ISSN 2229-5518

Compliant Mechanisms - An Overview

Poonam Tripathy

ABSTRACT: Compliant mechanisms provide specific benefits for usage that can address many of the problems with rigid mechanisms now in use. This paper gives a theoretical brief about compliant mechanisms and the vast concepts involved for scholars that need to grasp the concept before moving forth to mathematical modelling and analysis, and also help propel awareness and research on this engineering concept. This article examined the forms of optimizations and syntheses, as well as the benefits and drawbacks of compliant mechanisms. Various parts describe and apply the notion of pseudo-rigid bodies and pseudo-rigid body modelling. The author provided a concise overview of current material selection and modelling approaches, systematic synthesis methods that have been classified over time, as well as the issue of component fatigue life and dynamic analysis in compliant mechanisms. Compliant system manufacturing is also a crucial part in the development of such a product which has been briefed.

KEYWORDS: optimization, synthesis, compliant mechanisms, fatigue, FEM, non-linear, thermoplastics

1. INTRODUCTION

A simple observation of the natural entities around us distinguishes numerous examples of flexibility that seems to be the norm of nature, where rigidity is rare and compliance is abundant. Most systems designed by humans are kept rigid so the engineering design and calculation is simpler, as most variables that come with compliance are irrelevant. Hence flexibility in structures is more often than not avoided, but some examples of compliance persisted since early the stages of history-the bow and arrow- which was still modified to be rigid later with advancements in technology and physics. Traditional compliant mechanism designs usually have simple functionalities and do not require complex calculations (hence the persistence of those designs), but with improved numerical and simulation software, these complex compliant or flexible systems can be easily understood and designed with accurate numerical calculations. [4]

A standard mechanism is a mechanical structure with rigid part and joints that transforms input energy into output motion. Compliant mechanisms on the other hand, are single-piece flexible structures that transform input elastic deformation energy and displacement to an output motion transmission at another location. Some or all of its motion is achieved through the flexibility of its parts in designs specifically created for easy extension and contraction rather than conventional joints that rely on sliding and rolling interfaces. This resolves the limitations that come with handling complex joints and actuation such as friction, backlash, fatigue, weight savings, wear and losses and adds advantages of reduced individual parts, reduced maintenance requirements, increased precision, a greater favourable ratio between load capacity and mass, low sensitivity to contaminants and increased cleanliness of operation elimination of lubricants, ease of miniaturization, which allows the use of compliant mechanisms in small scale applications. The study of essential elasto-kinematic characteristics and movement behaviour only in certain loads or displacements is not trivial due to the unitary structure of compliant mechanisms and geometric non-linearity that occurs as a consequence of massive deformation. The material closure does not have to be interrupted, making most designs monolithic, therefore it also adds the benefit of absence of the assembly procedures with many of the designs even being entirely 3-D printed for use. This is possible mainly due to special designing techniques. Multiple rigid components, pin joints, and additional springs are substituted with a single mechanism with built-in flexible segments, Hence, saving space while also lowering the price of parts, materials, and assembly labour.[2]. Compliant mechanisms are especially well-suited for usage in form adaptive systems, which do not have a finite number of distinct variables like traditional gearboxes, but are characterised by dynamic surface changes due to an imprinting parameter.[3] They hold the possibility to revolutionize the fundamental perspective to achieving controlled motion.

2. ADVANTAGES OF COMPLIANT

MECHANISMS

A compliant mechanism may be evaluated for use in a certa in operation for a variety of reasons.
The following are a few of these benefits.

2.1. Part Count

Compliant mechanisms have the ability to drastically reduce the overall number of components needed to complete the task. Compliant gears have numerous intrinsic advantages due to their monolithic structure, including cheap cost, no backlash, ease of production, and scalability. Flexible components, rather than springs, pins, and conventional rigid hinges, can be used to minimise the part count. When compared to a rigid version of the same system, the number of components necessary for a compliant mechanism might be significantly fewer. Some designs may be made of injection-mouldable material and produced in a single piece, according to the specification.

2.2. Productions processes

As they adapt themselves to a variety of production techniques, compliant mechanisms can be straightforward to develop. Many compliant mechanisms may be made flat from plane sheets of material since their mobility is derived from flexible sections. Newer competitive goods must suit the market's rising expectations. They must be light, resource smart, long-lasting, and steady, with minimal audio emissions. At the same time, the items must be brought into the marketplace as fast as possible. Manufacturing compliant mechanisms can be done in a variety of ways, including injection moulding, machining, stamping, laser cutting, water jet cutting, 3D printing, and E DM.

2.3. Price

Compliant mechanisms are very cheap to construct as they have fewer components and simplified production methods. The decrease in part count can optimize manufacturing and minimize fabrication and assembly associated costs.

2.4. Precise motion

Traditional mechanisms often lose accuracy through to backlash and wear, whereas compliant mechanisms can reduce or eliminate this, allowing accurate movements. Physical pins and hinges sliding against each other produce mechanical wear, which inevitably rubs off or modifies the material, changing the structure and mobility of the rigid-body system.
Backlash is generated by interconnected component toleran ces, which in compliant mechanisms can be minimized or e radicated as there are fewer or even no interconnected parts . This concept is extensively been employed in instrument design. The adoption of compliant mechanisms can help minimize vibrations generated by the rotating and sliding joints of rigid-body systems.

2.5. Performance

Compliant mechanisms feature reduced moving joints, including rotating pins and sliding joints. As a consequence, friction and the requirement of lubricants is minimized. These really are advantageous qualities for applications in which the system is not easy to access, and for use in hostile conditions which may damage joints. This becomes particularly significant in space applications since lubrication tends to burn up in the absence of gravity.

2.6. Proportions

A further benefit of compliant mechanisms is their ease of miniaturisation; simple microstructures, actuators, and sensors are widely used, and several other micro - electromechanical systems (MEMS) show tremendous potential; the significant decrease in the overall part count and joints provided by compliant mechanisms is a huge asset in the manufacturing of micro structures.

2.7. Portability

Application of a compliant mechanism instead of a rigid-body equivalent can result in substantial weight savings which can help companies profit off the compliance measures, reducing weight and transportation costs for consumer items. In aviation and other sectors, this is a critical element.

2.8. Predictability

As compliant mechanisms depend on the displacement of flexible elements, energy is collected as strain in the flexible members (which can be released by transforming it), that is analogous to the potential energy inside a deformed spring, and it's dynamics can be incorporated into the design. A bow and arrow system is a fundamental illustration of this: as the archer draws the bow, potential energy is stored throughout it, which is subsequently converted to kinetic energy by the arrow. These energy-storage qualities may also be applied to design for specific force-deflection characteristics or to cause a system to gravitate toward specified locations.

3. CHALLENGES OF COMPLIANT

MECHANISMS

Although these obstacles may well be resolved, it is critical to understand the challenges and limits of compliant systems. This information is useful for evaluating which applications will gain the most from the use of compliant mechanisms.

3.1. Combination of complex systems

The most challenging aspect of studying and creating compliant systems is the relative complexity of doing so. The complexity of understanding and creating compliant systems may be the most significant obstacle as a deep understanding of system characterization and flexible element deformation is a necessary pre-requisite, along with a grasp of how they interplay in a dynamic network.

3.2. Non-linear equations

Generalized linear beam computations are no longer viable since numerous flexible components experience significant deflections. Large deformation generates geometric nonlinear effects, thus nonlinear algorithms must be employed to account for these. Because of such problems, numerous compliance systems have been built via trial-and-error methods throughout the earlier days. High deformations generate geometrical non - linearity, thus dynamic equations must be employed to accommodate most of these. Such approaches are only appropriate for extremely basic systems that execute very simple tasks, and they are frequently inefficient for several diverse uses. Although methods have been established to make compliant mechanism engineering easier, the constraints are still the same. Even with these advancements, compliant mechanism analysis and design is generally more complex than rigid-body mechanism design and analysis.

3.3. Energy storage

Flexible components store energy, which may be used to optimize mechanisms with springs, acquire specific force-deflection relations, and store the energy that is transmitted or converted by the system. Interestingly, energy storage in flexible members might be a detriment in particular applications.

3.4. Fatigue

For compliant systems, fatigue analysis is usually more important than that for rigid-body equivalents. Because compliant elements are frequently engaged periodically when a compliant system is operated, it is indeed vital to develop them with enough strain rate to accomplish their intended functions.

3.5. Limited motion

The resilience of the bending parts also limits the motion caused by compliant linkage deformation. A compliant coupling, obviously, cannot ensure a steady rotating motion like a pin junction could.

4. PSEUDO-RIGID BODY MODELS

Pseudo rigid body modelling are designs utilising kinematic models, that substitute flexible parts with rigid body connections. As the behaviour of deformable components is difficult to incorporate into a mechanism's kinematic analysis, they are transformed to rigid body linkages for simplicity of analysis. This makes the equations easier to understand. Hence, to ease the study and design of compliant mechanisms, the pseudo-rigid-body paradigm adopted. It is used to bridge the gap between compliant mechanism and rigid-body mechanism theories by offering a precise approach of modelling nonlinear displacement of flexible beams.

Pseudo-rigid-body modelling techniques for discrete flexible segments simplify calculating the deflections of bigger components, allowing their use in simulation of complex systems with flexible parts which is very helpful in simplifying the analysis and synthesis of compliant mechanisms. They compute the deformation of compliant mechanisms with spring components using well-known rigid-body kinematics concepts to simulate the material's elastic characteristics, the large body of knowledge available in the field of rigid body mechanics reinforce the concept and help unify compliant and rigid body theories, although care needs to be taken ensuring the accurate replication of the flexible element's behaviour. The pseudo-rigid-body model may be used in analysis to estimate kinematic motion, input requirements, and component stresses rapidly and effectively. It may also be used in prototyping a design and optimizing it for specific design objectives and subsequent refining using nonlinear finite element analysis methods, allowing systems to perform complex tasks. PRB models were customarily created for specific flexure el ement parameters and load-
dependent characteristics. Preliminary studies concentrated on certain sorts of flexible components.
Eventually, PRB models were developed for broader use. T hey were also customized for limiting positional synthesis a nd dynamic analysis as a result of the
approach's effectiveness.

The PRB modelling technique enables the application of well-known rigid-body analysis methods in the study of compliant systems (Salamon, 1989). Burns (1964) and Burns and Crossley (1968) modelled flexible connectors as a stiff connection five-sixths the span of the flexible section. Howell and Midha (1993) investigated compliant systems with short flexural pivots. As this flexible components' dimensions are relatively small compared to the stiff portions' dimensions, the torsional pivots are represented as kinematic links at the flexible section's midpoint. Torsional springs are utilised to indicate the stiffness of the component. As the relative length of the flexible component rises, the precision of this approach diminishes, and then a new approach is necessary for compliant mechanisms with longer torsional pivots. Closed-form elliptic-integral formulations were utilised by Howell and Midha (1992) to create deformation estimations for an originally straight, flexible component with linear material properties.

One of the most notable advantages of the PRB method is the reduction of differential beam equations to algebraic kinematic equations, which greatly lowers computing times. As a result, it has been developed and applied to a wide range of applications in compliant mechanisms and automation. By translating the differential equations involved with beam theory into algebraic equations, the PRB models decrease the computing demands involved with the study of compliant members. This implies that using them in design optimization will substantially minimize the time involved in development, particularly given the repeated assessments of the objective function required by most quantitative minimization techniques. Their versatility in application will allow designers to create systems with numerous materials or different length ranges while maintaining the same underlying structure.
As the data points in PRB models may reflect varied deformation properties, they can be effectively utilised for general design of compliant mechanisms with numerous forms of compliant parts. To do so, meanwhile, a structure for deriving and evaluating statics and kinematics equations must be developed, which should also ideally be largely automated for usage with computing. It has also been proven that PRB models offer rapid analytical periods which again will aid iterative development techniques. Curved beams are rarely used in compliant mechanism development despite their potential benefits. They can provide rigidity between their terminals in addition to a distinct geometrical form, whilst straight beams tend to behave rather like restrictions, indeed for the construction and operation of compliant mechanisms, integrating the pace and adaptability of PRB designs with the use of curved and as well as straight beams might be really useful.

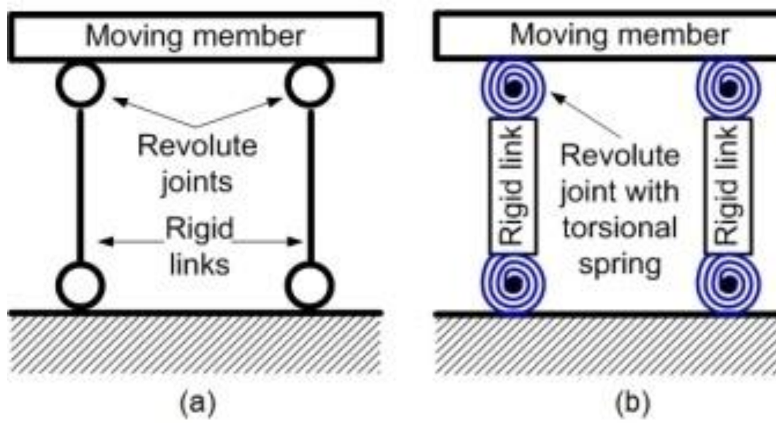

Figure 1. PRBM of a four-bar linkage mechanism to a partially-compliant **mechanism (linear spring) [25]**

The PRBM is a simple and quick way to analyse the performance of a compliant mechanism and in contrast to rigid-body systems, compliant flexure structures transmit mobility, energy, and even force based on the deflection characteristic of their own configuration. In a PRBM, pin joints known as characteristic pivots that connect directly to rigid-links. To simulate the flexure resistance of the beam, a torsional spring is installed at the usual pivot. PRB designs have largely subsequently been presented for fixed-guided beams, pinned-pinned curved segments, cross-strip pivots, and straight beams under deformation. The link between the loads operating on compliant mechanisms and then the distortion of the different components is closely interwoven with their own overall functioning and efficiency Matrix algebra and screw theory are frequently used to analyse deformations that are restricted to instantaneous or quite incredibly small quantities though when substantial distortion assessment is required however, beam theory, Finite Element Analysis (FEA), and as well as PRBM is utilised.
Dynamic analysis of PRBM structures will be briefed in further sections.

5. TYPES OF COMPLIANT SEGMENTS **AND EQUIVALENT** PRBMS

Compliant segments as well as rigid linkages and joints can be found in compliant mechanisms. Compliant mechanisms are classed as completely compliant or partially compliant depending on the type of links and joints in the mechanism.

Fully compliant mechanisms get all of their movements from deformations of compliant elements, whereas partially compliant mechanisms have one or more kinematic pairings as well as compliant segments (Howell, 2001). Fixed-pinned (Howell and Midha, 1995; Howell, 2001) compliant segments, fixed-guided compliant segments, and small-length flexural pivots are all examples of compliant segments.

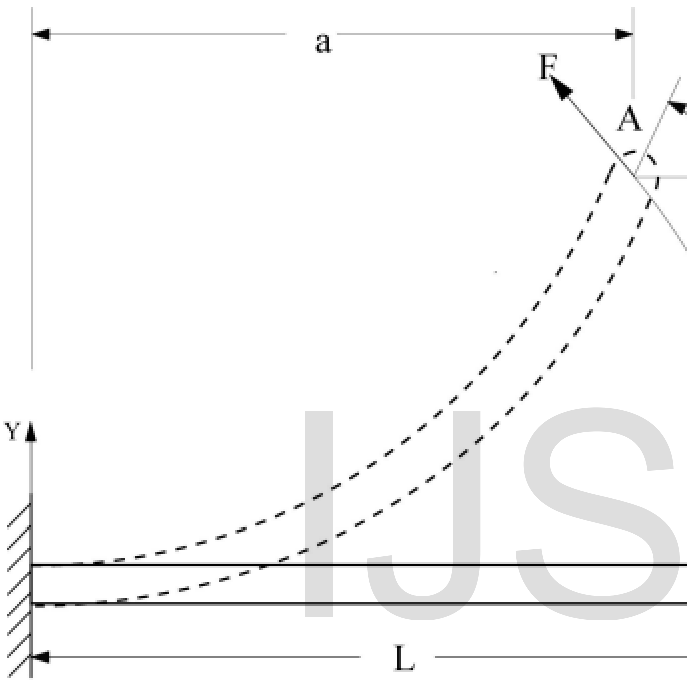

Figure 2. A Compliant Cantilever Beam with Large-Deflection [8]

5.1. Fixed-pinned compliant segment

Consider Figure 2, which depicts a flexible cantilever beam of length L with a constant cross-sectional area and linear material characteristics. The cantilever beam's free end follows a nearly circular route with a torque at the free end, according to large deflection elliptic integrals (Howell, 2001). The force-deflection characteristics of the beam are described by the torsional spring at the typical pivot, and this displacement route is simulated by two rigid linkages coupled at the characteristic pivot. Figure 3 shows the corresponding pseudo-rigid-body design for the fixed-pinned beam with force at the open edge.

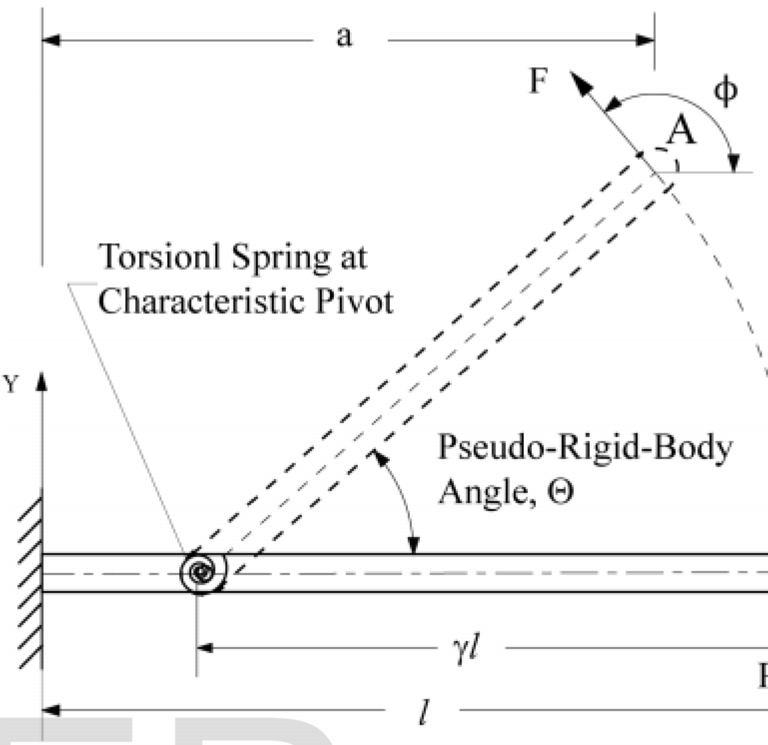

Figure 3. A Pseudo-Rigid-Body Model of Compliant Cantilever Beam with Large Deflection [8]

5.2. Fixed-guided compliant segment

Consider the loading conditions on a cantilever flexible beam as depicted in Figure 5. The resulting moment M6 must be present at the free end with the force P in order to retain a consistent beam-end angle. At its centerline, when the curvature reaches zero, the distorted form of the beam is anti-symmetric. As per the Euler-Bernoulli theory, the moment reaches zero at the mid-length. When only half of the beam is considered, it will have force P at one end and the same pseudo-rigid-body model as the fixed-pinned section (Howell, 2001). By merging the two antisymmetric one-half beam models as illustrated in Figure 6, the pseudo-rigid-body model for the whole beam may be produced; the PRBM is made up of three rigid links that are connected at two distinctive pivots by two torsional springs, one at each pivot.

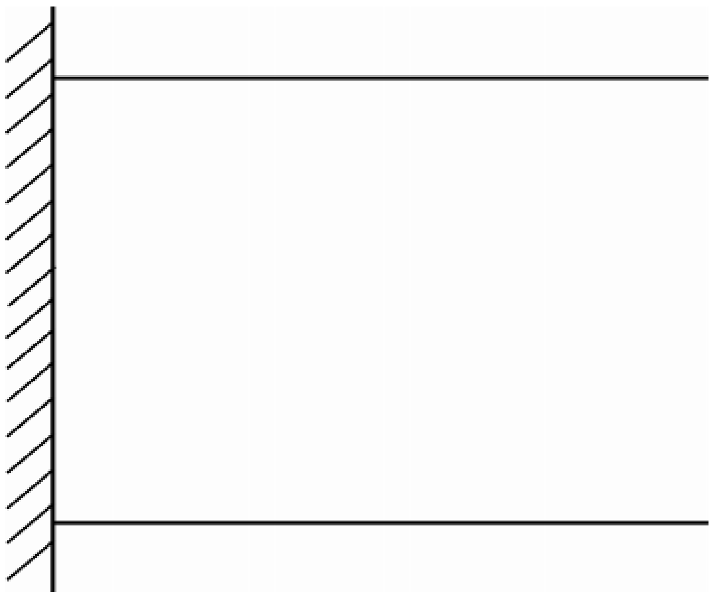

Figure 4. A Fully Compliant Mechanism (Howell**, 2001) [8]**

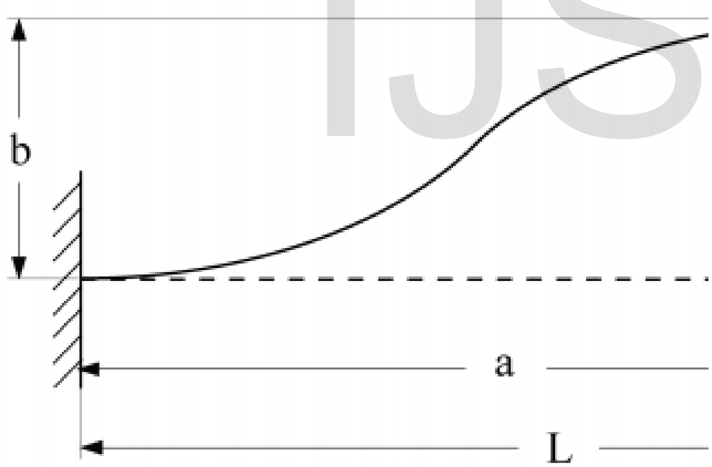

Figure 5. A Fixed-Guided Compliant Beam with Constant Beam-End Angle **[8]**

5.3. Small-length flexural pivot

Take a look at Figure 7, a cantilever beam, A short flexible section and a lengthy stiff portion make up the beam. Small length flexural pivots are made up of smaller flexible segments that are much shorter and more flexible than larger stiff segments. Longer segments are usually 10 times longer than shorter segments. (Howell, 2001). The motion of the system may be represented as two stiff links linked at a pin joint termed a typical pivot, as shown in Figure 8. As the deformation occurs at the compliant segment and is considerably less than the rigid segment's length, the typical pivot may be considered to be at the compliant segment's center (Howell, 2001). To evaluate the enormous deflections of the compliant mechanisms, Howell and Midha created the PRBM. A compliant beam is represented by a torsional spring at the pivot and is described as two rigid elements linked by a pin-joint or characteristic pivot. To integrate the compliant mechanism theory with the current idea of rigid body systems, PRBMs were created for a number of common compliant segment types, including Small-Length Flexural Pivot (SLFP), fixed-free, fixed-guided, and fixed-fixed segments. One of the main tasks in modelling a compliant beam's PRBM is to establish a typical pivot throughout its length, and recommendations and formulae are supplied for each conventional compliant segment type. In recent years, there has been an increased interest in testing the basic assumptions used in modelling the PRBM of an SLFP. Composite compliant beams include two segments: a flexible or compliant section and a rigid section. Small-length flexural pivots are present in these beams. Midha and Her were the ones who first proposed a basic approach for analysing compliant systems using rigid-body equivalent structures and torsional springs.

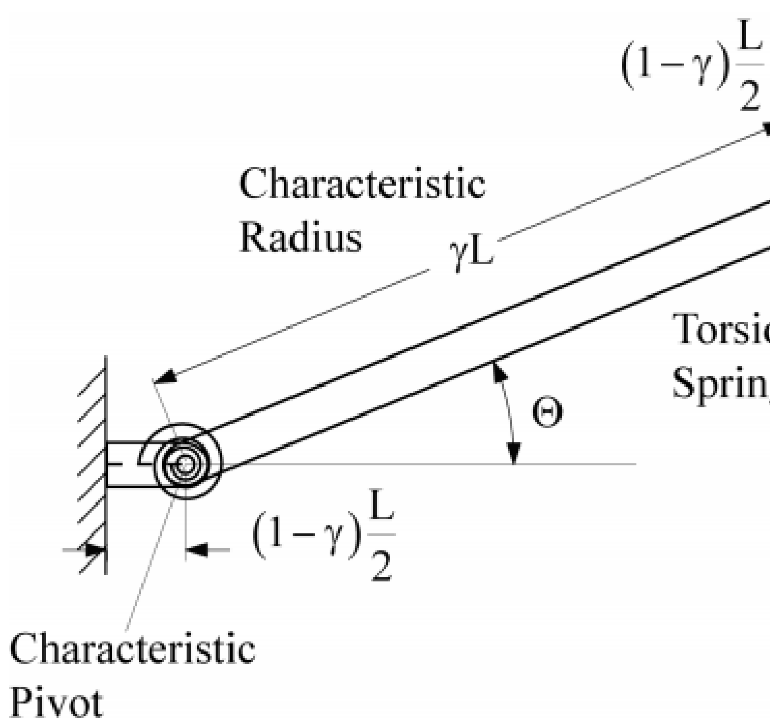

Figure 6. A Pseudo-Rigid-Body Model of Fixed-Guided

Compliant Beam with Constant Beam-End Angle [8]

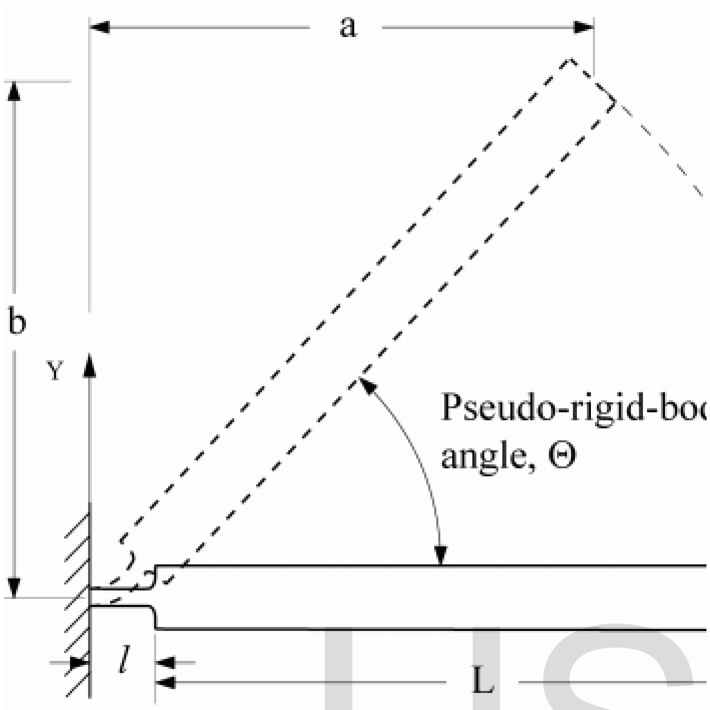

Figure 7. A Small-Length Flexural Pivot [8]

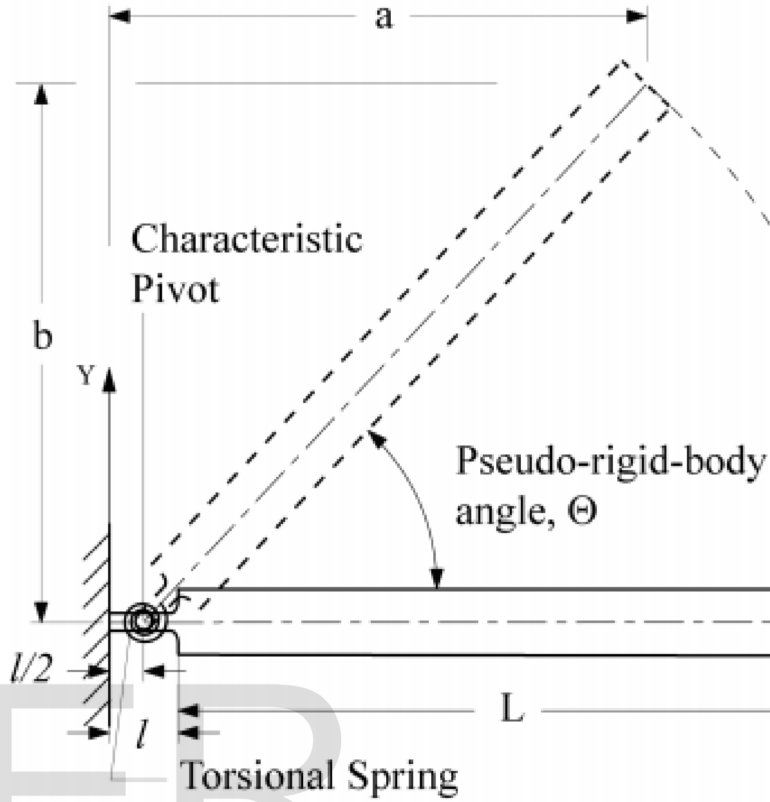

Figure 8. A Pseudo-Rigid-Body Model of a Small-Length Flexural Pivot **[8]**

6. TYPES OF **OPTIMIZATIONS**

The structure of the objective function, constraints (e.g., linear, nonlinear, convex), the nature of the variables (e.g. small, big), the regularity of the functions (e.g. differentiable, nondifferentiable), and etcetera may all be used to classify optimization issues (Nocedal and Wright, 1999). The constraints on the variables are one of the most significant classifications of optimization problems, with unconstrained optimization having no restrictions and constrained optimization having variables that are restricted in some way.

6.1. Unconstrained **Optimization**

The variables in this form of optimization are not restricted. The objective function, which is dependent on real variables and has no constraints, is minimised in this optimization. For the unconstrained optimization of smooth functions, there are a variety of methods accessible. They all need the user to specify a beginning point. It will be simple for a person with understanding of the problem's data set and applications set to provide a good point to start. It can sometimes be permissible to ignore the restrictions upon the variables while dealing with issues involving inherent limitations since they don't impact or interfere with the solution (Nocedal and Wright, 1999).

To proceed from the initial position to the next design position, usually algorithms employ either of two strategies: line search or trust region techniques. Numerical Optimization Nocedal and Wright, 1999, gives additional details about these techniques and unconstrained optimization.

6.2. Constrained Optimization

There are certain restrictions on the variables in this optimization, such as design size or shape constraints, or profitability expenditure constraints, and so on. These constraints might be simple limits on the variables, linear inequality constraints, or complicated nonlinear connections amongst those variables.

7. STRONGLY COUPLED VS. WEAKLY COUPLED SYSTEM

As per the number of equations and unknowns included in the system by energy/torque considerations, the kinematic equations and energy/torque equations are resolved either as a strongly coupled or weakly coupled set of equations in the synthesis with compliance approach. The kinematic and energy/torque equations can be solved separately in a weakly coupled system, but they are solved concurrently in a strongly coupled system. Coupling the kinematic and energy/torque equations, on the other hand, increases the system's complexity because they are both nonlinear.

8. COMPLIANT MECHANISM SYNTHESIS

Compliant mechanism synthesis has received far less attention than rigid-body mechanism synthesis. (Burns, 1964; Burns and Crossley, 1968) pioneered the kinetostatic synthesis of a four-bar mechanism with a flexible coupler connection. Ashok Midha pioneered work on compliant mechanism design approaches in 1980. The pseudo-rigid-body model, proposed by Howell and Midha (1994), represented compliant segments as comparable rigid linkages and torsion springs, making compliant mechanism synthesis considerably easier using existing rigid-body mechanism concepts. Mettlach and Midha (1996) developed a graphical synthesis approach for designing compliant mechanisms with a larger number of precision locations, based on Burmester theory. Murphy et al. (1994) used type synthesis techniques to build a graph theory-based method for designing alternative topologies of compliant mechanisms. To synthesise the pseudo-rigid-body four-bar mechanism with energy/torque requirements, Annamalai (2003) and Midha et al. (2004) employed the pseudo-rigid-body model idea. For energy, torque, and force parameters, Kolachalam (2003) and Midha et al. (2011) developed compliant single strip mechanisms. For limit position synthesis of a complaining four-bar mechanism with energy requirements, Dado (2005) proposed a variable parametric pseudo-rigid-body framework. Ananthasuresh's papers became the first to discuss design principles for distributed compliance (1994). In this scenario, rather than rigid-body kinematics, continuum solid mechanics techniques are employed. By employing the homogenization approach and using the displacement of one point as the goal specification, AnanthaSuresh (1994) employed the structural optimization methodology to create compliant mechanisms with dispersed compliance. Frecker et al. devised another structural optimization technique that uses mechanism deformation energy as its goal metric (1997). The four-bar mechanism with compliant coupler was created by Saggere and Kota (2001), they usually require specified shape change as well as rigid-body movement to generate displacement. Lu and Kota (2003) designed 30 shape deforming compliance mechanisms using load-path technique and evolutionary algorithms. Using the polynomial homotopy method, Su and McCarthy (2007) created a bi-stable four-bar compliant mechanism. Tari et Su (2011) proposed a complicated solution structure for compliant four-bar mechanism kinetostatic production.

8.1. Rigid Body Replacement Synthesis

The pseudo-rigid-body model of a generic compliant mechanism is utilised as a rigid-body mechanism whose measurements may be determined so that the final mechanism performs the necessary job in rigid-body replacement synthesis. By substituting corresponding compliant members and joints for stiff links and moveable joints, the rigid body mechanism is converted into a compliant mechanism. As a result, organisations who have existing rigid-body mechanisms that they would like to convert into compliant ones may use rigid-body replacement as a design method that can be reversed without trouble. It is the process of constructing compliant mechanisms by simply applying rigid-body equations to a pseudo rigid-body model without regard for the mechanism's energy storage properties and only kinematic equations are considered for the synthesis. This method is especially beneficial whenever a compliant mechanism is being used for traditional rigid body activities such as functional creation, route construction, and so on, sans taking into account energy storage within that system.

Compliant mechanisms may be evaluated using standard rigid-body mechanism analysis, which is a very important step in the evaluation of their performance. Pseudo-rigid-body model is the idea that links rigid-body mechanism analysis to compliant-body mechanism analysis, using the same links and joints as a rigid-body mechanism, a compliant mechanism that can achieve the same objective is created using the pseudo-rigid-body paradigm. As this synthesis may generate a range of answers which are legitimate only for rigid-body mechanism not for the compliant mechanism owing to certain practical geometry constrains, including the small-length flexural pivots that cannot rotate effectively, hence the primary objective in this method is to determine and assess the pseudo rigid-body model for the compliant mechanism and the iterative method apt for this approach.

8.2. Synthesis with **compliance** (kinetostatic synthesis):

Howell and Midha, 1994; Howell, 2001 presented this method of synthesis of compliant mechanisms. This method provides multiplicity of solutions along with expediency and accuracy of the solutions (Annamalai, 2003) and in complement to rigid-body kinematic formulations, takes into account energy storage properties of the flexible sections, hence both kinematic equations and static force equations are considered for the synthesis**.** Loop-closure formulae for the pseudo-rigid body system and energy equations are included in the synthesis, the loop-closure equations describe the mechanism's kinematic motion, whereas the energy/torque equations reflect the mechanism's compliance. The energy storage properties of the system can often be observed as energy stored in the sample as a function of input, required input torque and required input and output torque at each precision position (Howell, 2001). A mechanism intended for path creation with energies, torques, or tensions defined at precise locations can be used as an example of synthesis with compliance. As with the rigid-body replacement synthesis, this technique employs kinematic equations to generate a suitable pseudo-rigid-body prototype for the compliant mechanism. The energy equations are used to estimate the structural characteristics of the compliant segments based on either the stresses or input parameters. The kinematic equations are formed by the pseudo-rigid-body links and their orientations in extremely precise positions, whereas the energy equations are formed by the spring constants and deformations of springs affixed at the characteristic pivots in the pseudo-rigid-body system, resulting in two different kinds of variables, kinematic variables and energy variables. The former consists of pseudo rigid body model’s angles and length of the link corresponding to specific positions and the latter consists of undistorted torsional spring orientations and spring constants. Energy is retained in the flexible parts of compliant mechanisms as strain and it can be accounted using suitably rigid torsional springs at characteristic pivots of the PRBM.

In the pseudo-rigid-body framework, this energy may be compensated for by employing torsional springs with suitable rigidity values at typical transitions. The quantity of uncertain variables, i.e. kinematic variables (includes link lengths, angles of the pseudo-rigid-body model links) and energy variables (consists of spring constants, related to the stiffness coefficient, and undeflected spring parameters, related to the initial PRBM), brought into the system varies with the number of springs in the system.

The variables that are shared by the kinematic and energy equations induce coupling. A system is considered to be weakly coupled if its kinematic equations may be calculated independently of its energy equations (Howell, 2001) and the uncertain variables are mostly in par or more than the number equations, but when more equations are added into the system than uncertain variables, the kinematic and energy/torque equations are solved concurrently, and thus the system is strongly coupled. The kinematic and energy/torque equations are nonlinear systems, and coupling them enhances the system's difficulty and computing requirements as assigning appropriate values to a vast set of variables is difficult. If the connection between the kinematic and energy/torque equations could be minimised and the equations computed individually, it would be favourable. The solutions are extremely susceptible to the values provided to the free choices and initial estimations of the variables due to the non-linearity of the equations employed in the system. Even a little shift in their values will not result in solutions or may result in

ISSN 2229-5518

illogical solutions. Defining the energy values at all precision locations has been seen to over-constrain the energy equations, resulting in negative spring stiffness values (Annamalai, 2003). A compliant mechanism's incorporation of several sorts of compliant segments generates a large number of different designs (Howell, 2001). For example, all 18 potential configurations for a pseudo-rigid-body four-bar mechanism with various sorts of compliant segments including small-length flexural pivots, full-length compliant segments are shown in Figure 9 (Midha et al., 1997; Annamalai, 2003), depending on the number of springs employed. Three compliant mechanism setups are conceivable utilising various compliant segment types for pseudo-rigid-body connections with four torsional springs. Similarly, there are five compliant mechanism variants with three springs, eight with two springs, and two with one spring, for a total of 18 variations. It is up to each individual to determine which configuration is best for a given function, taking into account development and production limitations. Now at four pin joints of any pseudo-rigid-body four-bar mechanism, a maximum of four torsional springs can be connected. As a result, the synthesis issue boils down to figuring out which spring constants provide the same amount of stored energy in the mechanism as required. Potential energies contained within each torsional spring is used to determine the overall stored energy in the compliant mechanism at a precise location (Howell 1993; Annamalai, 2003).

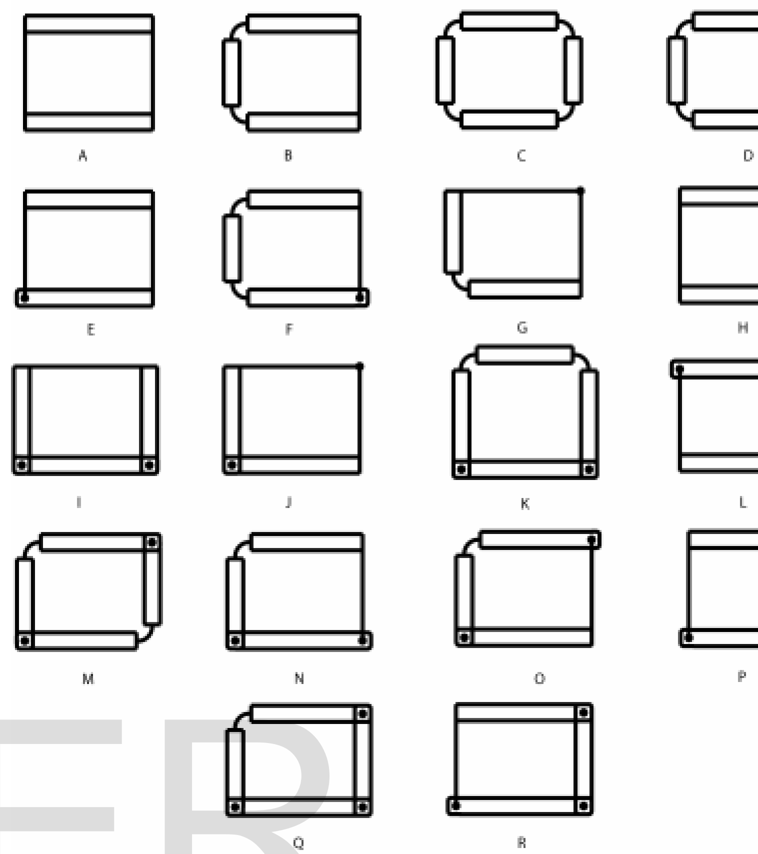

Figure 9. A Four-Bar Mechanism with Four Torsional Springs **at the Pivots [8]**

Another method for attempting to solve kinematic and energy equations has been proposed(Koli, Ashish B.), which solves kinematic and energy equations as weakly coupled systems only, making the method computationally simple and fast because the user only needs to designate appropriate estimates to a small number of variables (for kinematic variables only). Traditional nonlinear equation solving approaches, such as the Newton-Raphson algorithm, are utilised to solve the kinematic loop-closure equations, while constrained optimization is used to solve the energy equations. The user is not needed to make free choices while solving the energy equations since optimization is used, as a result, the result’s susceptibility to the values provided as free choices and initial estimations is decreased.

9. SYSTEMATIC SYNTHESIS OF COMPLIANT

MECHANISMS

Over the decades, there has been quite a development in the approaches for the systematic synthesis of compliant mechanisms, as observed, there are three approaches: the kinematics-based approach, the building blocks approach, and the topology optimization-based approach which includes the structural optimization-based approach, discretization topology optimization methods and some others like Homogenization Methods, continuum discretization, etcetera. [2][4]

9.1. Kinematics-**based** approach:

Compliant segments apply the rigid-body mechanism concepts in the kinematics-based approach, they are shown as a series of stiff links joined by pin joints, with torsional springs added to resist torsion. Depending on the kind of segment, the value of spring constants and where they'll be placed in the model are computed accordingly. The kinematics-based methodology is often ideally suited to mechanisms that experience massive, nonlinear deformations, but it necessitates the use of a known rigid-links mechanism to begin with. Two approaches are pretty similar to this method that determine their relationship to the pseudo-rigid body model; first is the Freedom and Constraints Topologies (FACT)- It makes use of Newtonian methodologies, specifically, each link is examined to achieve static equilibrium and only gives configurations (or topologies) for CMs, necessitating the employment of other techniques (e.g., dimensional synthesis, form optimization, and size optimization) to complete the design process. As a result, the complete mechanism's force system is established; second is the Rigid Body Replacement method, which substitutes a stiff body mechanism's corresponding system of pin joints, rigid links, and torsional springs for the flexible segments. Using the Rigid Body Replacement approach during the design phase, the Pseudo-Rigid-Body model is perhaps the most well-known approach.

9.2. Topology **optimization** approach:

Topology optimization is a method for determining the topology, shape, and size of a mechanism, by determining the best material distribution in a given design domain, beginning with a numerical topology optimization approach with a material domain to which external loads and supports are applied, and minimizing a given cost function while satisfying a set of constraints. It is not necessary to start with an established rigid link mechanism in this method. The compliance of the structure under the specified loads, subject to a volume restriction, is frequently the objective function. The design domain is represented via finite element metho d. Every finite element's material attribute (such as Young's modulus or cross-sectional dimensions) is regulated. Finite element analysis is closely linked to recently developed topology optimization techniques. An element will be deleted when its cross-sectional area reaches zero. As a result, certain members from the exhaustive set will be eliminated after the optimization method converges. The topology for the compliant mechanism is defined by the remaining portion. [2][3]

Approaches that employ link/beam components to discretize the candidate design domain are categorised as discretization topology optimization methods. There are generally two significant types of design domains in the discretization methods i.e., ground structures and continuum structures.
The ground structure or truss technique seems to be the most typical method in discretization topology optimization. In the design domain, it employs a comprehensive collection of truss or beam/frame components. In the design realm, it employs an extensive collection of truss or beam/frame components. Design variables define the particular cross-section. The ground structure can be created by connecting the nodal points of intersection. In topology optimization utilising the ground structure approach, the buckling constraint is significant since the members are frequently quite thin. When long and thin components are compressed, the influence of local buckling on structural instability can be significantly alleviated by incorporating the Euler buckling load constraint to extend the ground structure in the typical polymer construction. The efficiency of the output is improved by linearizing the buckling constraint term. It is generally accepted that the ideal structure is found in ground structures, thus the more rods there are, the more likely it is that the best solution may be found. The major issue in truss topology optimization is large-scale optimization, which is required when a high number of potential bars are utilised as design variables. Sokół (2011b, 2014) updated Gilbert and Tyas' (2003) adaptive ground structure method and expanded it to different load situations. Nodal locations may often not be optimum and reasonable, resulting in a ground structure geometry issue. Nodal points can be added to the final topology to reduce the number of bars and joints, but this will increase the computational cost. When optimising truss topology, conventional design domains are frequently rectangular or square (Sokó 2011a). When it comes to real-world challenges in engineering, random geometries are

more common. Each bar member should be penalised for joint length, according to Parkes (1975), so that the cost of producing the joints can be reduced. These methods have been extended by Pritchard, Gilbert, and Tyas (2005) to the optimization of huge three-dimensional truism topology problems, as well. As a result of Dobbs and Felton's (1969) work, the approach of alternating topology and geometry optimization has been widely used. Smith (1996, 1998) developed an interactive approach for generating ground structures in non-convex design regions using the finite element mesh technique and computer-aided design software and for unstructured environments, Zegard and Paulino (2014) employed the ground structure design and analysis technique but these techniques focus mostly on meshing irregular domains rather than how to organise the nodal locations reasonably, and meshing without taking external load vectors and displacement boundary conditions into account may lead to unsatisfactory optimization solutions. The method utilised to produce the ground structure is highly essential in truss topology optimization. According to Ge Gao et al (2016), the ground structure was generated by using nodal points created by solving the analogous static problem in the whole design area using homogenous isotropic material. A mathematical programming technique was employed by Achtziger (2006, 2007) in order to optimise truss geometry and topology concurrently, without the melting nodes effect. For the optimization of truss architecture, Wei et al. (2014) developed a stiffness spreading approach to move bar components independently. Variable densities and positions of components were utilised to create a continuum truss-like structure by Zhou (2011).

Topology optimization based on continuum discretization (e.g., bilinear quadrilateral element) is referred to as continuum topology optimization. Topology and form optimization are coupled with linear and nonlinear finite element analysis in continuum synthesis. Although this method is relatively new, it has been rapidly growing in the past decades. The homogenization method, the SIMP (solid isotropic material with penalization) method, the level set method, the ESO (evolutionary structural optimization) method, phase field methods, the MMC (moving morphable components) method, and genetic algorithms are examples of analytical techniques in continuum structural topology optimization. A process called homogenization was developed by Bendsoe et al in 1988, it involves splitting the project region into distinct micro-hole structural units to optimise the topology of continuum structures. [2][3]

9.3. Building-blocks approach

The building blocks method is built on the notion that a me chanism should be composed of several sub-mechanisms (basically building blocks). A range of flexible sub-mechanisms are first chosen for the design of CMs utilising this methodology. Their arrangement then optimised to ensure that the final CMs have the appropriate functionality.

10. MANUFACTURING TECHNIQUES

Compliant mechanisms are extremely complicated, requiring highly specialised and pricey production processes, with the best approach being the Wire Electro-Discharge. With very significant material losses and very lengthy and careful processing methods, it's possible to machine an object from a block of bulk material. Electro-erosive machining, chemical and electrochemical machining by abrasion, laser beam processes, electron beam, plasma, and waterjet are some of the unusual processing technologies that are successful when it comes to compliant mechanisms. As AM technologies, especially metallic powder bed techniques such as Selective Laser Melting, provide new opportunities, this paradigm is being questioned (SLM). Since they produce accurate and desirable outcomes in a short period of time, milling, 3D printing, and moulding have been employed as realisation methods for the project. Elastic coupling geometry as well as midway stiffness affect movement accuracy. All parts and stiff machinery must be precisely positioned, which requires heavy care. As of 2016, CSEM showed the possibility of high-performance compliant structures manufactured by additive manufacturing. They are experts in the computerised optimization of such mechanisms for AM and have developed a novel design concept for interlocking lattice flexures. even when deformed, the flexure components cross each other in this configuration, but they never contact. Due to the fact that flexure topologies may be developed from scratch or improved via additive manufacturing, this was only conceivable. To prepare thin films, configure thin films and create micro-structures, these micro-technologies require substrate processing. For example, mono and polycrystalline silicon; silicon nitride; nickel; copper; gold; epoxy resins; silicone and acrylic; polymethylmethacrylate; etc. are the most widely utilised materials. In addition to chemical vapour deposition and thermal oxidation techniques, thin films can also be created through spraying, ion implanting, thermal evaporation in vacuum, autocatalytic deposition, and plasma polymerization methods. Wet chemical erosion, dry etching, and lithography (photolithography, electronic lithography, nanolithography) or Roentgen lithography are some of the techniques used to create thin films. Anisotropy and selectivity-based micro-processing or the LIGA method can be used to create the micro-mechanisms. Compliant mechanisms are ideal for micro actuators and MEMS since they don't require any assembling procedure to operate. More emphasis should be paid to manufacturing-oriented topology optimization approaches for CM design. Topology-optimized CMs have a high rate of manufacturing rule breaches. Examples include internal holes or very thin components that cannot be manufactured by machining or injection moulding/casting. The manufacturing rules and solution methods must be taken into account while developing CMs utilising topology optimization.

11. FATIGUE AND **FAILURE** MODES AND PREVENTION

Fatigue is one of the most common causes of failure in mechanical constructions. Time-varying cyclic loads cause components to break at stress levels below the material's yield or absolute durability. Element fatigue breakdown is caused by the start and progression of a fracture until it renders unstable, at which point it progresses to abrupt failure. By segregating the fracture start and propagation phases, no technique could estimate the fatigue life using the damage value. The approaches for predicting crack commencement lives are mostly empirical and do not define the material damage. Because they are primarily curve fitting techniques, the stress- or strain-based procedures employed do not define the damage caused to the material. Because of the complicated nature of fatigue, which is impacted by numerous factors, statistical character of fatigue phenomena, and time-consuming fatigue testing, the total fatigue life, which is the sum of fracture initiation and crack propagation life, has become crucial to forecast. Despite the fact that several fatigue models have been created to address tiredness issues, their validity range is limited. Because of the limitations of this method, micromechanics models known as local methods based on continuum damage mechanics were developed (CDM). The local methods are based on the use of micromechanics fracture models in which stress/strain and damage at the crack tip are linked to the critical fracture parameters. These models are calibrated using material-specific parameters determined for a given material, and they may be considered to be free of geometry and loading mode, allowing them to be used to the evaluation of a component made of the same material and can be used to design devices that can withstand fluctuating stresses. Because fatigue failure is difficult to anticipate with any degree of accuracy, understanding how to predict and avoid fatigue failure is essential when building a system. Modelling the life of components stressed beyond their endurance threshold is at best a vague process, and estimating the fatigue life of mechanical and structural compliant systems exposed to arbitrarily differing stress cycle intensity (e.g., compliant automotive suspension and compliant aircraft structural components, etc.) is even more difficult, models like the continuum damage model are adopted for the modelling as the regular stress and strain life models fail.

The production and evolution of microdefects inside an originally flawless material is typically recognised as the damage process in polymers. The material remains the very same, however it's geometry changes its macroscopic characteristics. There are two stages to thermoplastic polymer fatigue failure. The material initially collects fatigue damages (during the beginning phase), which eventually leads to the creation of observable crazes. Craze development, which is both a localised yielding process and the initial stage of fracture formation in polymers, is widely thought to be one of the major sources of material degradation. The crazes continue to develop, produce cracks, and spread (i.e., they are in the propagation period) until they finally collapse. Crazes are caused by surface defects and abrasions, as well as interior cavities and impurities, and have a major impact on polymer distortion and overall mechanical behaviour. Kachanov was the first to present continuum damage mechanics, which he developed within the context of thermodynamics. It goes through the impact of microdefects on their following growth as well as stress and strain states in materials in a methodical manner. It's been used to study material fatigue and breakage. Decoupled numerical treatments of the dilatation and deviatoric portions of the distortion gradient are commonly used in finite element computations of virtually incompressible material designs.

The revised Basquin equation was utilised by Li et al. (1989) to calculate the life cycle till breakdown of a compliant rapid instrument servo. As per the equation, the fatigue life is determined by the akin reverse stress, fatigue concentrated stress factor, stress range, maximum stress, average stress, and tolerance threshold. For fatigue failure modelling of compliant mechanisms, Demirel et al. (2010) and Subaşi (2005) utilised the factor of safety stated in terms of variable stresses, tolerance threshold, mean stress element, and an intermittent stress element. The flexible part is anticipated to have an unlimited life if the stress situation is less than the two lines specified in the revised Goodman diagram for fatigue failure. Howell et al. developed a deterministic design technique for a bistable compliant slider-crank mechanism, with the goal of increasing the system's dependability in fatigue. Cannon et al. (2005) utilised the theoretical fatigue strength and the revised fatigue strength at periods represented as Marin adjustment factors to forecast the failure characteristic of a flexible end-effector for micro scribing. Damage evolution has been used by a multitude of scholars to forecast fatigue failures of mechanical configurations and elements. Building on Lemaitre's possibilities of dispersion, Jiang (1995) developed a defective development model for ductile metal strain fatigue. The fatigue life estimation equation and the aggregate fatigue damage criteria were then derived. Experiments were used for model validation. In this article, Shi et al. (2011) introduces a modified defect kinematics model to estimate the fatigue life of fibre strengthened polymer lamina and incorporated the singularity of stiffness matrix as the lamina failure criterion, ingeniously transforming the sophisticated nonuniform issue of polymer lamina fatigue into the analysis of single-variable isotropic damages for fibre and matrix. Akshantala et al. (2000) presented a fatigue-life prediction approach that combines a micromechanics-based fault evolution assessment with a semi-empirical fatigue breakdown criterion. The example studied was mixed ply composites under cyclic stress. For a variety of glass resin and carbon resin composite materials, the projected conclusions were evaluated by comparing to experimental results. Ping et al. (2003) presented a nonlinear continuum impact kinematics model to estimate the creep-fatigue life of a steam turbine rotor, which takes into consideration the impacts of complicated multiaxial stress and fatigue-creep coupling. The findings were contrasted to those of the linear aggregation theory, which had previously been used to estimate the life of steam turbine rotors. The nonlinear continuum damage kinematics model better represents the accumulation and evolution of damages than the linear aggregation theory, according to the analysis. Using dumb-bell test samples under uniaxial stress, Ali et al. (2010) studied the fatigue properties of rubber. The function of the strain range under cyclic loading was used to develop a continuum degradation model for predicting fatigue damage behaviour. Upadhyaya and Sridhara (2012) calculated the strain-controlled fatigue endurance of EN 19 steel and 6082-T6 aluminium alloy, taking into account both the starting and growth stages of the fracture.

Wang et al. and Mahmoud et al. utilised the theory of continuum damage mechanics to investigate fatigue degradation processes in elastomers, such as nucleation and early defect development.

Creep is a strain increase that happens in some materials when a stress boundary condition is applied repeatedly. Creep deformation severity is proportional to applied stress, temperature, and time. The plot of strain against time for one stress intensity, which remains constant during the test, can be used to demonstrate creep behaviour. The Voigt-Kelvin element is a mechanical analogue of creep deformation in a viscoelastic material that consists of a parallel connection of a spring and a dashpot. Both the Newtonian viscosity of the dashpot and the shear modulus of the Hookean spring influence the first response of the Voigt-Kelvin element. The spring and dashpot lengthen with time, representing creep deformation in a simple way. The dashpot provides preliminary resistance to creep experimental subject displacement, resulting in a strain reaction that is time-dependent. The spring's elongation is limited by the elongation of the dashpot when a load is applied to a Voigt-Kelvin component for the first time. A material with a very viscoelastic reactivity has a greater amplitude of the Newtonian viscosity of the dashpot within the Voigt-Kelvin component than a material with a nearly-elastic viscoelastic response.

When some materials are constrained in a deformed state and a strain boundary condition is applied continuously, stress relaxation ensues. The applied strain amplitude, temperature, and duration all have an impact on the severity of stress relaxation. Because compliant mechanisms are typically constructed with their operation depending on motion via deflection of their links, stress relaxation is more common than creep in them. The Maxwell element is a mechanical analogue of stress relaxation in a viscoelastic material that consists of a spring and a dashpot linked in series. The dashpot induces a time-dependent stress response in the stress release trial subject by providing an initial resistance to tension. The preliminary reaction of the Maxwell Element, like that of the Voigt-Kelvin Element for creep, is determined by the dashpot's Newtonian viscosity and the Hookean spring's shear modulus. For a material with a strongly viscoelastic response, the amplitude of the Newtonian viscosity of the dashpot within the Maxwell Element is greater than for a material with a nearly-elastic viscoelastic reaction. The initial deflection of a Maxwell Element is represented by the expansion of the spring when the load is applied. For a continuous strain, the spring retracting and dashpot expansion with time give a simple depiction of stress relaxation. As a result, the stress-strain relationship, also known as the stress relaxation modulus, reduces with time, as shown by a drop in stress while retaining the same level of strain. The stress oscillations in compliant systems are constant, which makes the stress-life model a good tool for analysing fatigue. The stress-life method of fatigue analysis is based on the connection between cyclic stress and the number of cycles required to reach failure, all stress excursions are assumed to be within the elastic limit in this method. The Wöhler fatigue curve represents the connection between stress and the number of loops to failure for a given material. The number of iterations to failure (N) vs stress amplitude is plotted on the Wöhler fatigue curve (S-N curve) (S). [118][119]

12. MODELLING METHODS

Models employing the finite element technique (FEM) are universally acknowledged to produce most high accuracy. The list of alternative deformation paths specified by various element types and the available degrees of freedom for the estimated model (1D/2D/3D) must be differentiated, so the more deformation directions and degrees of freedom a finite element has, the more precise a result for the entire model may be. As a consequence, the outcomes of 3D solid elements are the most realistic. Because of the short computation durations, beam elements, or 1D/2D FEM, can be a useful technique for designing compliant mechanisms. Particular 3D effects, such as lateral contraction, are, unfortunately, overlooked, despite the fact that they may be significant in certain aspect ratios. Even for ostensibly planar movements, large deflections, non-linear material characteristics, shear, lateral contraction, axial elongation, or 3-D phenomena such as stress concentration or anticlastic bending must be addressed, fortunately, 3-D mechanisms can be modelled and deformations and exo-plane loads can be predicted and calculated. The degree of precision is primarily determined by the discretization, element type, and degrees of freedom (1D/2D/3D). Higher discretization drives up the computation times. The rigid-body model (RB) has been praised for its ease of use in calculating compliant mechanisms. Because RBs are largely load-dependent, a second "design step" to evaluate a finished CM configuration may be required. Furthermore, the placement of the traditional hinges in the RB is classified by the characteristics of compliant member, so because deflection of the axis of rotation is not taken into account, the deformation is incorrect. It has also been demonstrated that a significant amount of deformation takes place inside the connections and framework of a compliant system. As there are limited tangible or usable instruments in the field for modelling and analysing processes in a precise way, time-consuming 3D FEM is frequently utilised.

There are a variety of approaches for flexure hinge and compliant mechanism evaluation and synthesis, in addition to FEM models. An analytical characterization of the distorted phase generally leads to a system of differential equations in the sense of a boundary value problem when significant deformations are addressed. As a result, approximation methods such as design equations generated from empirical data acquired via FEM or analytical models can be used. For significant deflections, closed form equations are seldom preferred. There are numerous advantages to adopting design equations, such as the explicit form, which allows for utilisation sans the use of specialized equipment and implementation without prior expertise. In addition, the findings are acquired quickly, making them appropriate for synthesis. Despite the benefits, most design equations are merely estimates for certain configurations and may only be viable within specified aspect ratio limits. Aside from that, they are rarely used in complicated systems.

For determining the deformation behaviour of compliant systems, other techniques rely on kinetostatic analysis. One, is the compliance matrix method, it may be used in which deformations at the end point of a flexure hinge are linked to displacements at one point of a rigid link and even the load on that link using this approach, it's beneficial as it simultaneously considers shear, bending and axial elongation, and it has simple formulation but it's validity is mostly limited to small deformations. A second example is the energy-based model like virtual work principles for kinetostatic analysis and is used to calculate the equilibrium in the system generally, where, the structural integrity of a compliant system is sustained by calculating the minimal potential energy condition of the system. Virtual work encompasses both the work performed on a system by a force during a virtual displacement and the work performed on a real displacement by a virtual force, although it doesn't predict the complete condition of deformation of the system, it does predict the forces, displacements and angles at defined joints, when applied to a continuum. The chained beam constraint model (CBCM) is another common approach that may be used to synthesise beam-based compliant mechanisms. When the CBCM has been produced, it can be used to optimise geometric variables for synthesis. Singular beams are discretized into a network of associated pieces in the CBCM paradigm. Various constraints, load situations, and even shear may be addressed, and a solution for a particular mechanism can be presented in an expressive manner, allowing for a quick computation. Geometric restrictions control the geometry of the beams. Thus, even originally curved beams may be computed by discretizing utilizing circular arc components or equivalent. Guimin Chen et. al consider the limitation of the CBCM approach to be that the precision of the results heavily depend on the discretization and is suitable for limited applications in software for arbitrary planar systems' analyses as the model is valid for a certain geometry but for use in other mechanisms it has to be derived anew. The non-linear beam theory (BT), sometimes referred to as the theory for large deformations of rod-like components, is a well-known analytical technique for estimating compliant systems or flexure hinges, it is versatile as arbitrary constraints, branching points and load cases by distributed forces can be assumed and shear and lateral contractions can be applied. Due to its continuity, the theory has already been demonstrated to be applicable to complicated compliant systems by subdividing it into sections and modelling of non-constant cross sections, curvatures and material properties is also possible. Each subsection can be treated with a different theory to suit it; hence, each section can be modelled according to their individual aspect ratios. When the model is regarded dimensionless, it is scalable and current methods can solve the BVP quickly and precisely, hence, it is ideal for parametric research and synthesis, as well as for use in graphical software applications with hypotheses in both 2D and 3D scenarios. Individual execution for distinct structures, on the other hand, can perhaps be complicated; in addition, mathematical formulae for cross-sections, curvatures, and the Young's modulus must be derived for every system. Solution tolerances, as well as beginning estimations, may have a significant impact on the efficiency and accuracy of the applied BVP-solver. Despite the vast range of theories and approaches present, the application of these theories and techniques to an engineer with minimal expertise designing compliant mechanisms may be restricted owing to a lack of simpler techniques.

The influences of shear, lateral contraction, and axial elongation can all be included in some of the theories described. Compliant mechanisms might contain a variety of centralized and dispersed compliant components with varied aspect ratios, these may necessitate taking into account various consequences. Some articled including that of Venkatasubramanian Kalpathy Venkiteswaran, united notions such as pure bending, shear, and lateral contraction in order to apply them separately or in combination to particular portions of a compliant mechanism utilising Beam Theory. In beam theory, concentrated and dispersed conformance, straight and curved segments, non-constant cross-sections, curvatures, and changing material characteristics are all valid in this approach and this technique is especially appropriate for execution in a software application because of the effectiveness of the analytical model and its flexibility to arbitrary plane mechanics. The deformed form of a compliant mechanism may be computed using beam theory in split seconds because to the processing power of contemporary numerical analysis tools. However, for some purposes, considering pure bending is inadequate since any geometry requirements can be used to diverge from Bernoulli's hypotheses.

12.1. Nonlinear **Static** Modelling Methods

When evaluating the functionality of compliant mechanisms, nonlinear FEM is often employed. To characterise the quasi-static behaviour of compliant mechanisms, many modelling approaches have come up, most notably the pseudo rigid body model, which integrates the beam theory with Castigliano's displacement theorem, and the linear finite element method (FEM), these methods are based on small displacement concept as greater input forces bring considerable differences between the linear simulation and the experimental data. As flexure hinge deformation increases with increased input forces, geometric nonlinearity must be considered. The cross-sectional variance of flexure hinges exacerbates the problem of building the nonlinear model for compliant mechanisms.
For a nonlinear analysis, Yang et al. proposed a co-rotational beam component and split a super-elastic parabolic flexure hinge into 20 components. Friedrich et al. suggested a three-node beam component and a five-element circumferential flexure hinge. The Euler–Bernoulli beams with Green stresses provide the basis for the aforementioned components. When a single flexure hinge is studied in the findings of nonlinear models in the paper by Xu et al., it was proved that the shear effect may be neglected [120]. Nonetheless, at the system level, the nonlinear behaviour of compliant mechanisms is influenced and the strains are objective measures rather than geometric measures. Modelling these tangent stiffness matrices need the derivatives of the form equations and their particular functions. Due to the fact that flexure hinges are not precise deformation form equations, they must be split into a number of components in order to achieve sufficient precision. The form functions must be updated as the degrees of freedom change. The variable thickness of irregular-shaped flexure hinges makes it more difficult to formulate accurate displacement shape functions, which increases computing work and risks divergence.

The quadrature beam element (QBE) was created lately to address the shortcomings of the mentioned beam elements, particularly in nonlinear analysis. The tangent stiffness matrix is calculated using the weak form quadrature element technique, which eliminates the requirement for form equations and relies solely on basic algebraic loading coefficient matrices, decreasing computational complexity. Because its strain measurements are geometric and objective, and the shear effect is taken into account, the geometrically precise beam theory, which is the foundation of the QBE, has gotten a lot of attention in nonlinear analysis. The formulation contains no translation or rotation update approximation, resulting in fewer load steps.

Guangbo hao et. al. summarized and explained the how all emerging static modelling methods and load displacement relations are related by a quadripartition classification:

Based on modelling dimension: It covers single-axis and multi-axis CMs, as well as linear and spatial motion CMs. Modelling becomes increasingly complex as the number of dimensions increases.

Based on modelling objects: Slender, short, multi-stable beams, serial/parallel/hybrid modules, and complicated systems are all modelled. For ease, the beam length is assumed to be greater than the width, ignoring the shearing force effects and cross-section deformation. Contrary to slender-beam modelling, short-beam modelling incorporates the shearing force and/or cross-section variation with distortion, which makes inference and solution more challenging. The beam modelling findings with coordinate transformations are used to model serial/parallel/hybrid modules, where each unit is made up of thin and/or short beams in a parallel, serial, or hybrid arrangement. The same may be said for the modelling of complicated systems, such as a decoupled multi-axis compliant parallel manipulator. In essence, desirable connections among input deformations and output deformations, or between input forces and output displacements, should be determined using actuation/control techniques like force control or displacement control when modelling a multi-input and multi-output CM.

Based on the form of solutions: Free-body-diagram (FBD) and energy methods are included. Decomposition into rigid phases and compliant units, followed by identification of distorted configurations, is the initial step in FBD's method to modelling. By solving all the equations related with load constraints on rigid stages, geometric compatibility conditions, and force-displacement correlations of the compliant modules, the (analytical) model of a CM is obtained. This simplifies the geometric compatibility criteria as well as any force-displacement connections between the units. However, utilising the FBD method to describe a complex CM is difficult since it is difficult to see the distorted geometries and reduce the intricate formulations. For both linear and nonlinear modelling of CMs with account of all applied external forces, there is another enhanced FBD-based technique called constraint-force-based (CFB) modelling technique. With apparent physical significance, the CFB method's quantity of expressions is decreased. Because a compliant component's elasticity is derived from its displacement, the elasticity of a compliant unit is proportional to its displacement. As a variable constraint force, this approach measures the displacement related constraint force. A nonlinear force - deformation correlation can be used to calculate the changeable constraint force of a basic compliant element. The summation of all variable constraint pressures from numerous compliant components linked to a rigid body in a parallel configuration is the variable constraint force on that rigid body. Furthermore, as exterior forces acting on CMs are independent of compliant element displacement, the CFB approach considers them to be continuous constraint forces. For all rigid stages in a balanced CM, the force equilibrium equations may be characterized by the related dynamic constraint forces and static constraint forces, with the analytical model being deduced from the force balance equations. As a result of the elimination of internal load factors, the energy approach can minimise the number of arbitrary parameters related to the FBD-based approach by applying the virtual work concept. It may be used to calculate strain energy of tiny elements or the work done by external loads using this energy technique. When it comes to calculating motion spectrum and fatigue life, nonetheless, the internal loads are equally important in CMs.

Based on modelling process: This consists of both numerical and PRBM solutions. Mathematical solutions are often achieved by means of methods such as numerical differential and elliptical integration. With the elliptical integration approach, nonlinearity of enormous geometrical magnitudes may be dealt with effectively. Using the Gauss-Chebyshev quadrature method, Saxena and colleagues solved these equations numerically. Banerjee et al. presented a non-linear shooting technique in numerical differential technique, where the boundary value equation is transformed to an initial value equation. When Chen et al. devised the chaining technique, they were able to solve both planer and spatial deformation problems simultaneously. For large deformation and multi-stable processes, this technique splits a flexible beam into a few parts and simulates each component using a medium large-range (spatial) beam constraint model (BCM). While the approximation accuracy of this method is defined by the quantity of sections, Saggere and Kota (2001) proposed one with an evenly decomposed flexible beam into at least three sections linked by at least three torsion springs with a stiffness that is entirely load-independent and can incorporate an array of loads. For high deflections of planar-motion beams, Su et al. presented a load-independent PRBM, which was confirmed to be accurate. Other PRBMs with prismatic joints have lately been suggested for Timoshenko beams to integrate the flexible elongation effect, as well as impacts from shearing force and cross-section variations. To cope with medium-large deflections, Awtar developed a closed-form analytical solution for planar displacement considering the distorted configuration for load-equilibrium circumstances (i.e., BCM), the transverse displacement is assumed to be less than 0.1 times the beam length in the mathematical solution.

12.2. Finite Element Method

For systems given in the form of partial differential equations, the finite element method (FEM) is one of the most powerful, flexible, and theoretically correct discretization approaches. In engineering, FEM is a computer method for obtaining estimated solutions to boundary value issues. The FEM method allows a continuous continuous domain to be discretized into a particular number of components and emphasises that the features of the continuous domain can be approximated by grouping the comparable attributes of discretized elements per node. As a result, the FEM has been intensively applied to solve a plethora of challenges in engineering and science and has swiftly evolved over time. Every individual component has its local coordinates system, by combining their stiffness matrices and element masses and they can be transformed to system global coordinates through transformations. For sizeable deformation of curved rodlike frameworks, the nonlinear theory is useful for analyzing the deflection interactions of compliant systems if the cross-section dimensions are lesser than the rod lengths, that can be solved using the well-known Euler-theorem Bernoulli's for a static problem of a slender configuration with a presumed axial inextensible line. The concept of Saint-Venant and Hooke's law may also be applicable. Particular phenomena related to notch flexure hinges, such as shear deflection, stress concentration, anticlastic bending, or manufacturing defects, must be anticipated, specifically for very thin hinges.

Natural frequencies, sensitivity analysis and strain response are unconventionally crucial for accurate modelling of a compliant system. Natural frequencies represent the mechanism's elastic characteristics, and the influence of design factors gleaned through natural frequency studies enhances mechanism performance and engineering. The natural frequency of the system is therefore directly defined by every element's mass and stiffness matrices. Similar inherent connections between the natural frequency of the system and other design factors may be derived. Because damping has relatively little effect on a system's natural frequencies, the undamped free vibration equation is employed to derive the system's natural frequencies and natural modes. Sensitivity analysis is a useful tool for predicting how different design factors impact the behaviour of compliant systems and it may be used to assist revision operations in fine-tuning design specifications to achieve preferred dynamic performance. Strain response is a critical metric for evaluating the dynamic design of compliant mechanisms. The strain reaction of a single position on a flexible link is a function of duration and may be computed using the system's elastic motion reaction.

12.3. Inverse Finite Element Model

For non-linear beams, the inverse finite element model was developed in 1988 by Cardona and Geradin, and in 2000 by Geradin and Cardona, respectively. For isotropic behaviour, inverse finite element models have been presented by Govindjee and Mihalic (1996) and (1998), and Yamada (1997), and recently extended to orthotropic materials by Fachinotti et al. (2008). IFEM is a finite element approach that uses only one solution of the equilibrium equations to find the undeformed shape of a mechanism. Cardona and Geradin's (1988) and Geradin and Cardona's (2000) inverse finite element models of non-linear beams were proposed by these authors. Government and Mihalic (1996), Govindjee and Mihalic (1998), and Yamada (1997) have presented inverse finite element models for the design of two-dimensional and three-dimensional isotropic elastic continuum bodies exposed to massive deformations with known deformed shape and loads. Fachinotti et al (2008). A systems's undeformed shape is determined by solving the equilibrium equations just once using IFEM. Facchinotti and coworkers created an orthotropic 3D elastic material using the IFEM. A 2D or 3D solid finite element analysis consumes vast amounts of computer resources. This is why Albanesi et al. developed the IFEM for elastic beams with significant deflections. A beam's initial form must be determined in order for it to achieve its design shape under the influence of service loads. There are direct uses for this approach in disciplines such as compliant system analysis, and it is a unique and original approach in this sector because there is no history of IFEM in the processes used to design compliant systems at this time. Because of these three assumptions, the nonlinear beam model is able to connect strain and stress measurements in deformed and undeformed configurations of the beam. the Newton-Raphson technique is used to solve the problem (Zienkiewicz and Taylor, 2000). For any technique based on the Newton Raphson algorithm, convergence of the inverse beam model is quadratic. To establish the manufacturing shape of the mechanism, we'll need to run it through a few tests. We'll need to know the system's deformed arrangement, how it's sustained and operated, and the characteristics of the material used to make it. Tests on mechanics - The distinctiveness of the solution, which is lost when an unstable equilibrium state (or critical point) is reached during deformation, and the validity of the elastic material behaviour hypothesis, as determined using the yield criteria best suited for the used material. When it comes to detecting key spots, the developer's knowledge is usually sufficient. As a formal check, the spectrum test can be used; and topological tests-absence of interpenetrated beam components. Sometimes, the mechanism must remain inside a design domain during the whole deformation. As a rule, mechanical and topological tests also apply to configurations corresponding to intermediate load stages. Using the inverse beam model, the difficulties may be recreated for numerous applications at a considerably reduced cost of computing. Due to its ability to exactly match item forms, this model has an obvious benefit. As a result, it is possible to decrease the concentration of stress in dispersed compliance methods as well. Even though the inverse analysis started with a properly deformed geometry, intersections of beam components appeared in undeformed arrangement in some situations. All of this might lead to a trial-and-error procedure, where you have to experiment with different beam cross-sections and materials until the junction vanishes. Implementing a basic contact issue of the form node-to-segment with consequence and soft contact algorithms with friction for 3D beams (Litewka, 2007) to apply solution constraints and remove impractical designs was presented by Puso and Laursen, 2004 as a potential solution to this challenge. The model's development when it transitions from deformed to undeformed state is unclear. Even if the convergence rate is quadratic, this might result in impractical designs since a beam component may slip outside of the required design domain at a given iteration. Finding an undeformed solution inside a given design space without inter-crossings is the greatest challenge when employing this technique. For example, passive microvalves proposed by Seidemann et al. (2001) to seal a 200 m microfluidic channel Kwang and Chong (2006) also presented similar check valves that just open to forward pressure showing diode-like characteristics), or the inversion analysis of shape memory alloy microgrippers with stress-optimized geometry proposed by Kohl et al. (2000) to design a high-flexibility structure.

13. **DYNAMIC** ANALYSIS OF COMPLIANT MECHANISMS

The influence of dynamic activity on compliant mechanism development is critical, particularly for high-speed operations. For dynamic analysis of compliant beams, finite element analysis and Nodal coordinate techniques appear to be appropriate. Partially compliant four-link systems, for example, are generally intended to give a certain motion. Kinematic synthesis is generally used to construct the morphology of connections and segments that give the desired displacement, velocity, or acceleration numerically. Kinematic synthesis produces a machine that can perform the desired behaviour geometrically, but does not consider stress or resilience. Following that, a stress analysis is necessary to develop a robust system that can fulfil both the motion needs as well as application-specific criteria such as material constraints. Some academics have recently focused on the dynamic study of compliant systems. Lyon et al. directly used the pseudo rigid-body model to examine the natural frequency of compliant parallel-guiding systems. The dynamic reactivity of compliant constant-force compaction devices was investigated analytically and experimentally by Boyle et al. Using the pseudo-rigid-body theory, created a novel dynamic model of compliant mechanisms based on the notion of dynamic equivalence. A planar compliant parallel-guiding system's natural frequency was determined. Both dynamic analysis and compliant mechanism designing need to be researched further in order to increase the functionality and operative precision of compliant mechanisms. Howell demonstrated two different stress estimates in a homogenous, fixed-free compliant section with zero and positive load factors. He found that the precision of the PRBM force-deflection analysis produced correct stress analysis answers after comparing stress calculations using PRBM, elliptic integral technique, and finite element analysis. Lobontiu et al. (2002) investigated flexure hinges and developed a dynamic model that included their inertia and damping characteristics. For system matrices, Rosner et al. introduced a Krylov subspace reduction reduced-order model. Lyon et al. and Boyle et al. created a dynamic design for a constant-force compression compliant mechanism using a PRB model and explored the first-order frequency of flexible parallel-guiding systems. Pei et al. presented a new approach for constructing PRB models for beam-based compliant mechanisms, in which a rigid bar with two pin-joints imitates the behaviour of parallel linear spring stage beams. Using the finite element approach, Li and Kota investigated the dynamics of compliant systems, looking at natural frequencies, modes, dynamic reaction, and frequency properties. Zhao et al. created a dynamic system for a compliant linear-motion system premised on the Lagrange equation, then studied the influence of material characteristics, form parameters, and geometric factors on natural frequency in order to provide numerical perspective to engineers. Based on the conventional 1R pseudo-rigid body model, Yu et al. created a dynamic system of compliant mechanisms. Li et al. investigated the dynamic evaluation of a curved-type compliant system using multi-revolute joints pseudo-rigid body model based on the mechanism's stiffness and curvature. The PRBM technique uses concise equations to forecast the beam-end coordinates with accuracy. Euler-Bernoulli beam theory cannot properly predict large deformation behaviour. The PRBM may be used to more correctly quantify deformation behaviour, especially the connection between beam-end deflection and imparted transverse force. The load factor is an important component to consider when computing moments and stresses. The axial force to transverse force ratio is called the load factor, If it is larger than zero, a compressive axial stress element lowers the tensile stress while increasing the compressive stress. The load angle is larger than 90 degrees for the associated load angle. If the load factor is below zero and the corresponding load angle is lower than 90 degrees, the tensile stress is raised while the compressive stress is lowered by a tensile axial stress element. As nonlinear beam analysis techniques are required for reliable computation of large deflections and beam-end coordinates to calculate the bending moment, bending stress calculations become complicated when applied to compliant sections. Closed form solutions derived from elliptic integral formulations produce reliable findings, but their complexity limits their applicability to simple boundary conditions. Nonetheless, they are frequently employed for analysis. The Chain Algorithm is a computational technique for evaluating compliant elements and structures in an efficient and exact manner. [171]

14. MATERIAL SELECTION

Selection of materials and stress analysis are typically complementary responsibilities in any technical device's development process. Due to the inherent necessity for compliance whilst preserving strength, material choice for compliant mechanisms is frequently more complicated. Any compliant mechanism's designer must manage environmental concerns, material strength, and compliance while keeping budget and machinability in mind. Developers of compliant mechanisms must conduct two tasks: kinetic synthesis to ensure that functionality requirements are satisfied, and load analysis and selection of materials to ensure that the system is durable. Performing the two design goals frequently necessitates several iterations and compromises.

Because of their low cost, light weight, capacity to hold energy, decreased wear owing to removal or decrease in the amount of kinematic couples, compliance, and manufacturability, engineering plastics are widely utilised to make compliant segments. Comparing the strength to the modulus of elasticity is one technique to assess a segment's strength and flexibility. Plastics show viscoelastic behaviour. When compared to materials with a low strength-to-modulus ratio, a polymer - based material with a high strength-to-modulus ratio may achieve greater strain amplitudes sans exceeding the stress limits of the material, making them an attractive choice to engineers. It is crucial assess the non - linearities in the material to analyse the performance and behaviour, which is a function of time. The viscoelastic or time dependant strain reaction adds complexity to the design phase of polymeric compliant mechanisms since the engineer must address the acceleration and velocity of each section additional to the forces.

Thermoplastic polymers like polypropylene are viscoelastic and have a time-dependent stress-strain relationship, whereas most technical materials are elastic solids (such as metals). Tensile testing at various strain rates is a simple way to detect viscoelastic properties, with certain constitutive laws, (like viscoelasticity and associative plasticity), the material seems to be in a nearly incompressible form. The resultant stress-strain graphs indicate changes in the stress-strain curve's amplitude and form, as well as fluctuations in the material's load capacity. A spring and a dashpot are used to simulate the elastic and viscous components of the viscoelastic responses, correspondingly. To handle creep and stress relaxation, linear viscoelastic materials' constitutive equations can be stated in terms of stress or strain. Plastics, in combination to the design challenges posed by viscoelasticity, do not function as predictably as a few other materials like metals and have significant limitations of fatigue, creep resistance, and stress relaxation resistance. As a result, material issues such as stress relaxation and creep behaviour must be adjusted with the beneficial energy storage afforded by displacement of a compliant link or segment. When compared to engineering plastics, metals have better creep resistance and stress relaxation characteristics at room temperature, making them appropriate for usage as metallic reinforcement inside a plastic composite. In order to minimise creep and fatigue breakdowns, Kuber et. al. proposed that a robust strengthening material be placed within a compliant housing made of a comparatively weak housing material. Kuber et. al. also proposed bi-material compliant sections as a means to increase fatigue and creep resistance, especially plastic shells with spring reinforcing steel. Polypropylene, for example, has a strength-to-modulus ratio of around 25 compared to 0.87 for hot rolled steel. IE-3075 is a popular prototype plastic with similar modulus and flexural strength to other engineering polymers, it is a stiff thermoset with material characteristics similar to acetal, a typical material in the building of compliant mechanisms. Because of these characteristics, it is a material worth considering in the field. It may also be cast at ambient temperature, eliminating the requirement for high-temperature injection moulding machines. Compressible (materials such as plastics generally show minor volumetric changes during deformation, like foams) and incompressible or nearly-incompressible (most plastics are almost incompressible) materials are the two types of hyper elastic materials with characteristic volumetric behaviour. At low temperatures, polymers flex elastically like glass; at high temperatures, they behave viscously like liquids; and at moderate temperatures, they behave like a rubbery solid would.

15. APPLICATIONS

Even NASA's cutting-edge space probe uses the same flexible, hinge-less compliant mechanism that's found in paper clips, nail-cutters, and smartphones. Compliant mechanisms are used in a wide range of high precision applications, including electrostatic suspensions, micro-robot and surgical instruments, different space research projects as well as car and architecture, as well as furniture and apparel. In robotics, self-adaptive systems are widely employed for gripping. Constraints in some contexts allow for substantial improvements in mechanism performance when using compliant mechanisms. In spite of the fact that many conventional mechanisms have been improved to a great degree. This spectacular method will fuel futuristic innovations in all generations to come with its potential that only been lightly tapped as of yet. A compliant mechanism would not have been conceivable without the use of tiny mechanical and electro-mechanical components (MEMS). We were saved from a worldwide disaster by the same mechanisms by keeping nuclear bombs from exploding. Design and analytical methodologies, as well as a rise in commercialized compliant mechanisms, have advanced the area of compliance mechanisms to a point where they may be used in critical areas.

Compliant robots, unlike traditional robots, are made from materials that are flexible and adaptable, drawing influence from biological beings. Soft robots may now be used in sectors such as health and manufacturing since they are safer when dealing with humans. If a rigid robot is large enough, some soft robotic mechanics can also be employed as an end-effector in it. To grasp and manipulate delicate things, soft robotic grippers have been created. These grippers provide minimal grab forces, which is critical to safely grasping fragile items. NASA's Goddard Space Flight Centre developed compliant cable technology. When used in structural connections, these mechanisms offer compliance and damping, allowing motion in the principal direction and selective motion in other directions. In this way, mating and contact surfaces are able to conform to one other. Material and settings are customised for each application. While disassembling, it's typically difficult to disengage without causing damage to the components. To enhance the effective stroke of piezoelectric actuators, compliant mechanical amplifiers are utilised. Aviation wings with compliant mechanisms might bend and twist as an one piece, removing the need for separate control surfaces like ailerons, spoilers and flaps to guide the aircraft's flight path. Construction becomes easier as a result. Because it decreases the radar cross-section, it improves stealth qualities, reduces weight and complexity, and enhances aircraft manoeuvrability. Pawl clutches with or without centrifugal ejection produce torque in one direction but allow freewheeling in the opposite. For example, pull-starts for tiny engines, bicycles, fishing reels, gear drives, and winches are all examples of one-way and two-way rotating applications for this type of motor. In contrast, centrifugal clutches designed as compliant devices do not require a large number of springs, pins, rivets, etc. Centrifugal force is used to move equipment when the hub (powered by a motor) spins the clutch to high speeds and the heavy sections engage with a drum. The flexible sections are built into this single moving element. Go karts, mini-bikes, trimmers, tillers, chainsaws, chippers, amusement rides, and agricultural and industrial machine couplings are examples of small and medium horsepower uses. Consistently designed bicycle brakes produce parallel motion, are visually appealing, and are recommended by professionals for their strength, improved control and even wear. As an example, vibration damping for power tools such as jackhammers and rivet guns can lead to repetitive motion injuries such as nerve damage and carpal tunnel syndrome due to the repetitive motions involved. As the tool is being used, it transmits vibrations that cause harm. This vibration can be reduced by using cable compliance technology. Small, compliant devices, Microelectromechanical Systems (MEMS) are used in mechanical and electrical applications in electrical technology. When it comes to manufacturing MEMS, computer chip manufacturing processes are used. A MEMS device is a device that can execute micro-manipulation operations by turning energy into a regulated motion. Since it is small it can also be applied to hand tools. Which have very interesting applications of compliant mechanisms and are used vastly. Medical tools for in-body surgery, hearing aids, airbag sensors, micro pumps, and optics and tilting mirrors for projection systems are all examples of MEMS applications in medicine. For lengthy periods of time, electrical connectors with near-constant-force (NCF) technology maintain a continuous interface. Faulty electrical contact integrity is responsible for the majority of computer hardware and vehicle electrical issues. Electricity connections and dependability are improved with the NCF electrical adapter. They may be used as switches, circuit breakers, clamps, snap hinges and closures, as well as for positioning mechanisms, among other things since they can be moved between two stable situations. There is no holding energy required for either posture despite the need for external force. The durability of plastic prototypes has been tested to over a million cycles. Using NCF compression mechanisms for compliant robotics, compression forces may be maintained at a near-constant level with a variation of just 2% in compression forces. In order to accommodate a wide range of travel patterns, several designs have been developed. This means that the possibilities are endless, as there is no known compression method for NCF on the market. Fitness goods, robot end effectors, tool holders, motor brush holders, wear test instruments, and safety devices might all benefit from NCF compression mechanisms. A joint prosthesis is a medical device that is costly and short-lived, with only the most expensive devices providing "human-like" responses. Unlike mechanical devices, the compliant junction has a nonlinear stiffness that rises as the cable bends. Flexible and rigid mechanisms must be able to accommodate the necessary displacement at the point of interest. When it comes to stiffness though, a compliant device should have the ability to withstand external force. When developing a compliant mechanism, flexibility and stiffness must be satisfied concurrently. To manufacture the flexible parts that make up compliant mechanisms using standard machining processes presents a number of problems in macro fabrication. Due to the recent development of a number of novel manufacturing processes (e.g., 3-axis computer numeric controlled milling and laser cutting), it is now possible to create a prototype of a compliant mechanism. These methods are nevertheless limited by the machine or the substance utilised. As a result, it's necessary to acquaint yourself with the machine and the material before moving on to the machining step. Heart, orthopaedic, rehabilitation, and biomechanics are some of the specialties in biomechanics engineering and simulation. For example, a knee, hip, or prosthetic heart valve may be made from one piece of material to make it easier to manufacture and more compatible with the natural flow of blood. There are technologies in development for MEMS that store information by transferring electrons from one location to another. This allows information that now requires a huge hard drive to be stored on an ultra-small storage device that is just several square millimetres in size. As a result, tasks such as reading and writing will be performed by a little robot arm from this small region. Many soft materials capable of changing 3D forms in response to stimuli are showing promise in the creation of soft robotics. Yet a task-oriented system for the design of structural structures comprising of such soft materials is far from being completed. For continuum topology optimization, this might be a good candidate for use in the future.

16. CONCLUSION

Many existing mechanisms are already highly optimised, but they still have fundamental flaws, and it's uncertain if further refinement of current designs will result in substantial performance gains. Compliant mechanisms have the ability to dramatically improve mechanism performance given the restrictions of an environment, and they offer a promising chance to modify the basic approach to producing controlled motion in unusual regions. Currently there are numerous challenges an engineer needs to deal with during the development of an efficient compliant mechanism design which is heavily under research.

Due to fundamental differences of the modelling methods and dynamic analysis from conventional mechanisms for designing rigid body replacements, compliant and pseudo rigid bodies is currently not as efficient for vast use the concept which is projected to change with growing interest in the benefits of compliant mechanisms. Material selection is generally understood but the detailed issues that rise with each metal or polymeric compounds and differences that come with production are also a major hold back. Compliant mechanisms have made a significant contribution to the design process in a variety of sectors, including adaptable buildings, transportation components, hand-held equipment, electronics, medical, and so on. It is a restriction of a compliant mechanism that it can only be utilised in situations with a restricted degree of motion or minor displacements.

17. REFERENCES

[1]. Howell, L.L., 2001, Compliant Mechanisms, John Wiley and Sons, Inc., New York, New York.

[2]. Solehuddin Shuib, M.I.Z. Ridzwan and A Halim Kadarman, “Methodology of Compliant Mechanisms and its Current Developments in Applications: A Review”, American Journal of Applied Sciences 4 (3): 160-167, 2007 ISSN 1546-9239, 2007

[3]. Fowler, R. & Howell, Larry & Magleby, Spencer. (2011). Compliant space mechanisms: A new frontier for compliant mechanisms. Mechanical Sciences. 2. 205-215. 10.5194/ms-2-205-2011.

[4]. Benliang Zhu, Xianmin Zhang, Hongchuan Zhang, Junwen Liang, Haoyan Zang, Hai Li, Rixin Wang, Design of compliant mechanisms using continuum topology optimization: A review, Mechanism and Machine Theory, Volume 143, 2020

[5]. Howell, L. L., A. Midha and T. W. Norton, 1996, "Evaluation of Equivalent Spring Stiffness for Use in a Pseudo-Rigid-Body Model of Large-Deflection Compliant Mechanisms," Journal of Mechanical Design, Trans. ASME, Vol. 118, No. 1, pp. 126- 131.

[6]. Howell, L.L., “Pseudo-rigid-body models and compliant mechanisms” 2003, http://research.et.byu.edu/llhwww

[7]. Howell, L.L., and Midha, A., 1993, “Compliant Mechanisms,” Section 9.10, Modern Kinematics: Development in the last Forty years, (Ed.: A.G. Erdman), John Wiley and Sons, Inc., New York, New York.

[8]. Koli, Ashish B., "A generalized approach for compliant mechanism design using the synthesis with compliance method, with experimental validation" (2013). Masters Theses. 7099.

[9]. Bapat, Sushrut Gangadhar, "On the design and analysis of compliant mechanisms using the pseudo-rigidbody model concept" (2015). Doctoral Dissertations. 2376.

[10]. Lusk, C., Using Pseudo-Rigid Body Models. Handbook of Compliant Mechanisms, 55–76, 2013

[11]. Venkatasubramanian Kalpathy Venkiteswaran, Hai-Jun Su, “Pseudo-Rigid-Body Models of Initially-Curved and Straight Beams for Designing Compliant Mechanisms”, Proceedings of ASME 2016 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference IDETC/CIE 2017

[12]. Ngoc Le Chau, Ngoc Thoai Tran, Thanh-Phong Dao, "An Optimal Design Method for Compliant Mechanisms", Mathematical Problems in Engineering, vol. 2021, Article ID 5599624, 18 pages, 2021. https://doi.org/10.1155/2021/5599624

[13]. Sebastian Linß, Stefan Henning and Lena Zentner, Modeling and Design of Flexure Hinge-Based Compliant Mechanisms, April 3rd 2019, DOI: 10.5772/intechopen.85224

[14]. Wenjing Wang, & Yueqing Yu., “Dynamic analysis of compliant

mechanisms using the finite element method", IEEE/ASME International Conference on Advanced Intelligent Mechatronics, 2008

[15]. Howell L.L. (2013) "Compliant Mechanisms" In: McCarthy J. (eds) 21st Century Kinematics. Springer, London.

[16]. Pradhan, K. K., & Chakraverty, S. (2019). Finite Element Method. Computational Structural Mechanics, 25–28. doi:10.1016/b978-0-12-815492-2.00010-1

[17]. Yu, Y., Howell, L. L., Lusk, C., Yue, Y., and He, M. (February 12, 2005). "Dynamic Modeling of Compliant Mechanisms Based on the Pseudo-Rigid-Body Model." ASME. J. Mech. Des. July 2005; 127(4): 760–765.

[18]. Guangbo Hao, Haiyang Li, Jingjun Yu, "A Brief Review on Nonlinear Modelling Methods and Applications of Compliant Mechanisms", Frontiers of Mechanical Engineering, May 2016

[19]. Xu, S., Zhu, X., Dong, Z., & Liu, P. (2018). Nonlinear modeling and analysis of compliant mechanisms with circular flexure hinges based on quadrature beam elements. Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science, 095440621880294. doi:10.1177/0954406218802945

[20]. Milojevic, Andrija & Pavlovic, Nenad & Milosevic, Milos & Tomic, Misa. (2013). New Software for Synthesis of Compliant Mechanisms.

[21]. Howell, L. L., and Midha, A. (March 1, 1996). "A Loop-Closure Theory for the Analysis and Synthesis of Compliant Mechanisms." ASME. J. Mech. Des. March 1996; 118(1): 121–125.

[22]. Yu, Y.-Q., Li, Q., & Xu, Q.-P. (2017). Pseudo-rigid-body dynamic modeling and analysis of compliant mechanisms. Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science, 232(9), 1665–1678. doi:10.1177/0954406217707547

[23]. Howell, L. L., Midha, A., and Norton, T. W. (March 1, 1996). "Evaluation of Equivalent Spring Stiffness for Use in a Pseudo-Rigid-Body Model of Large-Deflection Compliant Mechanisms." ASME. J. Mech. Des. March 1996; 118(1): 126–131.

[24]. Kuber, Raghvendra Sharadchandra, "Development of a methodology for pseudo-rigid-body models of compliant segments with inserts, and experimental validation" (2013). Masters Theses. 5363.

[25]. Midha, A., Norton, T. W., and Howell, L. L. (March 1, 1994). "On the Nomenclature, Classification, and Abstractions of Compliant Mechanisms." ASME. J. Mech. Des. March 1994; 116(1): 270–279.

[26]. Crews, Joshua Allen, "Methodology for analysis of stress, creep, and fatigue behavior of compliant mechanisms" (2016). Doctoral Dissertations. 2534.

[27]. Theddeus T. Akano, Omotayo A. Fakinlede, "Fatigue Failure Model for Polymeric Compliant Systems", International Scholarly Research Notices, vol. 2013, Article ID 321489, 11 pages, 2013.

[28]. Li, N., Liu, C., & Jiang, H. (2016), "Dynamics modeling for curved-type fully compliant mechanism based on pseudo-rigid-body modelling", 2016 IEEE International Conference on Mechatronics and Automation. doi:10.1109/icma.2016.7558924

[29]. Ezekiel G. Merriam, Jonathan E. Jones, Larry L. Howell, "Design of 3D-Printed Titanium Compliant Mechanisms", 42nd Aerospace Mechanisms Symposium, NASA GSFC, May 14-16, 2014

[30]. Daniel Lates, Marius Casvean, Sorina Moica, "Fabrication Methods Of Compliant Mechanisms", Procedia Engineering 181 221-225, 2017

[31]. Benliang Zhu, Xianmin Zhang, Hongchuan Zhang, Junwen Liang, Haoyan Zang, Hai Li, Rixin Wang, "Design of compliant mechanisms using continuum topology optimization: A review", Mechanism and Machine Theory, Volume 143, 103622, 2020

[32]. Arredondo-Soto M, Cuan-Urquizo E, Gómez-Espinosa A. A Review on Tailoring Stiffness in Compliant Systems, via Removing Material: Cellular Materials and Topology Optimization. *Applied Sciences*. 2021; 11(8):3538. https://doi.org/10.3390/app11083538

[33]. Alejandro E. Albanesi, Victor D. Fachinotti and Martin A. Pucheta, "A REVIEW ON DESIGN METHODS FOR COMPLIANT MECHANISMS", Mecánica Computacional Vol XXIX, November 2010

[34]. Spencer P. Magleby, David Smith, Larry L. Howell, Brian D. Jensen, "Metrics For Evaluation And Design Of Large Displacement Linear Motion Compliant Mechanisms", Journal of mechanical design, Jan 2012

[35]. Pandiyan, Arunkumar Gopal, "Design methods for compliant mechanisms used in new age industries- A review", Istrazivanja i Projektovanja za Privredu, January 2016

[36]. Jianing Wu, Shaoze Yan, Junlan Li, "Mechanism reliability of bistable compliant mechanisms considering degradation and uncertainties: Modeling and evaluation method", Applied Mathematical Modelling, July 2016

[37]. Gao, G., Liu, Z., Li, Y., & Qiao, Y. (2016). A new method to generate the ground structure in truss topology optimization. Engineering Optimization, 49(2), 235–251. doi:10.1080/0305215x.2016.1169050

[38]. Sui Y., Ye H., Peng X. (2006) TOPOLOGICAL OPTIMIZATION OF CONTINUUM STRUCTURE WITH GLOBAL STRESS CONSTRAINTS BASED ON ICM METHOD", In: LIU G., TAN V., HAN X. (eds) Computational Methods. Springer, Dordrecht. https://doi.org/10.1007/978-1-4020-3953-9_1

[39]. Crews, Joshua Allen, "Methodology for analysis of stress, creep, and fatigue behavior of compliant mechanisms" (2016). Doctoral Dissertations. 2534.

[40]. Fowler, R. & Howell, Larry & Magleby, Spencer. (2011). Compliant space mechanisms: A new frontier for compliant mechanisms. Mechanical Sciences. 2. 205-215. 10.5194/ms-2-205-2011.

[41]. Mart´ın A. Pucheta and Alberto Cardona, "SYNTHESIS AND OPTIMIZATION OF COMPLIANT MECHANISMS FROM A RIGID-BODY APPROACH", Mecánica Computacional Vol XXVII, November 2008

[42]. Fulei, Ma & Chen, Guimin, "Modeling Large Planar Deflections of Flexible Beams in Compliant Mechanisms Using Chained Beam-Constraint-Model",ASME Journal of Mechanisms and Robotics, 2016, 8. 10.1115/1.4031028.

[43]. Lionel Kiener, Hervé Saudan, Florent Cosandier, Gérald Perruchoud and Peter Spanoudakis, "Innovative concept of compliant mechanisms made by additive manufacturing", MATEC Web of Conferences 304, 07002 (2019)

[44]. Zirbel SA, Tolman KA, Trease BP, Howell LL (2016) Bistable Mechanisms for Space Applications. PLOS ONE 11(12): e0168218

[45]. S P Jagtap et al 2021 J. Phys.: Conf. Ser. 1969 012013

[46]. Ghoddosian, A., Riyahi Vezvari, M., Sheikhi Azqandi, M. and Karimi, M.A. (2018) 'Topology optimisation of the discrete structures with the minimum growing ground structure method', Int. J. Structural Engineering, Vol. 9, No. 1, pp.38–49.

[47]. E. G. Merriam, J. E. Jones, S. P. Magleby, and L. L. Howell, "Monolithic 2 DOF fully compliant space pointing mechanism", Mech. Sci., 4, 381–390, 2013

[48]. Liu, Pengbo & Yan, Peng. (2017). A modified pseudo-rigid-body modeling approach for compliant mechanisms with fixed-guided beam flexures. Mechanical Sciences. 8. 359-368. 10.5194/ms-8-359-2017.

[49]. Howell, L. L., A. Midha, and T. W. Norton, "Evaluation of Equivalent Spring Stiffness for Use in a Pseudo-Rigid-Body Model of Large-Deflection Compliant Mechanisms," Journal of Mechanical Design, Trans. ASME, Vol. 118, No. 1, March 1996, pp. 126-131.

[50]. Howell, L.L., "The Design and Analysis of Large Deflection Members in Compliant Mechanisms," MS Thesis, Purdue University, 1991.
[51]. Norton, T.W., "On the Nomenclature, Classification, and Mobility of Compliant Mechanisms," MS Thesis, Purdue University, 1991.
[52]. Midha, A., and I. Her, "Research Notes," Purdue University, 1985.
[53]. Midha, A., and Kuber R., "Closed-Form Elliptical Integral Solutions of Initially-Straight and Initially-Curved SmallLength Flexural Pivots", ASME 2014 International Design Engineering Technical Conferences, 2014.
[54]. Venkiteswaran, V.K., and Jun-Su, H., "Development of a 3-Spring Pseudo Rigid Body Model of Compliant Joints for Robotic Applications," ASME 2014 International Design Engineering Technical Conferences, 2014.
[55]. Jun Su, H., "A Pseudo-Rigid-Body 3R Model for Determining Large Deflection of Cantilever Beams Subject to Tip Loads", October 30, 2008. ASME Journal of Mechanisms and Robotics, p. 021008, vol 1, no.2, May, 2009
[56]. Lodagala, Vamsi & Karthik, Krutika & Midha, Ashok. (2015). Limitations in the Use of Small-Length Flexural Pivot in a Pseudo-Rigid-Body Model. V05AT08A026. 10.1115/DETC2015-47914.
[57]. Salamon, B.A., 1989, "Mechanical Advantage Aspects in Compliant Mechanisms Design," M.S. Thesis, Purdue University, West Lafayette, Indiana.
[58]. Burns, R.H., 1964, "The Kinetostatic Synthesis of Flexible Link Mechanisms," Ph.D. Dissertation, Yale University, New Haven, Connecticut.
[59]. Burns, R.H., and Crossley, F.R.E., 1968, "Kinetostatic Synthesis of Flexible Link Mechanisms," ASME Paper No. 68-Mech-36
[60]. Howell, L.L., and Midha, A., 1993, "Compliant Mechanisms," Section 9.10, Modern Kinematics: Developments in the Last Forty Years, (editor: A. Erdman), Wiley, New York, pp. 422-428.
[61]. Howell, Larry & Midha, Ashok. (1992). Parametric Deflection Approximations for End-Loaded, Large-Deflection Beams in Compliant Mechanisms. 157-166. 10.1115/DETC1992-0291.
[62]. Howell, L.L., and Midha, A., 1995, "Parametric Deflection Approximations for End-Loaded, Large-Deflection Beams in Compliant Mechanisms," ASME Journal of Mechanical Design, Vol. 117, No. 1, pp. 156-165.
[63]. Howell, L.L., and Midha, A., 1995, "Determination of the Degrees of Freedom of Compliant Mechanisms Using the Pseudo-Rigid-Body Model Concept," Proceedings of the Ninth World Congres on the Theory of Machines and Mechanisms, Milano, Italy, Vol. 2, pp. 1537-1541.
[64]. Howell LL. Compliant mechanisms (2001). John Wiley&Sons, Inc.:1-8.
[65]. Her, I., 1986, "Methodology for Compliant Mechanism Design", Ph.D. Dissertation, Purdue University, West Lafayette, Indiana.
[66]. Her, I., and Chang, J.C., 1994, "A Linear Scheme for the Displacement Analysis of Micropositioning Stages with Flexure Hinges,"Machine Elements and Machine Dynamics, DE-Vol. 71, 23rd ASME Biennial Mechanisms Conference, pp. 517-525.
[67]. Her, I., and Midha, A., 1987, "A Compliance Number Concept for Compliant Mechanisms, and Type Synthesis,"Journal of Mechanisms, Transmissions, and Automation in Design, Trans. ASME, Vol. 109, No. 3, pp. 348-355.
[68]. Her, I., Midha, A., and Salamon, B.A., 1992, "A Methodology for Compliant Mechanisms Design: Part II - Shooting Method and Application,"Advances in Design Automation, (Ed.: D.A. Hoeltzel), DE-Vol. 44-2, 18th ASME Design Automation Conference, pp. 39-45.
[69]. Nocedal, J., & Wright, S. J. (1999). Numerical optimization. New York: Springer.
[70]. Midha, A. & Turcic, D.. (1980). On the Periodic Response of Cam Mechanism With Flexible Follower and Camshaft. Journal of Dynamic Systems Measurement and Control-transactions of The Asme - J DYN SYST MEAS CONTR. 102. 10.1115/1.3149611.
[71]. Howell, Larry & Midha, Ashok. (1994). A Generalized Loop-Closure Theory for the Analysis and Synthesis of Compliant Mechanisms. 491-500. 10.1115/DETC1994-0293.
[72]. Howell, Larry & Midha, Ashok. (1994). The Development of Force-Deflection Relationships for Compliant Mechanisms. 501-508. 10.1115/DETC1994-0294.
[73]. Howell, Larry & Midha, Ashok. (1994). Evaluation of Equivalent Spring Stiffness for Use in a Pseudo-Rigid-Body Model of Large-Deflection Compliant Mechanisms. 405-412. 10.1115/DETC1994-0219.
[74]. Howell, Larry & Midha, A.. (1994). A Method for the Design of Compliant Mechanisms With Small-Length Flexural Pivots. Journal of Mechanical Design - J MECH DESIGN. 116. 10.1115/1.2919359.
[75]. Howell, Larry & Midha, Ashok. (1994). Development of force-deflection relationships for compliant mechanisms. American Society of Mechanical Engineers, Design Engineering Division (Publication) DE. 71.
[76]. Khanuja, Sukhwant & Mettlach, Gregory & Midha, Ashok. (1994). Graphical Synthesis of Optimal High-Performance Four-Bar Mechanisms. 229-237. 10.1115/DETC1994-0199.mett
[77]. Mettlach, Gregory & Midha, Ashok. (1996). Using Burmester Theory in the Design of Compliant Mechanisms. 10.1115/96-DETC/MECH-1181.
[78]. Murphy, Morgan & Midha, Ashok & Howell, Larry. (1994). Type Synthesis of Compliant Mechanisms Employing a Simplified Approach to Segment Type. 51-60. 10.1115/DETC1994-0178.
[79]. Annamalai, Y., 2003, "Compliant Mechanism Synthesis for Energy and Torque Specifications," M.S. Thesis, University of Missouri-Rolla.
[80]. Midha, A., Annamalai, Y., and Kolachalam, S. K., 2004, "A Compliant Mechanism Design Methodology for Coupled and Uncoupled Systems, and Governing Free Choice Selection Considerations" Proceedings of DETC'04 ASME Design Engineering Technical Conferences and Computers and Information in Engineering Conference, Salt Lake City, Utah, pp. 57579-1-9.
[81]. Kolachalam, S. K., 2003, "Synthesis of Compliant Single-Strip Mechanisms that may be Represented by a Dyad," M.S. Thesis, University of Missouri-Rolla.
[82]. Midha, A., Kolachalam, S. K., and Annamalai, Y., 2011, "On the Classification of Compliant Mechanisms and Synthesis Compliant of Single-Strip Mechanisms" Proceedings of the ASME 2011 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference, Washington, DC pp. 71400-1- 10.
[83]. Dado, M. H., 2000, "Variable Parametric Pseudo-Rigid-Body Model for LargeDeflection Beams with End Loads," International Journal of Non-Linear Mechanics, Vol. 36 pp. 1123-1133.
[84]. Dado, M. H., 2005, "Limit Position Synthesis and Analysis of Compliant 4-Bar Mechanisms with Specified Energy Levels Using Variable Parametric Pseudo-RigidBody Model," Mechanism and Machine Theory Vol. 40, No.8 pp. 977-992.
[85]. Frecker, M. I., Ananthasuresh, G. K., Nishiwaki, S., Kikuchi, N., and Kota, S., 1997, " Topological Synthesis of Compliant Mechanisms Using Multi-Criteria Optimization," Journal of Mechanisms, Transmissions, and Automation in Design, Trans. ASME, Vol. 119, pp. 238-245.
[86]. Ananthasuresh, G. K., 1994, "A New Design Paradigm for Micro-Electro-Mechanical Systems and Investigation on the Compliant Mechanism Synthesis," Ph.D. Thesis, University of Michigan, Ann Arbor.
[87]. Saggere, L., and Kota, S., 2001, "Synthesis of Planar, Compliant Four-Bar

Mechanisms for Compliant-Segment Motion Generation," Journal of Mechanisms, Transmissions, and Automation in Design, Trans. ASME, Vol. 123, No. 4, pp. 535-541.

[88]. Lu, K. J., and Kota, S., 2003, "Design of Compliant Mechanisms for Morphing Structural Shapes," Journal of Intelligent Systems and Structures, Vol. 14, pp. 379-391.

[89]. Su, H-J., and McCarthy, J. M., 2007, "Synthesis of Bistable Compliant Four-Bar Mechanisms Using Polynomial Homotopy," Journal of Mechanisms, Transmissions, and Automation in Design, Trans. ASME, Vol. 129, No.10, pp. 1094-1098.

[90]. Tari, H., and Su H-J., 2011, "A Complex Solution Framework for the Kinetostatic Synthesis of a Compliant Four-Bar Mechanism," Mechanism and Machine Theory, Vol. 46, pp. 1137-1152.

[91]. Howell, L. L., 1993, "A Generalized Loop-Closure Theory for the Analysis and Synthesis of Compliant Mechanisms," Ph.D. Dissertation, Purdue University.

[92]. Gilbert, Matthew & Tyas, Andrew. (2003). Layout optimization of large-scale pin-jointed frames. Engineering Computations. 20. 1044-1064. 10.1108/02644400310503017.

[93]. Sokół, T., & Rozvany, G.I. (2013). On the adaptive ground structure approach for multi-load truss topology optimization.

[94]. Gilbert, M., Darwich, W., Tyas, A., & Shepherd, P. (2005). Application of Large-scale Layout Optimization Techniques in Structural Engineering Practice. Paper presented at 6th World Congresses of Structural and Multidisciplinary Optimization, Rio de Janeiro, Brazil.

[95]. Sokół, T. (2011). A 99 line code for discretized Michell truss optimization written in Mathematica. Structural and Multidisciplinary Optimization, 43, 181-190.

[96]. Sokół, T., & Lewiński, T. (2011). Optimal design of a class of symmetric plane frameworks of least weight. Structural and Multidisciplinary Optimization, 44, 729-734.

[97]. Dobbs, M.W., & Felton, L.P. (1969). Optimization of Truss Geometry. Journal of the Structural Division, 95, 2105-2118.

[98]. Smith, O. P. P. D. S. (1996). Optimal Truss Topology Design: Generation of Ground Structures & Local Stability Constraints.

[99]. da Silva Smith, O. (1998), "Generation of ground structures for 2D and 3D design domains", Engineering Computations, Vol. 15 No. 4, pp. 462-500.

[100]. Zegard, T., Paulino, G.H. GRAND — Ground structure based topology optimization for arbitrary 2D domains using MATLAB. Struct Multidisc Optim 50, 861–882 (2014).

[101]. Achtziger, W., Stolpe, M. Truss topology optimization with discrete design variables—Guaranteed global optimality and benchmark examples. Struct Multidisc Optim 34, 1–20 (2007).

[102]. Achtziger, Wolfgang. (2006). Simultaneous Optimization of Truss Topology and Geometry, Revisited. 10.1007/1-4020-4752-5_41.

[103]. Stolpe, Mathias & Achtziger, Wolfgang & Kawamoto, Atsushi. (2006). A Concept for Global Optimization of Topology Design Problems. 10.1007/1-4020-4752-5_19.

[104]. Martin Philip Bendsøe, Noboru Kikuchi, Generating optimal topologies in structural design using a homogenization method, Computer Methods in Applied Mechanics and Engineering, Volume 71, Issue 2, 1988, Pages 197-224, ISSN 0045-7825

[105]. Kachanov, L.M., & Krajcinovic, D. (1986). Introduction to continuum damage mechanics.

[106]. P. Li, N.J. Marchand, B. Ilschner, Mater. Sci. Eng. A 119 (1989) 41.

[107]. Burak Demirel, M. Tolga Emirler, Ümit Sönmez1, Ahmet Yörükoğlu, Semicompliant Force Generator Mechanism Design for a Required Impact and Contact Forces, Journal of Mechanisms and Robotics, ASME NOVEMBER 2010, Vol. 2 / 045001-1

[108]. Levent Subaşi, Synthesis Of Compliant Bistable Four-Link Mechanisms For Two Positions, A Thesis Submitted To The Graduate School Of Natural And Applied Sciences Of Middle East Technical University, November 2005

[109]. Bennion R. Cannona ,Todd D. Lillian, Spencer P. Magleby, Larry L. Howell, Matthew R. Linford, A compliant end-effector for microscribing, Precision Engineering 29 (2005) 86–94

[110]. M. Jiang, "A damaged evolution model for strain fatigue of ductile metals," Engineering Fracture Mechanics, vol. 52, no. 6, pp. 971–975, 1995.

[111]. W. Shi, W. Hu, M. Zhang, and Q. Meng, "A damage mechanics model for fatigue life prediction of fiber reinforced polymer composite lamina," Acta Mechanica Solida Sinica, vol. 24, no. 5, pp. 399–410, 2011.

[112]. N. V. Akshantala and R. Talreja, "A micromechanics based model for predicting fatigue life of composite laminates," Materials Science and Engineering A, vol. 285, no. 1-2, pp. 303– 313, 2000.

[113]. J. J. Ping, M. Guang, S. Yi, and X. SongBo, "An Effective continuum damage mechanics model for creep-fatigue life assessment of a steam turbine rotor," International Journal of Pressure Vessels and Piping, vol. 80, no. 6, pp. 389–396, 2003.

[114]. A. Ali, M. Hosseini, and B. Sahari, "Continuum damage mechanics modeling for fatigue life of elastomeric materials," International Journal of Structural Integrity, vol. 1, no. 1, pp. 63– 72, 2010.

[115]. Y. S. Upadhyaya and B. K. Sridhara, "Fatigue crack initiation and propagation life prediction of materials," in Proceedings of the International Conference on Mechanical, Electronics and Mechatronics Engineering (ICMEME '12), Bangkok, Thailand, 2012.

[116]. B. Wang, H. Lu, and G. Kim, "A damage model for the fatigue life of elastomeric materials," Mechanics of Materials, vol. 34, no. 8, pp. 475–483, 2002.

[117]. W. E. Mahmoud, S. A. Mansour, M. Hafez, and M. A. Salam, "On the degradation and stability of high abrasion furnace black (HAF)/acrylonitrile butadiene rubber (NBR) and high abrasion furnace black (HAF)/graphite/acrylonitrile butadiene rubber (NBR) under cyclic stress-strain," Polymer Degradation and Stability, vol. 92, no. 11, pp. 2011–2015, 2007

[118]. X. Y. Gong and R. Moe, "On stress analysis for a hyperelastic material," in Proceedings of the ANSYS Conference, 2002.

[119]. N. D. Mankame and G. K. Ananthasuresh, "A novel compliant mechanism for converting reciprocating translation into enclosing curved paths," Journal of Mechanical Design, vol. 126, no. 4, pp. 667–672, 2004.

[120]. Yang, Miao & Du, Zhijiang & Dong, Wei. (2016). Modeling and analysis of planar symmetric superelastic flexure hinges. Precision Engineering. 46. 10.1016/j.precisioneng.2016.04.012.

[121]. Su, H.-J., 2009, "A Pseudo rigid-Body 3r Model for Determining Large Deflection of Cantilever Beams Subject to Tip Loads". Journal of Mechanisms and Robotics, 1(2), Jan., pp. 021008–021008.

[122]. Awtar, S., and Sen, S., 2010, "A Generalized Constraint Model for Two-Dimensional Beam Flexures: Nonlinear Load-Displacement Formulation", Journal of Mechanical Design, Vol. 132:081008.

[123]. Sen, S., Awtar, S., 2013, "Nonlinear Strain Energy Formulation of a Generalized Bisymmetric Spatial Beam for Flexure Mechanism Analysis", Journal of Mechanical Design, 136(2):021002-021002-13. doi:10.1115/1.4025705.

[124]. Saxena, A., and Kramer, S. N., 1998, "A Simple and Accurate Method for Determining Large Deflections in Compliant Mechanisms Subjected

to End Forces and Moments", Journal of Mechanical Design, Vol. 120(3): 392–400.

[125]. Kumar, R, 2004. "Techniques Based on Genetic Algorithms for Large Deflection Analysis of Beams", Sadhana, Vol. 29(6): 589–604.

[126]. Banerjee, A, et al., 2007, "Large Deflection of Cantilever Beams with Geometric Non-linearity: Analytical and Numerical Approaches", International Journal of Non-Linear Mechanics, Vol. 43(5): 366–376.

[127]. Ma, F., and Chen, G., 2015, "Modeling Large Planar Deflections of Flexible Beams in Compliant Mechanisms Using Chained Beam-Constraint-Model", Journal of Mechanisms and Robotics, doi:10.1115/1.4031028

[128]. Morsch, F. M., Tolou, N., Herder, J. L., 2009, "Comparison of Methods for Large Deflection Analysis of a Cantilever Beam under Free End Point Load", Proceedings of the ASME 2009 International Design Engineering Technical Conferences &Computers and Information in Engineering Conference, August 30–September 2, 2009, San Diego, California, US.

[129]. Govindjee S, Mihalic PA. Computational methods for inverse finite elastostatics. Computer Methods in Applied Mechanics and Engineering 1996; 136:47–57.

[130]. Govindjee S, Mihalic PA. Computational methods for inverse deformations in quasi-incompressible finite elasticity. International Journal for Numerical Methods in Engineering 1998; 43:821–838.

[131]. Koishi M, Govindjee S. Inverse design methodology of a tire. Tire Science and Technology 2001; 29(3):155–170.

[132]. Yamada T. Finite element procedure of initial shape determination for hyperelasticity. Structural Engineering and Mechanics 1997; 6(2):173–183.

[133]. Govindjee S. Finite deformation inverse design modeling with temperature changes, axis-symmetry and anisotropy. Technical Report UCB/SEMM-1999/01, University of California, Berkeley, 1999.

[134]. Lu J, Zhou X, Raghavan ML. Computational method of inverse elastostatics for anisotropic hyperelastic solids. International Journal for Numerical Methods in Engineering 2007; 69:1239–1261.

[135]. Fachinotti V, Cardona A, Jetteur Ph. Finite element modelling of inverse design problems in large deformations anisotropic hyperelasticity. International Journal for Numerical Methods in Engineering 2008; 74:894–910.

[136]. Zhou X, Lu J. Inverse formulation for geometrically exact stress resultant shells. International Journal for Numerical Methods in Engineering 2008; 74:1278–1302. 10. Cardona A, Géradin M. A beam finite element non-linear theory with finite rotations. International Journal for Numerical Methods in Engineering 1988; 26:2403–2438.

[137]. Géradin M. Cardona A. Flexible Multibody Dynamics. A Finite Element Approach. Wiley: New York, 2000.

[138]. Bathe K-J, Bolourchi S. Large displacement analysis of three-dimensional beam structures. International Journal for Numerical Methods in Engineering 1979; 14:961–986.

[139]. Simo JC, Vu-Quoc L. A three-dimensional finite strain-rod model. Part II: computational aspects. Computer Methods in Applied Mechanics and Engineering 1986; 58:79–116.

[140]. Dvorkin EN, Oñate E, Oliver J. On a non-linear formulation for curved Timoshenko elements considering large displacements/rotations increments. International Journal for Numerical Methods in Engineering 1988; 26:1597–1613.

[141]. Lan C-C, Cheng Y-J. Distributed shape optimization of compliant mechanisms using intrinsic functions. Proceedings of the ASME 2007 International Design Engineering Technical Conferences & Computer and Engineering Conference, Las Vegas, NV, U.S.A., September 2007.

[142]. Zienkiewicz OC, Taylor RL. The Finite Element Method, Volume 2: Solid and Structural Mechanics (5th edn). Butterworth-Heinemann: London, 2000.

[143]. Shampine LF. Vectorized adaptive quadrature in Matlab. Journal of Computational and Applied Mathematics 2008; 211:131–140.

[144]. Crane NB. Compliant centrifugal clutches: design, analysis and testing. Master's Thesis, Department of Mechanical Engineering, Brigham Young University, 1999.

[145]. Weight RG. High-torque capacity compliant centrifugal clutches. Master's Thesis, Department of Mechanical Engineering, Brigham Young University, 2004.

[146]. Litewka, Przemyslaw & Wriggers, Peter. (2002). Contact between 3D Beams with Rectangular Cross-Sections. International Journal for Numerical Methods in Engineering. 53. 2019 - 2041. 10.1002/nme.371.

[147]. Litewka, Przemyslaw (2007), Smooth Frictional Contact between Beams in 3D, Springer, Dordrecht, 978-1-4020-6405-0

[148]. Puso, Mike & Laursen, Tod. (2004). A mortar segment-to-segment contact method for large deformations. Computer Methods in Applied Mechanics and Engineering. 193. 601-629. 10.1016/j.cma.2003.10.010.

[149]. Seidemann, V. & Bütefisch, S. & Bütgenbach, S.. (2002). Fabrication and Investigation of Inplane Compliant su8 structures for MEMS and Their Application to Micro Valves and Micro Grippers. Sensors and Actuators A. 2316. 1-6.

[150]. Oh, Kwang & Ahn, C.. (2006). A Review of Microvalves. J. Micromech. Microeng. 16. 13-39. 10.1088/0960-1317/16/5/R01.

[151]. Kohl M, Just E, Fleging WP, Miyazaki S (2000) SMA microgripper with integrated antagonism. Sens Actuators 83:208–213

[152]. Boyle, Cameron. (2021). A Closed-Form Dynamic Model Of The Compliant Constant-Force Mechanism Using The Pseudo-Rigid-Body Model.

[153]. Boyle, Cameron & Howell, Larry & Magleby, Spencer & Evans, Mark. (2003). Dynamic modeling of compliant constant-force compression mechanisms. Mechanism and Machine Theory. 38. 1469-1487. 10.1016/S0094-114X(03)00098-3.

[154]. Lyon, Scott & Evans, Mark & Erickson, Paul & Howell, Larry. (1997). Dynamic Response of Compliant Mechanisms Using the Pseudo-Rigid-Body Model. 10.1115/DETC97/VIB-4177.

[155]. Lyon, Scott & Erickson, Paul & Evans, M. & Howell, Larry. (1999). Prediction of the First Modal Frequency of Compliant Mechanisms Using the Pseudo-Rigid-Body Model. Journal of Mechanical Design - J MECH DESIGN. 121. 10.1115/1.2829459.

[156]. Lobontiu, N., Paine, J. S., Garcia, E., and Goldfarb, M., 2000, "Corner-Filleted Flexure Hinges," ASME J. Mech. Des., 123(3), pp. 346–352.

[157]. Lobontiu, N., 2002, Compliant Mechanisms: Design of Flexure Hinges, CRC Press, Boca Raton, FL.

[158]. Rosner, M., Lammering, R., and Friedrich, R., 2015, "Dynamic Modeling and Model Order Reduction of Compliant Mechanisms," Precis. Eng., 42(5), pp. 85–92.

[159]. Lyon, S. M., Evans, M. S., Erickson, P. A., and Howell, L., 1997, "Dynamic Response of Compliant Mechanisms Using the Pseudo-Rigid Body Model," ASME Paper No. DETC97/VIB-4177.

[160]. Lyon, S., Erickson, P., Evans, M., and Howell, L., 1999, "Prediction of the First Modal Frequency of Compliant Mechanisms Using the Pseudo Rigid-Body Model," ASME J. Mech. Des., 121(2), pp. 309–313.

[161]. Pei, X., Yu, J., Zong, G. H., and Bi, S., 2010, "An Effective Pseudo-RigidBody Method for Beam-Based Compliant Mechanisms," Precis. Eng., 34(3), pp.

634–639.
[162]. Li, Z., and Kota, S., 2002, "Dynamic Analysis of Compliant Mechanisms," ASME Paper No. DETC2002/MECH-34205.
[163]. Zhao, H., Bi, S., and Pan, B., 2015, "Dynamic Analysis and Experiment of a Novel Ultra-Precision Compliant Linear-Motion Mechanism," Precis. Eng., 42, pp. 352–359.
[164]. Yu, Y.-Q., Howell, L. L., Lusk, C., Yue, Y., and He, M.-G., 2005, "Dynamic Modeling of Compliant Mechanisms Based on the Pseudo-Rigid-Body Model," ASME J. Mech. Des., 127(4), pp. 760–765.
[165]. Li, N., Liu, C., and Jiang, H. Y., 2016, "Dynamics Modeling for Curved-Type Fully Compliant Mechanism Based on Pseudo-Rigid-Body Modeling," IEEE International Conference on Mechatronics and Automation (ICMA), Harbin, China, Aug. 7–10, pp. 2302–2307.
[166]. Venkiteswaran, V. K., and Su, H.-J., 2015, "A Parameter Optimization Framework for Determining the Pseudo-Rigid-Body Model of Cantilever-Beams," Precis. Eng., 40, pp. 46–54.
[167]. She, Y., Meng, D.-S., Shi, H.-L., and Su, H.-J., 2015, "Dynamic Modeling of a 2D Compliant Link for Safety Evaluation in Human-Robot Interactions," IEEE International Conference on Intelligent Robots and Systems (IROS), Hamburg, Germany, Sept. 28–Oct. 2, pp. 3759–3764.
[168]. Howell, L. L., and Midha, A., 1995, "Parametric Deflection Approximations for End-Loaded, Large-Deflection Beams in Compliant Mechanisms," ASME J. Mech. Des., 117(1), pp. 156–165.
[169]. Su, H. J., 2009, "A Pseudo Rigid-Body 3R Model for Determining Large Deflection of Cantilever Beams Subject to Tip Loads," ASME J. Mech. Rob., 1(2), p. 021008.
[170]. McInroy, J. E., and Hamann, J. C., 2000, "Design and Control of Flexure Jointed Hexapods," IEEE Trans. Rob. Autom., 16(4), pp. 372–381.
[171]. Na Li, Hai-Jun Su, Xian-Peng Zhang, Accuracy Assessment of Pseudo-RigidBody Model for Dynamic Analysis of Compliant Mechanisms, Journal of Mechanisms and Robotics, October 2017, Vol. 9 / 054503-1