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COMPLIANT MECHANISM ROAD BICYCLE BRAKE: A RIGID-BODY REPLACEMENT CASE STUDY

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ABSTRACT

The design of high-performance bicycle brakes is complicated by the competing design objectives of increased performance and low weight. But this challenge also provides a good case study to demonstrate the design of compliant mechanisms to replace current rigid-link mechanisms. This paper briefly reviews current road brake designs, demonstrates the use of rigid-body replacement synthesis to design a compliant mechanism, and illustrates the combination of compliant mechanism design tools. The resulting concept was generated from the modified dual-pivot brake design and is a partially compliant mechanism where one pin has the dual role of a joint and a mounting pin. The pseudo-rigid-body model, finite element analysis, and optimization algorithms are used to generate design dimensions, and designs are considered for both titanium and E-glass flexures. The resulting design has the potential of reducing the part count and overall weight while maintaining a performance similar to the benchmark.

1 Introduction

A driving factor in the high-performance bicycle component industry is to increase device performance and decrease the overall weight. These competing objectives make it difficult to create new bake designs, and the current state-of-the-art designs have already been highly optimized over the years. It may be possible to address these challenges by introducing the advantages of compliant mechanisms to create novel designs that maintain the benchmark performance while reducing weight. This problem also makes a strong case study for rigid-body replacement synthesis to illustrate how the design approach can be used in an application and to take advantage of the benefits of compliant mechanisms.

Compliant mechanisms achieve motion or force transmission through the deflection of flexible members [1]. 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 that improved the performance of a mountain bicycle and BMX brake, a rear derailleur [2], and a clipless pedal [3]. This paper will incorporate compliant mechanisms to design a novel road bicycle brake.

2 Background

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

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 [4]. This advancement of compliant mechanisms has lead to a development of design

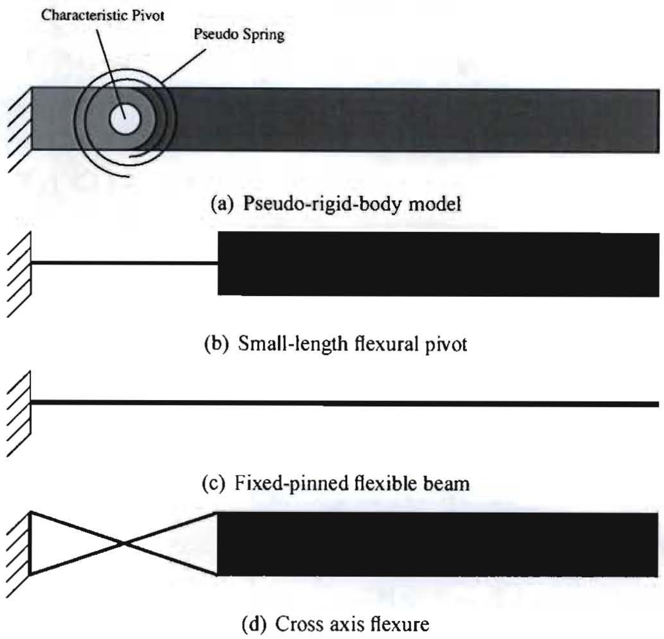


Figure 1. The (a) pseudo-rigid-body model concept for a (b) small-length flexural pivot, (c) fixed-pinned flexible beam, and (d) cross-axis flexure

methodologies [5–12], one being rigid-body replacement synthesis, and a development of a reference library for compliant mechanisms [13].

2.1.1 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 [14]. 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 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).

2.1.2 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 [13]. The classification scheme categorizes compliant designs by three approaches, with the primary approach being functionality. This approach will be used here 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. [15] 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.

2.2 Self-Centering Mechanisms

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. [16] 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.

3 Rigid-Body Brake Designs

The industry for road bicycle components is 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 when 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 work is to present a new linkage configuration that will inherently use less material, and remove the need for assembled components. An understanding of the existing brake designs and their advantages and disadvantages is requisite, for benchmarking 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 1.

3.1 Synthesis of Alternative Configurations

The modified single pivot and modified dual pivot rigid-body brake designs shown in Table 1 function with a higher kinematic pair (i.e. cam). These designs do not facilitate a greater number of compliant configuration counterparts to be formed by rigid-body replacement synthesis because most compliant element designs are established from 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. [17] 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 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 2.

4 Rigid-Body Replacement Synthesis

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

4.1 Compliant Counterparts

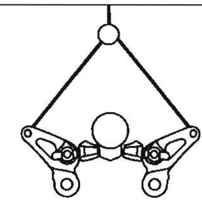
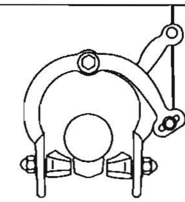
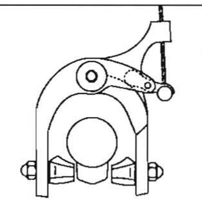
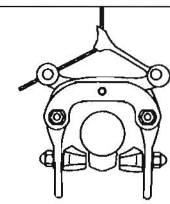
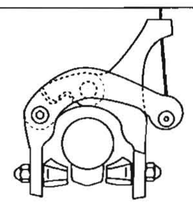
Rigid-body replacement synthesis treats the rigid-body linkage configuration as a pseudo-rigid-body model. That is, the rigid links and pin joints can be replaced with a compliant element that has similar motion. This section describes what type of compliant elements would be beneficial in replacing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

4.1.1 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 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 [18]. 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 [15] that match these criteria, some possible candidates are the cross-axis flexure and the tubular cross-axis flexure [19]. The resulting compliant replacement possibilities can be found in Table 2.

4.1.2 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 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 [15] 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 2.

Table 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	

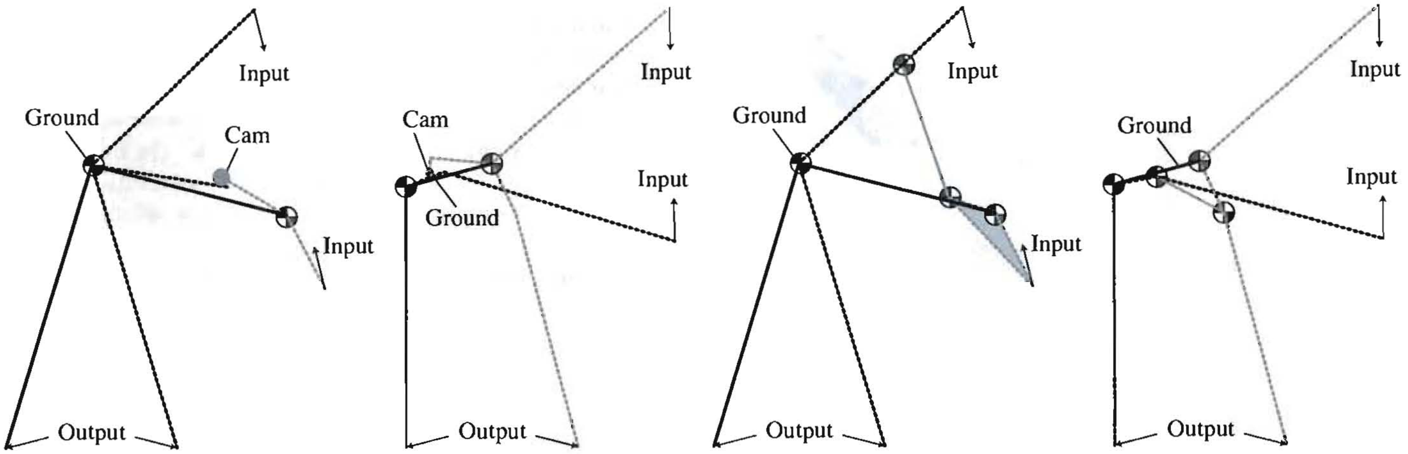
A challenge with this rigid-body design, according to the postulates for grasping mechanisms [16], the mechanism needs at least two degrees of freedom. The mechanism shown in Figure 2(d) is a four bar mechanism and has one degree of freedom. The second degree of freedom is accomplished through system compliance. For example, a compliant beam can achieve a second degree of freedom by entering into another mode of motion.

4.2 Selection

A preliminary finite element analysis was conducted on the design configurations found in Table 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. To accommodate this mounting pin, the design that was selected the linkage design where the ground link has rigid pin joints (including the mounting pin) and the coupler is a compliant fixed-fixed beam.

5 Compliant Brake Design

The resulting compliant design concept originated from the modified dual-pivot brake (see Table 1). This design was then



(a) Modified cam single-pivot schematic (b) Modified cam dual-pivot schematic (c) Modified linkage single-pivot schematic (d) Modified linkage dual-pivot schematic

Link 1 Link 3
Link 2 Link 4
Revolute

Figure 2. Configurations of ((a)-(b)) higher-order and ((c)-(d)) lower-order kinematic paris

Table 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

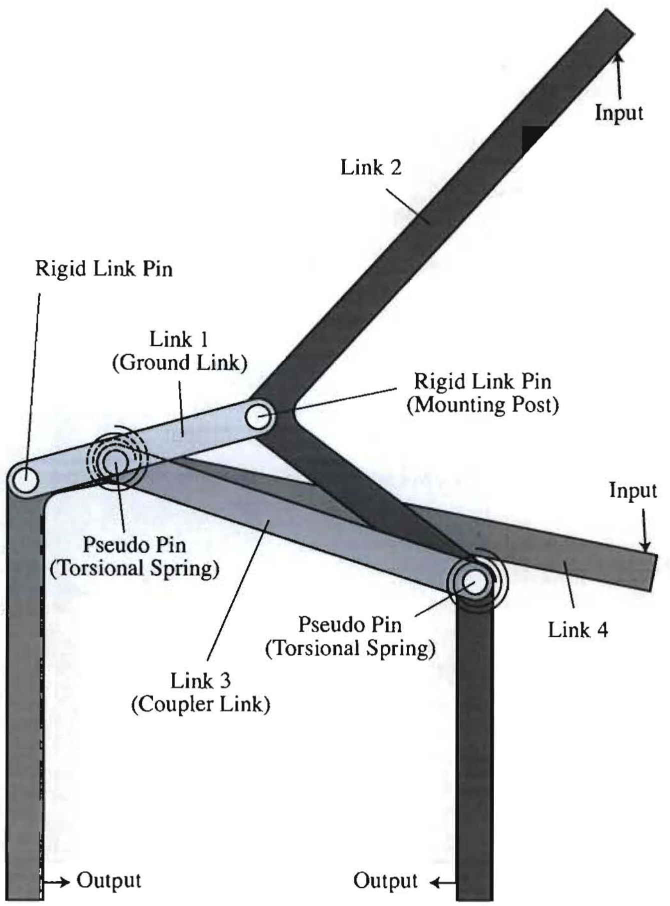


Figure 3. Pseudo rigid-body model of the compliant road bicycle brake concept

transformed from a higher-order kinematic pair to lower-order kinematic pairs (see Figure 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 mechanism 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 2). The pseudo-rigid-body model of this concept is shown in Figure 3.

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 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.

$$MA = \frac{(F_{out})_{average}}{(F_{in})_{average}} \tag{1}$$

By using the principle of virtual work [1] 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 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 3). These dimensions correspond to Figure 4.

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

$$\frac{(F_{out})_1}{(F_{out})_2} = -1 \tag{2}$$

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 was 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

$$V = \frac{1}{2}K(\theta - \theta_o)^2 \tag{3}$$

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) [1]. It is noted that the bicycle brake's flexible beam will not undergo a fixed-guided deflection, but that it will give an approximation

Table 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

for a closed form solution used by the optimization routine. The spring stiffness for the fixed-guided beam is

$$K = 2\gamma K_\theta \frac{EI}{l} \tag{4}$$

where γ and K_θ can be approximated as constants,

$$\gamma = 0.8517 \tag{5}$$

$$K_\theta = 2.65 \tag{6}$$

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

$$I = \frac{bh^3}{12} \tag{7}$$

and l is the length of the flexible segment.

$$l = \frac{L_3}{\gamma} \tag{8}$$

The cross section dimensions of the flexible beam are indicated in Table 4. Titanium and E-glass are evaluated for the flexible segment and their material properties are indicated in Table 5. The resulting optimized link lengths and angles are listed in Table 3, with these dimensions corresponding to those indicated in Figure 4. The number of flexures listed in Table 4 refers to the number of flexures, for the thickness listed, that are stacked together like leaf springs. This stacking of flexures allows an increased overall stiffness without increasing stress.

Table 4. Cross sectional geometry

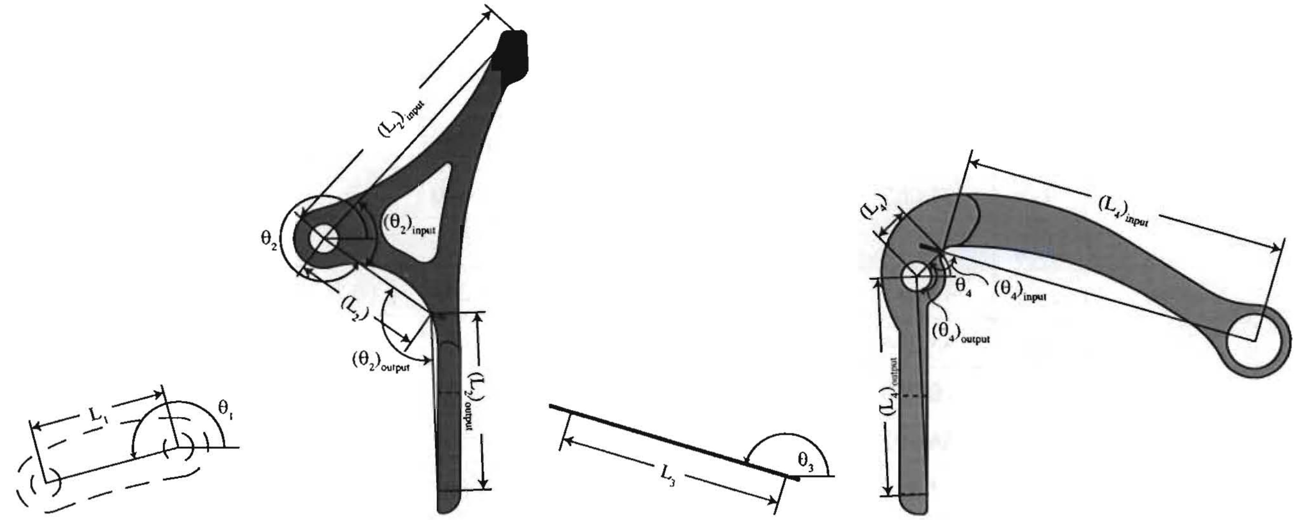
	Titanium	E-Glass
Number of flexures	10	1
Width (mm)	8	8
Thickness (mm)	0.508	2.5

5.2 Analysis

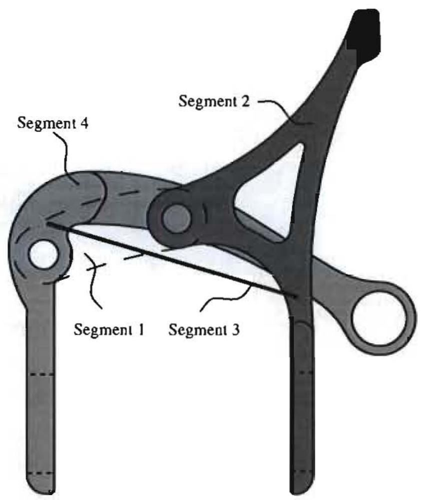
The deformation and stresses were analyzed for the optimized brake dimensions. The fatigue strength for the flexible segment was estimated as listed in Table 6. A commercial finite element analysis software (ANSYS) was used to compute the deformation and stress of the compliant mechanism. To simulate 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 by determining the reaction force from the displacement and applying the opposite direction force to the applied force. 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 7. The stress distribution and deflection are shown in Figure 6. These results predict that the bicycle brake operates within the desired deflection and prescribed allowable stress.

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



(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 4. Assembly and dimensions of bicycle brake

Table 6. Fatigue strength

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

is within 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 are feasible for this design.

One issue relating to this design is that the output links undergo an actuation rate ratio of 1.26 (see Table 7). However, it is noted the benchmark has a deflection rate ratio of 1.2. Thus, the compliant bicycle brake behaves similar to the benchmark brake.

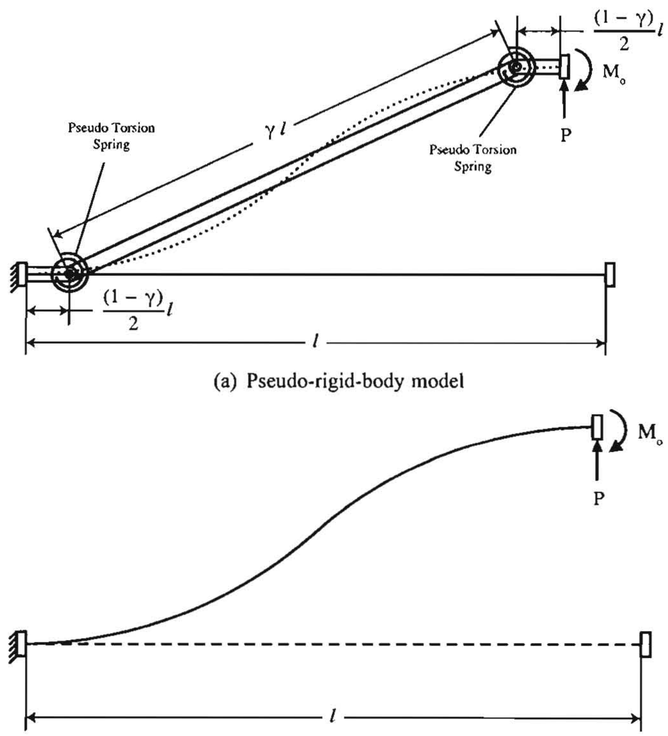
The potential for weight reduction comes from the removal of material by eliminating the cam and cam follower surface of the benchmark. Also, this concept removes the need of four accessory components, thus reducing assembly and further reducing weight. A preliminary demonstration prototype of this design is shown in Figure 7. An industrial design concept is illustrated in Figure 8.

Table 5. Material properties

	Titanium(Ti-5A1-2.5Sn annealed) [1]	E-Glass [2]
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

Table 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



(a) Pseudo-rigid-body model

(b) Fixed-guided beam

Figure 5. Fixed-guided flexible beam pseudo-rigid-body model

6 Conclusion

This cases study demonstrated how a designer implement type synthesis and rigid-body replacement synthesis in developing a design that maintains the benchmark performance, and also has the potential of lower weight and reduced assembly by the removal of accessory components. These are important design characteristics for high-performance bicycle component, where the industry is motivated to increase performance and decrease the overall weight of devices.

7 Acknowledgements

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

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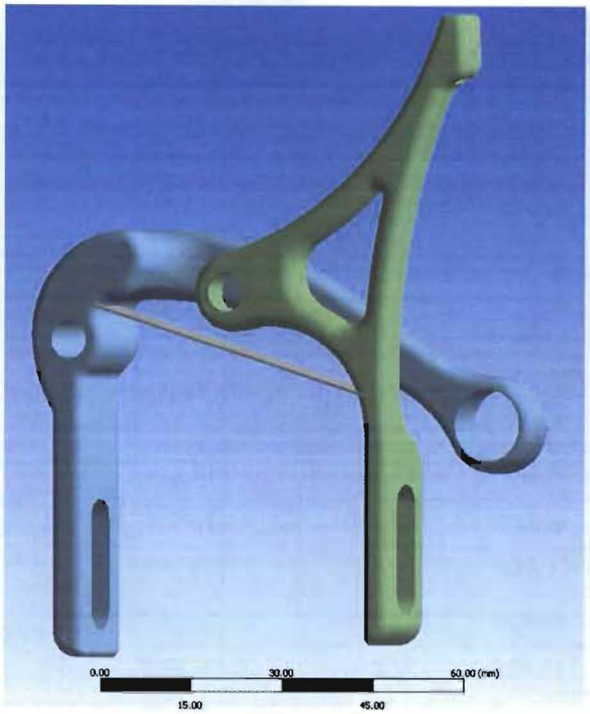
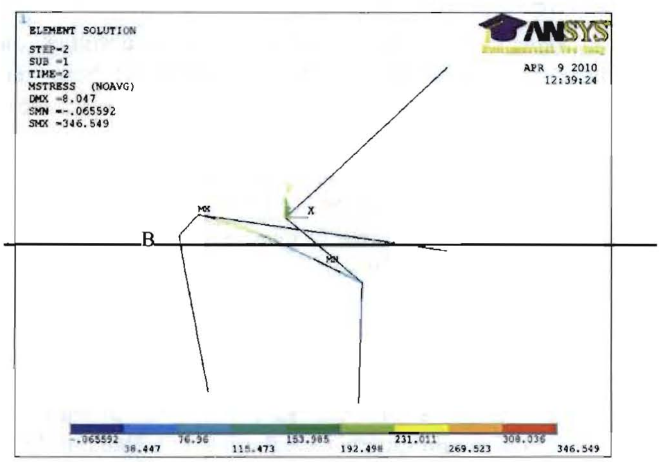
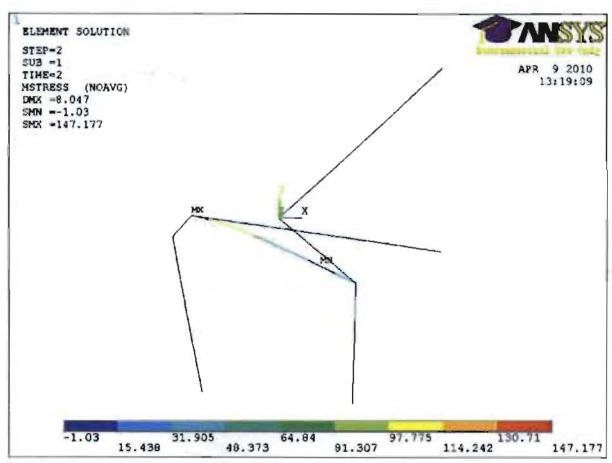


(a) 3D originial

(b) 3D deformation



(c) Titanium flexure operating stress



(d) E-Glass flexure combined operating stress

Figure 6. FEA results from ANSYS for the (c) titanium flexure and (d) the e-glass flexure

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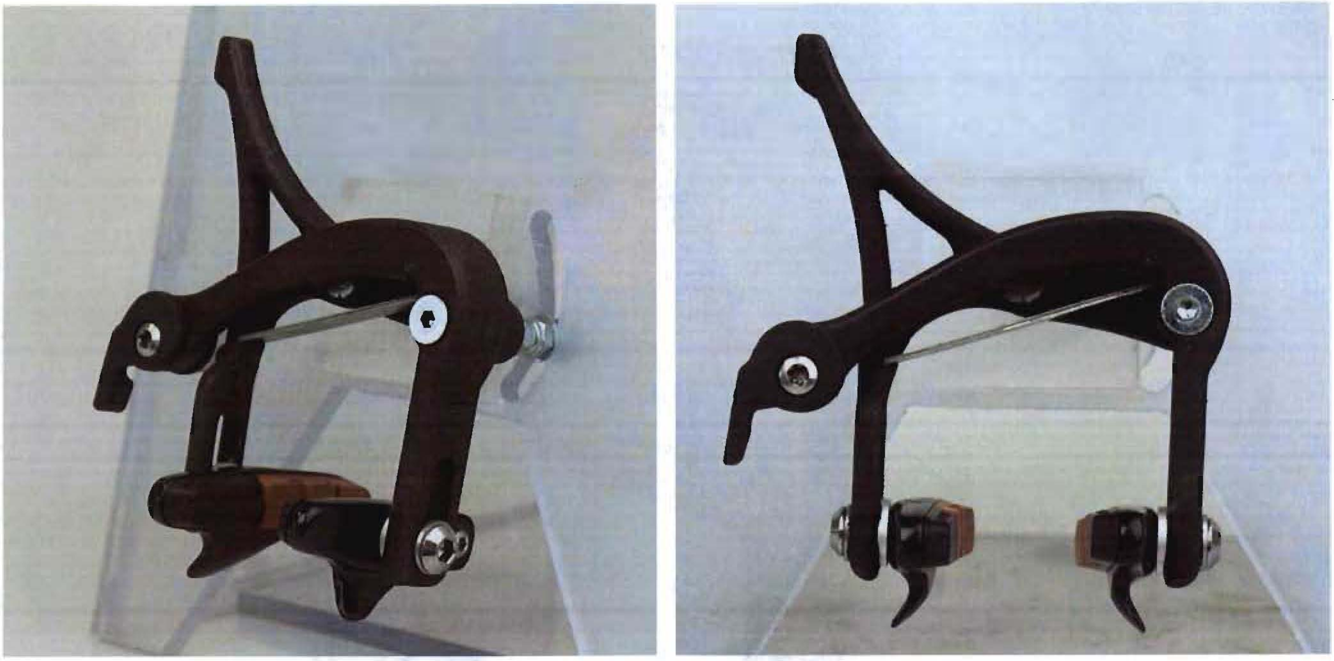

(a) Isometric view

(b) Front view

Figure 7. Prototype of the compliant road bicycle brake

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(a) Isometric view

(b) Front view

(c) Back view

Figure 8. Concept of the compliant road bicycle brake