM-6, in Appendix A.1) or by replacing the rigid parallel-guided mechanism with the existing compliant parallel-guided mechanism (see *Parallel Guided* mechanism, #M-24, in Appendix A.2). It is shown that the resulting element replacement and mechanism replacement approaches may result in similar configurations; however, these different approaches can also result in very different design concepts.

The advantages for this type of design is that it is based upon the existing cup holder design. The difference is this compliant design has only three parts compared to the rigid-body design which has eight parts. This design concept accomplishes this by using the compliant beam to replace the functionality of the input lever and torsional springs. However, a disadvantage for this design is stress relaxation due to the mechanism being stored in the deflected configuration. Thus, further design and analysis of this concept may be necessary to overcome stress relaxation

(a) Stored configuration



(b) Actuated (manufactured) configuration

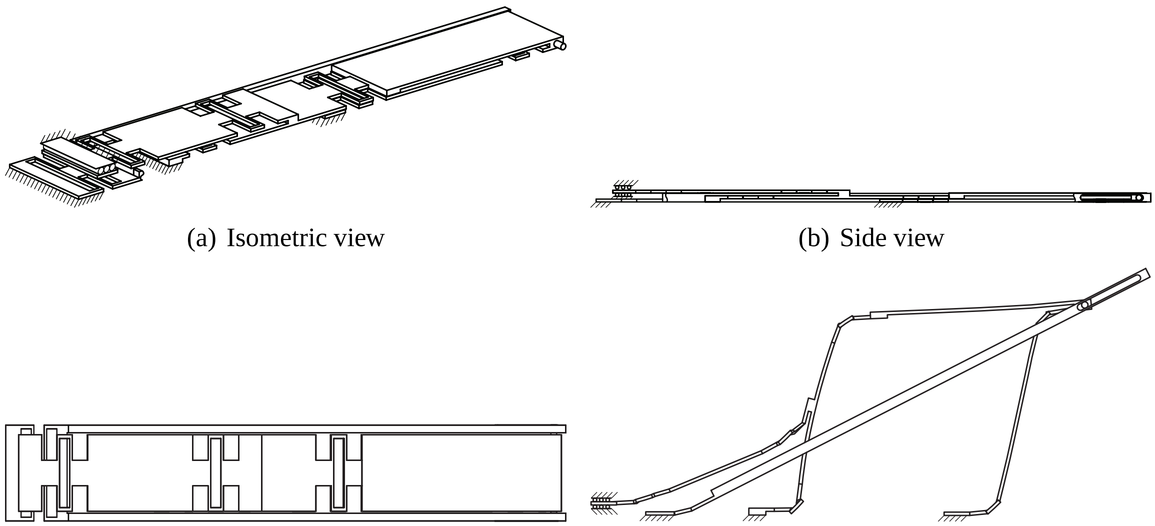
Figure 4.10: Prototype of the compliant beam mechanism design

Overall, this design concept could be a possible candidate to replace the existing rigid-body design. The prototype for this concept is shown in Figure 4.10.

Compliant Lamina Emergent Mechanism Design

The compliant lamina emergent mechanism design (see Figure 4.11) is based upon RBD2. The conversion process for this compliant design could be achieved by replacing the elements on the rigid parallel-guided mechanism with outside LET joints (see *Outside LET Joint* element, #EM-24, in Appendix A.1) or by replacing the rigid parallel-guided mechanism with the existing planar compliant mechanism (see *Multi-Layer Parallel-Guided LEM*, #M-27, in Appendix A.2).

This type of design encompasses the advantages associated with lamina emergent mechanisms (LEMs). A lamina emergent mechanism is a mechanism that is fabricated with sheet goods



(a) Isometric view (b) Side view

(c) Top view (d) Deflection

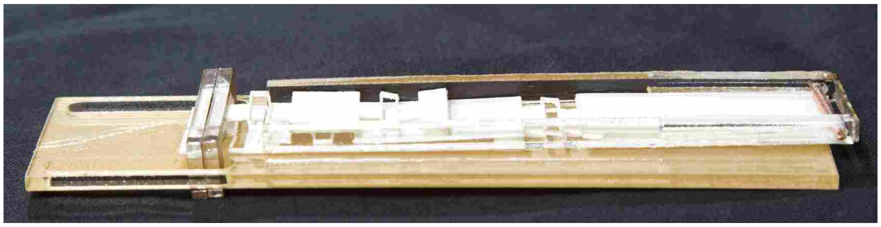
Figure 4.11: Compliant lamina emergent mechanism concept

and has motion out of the plane of fabrication [59]. Advantages of these types of mechanisms are that they can be fabricated by using less expensive manufacturing techniques. Another advantage is this concept has the possibility of using multiple layers. Thus, the mechanism can perform a complex motion with a minimal footprint. The disadvantage of multi-layer LEMs is the assembly of individual layers.

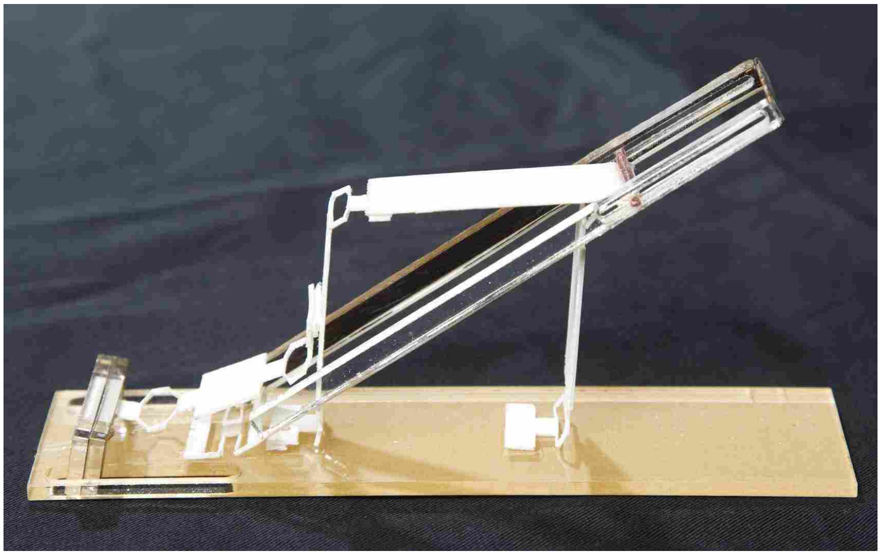
This design is different than the current rigid-body design, but has similar functionality as the cup holder. It also utilizes the mechanism that is used to remove the cup holder from its stored position, causing the mechanism to emerge from its planar configuration. A prototype for this design concept is shown in Figure 4.12.

4.5 Conclusion

A compliant mechanism classification scheme has been developed to help engineers become more familiar with compliant mechanisms and to find compliant designs they can incorporate into their own applications. This chapter used a compliant design methodology, rigid-body replacement synthesis, to illustrate how to use the classification scheme and to design compliant mechanisms.



(a) Stored (manufactured) configuration



(b) Actuated configuration

Figure 4.12: Prototype of the lamina emergent mechanism design

The design approach proposed in this chapter demonstrates two possible techniques to replace a rigid-body mechanism. One technique is to replace the rigid-body elements with a compliant counterpart. The other is to replace the rigid-body mechanism with a compliant mechanism that was designed for a similar application or functionality.

A classification scheme that categorizes compliant designs by three different factors can be useful for multiple design processes. Engineers could then employ these processes in conjunction with the scheme to incorporate compliance into their designs. The examples shown in this chapter indicate one design approach an engineer can employ that utilizes the compliant mechanism classification scheme.

CHAPTER 5. COMPLIANT ROAD BICYCLE BRAKE

5.1 Introduction

A driving factor in the bicycle component industry is to increase device performance and decrease the overall weight. These criteria have led to advanced materials and novel designs. One way to achieve these objectives is to decrease the number of components in the overall design. The purpose of this work is to use the advances in compliant mechanisms theory to develop a bicycle brake with the potential for low weight and high performance (i.e. compact design with a consistent mechanical advantage). This will be performed by incorporating a library of compliant mechanisms with rigid-body replacement synthesis.

Compliant mechanisms achieve motion or force transmission through the deflection of flexible members [14]. Using compliant mechanism theory to design a bicycle brake that achieves motion through the deflection of compliant members has a potential to decrease the number of parts and lower the overall weight, while maintaining or improving the performance. Compliant mechanism theory has previously been used to design bicycle components improving the performance of a mountain and BMX brake and a rear derailleur [43], and a clipless pedal [60]. This chapter will incorporate compliant mechanisms to design a novel road bicycle brake.

5.2 Background

This section provides a review of information related to (1) compliant mechanism design methods, and (2) the necessary functionality of brakes.

5.2.1 Compliant Mechanism Design Methods

The maturation of the analysis and synthesis of compliant mechanisms continues to improve allowing compliance to be incorporated into commercial products [2]. This advancement of

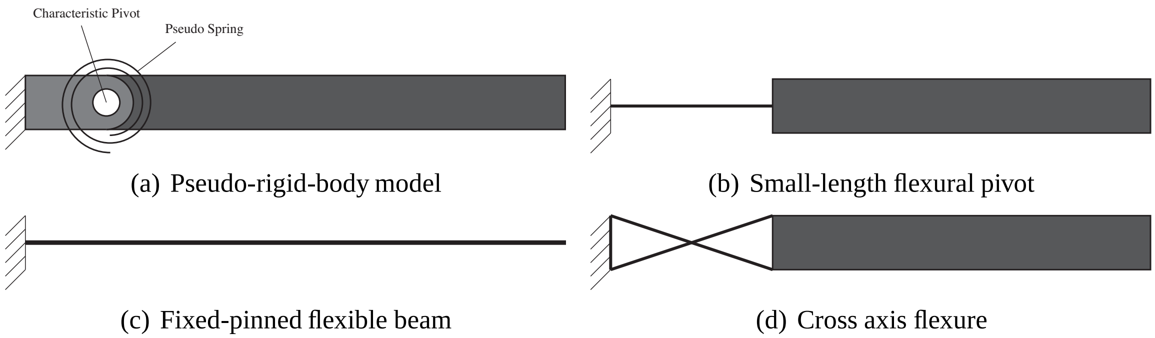


Figure 5.1: The (a) pseudo-rigid-body model concept for a (b) small-length flexural pivot, (c) fixed-pinned flexible beam, and (d) cross-axis flexure

compliant mechanisms has lead to a development of design methodologies [3, 22, 23, 47–51], one being rigid-body replacement synthesis, and a development of a reference library for compliant mechanisms [20].

Rigid-Body Replacement Synthesis

Rigid-body replacement synthesis involves designing or identifying a rigid-body mechanism that accomplishes the desired function and then converting the design into a compliant mechanism [21]. This conversion process can be achieved by two approaches. The first approach is to decompose a complex mechanism into mechanisms that have a simpler function, and then replacing the respective mechanism with a compliant counterpart. The second approach is to replace the rigid-body elements with a compliant counterpart. Permutations of compliant mechanisms can be found by using type synthesis.

The rigid-body mechanism in rigid-body replacement synthesis can be referred to as the pseudo-rigid-body model. The pseudo-rigid-body model predicts the deflection path of flexible segments by modeling it with characteristic pivots (i.e. rigid links attached at pin joints with torsion springs), see Figure 5.1.

A major challenge associated with rigid-body replacement synthesis is that while rigid-body mechanisms' kinematics and kinetics can be decoupled, the kinematics and kinetics of compliant mechanisms are highly coupled. One technique to overcome this challenge is to design a rigid-body mechanism for the general motion of the mechanism, then convert the mechanism into a compliant mechanism. This compliant counterpart could then be improved by using optimization techniques to obtain the desired forces and motion through finite element analysis (FEA).

Library of Compliant Designs

Previous work establishes a classification scheme for the purpose of helping engineers find existing compliant designs that they can incorporate into their own applications [20]. The classification scheme categorizes compliant designs by three approaches, with the primary approach being functionality. This approach will be used throughout this chapter because the functionality classification approach categorizes compliant designs into respective classes that work well with rigid-body replacement synthesis. The functionality approach separates compliant designs into *Elements of Mechanisms* and *Mechanisms* and are then subdivided into subcategories, classes, and subclasses, according to their respective function.

Olsen et al. [44] have illustrated how this classification scheme could be used as a basic framework for a library of designs that could be incorporated into the design process. This is done by using the functionality classification approach in conjunction with rigid-body replacement synthesis to design a mechanism that has flexible segments.

5.2.2 Self-Centering Mechanism

The kinematics of a mechanical brake system requires two characteristics to achieve a good design: (1) the shoes (pads) should self-center about the rim during the actuation process, and (2) the forces should be balanced on the rim. Brooks et al. [61] presented four design principles (postulates) to accomplish these objectives for a mechanical brake system, and also provided a design procedure that utilize these postulates. This work will focus on postulates one and three, which are:

> **Postulate 1:** A minimum of two degrees of freedom are required in the brake mechanism, in order to exhibit simultaneous centering and balanced reaction force characteristics.
>
> **Postulate 3:** To maintain the braking links in a stable equilibrium "off" position, at least one potential energy storage device is required for each degree of freedom in the mechanism.

5.3 Rigid-Body Brake Designs

The industry for road bicycle components is a fairly large and competitive, where many providers try to produce a high performance device with minimal weight. This is especially true for brake systems, where high performance is required due to the high loads the brakes undergo while being actuated, while maintaining a minimal weight. There are, however, few rigid-body linkage designs that have been established to achieve this objective, where most design variables are focused on material selection, accessory functions, and integrated components. Thus, the novelty of these designs are not contingent on their kinematic and kinetic functions.

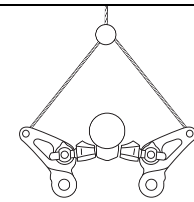
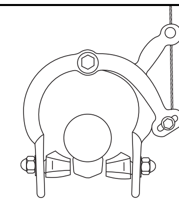
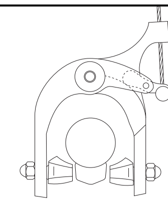
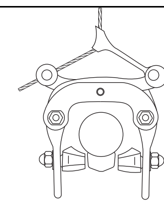
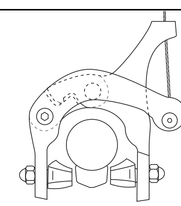
The purpose of this research is to present a new linkage configuration that will inherently use less material, and remove the need for integrated components. To begin, an understanding of the existing brake designs with their advantages and disadvantages is requisite, to better understand a benchmark for the compliant bike brake. There are primarily five rigid-body linkage designs, which will be referred to as: (1) cantilever, (2) single pivot, (3) modified single pivot, (4) dual pivot, and (5) modified dual pivot. A schematic of these designs along with their advantages and disadvantages are shown in Table 5.1.

5.3.1 Synthesis of Alternative Configurations

The modified single pivot and modified dual pivot rigid-body brake designs shown in Table 5.1 function with a higher kinematic pair (i.e. cam). These designs do not allow a greater number of compliant configuration counterparts to be formed by rigid-body replacement synthesis because most compliant element designs are established for lower kinematic pairs. Thus, by transforming the higher order kinematic pairs to equivalent combinations of lower order pairs, more compliant permutations can be found.

Titus et. al. [52] gave a list of transformation laws for basic kinematic chains. The fourth law is helpful in converting existing rigid-body designs into alternative configurations that have a similar function. The fourth law states that a "removal of a pin-connected binary link and substitution of a higher pair joint for the binary link and its 2 lower pair joints will not change the degrees of freedom." The opposite is also true, where a binary link substituted for a higher pair joint will not affect the degrees of freedom. This law is helpful for the modified single pivot and

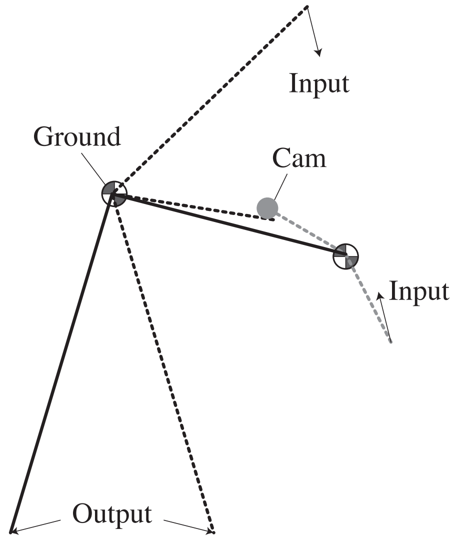
Table 5.1: Road bicycle brake comparison

NAME	*ADVANTAGES*	*DISADVANTAGES*	*IMAGE*
Cantilever	• Reduced number of parts • Force balanced • Free of debris	• Cable housing • Two mounts	
Single Pivot	• Reduced number of parts • Less expensive to fabricate	• Rotate about mount • Mechanical Advantage	
Modified Single Pivot	• Mechanical advantage • High performance	• Rotate about mount • Number of parts • Varying mechanical advantage	
Dual Pivot	• Force Balance • Less expensive to fabricate	• Cable housing • Number of parts • Number of attachment points	
Modified Dual Pivot	• Reduced number of parts • Compact • High performance	• Force Balanced • Lever arm rotation	

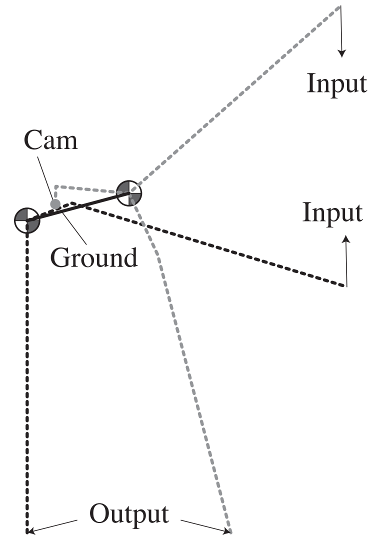
modified dual pivot brake designs, where their cams can be replaced with a binary link which will be more advantageous in converting the design into a compliant counterpart. By utilizing this law, equivalent configurations for the modified single and dual pivot designs are shown in Figure 5.2.

5.4 Rigid-Body Replacement Synthesis

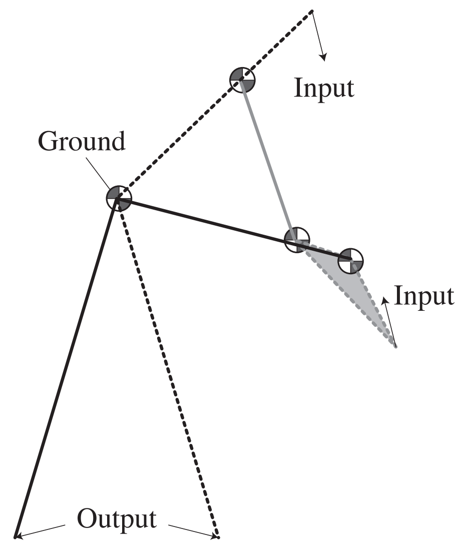
In preparation for rigid-body replacement synthesis a screening matrix was performed on the rigid-body designs of Table 5.1 based upon multiple criteria, including the designs' (1) eligi-



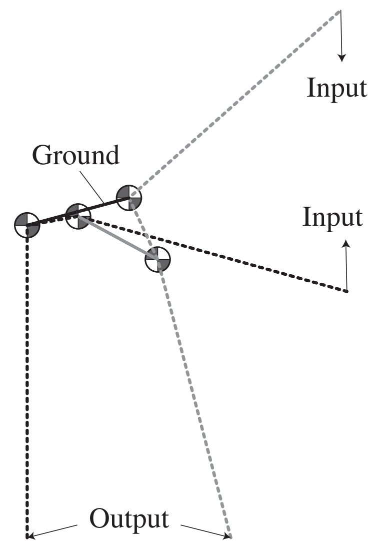
(a) Modified cam single-pivot schematic



(b) Modified cam dual-pivot schematic



(c) Modified linkage single-pivot schematic



(d) Modified linkage dual-pivot schematic

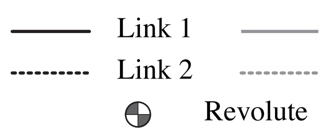


Figure 5.2: Configurations of ((a)-(b)) higher-order and ((c)-(d)) lower-order kinematic paris

bility to be converted into a compliant mechanism, (2) target mechanical advantage, (3) number of parts, (4) ability to self-center, and (5) angular deflections. The resulting design that is most eligible for conversion based upon the criteria is the modified dual pivot. As established in section 5.3.1 there are two possible rigid-body configurations associated with this concept: the higher-order pair design (Cam Design) and the lower-order pair design (Linkage Design).

5.4.1 Compliant Counterparts

Rigid-body replacement synthesis treats the rigid-body linkage configuration as a pseudo-rigid-body model. That is, the rigid links and pin joints can be replaced with a compliant element that has similar motion. This section describes what type of compliant elements would be most beneficial to replace the rigid-body elements.

Cam design

In the modified dual-pivot design there are three main rigid components: a torsion spring, two pin joints attached to the ground link, and a cam. This brake design has one degree of freedom, which contradicts postulate 1 of section 5.2.2. However, the spring in this device plays an important role in that it keeps the cam in contact with the cam surface and when one pad makes contact with the rim, the cam is removed from the contact surface to achieve its second degree of freedom. This behavior makes it a metamorphic mechanism [16]. It is imperative in rigid-body replacement synthesis that the compliant element that replaces the pin (attached to the torsion spring) helps maintain this function.

Rigid-body replacement synthesis for this concept (modified dual pivot with a cam) allows the two pin joints to be replaced by a compliant element. As compliance achieves energy storage through deflection, it can remove the need for the torsional spring. It is also important to note that the compliant elements that replace the pin joints need to have a high off-axis stiffness due to the high loads experienced during braking. Other requirements are that the element should be able to undergo large deflections for compliant mechanisms and be compact. By examining a library of compliant elements [44] that match these criteria, some possible candidates are the cross-axis

Table 5.2: Rigid-body replacement options

	Ground Link		Coupler Link
	Pin 1	*Pin 2*	
Cam Design	rigid pin joint	rigid pin joint	cam
	tubular cross-axis flexure	tubular cross-axis flexure	cam
	tubular cross-axis flexure	cross-axis flexure	cam
	cross-axis flexure	tubular cross-axis flexure	cam
	cross-axis flexure	cross-axis flexure	cam
Linkage Design	rigid pin joint	rigid pin joint	fixed-fixed beam
	tubular cross-axis flexure	tubular cross-axis flexure	fixed-fixed beam
	tubular cross-axis flexure	cross-axis flexure	fixed-fixed beam
	cross-axis flexure	tubular cross-axis flexure	fixed-fixed beam
	cross-axis flexure	cross-axis flexure	fixed-fixed beam
	rigid pin joint	rigid pin joint	fixed-pinned beam
	tubular cross-axis flexure	Tubular cross-axis flexure	fixed-pinned beam
	tubular cross-axis flexure	cross-axis flexure	fixed-pinned beam
	cross-axis flexure	tubular cross-axis flexure	fixed-pinned beam
	cross-axis flexure	cross-axis flexure	fixed-pinned beam

flexure and the tubular cross-axis flexure [56]. The resulting compliant replacement possibilities can be found in Table 5.2.

Linkage design

This mechanism design is similar to the cam design described in the previous section, but it has a binary link that replaces the cam (see Figure 5.2). Thus, the compliant replacements for the ground pins are similar to the cam design, but the binary coupler link can easily be converted into a compliant equivalent. The requirements for this type of compliant element replacement is that they should be able to undergo large rotations, be compact, and may be required to have energy storage if the ground pins are rigid-link joints. By examining a library for compliant elements [44] that fit this criteria, two possible candidates for the coupler link are a fixed-fixed and fixed-pinned compliant beam. The resulting compliant replacements can be found in Table 5.2.

A challenge with this rigid-body design, according to the first postulate (see section 5.2.2), the mechanism needs at least two degrees of freedom. The mechanism shown in Figure 5.2(d) is a four bar mechanism and has one degree of freedom. The second degree of freedom is accomplished through system compliance. For example, a compliant beam can achieve a second degree of freedom by entering into another mode of motion (i.e. buckling).

5.4.2 Selection

A preliminary finite element analysis was conducted on the design configurations found in Table 5.2. The purpose of this analysis was to determine which configuration provided a sufficient amount of energy storage through actuation, while maintaining minimal stresses. It was found that the fully compliant designs (i.e. no rigid pin joints), and the configurations where the ground link has rigid pin joints and a fixed-fixed coupler link would result in designs that will perform similar to the benchmark. A challenge with a fully compliant brake design is the mounting pin, where one ground pin has a dual role as a rigid pin joint and also the mounting point. The design that will be featured in the rest of this chapter is the linkage design where the ground link has rigid pin joints and the coupler is a compliant fixed-fixed beam.

5.5 Compliant Brake Design

The resulting compliant design concept originated from the modified dual-pivot brake (see Table 5.1). This design was then transformed from a higher-order kinematic pair to lower-order kinematic pairs (see Figure 5.2(d)), which proves to be a better candidate for rigid-body replacement synthesis. By using type synthesis, permutations of compliant configurations were generated. After creating a screening matrix, the compliant mechanisms concept that proved to be the most advantageous was one where the ground link has rigid pin joints and the coupler link is a compliant fixed-fixed beam (first row of “Linkage Design” in Table 5.2). The pseudo-rigid-body model of this concept is shown in Figure 5.3.

5.5.1 Optimization

The next step in the design process was to optimize the mechanism for mechanical advantage and force balance. Two separate optimization problems were solved, with objectives of maximizing the mechanical advantage and the force balance. The brake design is required to fit

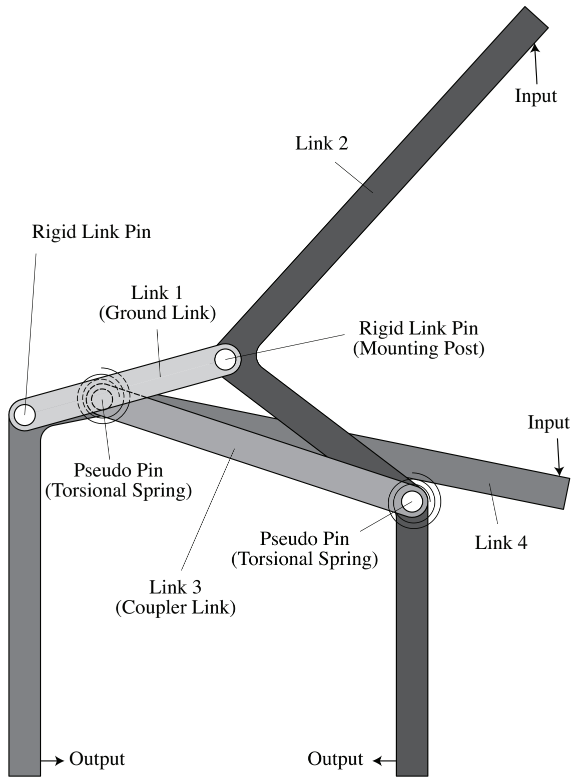


Figure 5.3: Pseudo rigid-body model of the compliant road bicycle brake concept

in a specified envelope, thus the input/output and mounting points are constrained to a relative location, so the design variables are the placement of the second ground (non-mounting) rigid pin joint and the characteristic pivot locations of the compliant coupler.

The mechanical advantage and force balance equations were derived using the principle of virtual work. The mechanical advantage derivation did not include the pseudo torsion springs, because its focus was to find the kinematic mechanical advantage. The mechanical advantage (MA) for this multi-degree-of-freedom mechanism is described as the ratio of the average output forces to the average input forces.

$$MA = \frac{(F_{out})_{average}}{(F_{in})_{average}} \tag{5.1}$$

By using the principle of virtual work [14] it was found that the primary design variable for mechanical advantage is the location of the second rigid pin joint of the ground link, where the location of the coupler characteristic pivots have a negligible effect. This is helpful because the

Table 5.3: Optimized bicycle brake values

Pseudo-Rigid-Body Model				Compliant Mechanism			
Link Lengths (*mm*)		Link Angles (°)		Link Lengths (*mm*)		Link Angles (°)	
L_1	30.245	θ_1	190.000	L_1	30.245	θ_1	190.000
L_2	25.000	θ_2	324.204	L_2	28.670	θ_2	326.476
$(L_2)_i$	61.936	$(\theta_2)_i$	85.000	$(L_2)_i$	61.936	$(\theta_2)_i$	82.727
$(L_2)_o$	35.881	$(\theta_2)_o$	135.402	$(L_2)_o$	34.248	$(\theta_2)_o$	127.482
L_3	43.294	θ_3	161.517	L_3	50.934	θ_3	161.517
L_4	10.000	θ_4	25.814	L_4	7.740	θ_4	45.976
$(L_4)_i$	66.991	$(\theta_4)_i$	140.000	$(L_4)_i$	70.807	$(\theta_4)_i$	119.606
$(L_4)_o$	44.886	$(\theta_4)_o$	111.316	$(L_4)_o$	44.886	$(\theta_4)_o$	131.478

force balance optimization routine has one less design variable and can be more dependent on the geometry and placement of the compliant coupler link. The resulting mechanical advantage is 1.25 for the optimized link lengths and angles (see Table 5.3). These dimensions correspond to Figure 5.4.

The next optimization routine was to optimize the force balance, constrained for the given mechanical advantage listed above. Force balancing refers to having the output forces equal,

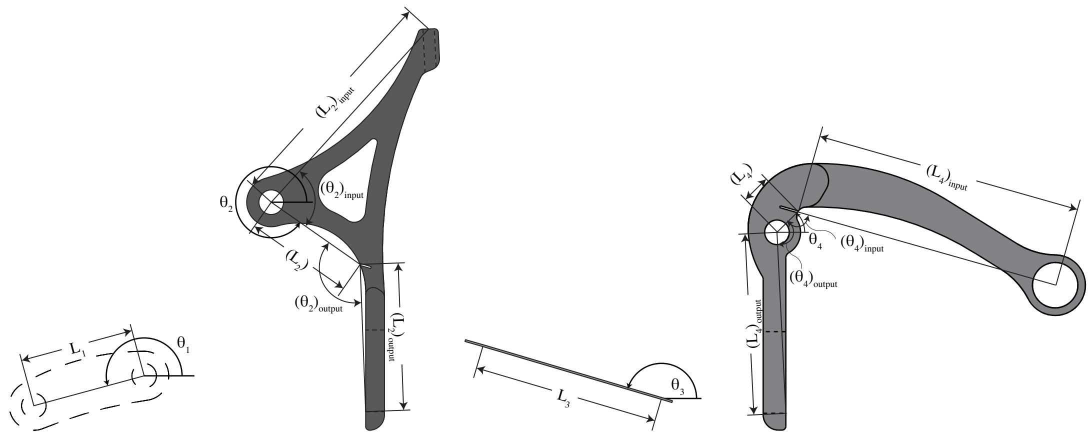
$$\frac{(F_{out})_1}{(F_{out})_2} = -1 \tag{5.2}$$

so the pads will have have equal wear when actuated and have a similar actuation rate. The output forces were found by using the principle of virtual work. The design variables were only the placement of the flexible fixed-fixed beam's characteristic pivots and the pseudo torsion springs' potential energy.

The potential energy equation needed for the principle of virtual work for the pseudo torsion springs is

$$V = \frac{1}{2}K(\theta - \theta_o)^2 \tag{5.3}$$

where V is the potential energy, K is the torsion spring constant for the pseudo springs, and $(\theta - \theta_o)$ is the angular deflection. The spring constant for the pseudo torsion springs can be approximated by the fixed-guided beam equations (see Figure 5.5) [14]. It is noted that the bicycle brake's flexible

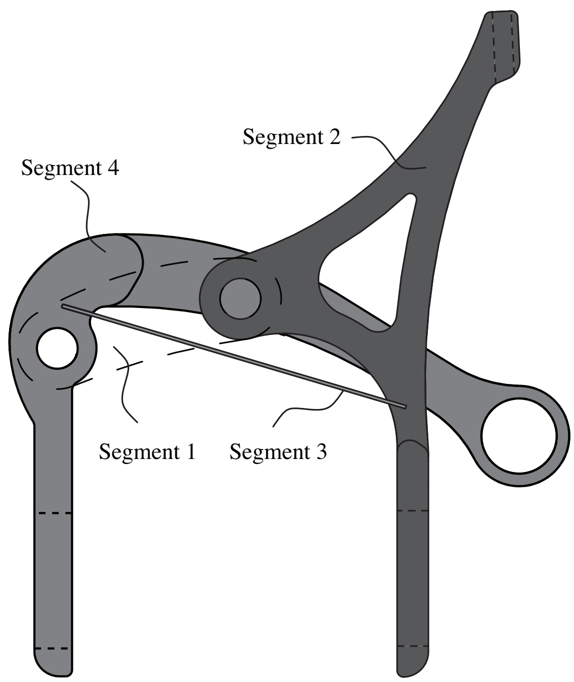


(a) Ground link (segment 1)

(b) Right side rigid segment (segment 2)

(c) Compliant segment (segment 3)

(d) Left side rigid segment (segment 4)



(e) Brake assembly (back view)

Figure 5.4: Assembly and dimensions of bicycle brake

Table 5.4: Cross sectional geometry

	Titanium	E-Glass
Number of flexures	10	1
Width (mm)	8	8
Thickness (mm)	0.508	2.5

beam will not undergo a fixed-guided deflection, but that it will give an approximation for a closed form solution used by the optimization routine. The spring stiffness for the fixed-guided beam is

$$K = 2\gamma K_\theta \frac{EI}{l} \tag{5.4}$$

where γ and K_θ can be approximated as constants,

$$\gamma = 0.8517 \tag{5.5}$$

$$K_\theta = 2.65 \tag{5.6}$$

E is Young's modulus, I is the moment of inertia,

$$I = \frac{bh^3}{12} \tag{5.7}$$

and l is the length of the flexible segment.

$$l = \frac{L_3}{\gamma} \tag{5.8}$$

The cross section dimensions of the flexible beam are indicated in Table 5.4. Titanium and E-glass are used for the flexible segment and their material properties are indicated in Table 5.5. The resulting optimized link lengths and angles are listed in Table 5.3, with these dimensions corresponding to those indicated in Figure 5.4. The number of flexures listed in Table 5.4 refers to the number of flexures, of the thickness listed, that are stacked together like leaf springs. This stacking of flexures allows an increased overall stiffness without increasing stress.

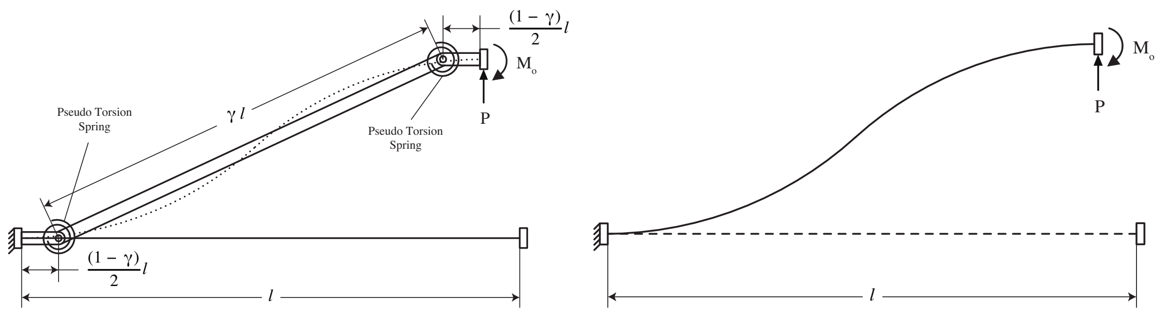


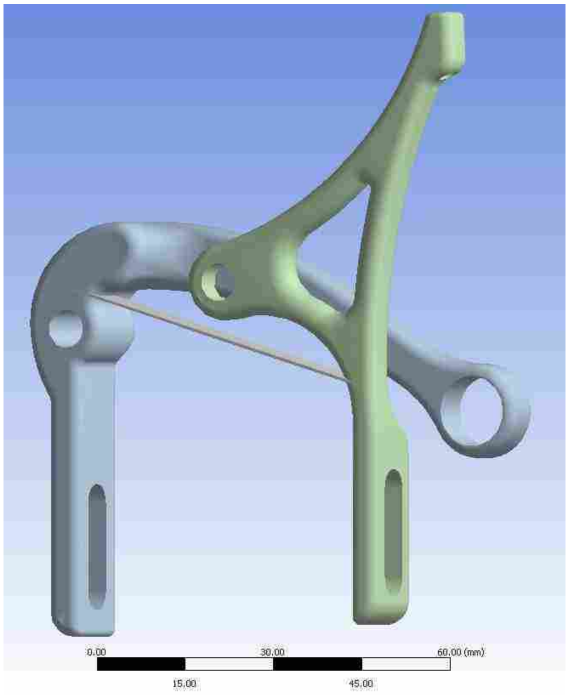
Figure 5.5: Fixed-guided flexible beam pseudo-rigid-body model

Table 5.5: Material properties

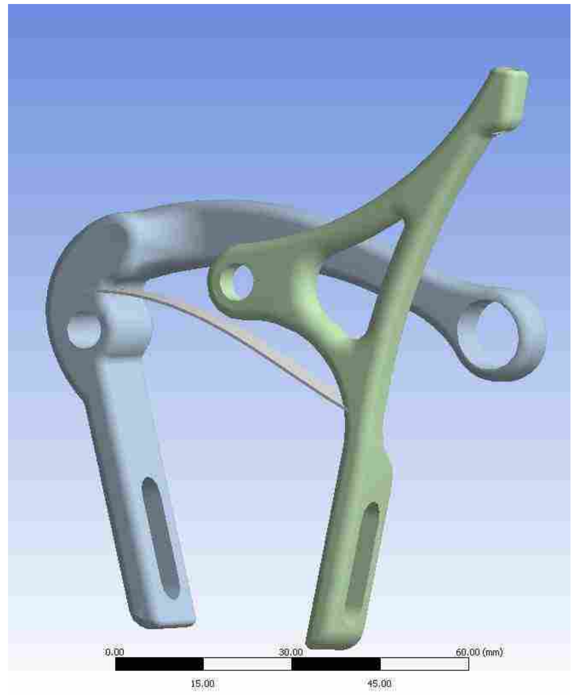
	Titanium(Ti-5A1-2.5Sn annealed) [14]	E-Glass [43]
Layer configuration	NA	w,o,o,w
Young's Modulus (GPa)	114	9.9
Tensile Yield Strength (MPa)	779	1,800
Ultimate Tensile Strength (MPa)	827	3,400

5.5.2 Analysis

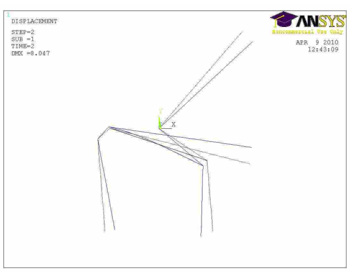
The deformation and stresses were analyzed for the optimized brake dimensions. The fatigue strength for the flexible segment was estimated as listed in Table 5.6. A commercial finite element analysis software (ANSYS) was used to compute the deformation and stress of the compliant mechanism. To simulate a cable tension a vertical displacement was applied to the 'left-side rigid segment' input arm and a vertical force was applied to the 'right-side rigid segment' input arm. The applied force was found iterating a approach to determine the reaction force from the displacement and applying the opposite direction force to the applied force (see Appendix B). The maximum operating deflection and the associated stresses for the mechanism were analyzed. These results and also the shoe deflections (output location) are indicated in Table 5.7. The stress distribution and deflection are shown in Figure 5.6. These results predicts that the bicycle brake operates within the desired deflection and prescribed allowable stress.



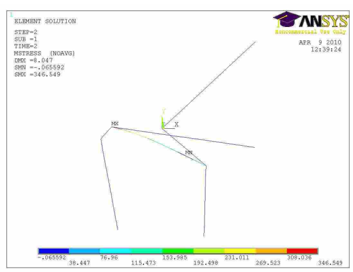
(a) 3D originial



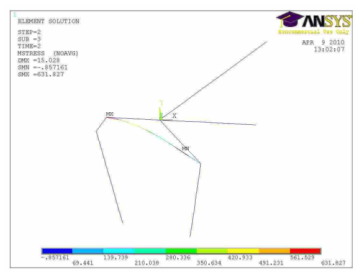
(b) 3D deformation



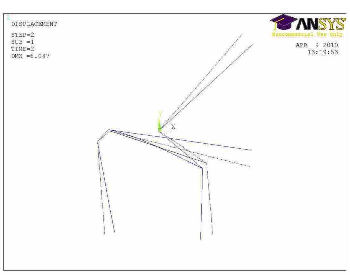
(c) Titanium flexure deformation



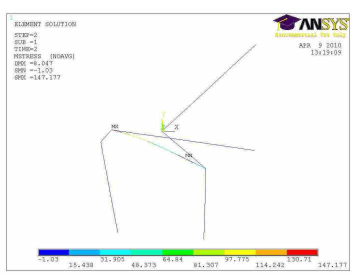
(d) Titanium flexure operating stress



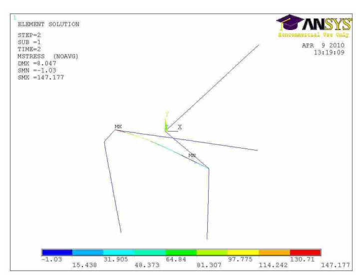
(e) Titanium flexure maximum stress



(f) E-Glass flexure deformation



(g) E-Glass flexure combined operating stress



(h) E-Glass flexure maximum stress

Figure 5.6: FEA results from ANSYS for the (c)-(e) titanium flexure and (f)-(g) the e-glass flexure

Table 5.6: Fatigue strength

	Titanium	E-Glass
Number of cycles	$25x10^3$	$25x10^3$
Safety Factor	1	1
Fatigue Strength (*MPa*)	630.981	2,594

Table 5.7: FEA results

	Material	**Deflection (*mm*)**		**Stress (*MPa*)**	**Cable Tension (*N*)**
		Left	*Right*		
Operation	Titanium	4.779	-5.804	346.549	14.829
	E-Glass	4.779	-5.842	147.177	15.372
	Benchmark	4.700	-5.640	NA	15.569
Maximum Deflection	Titanium	8.859	-11.959	631.827	25.703
	E-Glass	11.716	-17.025	342.876	31.610

5.5.3 Discussion

A compliant bicycle brake concept was developed that has the potential for weight reduction and performs similarly to the benchmark (modified dual pivot design). The brake undergoes the desired operating deflection and the flexible beam's stress is below the allowable fatigue strength. This analysis was performed on two materials: titanium and e-glass. The e-glass version results in fewer flexures, than the titanium version, to perform the same as the benchmark. Thus, different materials could be used for this design concept as long as it meets the engineers specification and is aesthetically pleasing.

An issue relating to this design is the actuation rate of the output links, where they undergo a ratio of 1.26 (see Table 5.7). However, it is noted the benchmark has a deflection rate ratio of 1.2. Thus, it can be seen that the compliant bicycle brake behaves similar to the benchmark brake.

The potential for weight reduction comes from the removal of material by eliminating the cam and cam follower surface of the benchmark. Also, this concept removes the need of four accessory components, thus reducing assembly and further reducing weight. A preliminary

(a) Isometric view (b) Front view

Figure 5.7: Prototype of the compliant road bicycle brake

demonstration prototype of this design is shown in Figure 5.7. An industrial design concept is illustrated in Figure 5.8.

5.6 Conclusion

The bicycle component industry is motivated to increase performance and decrease the overall weight of devices. This chapter has used compliant mechanism theory to integrate compliance into a road bicycle brake, and the resulting design has a potential of reducing the overall weight of the device while maintaining a desired performance.

The resulting compliant bicycle brake developed in this chapter proved to maintain the benchmark performance, and also has the potential of lower weight and reduced assembly by the removal of four accessory components.

5.7 Acknowledgements

The graphics for the brake concept was provided by Larrin Wada, and the prototype was fabricated by Nathan Llewellyn.

(a) Isometric view

(b) Front view

(c) Back view

Figure 5.8: Concept of the compliant road bicycle brake

CHAPTER 6. CONCLUSION AND RECOMMENDATIONS

6.1 Conclusion

Compliant mechanisms represent a relatively new discipline in machine design. As such, engineers may not be familiar with implementing compliance into designs. This unfamiliarity could be contributed to the lack of a readily available reference library of compliant designs, whereas there are an abundance of reference libraries available for rigid-body mechanisms.

The purpose of this thesis is to help engineers to become more familiar with compliant mechanisms and how to implement them into their respective applications and to provide a foundation for future compliant mechanism design reference. A classification scheme was developed to categorize compliant designs. The classification scheme categorizes compliant designs by using three different design approaches, which are: their (1) function, (2) application, and (3) fabrication constraints. The functionality classification approach was developed for engineers to use their basic understanding of rigid-body mechanisms to find a compliant counterpart. The application and fabrication classification approaches were developed to organize existing compliant designs for their application and fabrication constraints, respectively.

This thesis has also provided a framework for a reference library of compliant designs based upon the classification scheme. This thesis further establishes how the classification scheme and a library of compliant designs could be used with design methodologies. It illustrated this concept by using rigid-body replacement synthesis to redesign an automobile cup holder, where the classification scheme and library of compliant designs could be used in the conversion process. This was demonstrated by two approaches: replace the rigid-body mechanism with a compliant mechanism that has a similar functionality, and replace the rigid-body elements with a compliant counterpart.

A classification scheme that categorizes compliant designs by three different methods can be useful in design. Engineers could incorporate compliance into their applications using a reference library, and thus become more familiar with compliant mechanisms and utilize their advantages.

6.2 Recommendations

This thesis presents a classification scheme to categorize compliant designs to be used by engineers in mechanical design. Further research could investigate the proposed classification scheme and its relationship to ontological research [62, 63]. This work can contribute to mechanism design and future versions of this work. This thesis also includes a condensed compliant mechanism library that uses the devised framework, but does not contain a complete library of compliant designs. Further recommendations are to gather existing compliant designs and categorize them according to the classification scheme. As more designs are gathered it may be prudent to establish additional classes to appropriately categorize each design into the reference library. Then by using the existing compliant designs compile them into a compendium for a resource engineers can refer to while implementing compliance into their designs.

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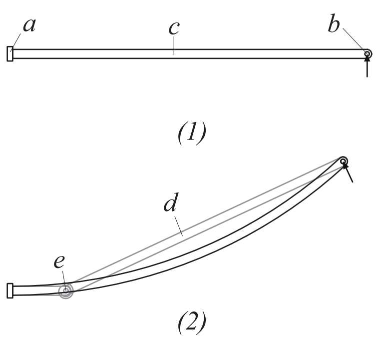
APPENDIX A. LIBRARY OF COMPLIANT DESIGNS

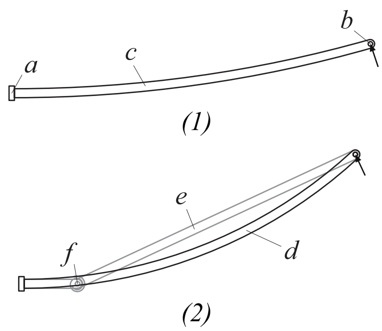
A.1 Elements of Mechanisms

A.1.1 Flexible Elements

Beams

EM-1	Fixed-pinned	FE FB
	This element is a cantilever beam with a force or moment at the free end. It can be modeled using the psuedo-rigid-body model, which approximates the flexible element as a rigid-link with a torsional spring. [14] *(1)* Segment *a* is fixed, segment *b* is pinned, and segment *c* is the flexible beam. *(2)* Segment *c* in the deflected position, with its pseudo-rigid-body link, *d*, and torsion spring, *e*.	

EM-2	Fixed-pinned Initially Curved	FE FB

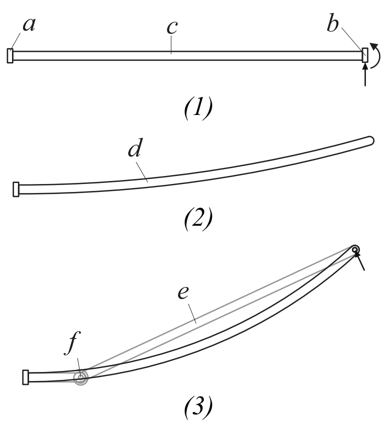


This element is an initially-curved cantilever beam with a force or moment at the free end. By using the Bernoulli-Euler equation (curvature is proportional to the moment) a moment can be applied as being an initially-curved beam. This element can be modeled using the psuedo-rigid-body model, which approximates the flexible element as a rigid-link with a torsional spring. [14]

(1) Segment *a* is fixed, segment *b* is pinned, and segment *c* is the flexible beam.

(2) Segment *c* in the deflected position, *d*, with its pseudo-rigid-body link, *e*, and torsion spring, *f*.

EM-3	Fixed-Fixed Case I	FE FB



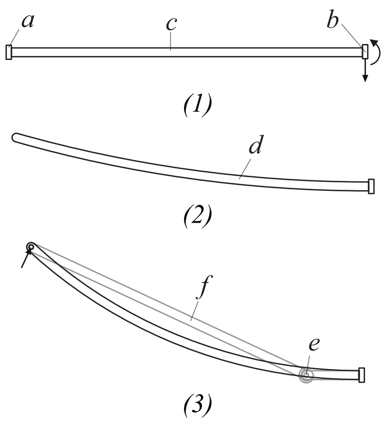
This element is a beam fixed at both ends. This case occurs when the force and moment are in the same direction, which can be modeled using a fixed-pinned initially curved beam. This element can be modeled using the psuedo-rigid-body model, which approximates the flexible element as a rigid-link with a torsional spring. [14]

(1) Segment *a* and *b* are fixed, and segment *c* is the flexible beam.

(2) Segment *c* is modeled as an initial curvature beam, *d*.

(3) Segment *c* in the deflected position, with its pseudo-rigid-body link, *e*, and torsion spring, *f*.

EM-4	Fixed-Fixed Case II	FE FB



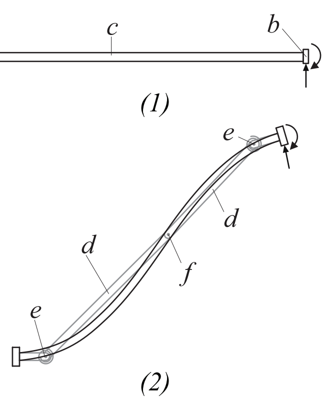
This element is a beam fixed at both ends. This case occurs when the force and moment are in opposite directions, but there is no inflection point. This case can be modeled using Fixed-Fixed Case I, with the ground switched to the opposite side of the beam. This element can be modeled using the psuedo-rigid-body model, which approximates the flexible element as a rigid-link with a torsional spring. [14]

(1) Segment *a* and *b* are fixed, and segment *c* is the flexible beam.

(2) Segment *c* is modeled as an initial curvature beam, *d*.

(3) Segment *c* in the deflected position, with its pseudo-rigid-body link, *e*, and torsion spring, *f*.

EM-5	Fixed-Fixed Case III	FE FB

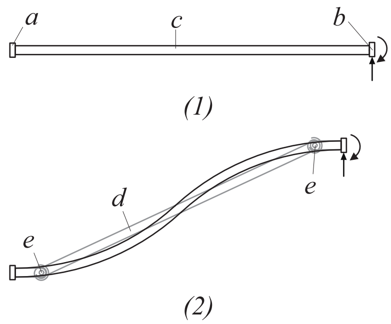


This element is a beam fixed at both ends. This case occurs when the force and moment are in opposite directions causing an inflection point. This element can be modeled using the psuedo-rigid-body model, which approximates the flexible element as rigid-links with torsional springs. [14]

(1) Segment *a* and *b* are fixed, and segment *c* is the flexible beam.

(2) Segment *c* in the deflected position, with its pseudo-rigid-body links, *d*, torsion springs, *e*, and pin *f*.

EM-6	Fixed Guided	FE FB

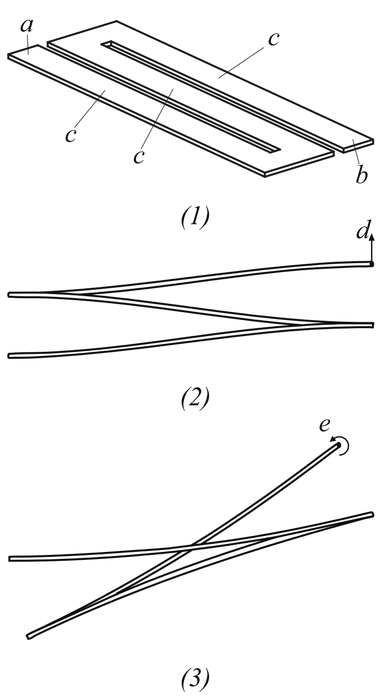


This element is a beam fixed at both ends and is a special case for a fixed-fixed case III beam. This occurs when one end goes through a deflection such that the angular deflection at the end remains constant, and the beam shape is antisymmetric about the center. It can be modeled using the pseudo-rigid-body model, which approximates the flexible element as rigid-links with torsional springs. [14]

(1) Segment *a* and *b* are fixed, and segment *c* is the flexible beam.

(2) Segment *c* in the deflected position, with its pseudo-rigid-body link, *d*, and torsion springs, *e*.

EM-7	Switch Back	FE FB



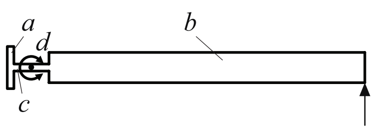
This element is a lamina emergent switch-back flexure because it is fabricated in a plane but has motion outside of the fabrication plane. It is flexible because of the increased length; yet still maintains a compact form. The switch-back can be treated as a fixed-pinned or fixed-fixed element, depending on the boundary conditions. [19]

(1) Segments *a* and *b* are attached to a mechanism. Segment *c* allows flexibility because of its increased length.

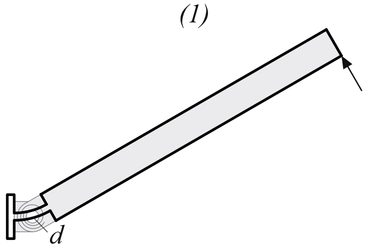
(2) Deformed configuration of a fixed-guided deflection in the *d* direction.

(3) Deformed configuration of a moment *e* on the end.

EM-8	Small Length Flexural Pivot	FE FB



(1)



(2)

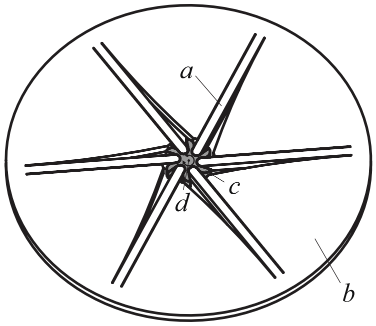
This element is a small length flexural pivot. This element can be approximated as a rigid link and a torsion spring by using the pseudo-rigid-body model, where a general rule is that the length of the flexure is much smaller than the rigid segment length. [14]

(1) Rigid segment *a* and *b* are connected by the small length flexure *c*. This mechanism rotates about the *d* axis.

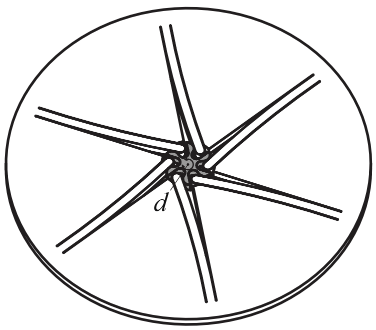
(2) Deformed configuration of rotation about the *d* axis.

Revolute

EM-9	Compliant contact-aided revolute	FE FR



(1)



(2)

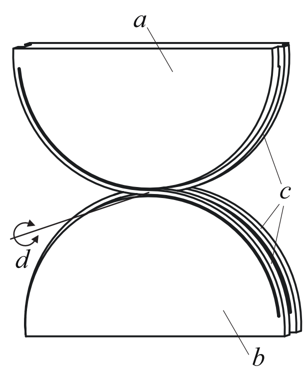
This element is a compliant contact-aided revolute (CCAR) joint. It is a planar element capable of performing functions similar to bearings and helical springs. This element can be fabricated at the micro or macro scale, and can withstand high off-axis loads. [35]

(1) Rigid segment *b* rotates around the rigid segment *c* about the *d* axis. Flexible segments, *a*, provide the energy storage and remain in contact with rigid segment *c*.

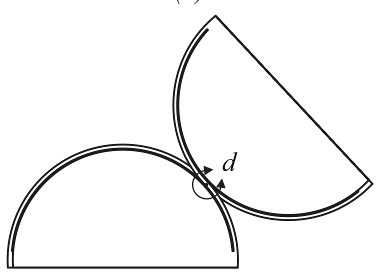
(2) Deformed configuration of rotation about the *d* axis.

Hinge

EM-10	**CORE**	FE FRH



(1)



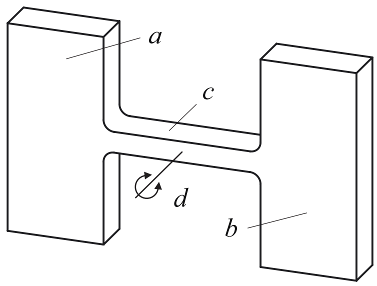
(2)

This element is a compliant rolling-contact element (CORE) that is designed for compression loads. The CORE connects two rigid links using flexible strips that pass between these rigid-link surfaces, and are attached to the links at the flexure ends. This element is unique such that the axis of rotation changes, which is located at the contact point. [35]

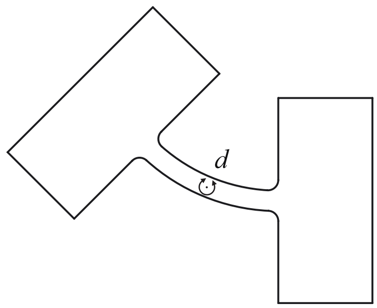
(1) Rigid segments *a* and *b* remain in contact with each other through the flexible segments, *c*, while the axis of rotation is at the contact point *d*.

(2) Deformed configuration of rotation about the contact surface.

EM-11	Small Length Flexure	FE FRH



(1)



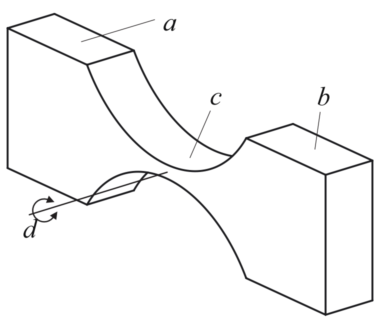
(2)

This element is a small length flexural pivot. A small length flexural pivot is defined as a segment that is significanly shorter and more flexible than its surrounding segments. [14]

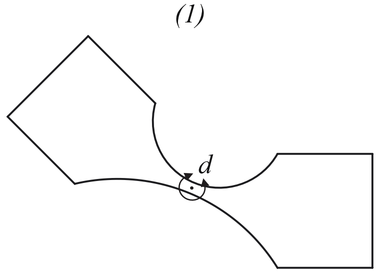
(1) Rigid segments *a* and *b* are connected by the flexible segment *c* which rotates about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-12	Living Hinges	FE FRH



(1)



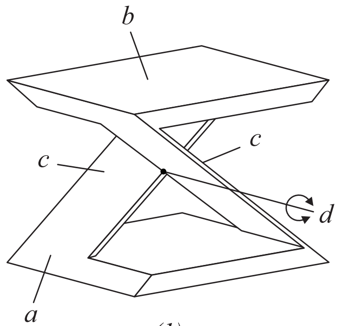
(2)

This element is a special form of a small length flexural pivot, where the flexure is extremely short and thin. This element offers little resistance throughout its deflection. [14]

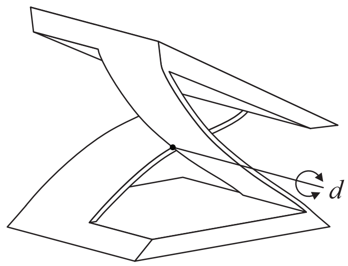
(1) Rigid segments *a* and *b* are connected by the living hinge segment *c* which rotates about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-13	Cross-Axis Flexural Pivot	FE FRH



(1)



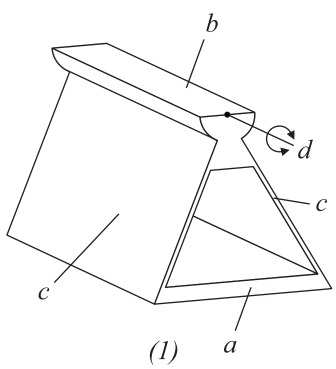
(2)

This element is a cross-axis flexural pivot because it has two flexible beams at an angle. The lengths of the flexible beams are increased because of their angle, but they do not increase the total effective length of the pivot. [14, 56]

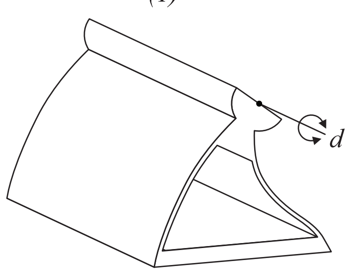
(1) Rigid segments *a* and *b* are connected by the flexible segments *c* causing rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-14	Double Blade Rotary Pivot	FE FRH



(1)



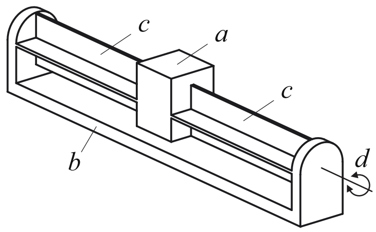
(2)

This element's axis of rotation remains parallel to the plane ground.

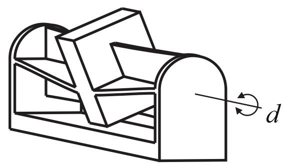
(1) Rigid segments *a* and *b* are attached to a mechanism. This element achieves compliance by the flexible segments *c*, causing rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-15	Large Deformation	FE FRH



(1)



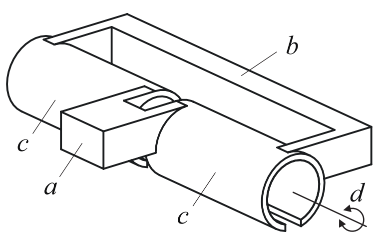
(2)

This element is designed for large rotations with high off-axis stiffness. The geometry of the cross-plates allow a high degree of flexibility in torsion. [34]

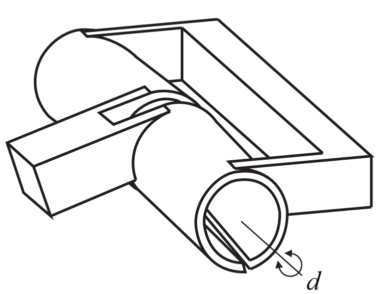
(1) Rigid segments *a* and *b* are attached to the two cross-plates *c* which rotate about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-16	Split-Tube Flexures	FE FRH



(1)



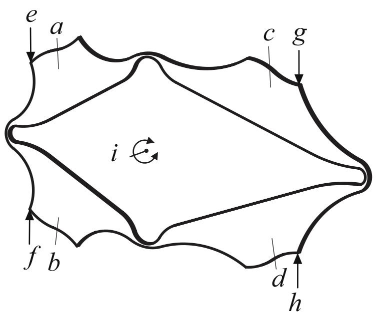
(2)

This element is designed for large rotations with high off-axis stiffness. The spit-tube flexures rely on torsion for their flexility. [36]

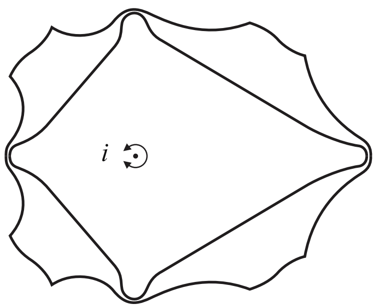
(1) Rigid segments *a* and *b* are attached to the split-tube flexures *c* which rotate about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-17	**Deltoid Q-Joint**	FE FRS



(1)



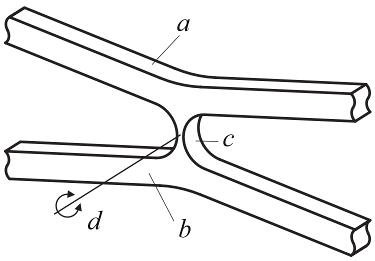
(2)

This element is a deltoid-type Q-joint. It is constructed when each rigid segment in the quadrilateral is made adjacent to a segment of equal length. [14]

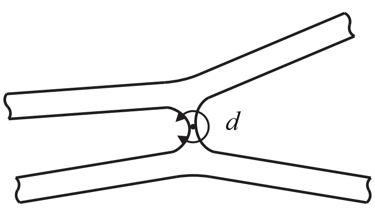
(1) Rigid segments *a* and *b*, and rigid segments *c* and *d* are equal length, respectively. When the rigid segments *a* and *b* deform in the *e* and *f* directions, respectively, the rigid segments *c* and *d* deform in the *g* and *h* directions, respectively. This element appears to rotate about the *i* axis.

(2) Deformed configuration. The angle between rigid segments *a* and *b*, and *c* and *d*, respectively, decreases.

EM-18	**Scissor Hinge**	FE FRS



(1)



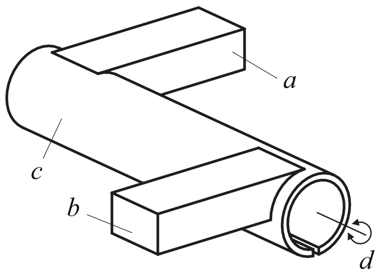
(2)

This element allows a scissor action by use of a small flexure placed in the middle of rigid segments.

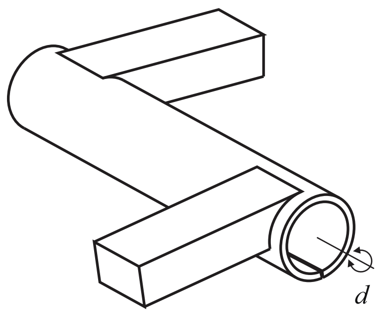
(1) Rigid segments *a* and *b* are connected by the flexible segment *c* and rotate about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-19	**Split-Tube Flexures**	FE FRT



(1)



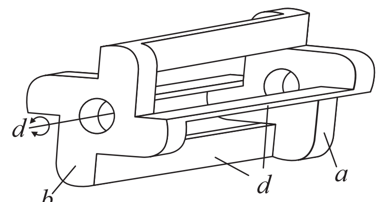
(2)

This element is a split-tube flexure. It is compliant in the desired axis of rotation, but stiff in its other axes. [36]

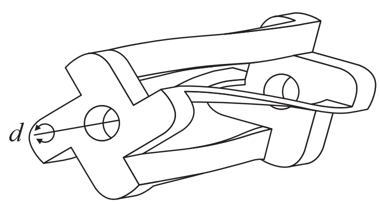
(1) Rigid segments *a* and *b* are attached to the split-tube flexure *c*, which rotates about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-20	Quadra Blade Rotary	FE FRT



(1)



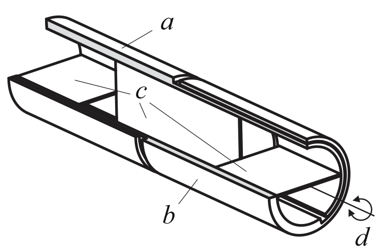
(2)

This element uses beams in a circular array, which allows rotation. When the rotation is large, the length of the element retracts.

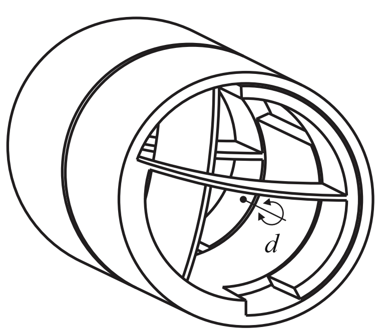
(1) Rigid segments *a* and *b* are attached to a mechanism. Flexible segments *c* allow rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-21	Tubular Cross-Axis Flexural Pivot	FE FRT



(1)



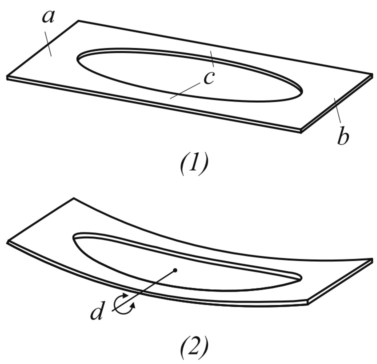
(2)

This element is a cross-axis flexural pivot because it has two flexible beams at an angle. The lengths of the flexible beams are increased because of their angle, but they do not increase the total effective length of the pivot. [14, 56]

(1) Rigid segments *a* and *b* are connected by the flexible segments *c* causing rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-22	Reduced Inside Area Joint	FE FRL

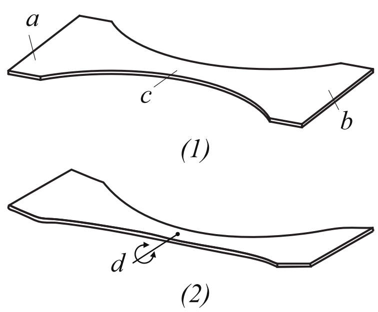


This element's inside area is reduced, allowing greater flexibility. It is suited for applications where angular rotation is desired. This element can also be fabricated in a single plane (lamina emergent). [19]

(1) Rigid segments *a* and *b* are attached to a mechanism. Segments *c* are flexible compared to the rest of the segments because of the reduced cross-sectional area, which allows rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-23	Reduced Outside Area Joint	FE FRL

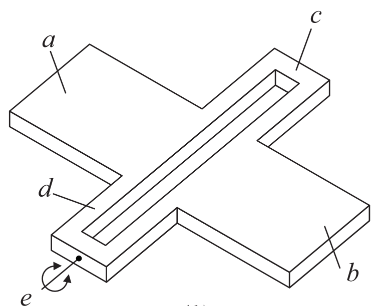


This element's outside areas are reduced, allowing greater flexibility. It is suited for applications where angular rotation is desired. This element can be fabricated in a single plane (lamina emergent). [19]

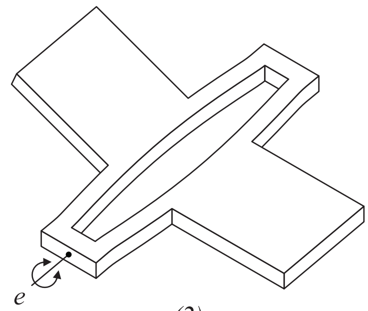
(1) Rigid segments *a* and *b* are attached to a mechanism. Segment *c* is flexible compared to the rest of the segments because of the reduced cross-sectional area, which allows rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-24	Outside LET Joint	FE FRL



(1)



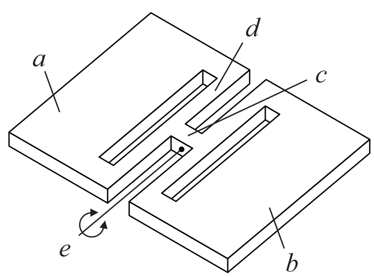
(2)

This element is a lamina emergent torsion (LET) joint. It is suited for applications where large angular rotation is desired, but high off-axis stiffness is not critical. It can be fabricated in a single plane. [37]

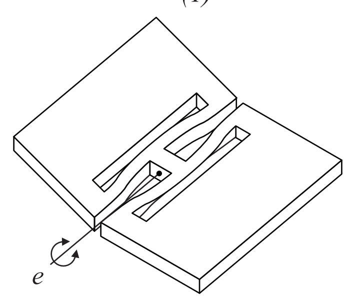
(1) Rigid segments *a* and *b* are attached to a mechanism. The flexible segments *c* and *d* are in bending and torsion, respectively, causing a rotation about the *e* axis.

(2) Deformed configuration of rotation about the *e* axis.

EM-25	Inside LET Joint	FE FRL



(1)



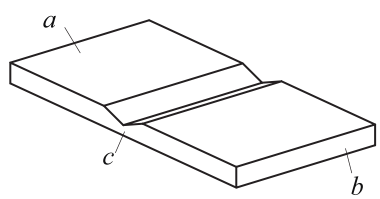
(2)

This element is a lamina emergent torsion (LET) joint. It is suited for applications where large angular rotation is desired but high off-axis stiffness is not critical. It can be fabricated in a single plane. [37]

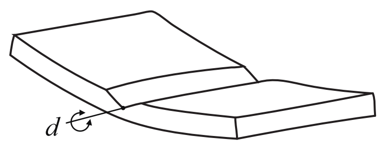
(1) Rigid segments *a* and *b* are attached to a mechanism. The flexible segments *c* and *d* are in bending and torsion, respectively, causing a rotation about the *e* axis.

(2) Deformed configuration of rotation about the *e* axis.

EM-26	Notch Joint	FE FRL



(1)



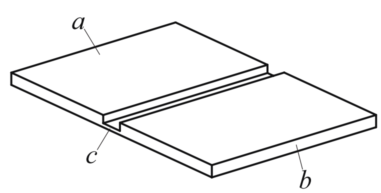
(2)

This element is designed for angular rotation and can be fabricated in a single plane (lamina emergent). The reduced thickness of this element allows for greater flexibility. [19]

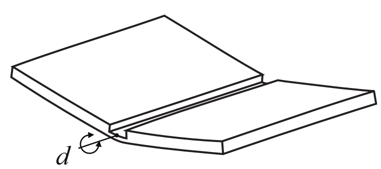
(1) Rigid segments *a* and *b* are attached to a mechanism. Segment *c* allows flexibility because of its reduced thickness, allowing rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-27	Groove Joint	FE FRL



(1)



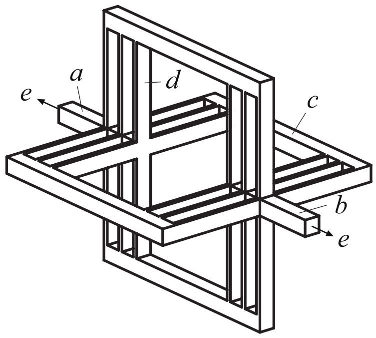
(2)

This is a lamina emergent groove joint. It is suited for applications where angular rotation is desired. The reduced thickness allows greater flexibility. [19]

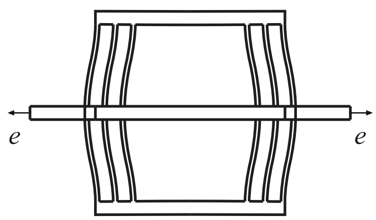
(1) Rigid segments *a* and *b* are attached to a mechanism. Segment *c* allows flexibility because of its reduced thickness, allowing rotation about the *d* axis.

(2) Deformed configuration of rotation about the *d* axis.

EM-28	**Leaf Spring Translational Joint**	FE FT



(1)



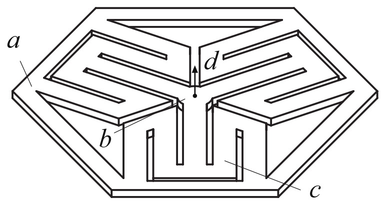
(2)

This element is a leaf spring translational joint that is designed to have high off-axis stiffness. This element has a relatively large range of motion. [34]

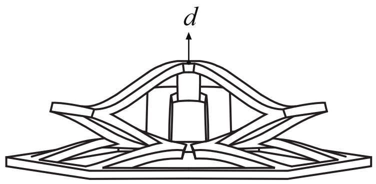
(1) Rigid segments *a* and *b* are attached to a mechanism. If the rigid segment *a* is fixed then the rigid segment *b* translates in the *e* direction. Segments *c* are rigid and segments *d* are flexible.

(2) Deformed configuration of translation in the *e* direction.

EM-29	LEM Translator	FE FTL



(1)



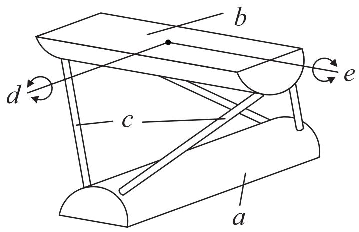
(2)

This element is a lamina emergent translator. It utilizes switch-back beams for a compact translational motion. This element can be fabricated in a single plane.

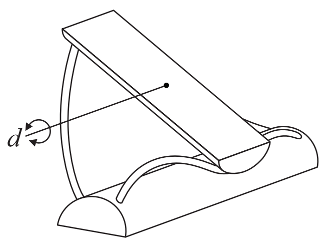
(1) Rigid segments *a* and *b* are attached to a mechanism. The flexible segments *c* are switch-back beams that allow flexibility, which is able to translate in the *d* direction

(2) Deformed configuration of translation in the *d* direction.

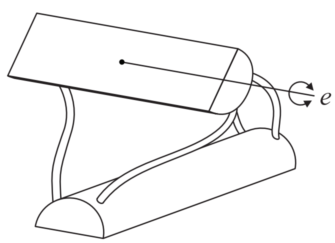
EM-30	**Ortho Skew Double Rotary**	FE FU



(1)



(2)



(3)

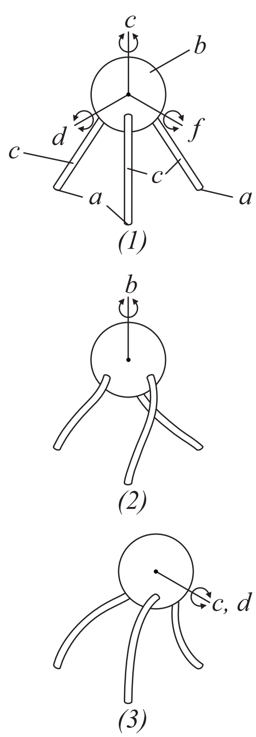
This element's axes of each of the four constraints intersect both lines of rotation.

(1) Rigid segment *a* and *b* are attached to a mechanism. The flexible constraints *c* allow rotation about the *d* and *e* axis.

(2) Deformed configuration of rotation about the *d* axis.

(3) Deformed configuration of rotation about the *e* axis.

EM-31	Tripod Spherical Joint	FE FU



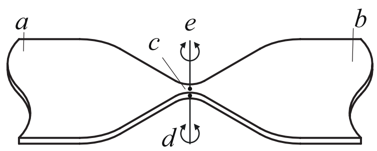
This element has three orthogonal rotational degrees of freedom. This flexure emulates the degrees of freedom of a spherical ball joint.

(1) Rigid segments *a* and *b* are attached to a mechanism. Rigid segment *b* is able to rotate about the *d*, *e* and *f* axis, by the flexible constraints *c*.

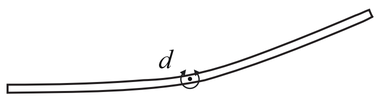
(2) Deformed configuration of rotation about the *d* axis.

(3) Deformed configuration of rotation about the *e* or *f* axis.

EM-32	Reduced Outside Area Joint	FE FUL



(1)



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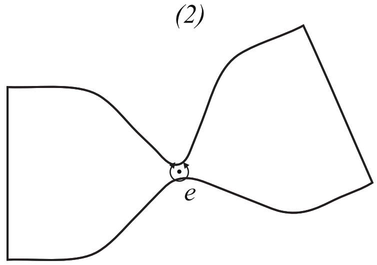


(3)

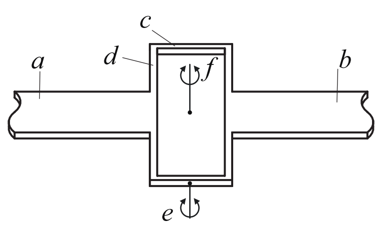
This element is a unique outside reduced area joint, such that the width of the reduced area is similar to its thickness. This reduces the off-axis stiffness and the rotational element becomes a universal element. This element can be fabricated in a single plane. [19]

(1) Rigid segments *a* and *b* are attached to a mechanism. Segment *c* is flexible compared to the rest of the segments because of the reduced cross-sectional area, which allows rotation about the *d* and *e* axes.

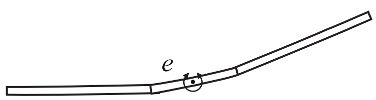
(2) Deformed configuration of rotation about the *d* axis.

(3) Deformed configuration of rotation about the *e* axis.

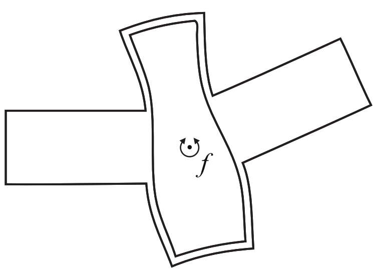
EM-33	Outside LET Joint	FE FUL



(1)



(2)



(3)

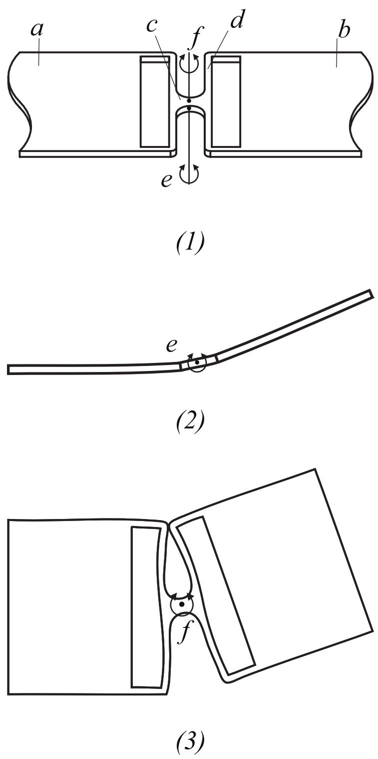
This element is a unique inside lamina emergent torsion (LET) joint where the torsional hinges are on the outside of the element. In this case, the off-axis stiffness is reduced, allowing the rotational element to become a universal element. This element can be fabricated in a single plane. [37]

(1) Rigid segments *a* and *b* are attached to a mechanism. The flexible segments *c* and *d* are in bending and torsion, respectively, causing a rotation about the *e* and *f* axes.

(2) Deformed configuration of rotation about the *e* axis.

(2) Deformed configuration of rotation about the *f* axis.

EM-34	**Inside LET Joint**	FE FUL



This element is a unique inside lamina emergent torsion (LET) joint where the torsional hinges are on the inside of the element. In this case, the off-axis stiffness is reduced, allowing the rotational element to become a universal element. This element can be fabricated in a single plane. [37]

(1) Rigid segments *a* and *b* are attached to a mechanism. The flexible segments *c* and *d* are in bending and torsion, respectively, causing a rotation about the *e* and *f* axes.

(2) Deformed configuration of rotation about the *e* axis.

(2) Deformed configuration of rotation about the *f* axis.

A.1.2 Rigid-Link Joints

Revolute

<table>
<tr><th>EM-35</th><th>Revolute Joint</th><th>RLJ
RR</th></tr>
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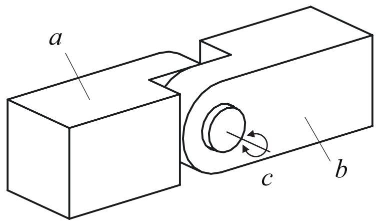
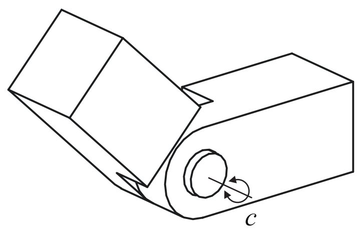


(1)



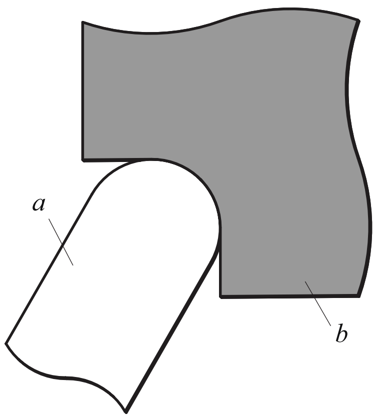
(2)

This element is a lower kinematic pair that provides one rotational degree of freedom between connected links. [19]

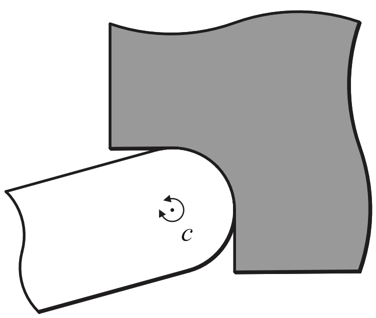
(1) Rigid segments *a* and *b* rotate about the *c* axis.

(2) Deformed configuration of rotation about the *c* axis.
</td></tr>
</table>

EM-36	Passive Joint	RLJ RR



(1)



(2)

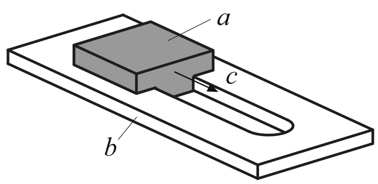
This element allows rotation between two rigid segments without using a traditional pin joint. These segments need to be in contact to operate. [14]

(1) Rigid segments *a* and *b* are to remain in contact, allowing rotation about the *c* axis.

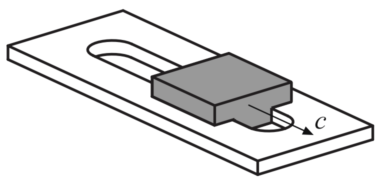
(2) Deformed configuration of rotation about the *c* axis.

Prismatic

EM-37	**Prismatic Joint**	RLJ RP



(1)



(2)

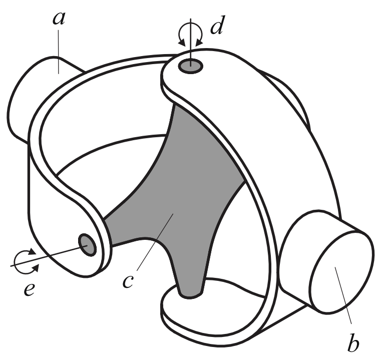
This element is a lower kinematic pair that provides one translational degree of freedom between connected links. [19]

(1) Rigid segment *a* translates in the *c* direction on rigid segment *b*.

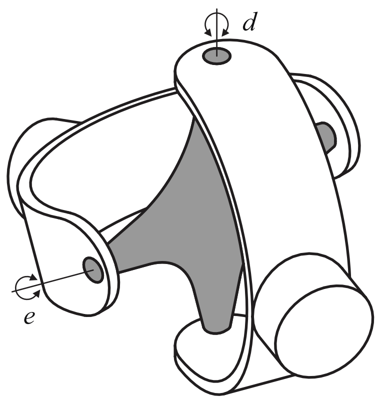
(2) Deformed configuration of translation in the *c* direction.

Universal

EM-38	**Universal Joint**	RLJ RU



(1)



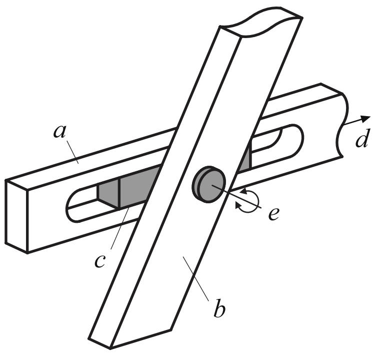
(2)

This element provides two rotational degrees of freedom between connected links. [19]

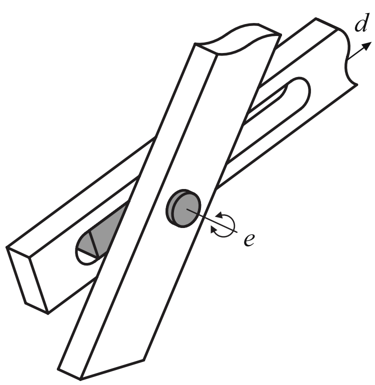
(1) Rigid segments *a* and *b* rotate around the *d* and *e* axes, by rigid segment *c*.

(2) Deformed configuration of rotation about the *d* and *e* axes.

EM-39	**Half Joint**	RLJ RO



(1)



(2)

This element provides one rotational and one translational degree of freedom between connected links. The axis of rotation is orthogonal to the direction of translation. [19]

(1) Rigid segment *a* translates in the *d* direction and rotates about the *e* axis on rigid segement *c*, which is attached to ground, *b*.

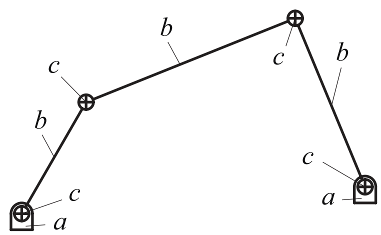
(2) Deformed configuration of translation in the *d* direction and rotation about the *e* axis.

A.2 Mechanisms

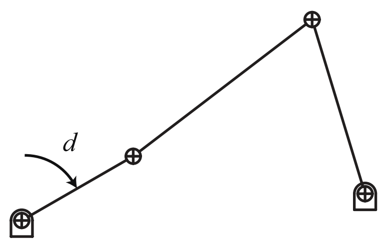
A.2.1 Basic

Four-Bar Mechanism

M-1	**Four-Bar**	BA BF



(1)



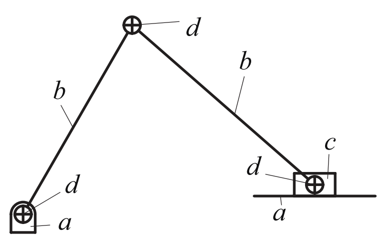
(2)

A four-bar linkage is one of the most simple mechanisms that has one degree of freedom. The configurations and functionality of this type of mechanism are shown by the Grashof criterion and Barker's complete classification for four-bar mechanisms. [17]

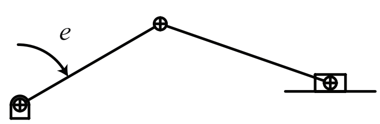
(1) Rigid bodies *a* are fixed. Rigid bodies *b* are binary links connected by revolute elements, *c*.

(2) Deformed configuration of input *d*.

M-2	**Crank Slider**	BA BF



(1)



(2)

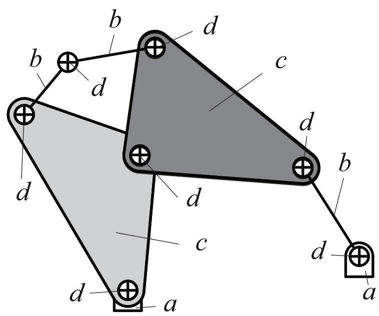
The crank slider is a unique four-bar mechanism that has a one degree of freedom of translation.

(1) Rigid bodies *a* are fixed. Rigid bodies *b* are binary links connected by a prismatic element, *c*, and revolute elements, *d*.

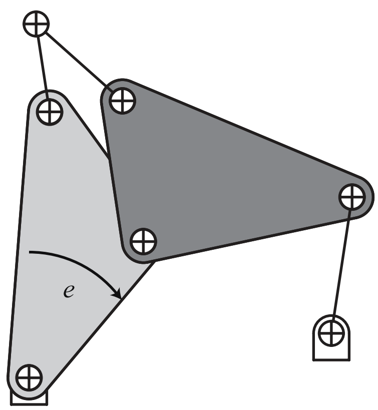
(2) Deformed configuration of input *e*.

Six-Bar Mechanism

M-3	Watt Inversion I	BA BS



(1)



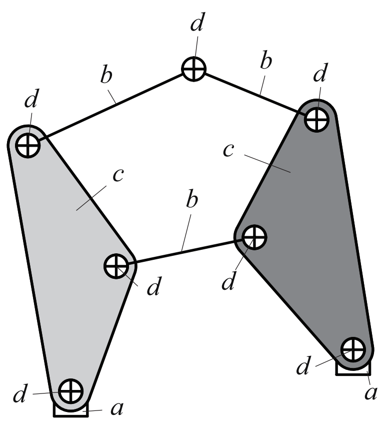
(2)

A Watt Mechanism is a six-bar mechanism characterized as having its two ternary links connected. This mechanism is in the inversion I configurations.

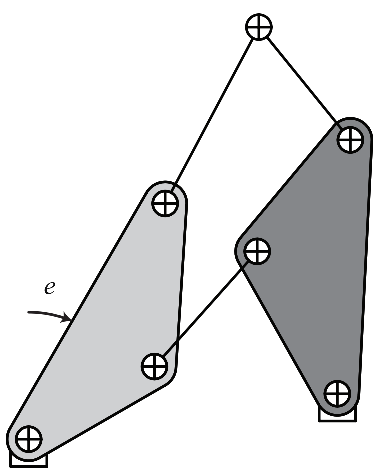
(1) Rigid bodies *a* are fixed. Rigid body binary, *b*, and ternary, *c*, links are connected by revolute elements, *d*.

(2) Deformed configuration of input *e*.

M-4	**Stephenson Inversion I**	BA BS



(1)



(2)

A Stephenson Mechanism is a six-bar mechanism characterized as having its two ternary links separated by a binary link. This mechanism is in the inversion I configurations.

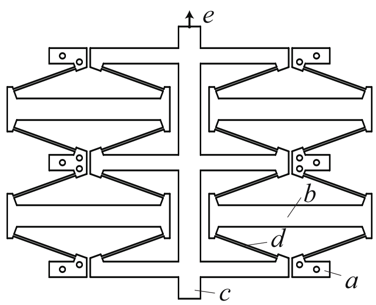
(1) Rigid bodies *a* are fixed. Rigid body binary, *b*, and ternary, *c*, links are connected by revolute elements, *d*.

(2) Deformed configuration of input *e*.

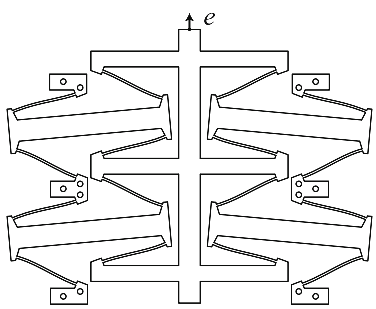
A.2.2 Kinematics

Translational

M-5	X Bob	KM TS



(1)



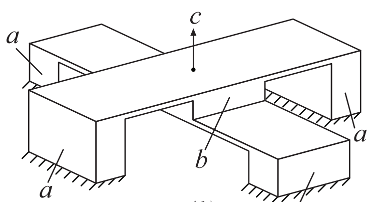
(2)

This mechanism is a fully compliant linear-motion mechanism with high off-axis stiffness. The design is based on multiple Roberts four-bar approximate straight-line mechanism, and by using symmetry. [64]

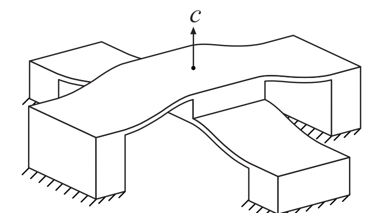
(1) Rigid bodies *a* are fixed. Segments *b* are rigid. Rigid body *c* translates in the *e* direction by flexible segments *d*.

(2) Deformed configuration of translation in the *e* direction.

M-6	Precision Cross Bladed Translator	KM TSP



(1)



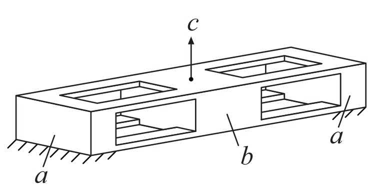
(2)

This mechanism is suited for precision applications where only one translational degree of freedom is required. All other motions are constrained. The translational degree of freedom is orthogonal to the plane of the ground.

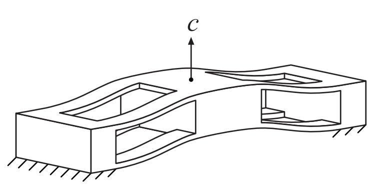
(1) Rigid bodies *a* are fixed. Rigid body *b* is free to translate in the *c* direction.

(2) Deformed configuration when translating in the *c* direction.

M-7	Parallel Blade Translator	KM TSP



(1)



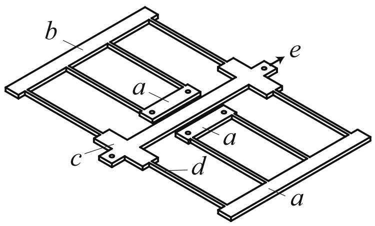
(2)

This mechanism is suited for precision applications where only one translational degree of freedom is required. All other motions are constrained. The translational degree of freedom is orthogonal to the ground plane.

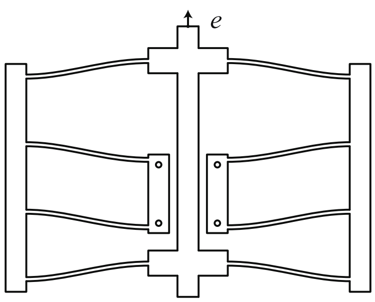
(1) Rigid bodies *a* are fixed. Rigid body *b* is free to translate in the *c* direction.

(2) Deformed configuration when translating in the *c* direction.

M-8	Parallel Translator	KM TSL



(1)



(2)

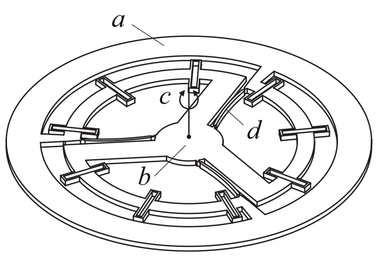
This mechanism has mirrored and folded parallel-guiding mechanisms. This mechanism allows a translational degree of freedom with high off-axis stiffness.

(1) Rigid bodies *a* are fixed. Segments *b* are rigid. Rigid body *c* translates in the *e* direction by flexible segments *d*.

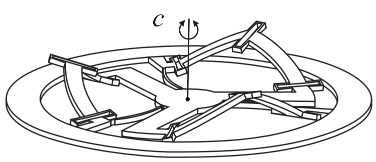
(2) Deformed configuration of translation in the *e* direction.

Rotational

M-9	Rotational LEM	KM RT



(1)



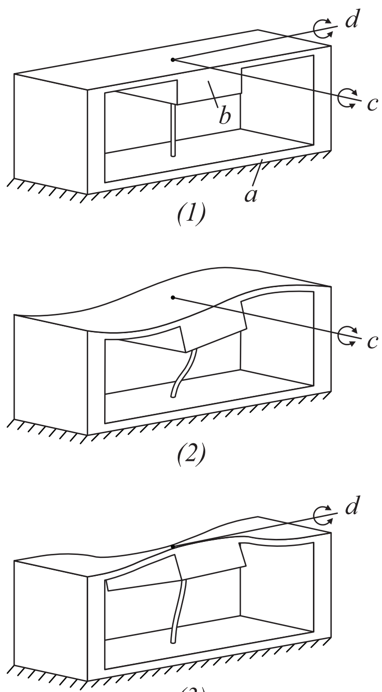
(2)

This is a spherical lamina emergent mechanism (LEM), which emerges out of plane when rotated. [37]

(1) Rigid body *a* is fixed. Rigid body *b* rotates about the *c* axis. The mechanism's rotation is constrained by the flexible segments *d*.

(2) Deformed configuration of rotation about the *c* axis.

M-10	Precision Constraint Rotator	KM RTP



(1)

(2)

(3)

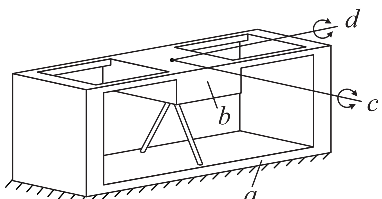
This mechanism is suited for precision applications where two orthogonal rotational degrees of freedom are required. The orthogonal rotational degree of freedom are parallel to the plane of the ground.

(1) Rigid body *a* is fixed. Rigid body *b* rotates about the *c* and *d* axes.

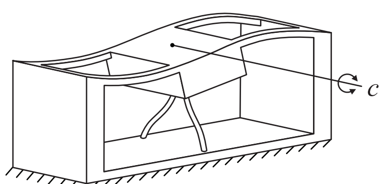
(2) Deformed configuration of rotation about the *c* axis.

(3) Deformed configuration of rotation about the *d* axis.

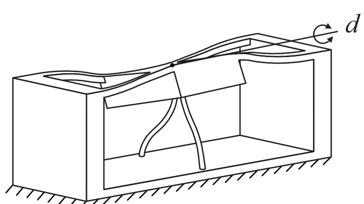
M-11	Precision Constraint Rotator	KM RTP



(1)



(2)



(3)

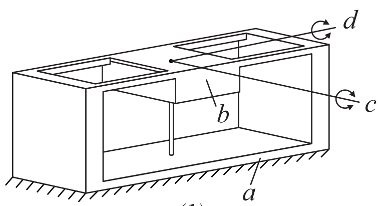
This mechanism is suited for precision applications where two orthogonal rotational degrees of freedom are required. The rotational degrees of freedom are parallel to the plane of the ground.

(1) Rigid body *a* is fixed. Rigid body *b* rotates about the *c* and *d* axes.

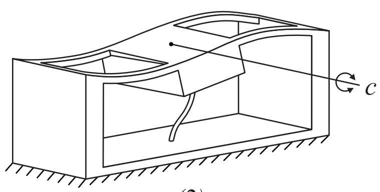
(2) Deformed configuration of rotation about the *c* axis.

(3) Deformed configuration of rotation about the *d* axis.

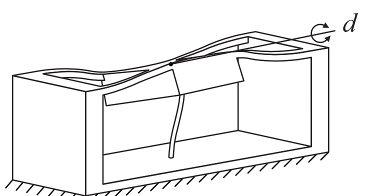
M-12	Precision Constraint Rotator	KM RTP



(1)



(2)



(3)

This mechanism is suited for precision applications where two orthogonal rotational degrees of freedom are required. The orthogonal rotational degrees of freedom are parallel to the plane of the ground.

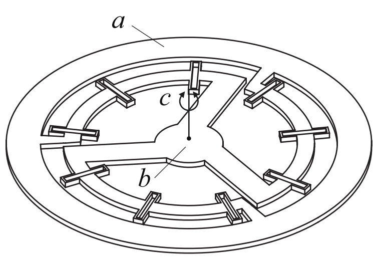
(1) Rigid body *a* is fixed. Rigid body *b* rotates about the *c* and *d* axes.

(2) Deformed configuration of rotation about the *c* axis.

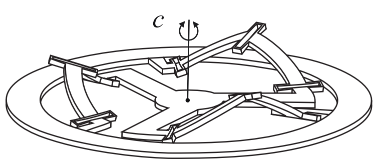
(3) Deformed configuration of rotation about the *d* axis.

Large Motion Path

M-13	Rotational LEM	KM RTL



(1)



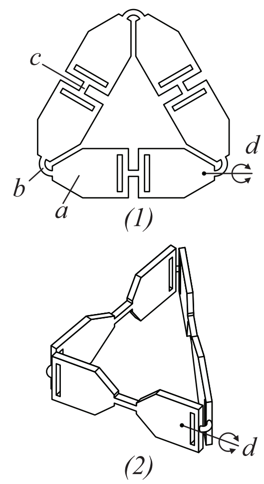
(2)

This is a spherical lamina emergent mechanism (LEM) that emerges out of the manufactured plane when rotated. [37]

(1) Rigid body *a* is fixed. Rigid body *b* rotates about the *c* axis.

(2) Deformed configuration of rotation about the *c* axis.

M-14	Bricard 6R (LEM)	KM RTL

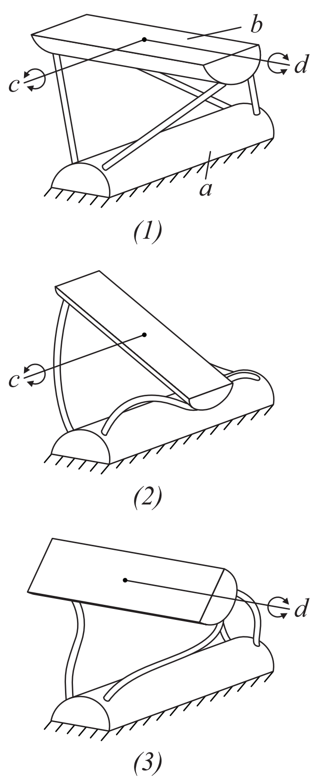


This is a Bricard 6R fully compliant lamina emergent mechanism (LEM). This mechanism allows infinite rotation. [19, 65]

(1) Rigid body *a* rotates by small length flexure *b* and the LET joint *c*.

(2) Deformed configuration of rotation, *d*.

M-15	Ortho Skew Double Rotary	KM RTL



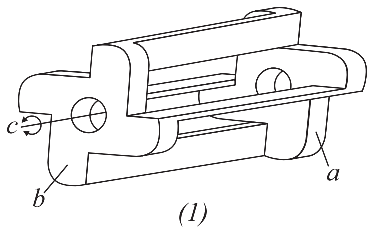
This mechanism is suited for precision applications where two skew orthogonal rotational degrees of freedom are required. The axes of each of the four constraints intersect both lines of rotation.

(1) Rigid body *a* is fixed. Rigid body *b* rotates about the *c* and *d* axes.

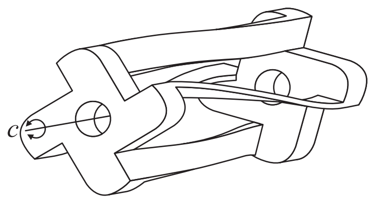
(2) Deformed configuration of rotation about the *c* axis.

(3) Deformed configuration of rotation about the *d* axis.

M-16	Quadra Blade Rotary	KM TR



(1)



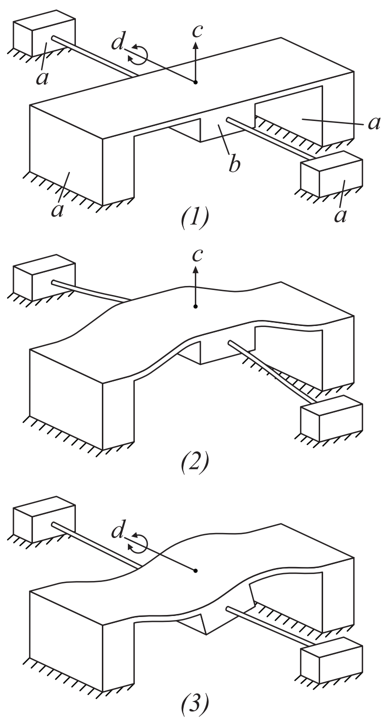
(2)

This mechanism is suited for precision applications where a single rotational degree of freedom is required. If the rotation is large the stage will retract toward the ground with a translation as well. The axis of rotation is perpendicular to the ground plane.

(1) Rigid body *a* is fixed. Rigid body *b* rotates about the *c* axis.

(2) Deformed configuration of rotation about the *c* axis.

M-17	Precision Cross Bladed Translator	KM TRP



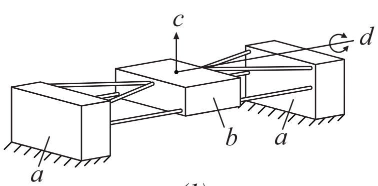
This mechanism is suited for precision applications where a translational and rotational degrees of freedom are required. The translational and rotational degrees of freedom are orthogonal. The translational degree of freedom is orthogonal to the plane of the ground. The rotational degree of freedom is parallel to the plane of the ground.

(1) Rigid bodies *a* are fixed. Rigid body *b* translates in the *c* direction and rotates about the *d* axis.

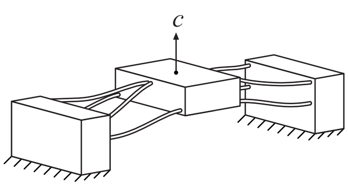
(2) Deformed configuration of translation in the *c* direction.

(3) Deformed configuration of rotation about the *d* axis.

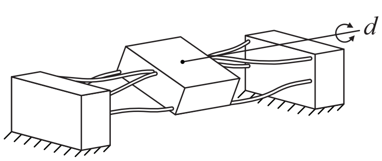
M-18	**Crossed Constraint Translator and Rotator**	KM TRP



(1)



(2)



(3)

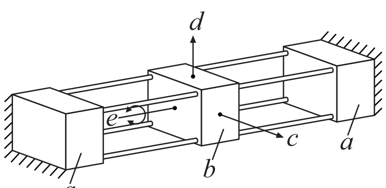
This mechanism is suited for precision applications where both a translational and rotational degree of freedom is required. The translational degree of freedom is orthogonal to the axis of the rotational degree of freedom. The translational degree of freedom is orthogonal to the plane of the ground and the rotational degree of freedom axis is parallel to the plane of the ground.

(1) Rigid bodies *a* are fixed. Rigid body *b* translates in the *c* direction and rotates about the *d* axis.

(2) Deformed configuration of translation in the *c* direction.

(3) Deformed configuration of rotation about the *d* axis.

M-19	**Precision Octa Parallel Symmetric Constraint**	KM TRP



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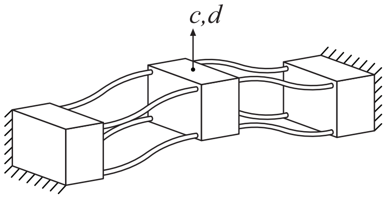

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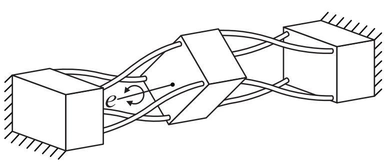


(3)

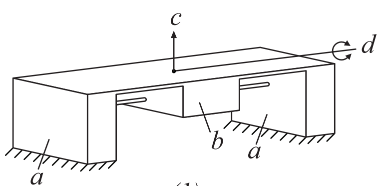
This mechanism is suited for precision applications where two orthogonal translational degrees of freedom and one rotational degree of freedom are required. All of these degrees of freedom are orthogonal. The two translations are parallel to the ground plane and the rotation is perpendicular to the ground plane.

(1) Rigid bodies *a* are fixed. Rigid body *b* translates in the *c* and *d* direction and rotates about the *e* axis.

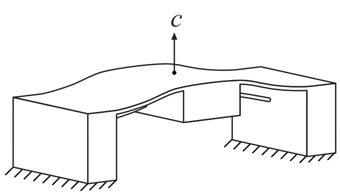
(2) Deformed configuration of translation in the *c* or *d* direction.

(3) Deformed configuration of rotation about the *e* axis.

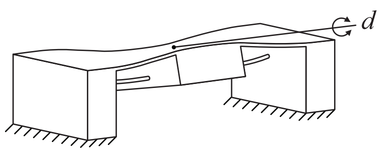
M-20	Parallel Blade Constraint	KM TRP



(1)



(2)



(3)

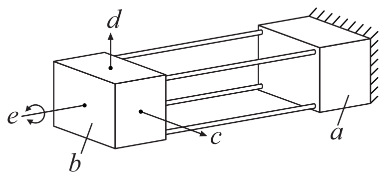
This mechanism is suited for precision applications where rotational and translation degrees of freedom are required. The rotational and translational degrees of freedom are orthogonal. The translational degree of freedom is perpendicular to the plane of the ground and the rotational degree of freedom is parallel to the plane of the ground.

(1) Rigid bodies *a* are grounded. Rigid body *b* translates in *c* direction and rotates about the *d* axis.

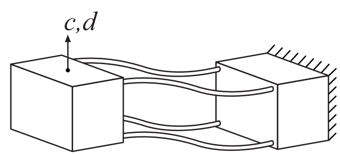
(2) Deformed configuration of translation in the *c* direction.

(3) Deformed configuration of rotation about the *d* axis.

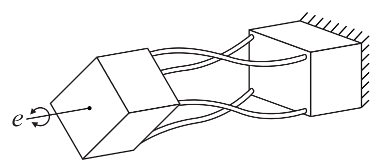
M-21	Quadra Parallel Constraint	KM TRL



(1)



(2)



(3)

This mechanism is suited for precision applications where two orthogonal translational degrees of freedom and one rotational degree of freedom are required. All of these degrees of freedom are orthogonal. The two translations are parallel to the plane of the ground and the rotation is perpendicular to the plane of the ground. The rotation will cause the rigid body to retract toward the ground with an undesired translation if the rotation is not small enough.

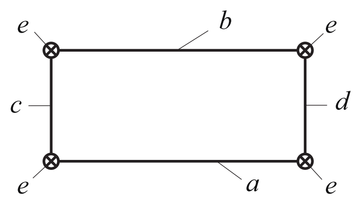
(1) Rigid body *a* is grounded. Rigid body *b* may translate in the *c* and *d* directions and rotate about the *e* axis.

(2) Deformed configuration of translation in *c* or *d* direction.

(3) Deformed configuration of rotation about the *e* axis.

Parallel Motion

M-22	4-Bar Parallel Guider	KM PM



(1)

(2)

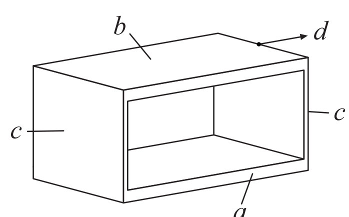
A 4-bar, parallel-guiding mechanism is a mechanism whose two opposing links remain parallel throughout the mechanism's motion. This design can have multiple configurations based upon its synthesis. [14, 22, 33]

(1) Rigid body *a* is fixed. Points *b*, *c*, *d*, and *e* provide a pivot rotation by either a flexible or rigid element.

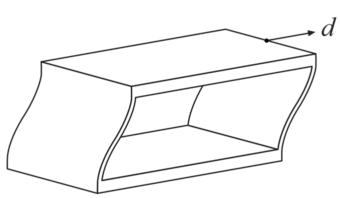
(2) Deformed configuration in the *f* direction.

Precision

M-23	Parallel Guiding	KM PM



(1)



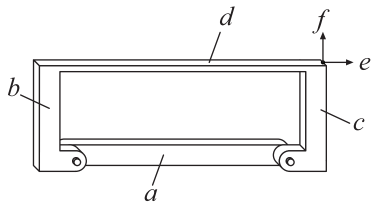
(2)

This mechanism achieves this motion by two fixed-guided beams. [33, 66]

(1) Rigid body *a* is fixed. Rigid body *b* translates in the *d* direction by the flexible fixed-guided beams *c*.

(2) Deformed configuration of translation in the *d* direction.

M-24	Parallel Guided	KM/KN PML/ES



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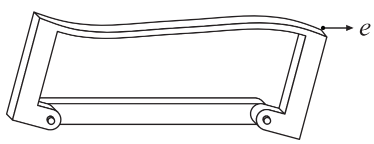

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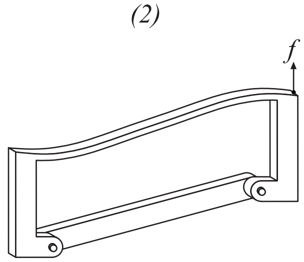


(3)

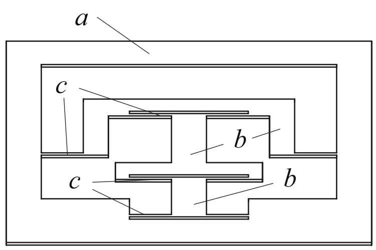
This mechanism's links remain parallel throughout the mechanism's motion and is capable of large deflections with energy storage. [43]

(1) Rigid body *a* is a rigid link. Rigid bodies *b* and *c* are rigid segments. Segment *d* is a fixed-guided beam.

(2) When rigid body *a* is fixed, the mechanism deforms in the *e* direction.

(3) When rigid body *b* is fixed, the mechanism deforms in the *f* direction.

M-25	Parallel-Guided LEM	KM PML



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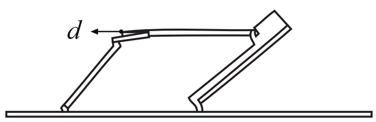


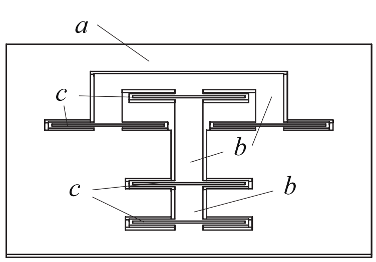
(2)

This is a lamina emergent parallel-guiding mechanism. It achieves its motion through torsion elements and LET joints. This mechanism can be fabricated in a single plane. [59]

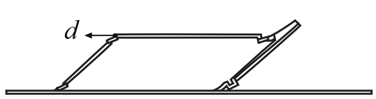
(1) Rigid body *a* is fixed. Rigid bodies *b* are rigid-link segments. Flexible segments *c* provide the rotational motion.

(2) Deformed configuration in the *d* direction.

M-26	Parallel-Guided LEM	KM PML



(1)



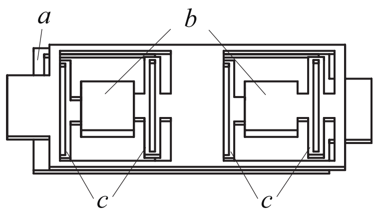
(2)

This is a lamina emergent parallel-guiding mechanism. It achieves its motion through torsion elements and LET joints. This mechanism can be fabricated in a single plane. [59]

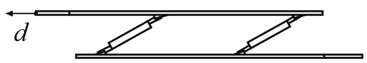
(1) Rigid body *a* is fixed. Rigid bodies *b* are rigid-link segments. Flexible segments *c* provide the rotational motion.

(2) Deformed configuration in the *d* direction.

M-27	Multi-Layer Parallel-Guided LEM	KM PML



(1)



(2)

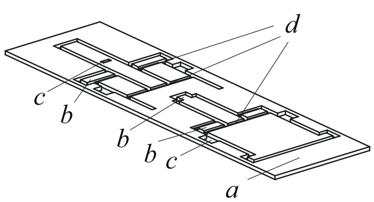
This is a multi-layer, lamina emergent, parallel-guiding mechanism. It achieves its motion through torsion elements and LET joints. This mechanism can be fabricated in a single plane. [59]

(1) Rigid body *a* is fixed. Rigid bodies *b* are rigid-link segments. Flexible segments *c* provide the rotational motion.

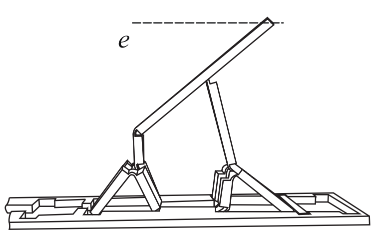
(2) Deformed configuration in the *d* direction.

Straight Line

M-28	Hoeken (LEM)	KM SL



(1)



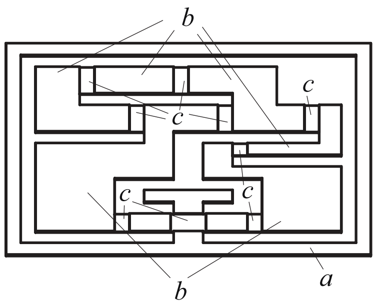
(2)

This is a fully compliant lamina emergent Hoeken mechanism that was designed using the compliant ortho-planar metamorphic mechanism (COPMM) technique. A Hoeken mechanism produces a straight line through part of its motion. This mechanism can be fabricated in a single plane. [30, 67]

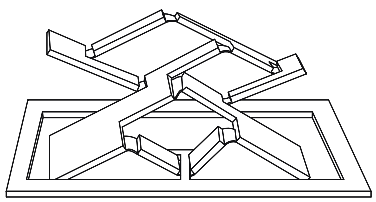
(1) Rigid body *a* is fixed. Segment *b* is inserted into segment *c* during assembly. Segments *d* allows flexibility.

(2) Assembled configuration of mechanism. The end point traces through a near straight line, *e*.

M-29	Pantograph (LEM)	KM/KN SA/FA



(1)



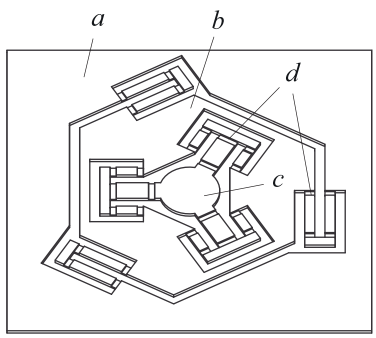
(2)

A pantograph mechanism is a multi-degree-of-freedom device used for scaling force or motion. This mechanism is designed to be lamina emergent. [4]

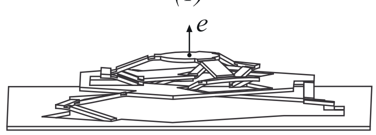
(1) Rigid body *a* is fixed. Segments *b* are significantly rigid and segments *c* allow flexibility.

(2) Deformed configuration.

M-30	Multiple Stage Platform	KM/KN SP/ES



(1)



(2)

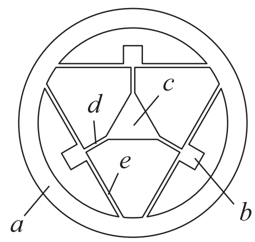
This mechanism is similar to ortho-planar springs, but it uses a multi-stage platform to raise its platform. [4]

(1) Rigid body *a* is fixed. Rigid bodies *b* and *c* are platforms. Segments *d* are the flexible segments allowing platform *c* to translate in the *e* direction.

(2) Deformed configuration after translation in the *e* direction.

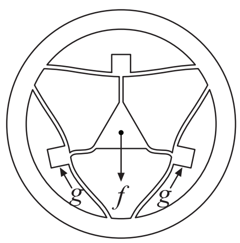
M-31	HexFlex	KM SPP

The HexFlex™ is a single-layer, multi-axis spatial positioning control mechanism, which can be used for both macro and micro applications that require precision positioning. [39]



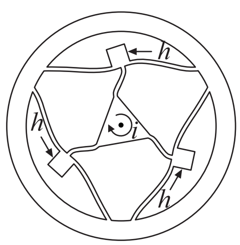
(1)

(1) Rigid body *a* is fixed. Rigid bodies *b* are the actuator action tabs. Rigid body *c* is the motion stage. Flexible elements, *d* and *e*, allow infinitesimal motions.



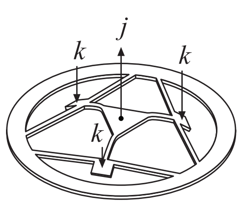
(2)

(2) Deformed configuration by planar displacement of the actuator tabs in the *g* direction, which causes the motion stage to displace in the *f* direction.



(3)

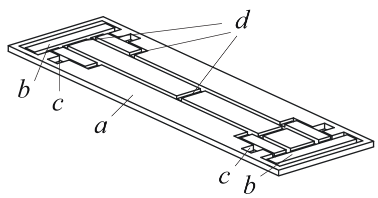
(3) Deformed configuration by planar displacement of the actuator tabs in the *h* direction, which causes the motion stage to rotate about the *i* axis.



(4)

(4) Deformed configuration by orthogonal displacement of actuator tabs in the *k* direction, which causes the motion stage to translate in the *j* direction.

M-32	Lamina Emergent 4 Bar	KM MM



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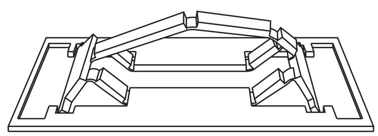


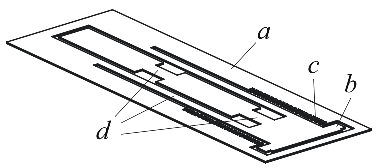
(2)

This is a lamina emergent four-bar mechanism that was designed using the compliant ortho-planar metamorphic mechanism (COPMM) technique, allowing the mechanism to be raised from the initial plane of fabrication by using a system of redundant link structures. [67]

(1) Rigid body *a* is fixed. Segments *b* are inserted into segments *c* during assembly. Segments *d* allow flexibility from the manufactured state to the configured state.

(2) Assembled configuration of mechanism.

M-33	Bistable Locking COPMM	KM/KN MM/SBB



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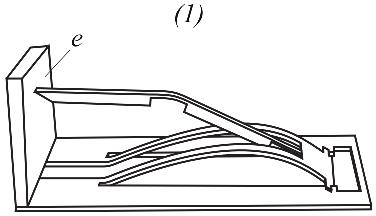
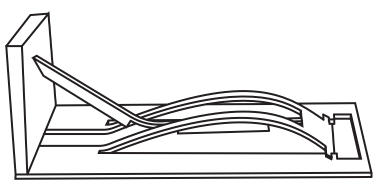


(2)



(3)

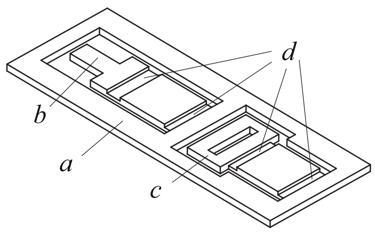
This is a bistable locking compliant ortho-planar metamorphic mechanism (COPMM). [67]

(1) Rigid body *a* is fixed. Segment *b* is inserted into segment *c* during assembly. Segments *d* allow flexibility.

(2) Assembled and stable configuration of mechanism. Rigid body *e* is attached to a mechanism during assembly.

(3) Deformed and stable configuration of mechanism.

M-34	COPMM Bistable Switch	KM/KN MM/SBB



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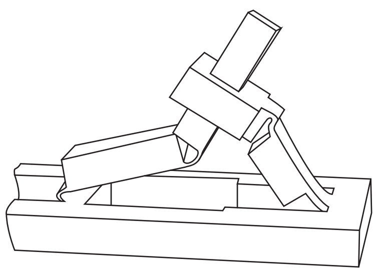
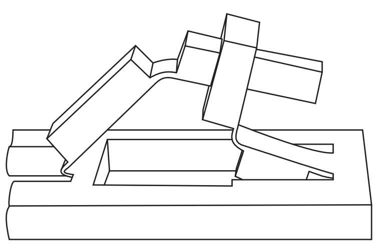


(2)



(3)

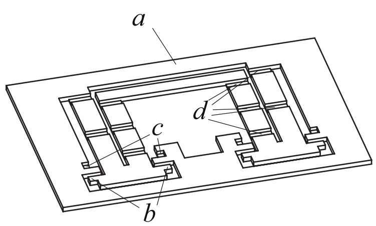
This is a compliant ortho-planar metamorphic mechanism (COPMM) that is a bistable switch. It is based on a fully compliant switch, and was redesigned by using the COPMM technique. This switch can be manufactured in a single plane and is assembled out of the plane for operation. [67]

(1) Rigid body *a* is fixed. Segment *b* is inserted into segment *c* during assembly. Segments *d* allow flexibility.

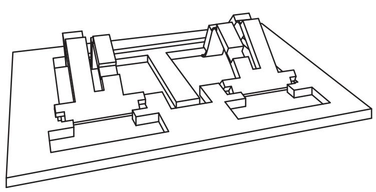
(2) Assembled and stable configuration of mechanism.

(3) Deformed and stable configuration of mechanism.

M-35	**Bistable COPMM**	KM/KN MM/SBB



(1)



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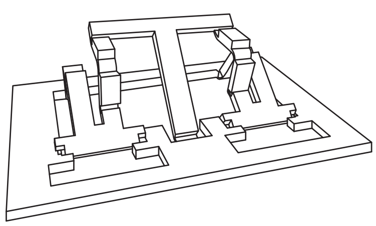


(3)

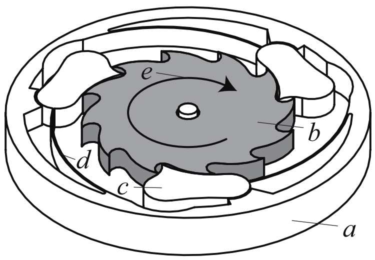
This is a fully compliant ortho-planar metamorphic mechanism (COPMM) that is bistable. It is based upon a closed loop 6-bar to a bistable non-grashoffian 4-bar.

(1) Rigid body *a* is fixed. Segment *b* is inserted into segment *c* during assembly. Segments *d* allow flexibility.

(2) Assembled and stable configuration of mechanism.

(3) Deformed and stable configuration of mechanism.

M-36	**Overrunning Ratchet Clutch**	KM RC



(1)

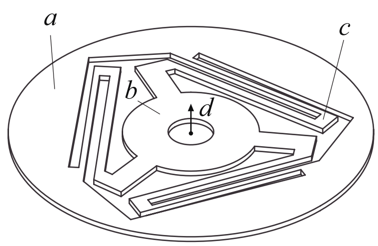
This mechanism is an over-running ratchet and pawl clutch with centrifugal throw-out. An important factor in the design is the use of passive joint elements that allow rotation of the pawls. [68]

(1) Rigid body *a* is fixed. Rigid-body *b* rotates in the *e* direction. Rigid-bodies *c*, the pawls, prevents rotation in the opposite direction. The pawls are able to deflect by using the flexible segments *d* and resist motion by using a passive element. The extra mass on the pawls, *c*, allows the centrifugal throw-out.

A.2.3 Kinetics

Energy Storage

M-37	**Ortho–Planar Spring**	KN ES



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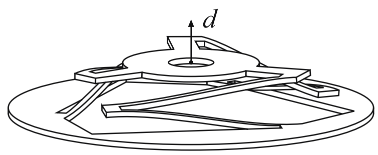

(2)

This mechanism is an ortho-planar spring that operates by raising and lowing its platform to the base. The benefit of this mechanism is it achieves this motion without rotation, which eliminating problems of rotational sliding against adjoining surfaces and has less sensitive variation in assemblies. [15, 69]

(1) Rigid body *a* is fixed. Rigid body *b* is the platform, which translates in the *d* direction through the flexible switch backs, *c*.

(2) Deformed configuration of translation in the *d* direction.

See also

Name	*Reference Index*	*Categorization Index*
Parallel Guided	M-24	KM/KN PML/ES
Multiple Stage Platform	M-30	KM/KN SP/ES

<table>
<tr><td>M-38</td><td>Gripper Hook</td><td>KN
ESC</td></tr>
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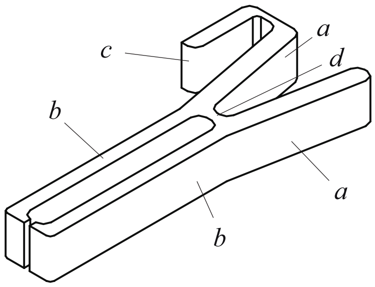

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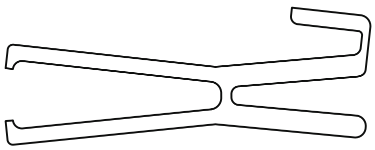

(2)

</td><td>

This mechanism uses a small length flexure as the pivot and the energy storage device. [70]

(1) When rigid-bodies *a* come together, rigid-bodies *b* separate by the flexible segment *d*. Rigid-body *c* is a hooking device.

(2) Deformed configuration.

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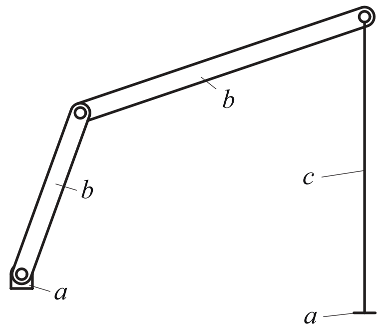
M-39	Unistable	KN SB



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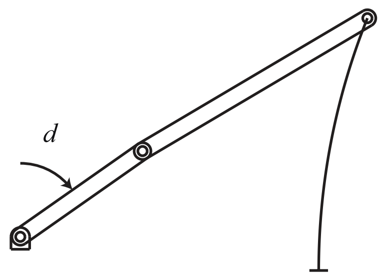


(2)

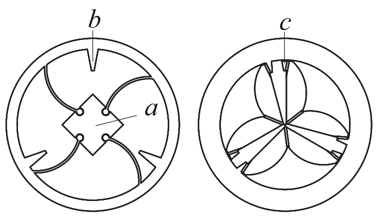
This mechanism has a cantilever beam that forces the mechanism into a single stable position when no input is applied.

(1) Rigid-bodies *a* are fixed. Rigid bodies *b* are binary links. Flexible segment *c* utilizes energy transfer to hold the mechanism in this current configuration when not input is applied.

(2) Deformed (unstable) configuration of input *d*.

Bistable

M-40	Bistable Button	KN SBB



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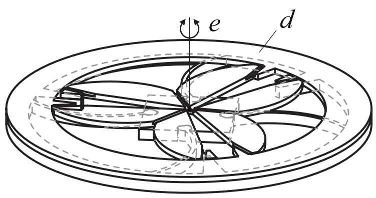

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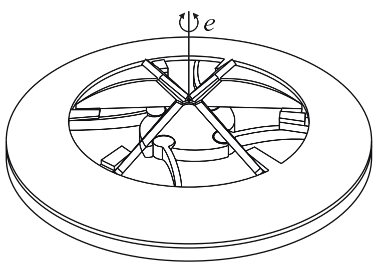


(3)

This is a multi-layer bistable mechanism. It operates from a planar configuration into a spherical configuration. [71]

(1) Rigid body *a* is fixed. Rigid-body segments *b* are attached to rigid segments, *c*.

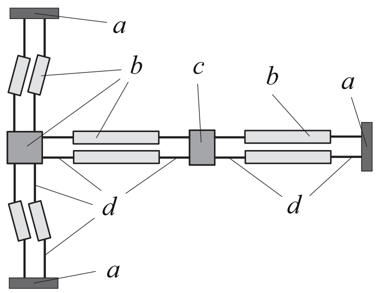
(2) Assembled configuration of mechanism. Rigid-body *d* rotates about the *e* axis.

(3) Deformed configuration of mechanism where rigid-body *d* rotates about the *e* axis.

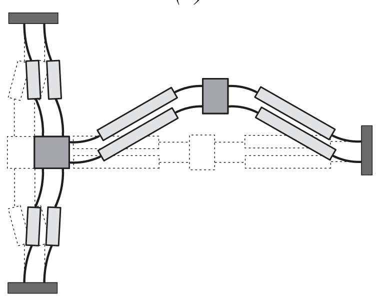
See also

Name	*Reference Index*	*Categorization Index*
Bistable Locking COPMM	M-33	KM/KN MM/SBB
COPMM Bistable Switch	M-34	KM/KN MM/SBB
Bistable COPMM	M-35	KM/KN MM/SBB

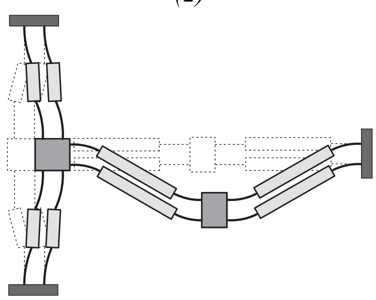
M-41	Dancing Tristable	KN SBM



(1)



(2)



(3)

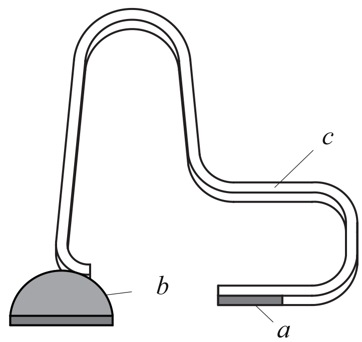
This mechanism employs two bistable mechanisms orthoginally to achieve three stable equilibrium positions. [72]

(1) Rigid bodies *a* are fixed. Rigid-bodies *b* and *c* are significantly rigid to flexible segments *d*. The shuttle *c* has three stable positions.

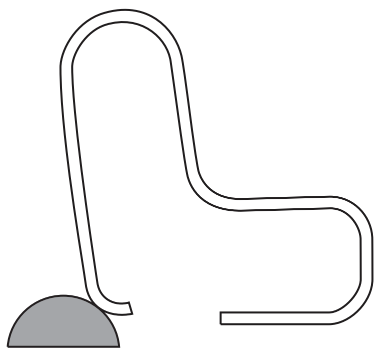
(2) Deformed and stable configuration.

(3) Deformed and stable configuration.

M-42	Electrical Connector	KN CF



(1)



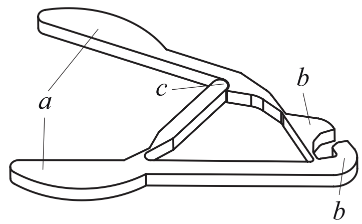
(2)

This mechanism is a constant-force eletric connector (CFEC). This mechanism uses a contact cam surface and geometry to maintain a constant optimal force so fretting or adhesive wear will less likely occur. [73]

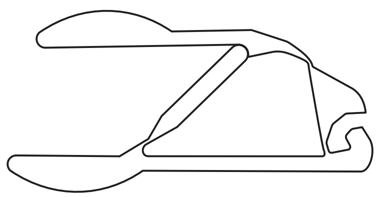
(1) Rigid body *a* is fixed. Rigid body *b* is the cam contact surface. The flexible segment *c* is the electrical connector.

(2) Deformed configuration.

M-43	**Pliers**	KN FA



(1)



(2)

This mechanism is a fully compliant plier that, in theory, will have an infinite mechanical advantage through part of its motion. [74]

(1) Rigid segments *a* are the input levers, and rigid-bodies *b* are the output levers (where the force is amplified). Point *c* is a passive element.

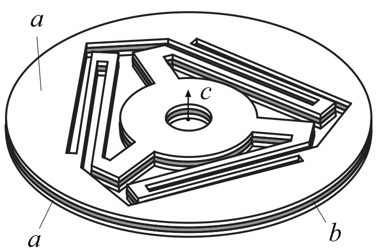
(2) Deformed configuration.

See also

Name	*Reference Index*	*Categorization Index*
Pantograph (LEM)	M-29	KM/KN SA/FA

Dampening

M-44	Dampening Ortho–Planar Spring	KN ES



(1)

This mechanism dampens an ortho-planar spring by utilizing a viscoelastic constrained-layer for dampening. This is to reduce the free response oscillations of the spring and suppress resonance responses. [75]

(1) Compliant mechanisms *a* are ortho-planar springs. They are separated by a viscous material, *b*, that allows dampening in an oscillating form in the *c* direction.

APPENDIX B. ANSYS BATCH FILE

The following ANSYS batch file can be used to perform finite element analysis to determine the deflection and stresses (see Figure 5.6) of the bicycle brake concept.

```
Finish
/Clear
/Begin
/Filname,BikeBrake,1
/Prep7
!Constants
n=1 !Titanium n=2 and E-Glass =1
t=2.5 !Titanium t=2.5 and E-Glass =0.508
b=8
a=b*n*t
i=b*n*(t)**3/12
!Material Constants
E=9.9e3. !Titanium E=144e3 and E-Glass =9.9e3
v=0.3
!Set Element Type
Et,1,Beam3
!Real Constants
R,1,a,i,t , , , ,
R,2,1e6,1e9,1, , , ,
!Material Properties
Mp,Ex,1,E
```

```
Mp,Prxy,1,v
!Key points
K,1, 40.464 , 46.891 , 0
K,2, 0 , 0 , 0
K,3, 23.358 , -15.650 , 0
K,4, 26.265 , -50 , 0
K,5, 44.164 , -17.322 , 0
K,6, -23.863 , 0.131 , 0
K,7, -29.786 , -5.252 , 0
K,8, -26.270 , -50 , 0
!Create lines
L,1,2
L,2,3
L,3,4
L,5,6
L,6,7
L,7,8
L,3,6
!Create Mesh
Esize,,500
Real,1
Lmesh,7
Esize,,5
Real,2
Lmesh,1
Lmesh,2
Lmesh,3
Lmesh,4
Lmesh,5
Lmesh,6
```

```
!Non-linear analysis
iterat=30
steps=3
ffirst=5
*Do,i,1,iterat,1
Finish
/Solu
Antype,0
Nlgeom,on
*Do,j,1,steps,1
Dk,2,Ux,0
Dk,2,Uy,0
Dk,7,Ux,0
Dk,7,Uy,0
Dk,5,Uy,8*j/steps
F,502,Fy,-ffirst
Lswrite,j
*Enddo
Lssolve,1,steps,1
Finish
/Post1
Set,last
*Get,reactfy,Node,517,Rf,fy
*Set,ffirst,reactfy
Finish
*EndDo
```