A Survey on Recent Development of Large-Stroke Compliant Micropositioning Stage

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Abstract: Compliant micropositioning mechanism plays an important role in precise positioning and alignment applications. Unlike conventional guiding mechanism, compliant mechanism delivers motion through the deformation of flexible components. As a result, the movement of flexible mechanism is continuous and the structure is more compact, which makes it a hot research topic in the field of precision machine design. As the increase of demands for micropositioning stage providing large motion range and high precision, this paper conducts a state-of-the-art survey on recent development on large-stoke compliant microopsitioning stages. First, a basic introduction of the compliant micropositioning stage is presented. The type synthesis and mechanism structure design of the flexible component are reported, which are the decisive factors determining the stage performance. Then, the design method of flexure hinge is introduced. Afterwards, the development of large-stroke compliant mechanism in recent years is reviewed, and a comparison study of several typical large-stroke compliant micropositioning stage is carried out. The existing issues and future work on the large-stroke compliant micropositioning stage are summarized.

Keywords: Micro/Nano-positioning stage, compliant mechanism, flexure mechanism, large stroke, large workspace, mechanical design.

1. INTRODUCTION

With the development of micro-science and technology, micropositioning system plays an important role in the microelectronics engineering, aerospace technology, biological engineering, and many other fields [1-8]. It directly affects the whole machinery industry technology and technique development. In addition, it determines the quality of machining accuracy and performance of products, and also affects the reliability and stability of the device.

Micropositioning stage is the dominant equipment to achieve precise positioning and displacement [9]. The stage relies on specific mechanical structure to provide displacement. It can be divided into rail and non-rail based positioning stage [10]. The rail-based micro-positioning stage belongs to rigid-body mechanism, which achieves the movement or energy transfer through the kinematic pair (rotation or sliding pair). In contrast, non-rail micropositioning stage is a compliant mechanism, which works based on the elastic deformation of the flexible components [8, 11-18]. As compared with rigid-body mechanism, compliant mechanism has some advantages in terms of easy manufacturing, more compact structure, light weight, and simple assembly. Moreover, it does not exhibit backlash and friction. Hence, compliant mechanism is capable of smooth motion with high resolution.

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Compliant mechanism-based non-rail micropositioning stage is currently a hot research topic. A lot of different flexible displacement mechanisms have been proposed in the literature [12-23]. Without loss of generality, in this work, a compliant mechanism is called a large-stroke micropositioning mechanism if it can provide a stroke large than 1 mm in each working axis. Correspondingly, a compliant mechanism whose stroke is less than 1 mm is called a small-stroke micropositioning mechanism. As far as the type of flexure hinge is concerned, the notch-type hinge is difficult to deliver a large-stroke movement because of the limited flexible deformation. In order to achieve a large stroke of motion, the physical size of the notch hinge-based compliant mechanism is usually large, which will violate the compactness requirement of the mechanical design.

Nowadays, the mechanism based on pseudo-symmetrical flexible structure or quadrilateral flexible structure is widely adopted in the design of compliant mechanism. According to the results of previous analysis, the motion accuracy of such mechanism is generally about 1% of its workspace [54]. In the symmetrical structure, there is no strict compensation for coupling displacement. It just relies on the hypothesis that the same characteristics of the flexible hinge deformation is consistent to ensure that the deformations of flexible hinges are identical. So, such structures usually exhibit a small amount of coupling displacement.

Currently, the design of micropositioning mechanism is more concentrated on the small-stroke

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mechanism. In order to eliminate the coupling motion induced by mechanism deformation, a designer usually employs a redundant symmetric structure. The performance of the designed stage is ultimately determined by the mechanical structure. In contrast, the work on design of micropositioning stages with large stroke (greater than 1 mm) is relatively limited in the literature [24, 25].

2. BASIC DESIGN METHOD OF FLEXURE HINGE

Through the development in the past two decades, compliant mechanism has become an important branch of modern mechanisms and it is widely applied in precision engineering and robotics. Although a series of design methods and theories of compliant mechanism have emerged, its design still poses a great challenge. In this work, a comprehensive overview of design methods is provided for compliant mechanisms used for precision applications, which are usually called flexure mechanisms. This section dictates the background and briefly introduces several main approaches to the design and analysis of flexure mechanisms, including pseudo-rigid-body model [26], structural matrix model [27], constraint-based design [28-31], freedom and constraint topology synthesis based on screw theory [32-35], building block approach [36-39], and structural optimization [11].

2.1. Pseudo-Rigid-Body Model

Pseudo-rigid-body model approach was first proposed by Howell [11]. He performed a variety of basic pseudo-rigid element modeling, and laid a theoretical foundation for the development of pseudo-rigid body model, i.e., using two-link hinge and applied torsion to simulate the deformation of cantilever. By establishing the relations of hinge point and spring constant under different loading conditions, the deformation of the flexible beam with rigid-rod displacement is approximated. Therefore, the motion characteristics of flexible rod is represented by the rigid rod simulation with hinge, whose stiffness characteristics is used to describe the additional spring constant, as shown in Figure **1** [11]. With pseudo-rigid body model, a bridge can be built between the flexible body and rigid bodies, and the reciprocal relationship can be found.

2.2. Structural Matrix Model

Because compliant mechanisms often contain a lot of flexible units or modules, especially the complex space flexible mechanism, their deformation under load is complicated (such as space deformation). So, it is hard to establish a relatively accurate and effective planar pseudo-rigid-body model. Hence, the pseudo-rigid-body model needs to be extended to space mechanism. Inspired by finite-element method, flexible hinge mechanism can be "shaping". It is different from the ordinary rigid body, and also different from distributed flexible mechanism. Thereby, dividing the flexible bodies into pieces, the mechanism unit or module and other parts of the flexible can be treated separately. The former is a multidimensional flexibility of the hinge, while the latter is rigid rod. By spatial coordinate transformation, a space flexibility model can be obtained to represent the overall system. This model is the core to study the performance of various issues such as flexible rigidity, accuracy, etc., as shown in Figure **2** [24].

2.3. Constraint-Based Design

The basic idea of the constraint-based design is to use ideal flexible constraints as the basic unit to

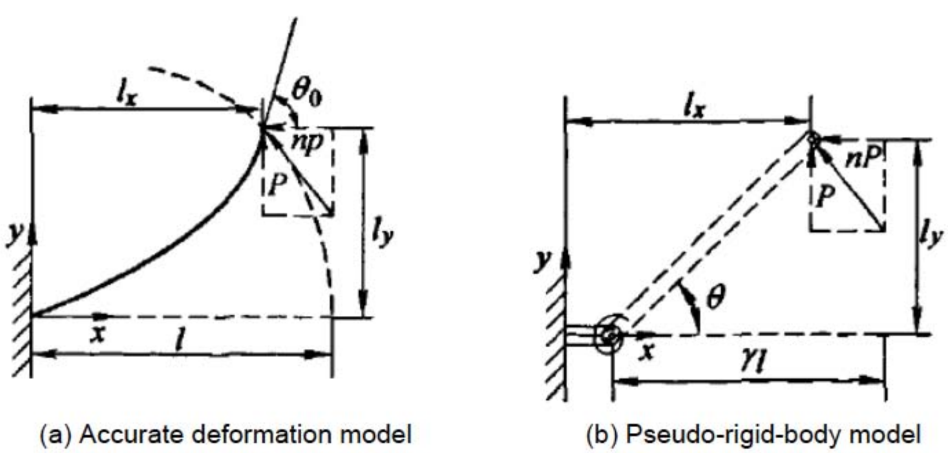


Figure 1: Pseudo-rigid-body model [11].

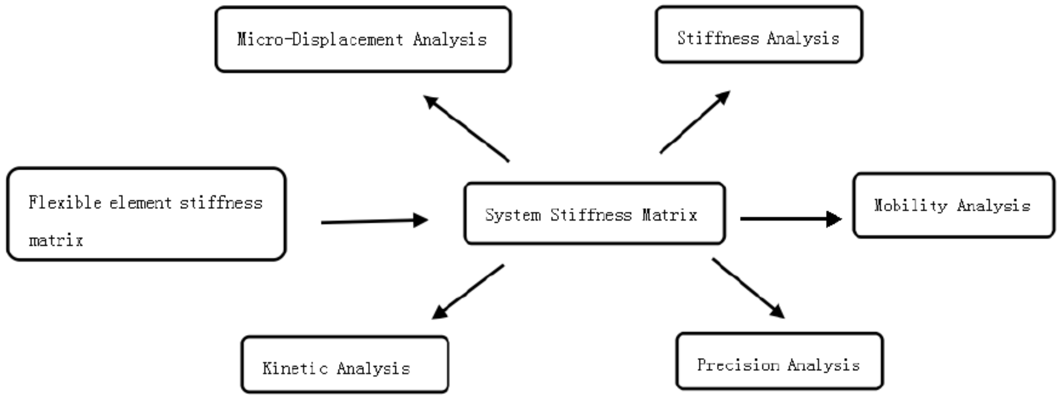


Figure 2: The basic idea of the structure matrix method [24].

construct a flexible mechanism. This idea comes from Maxwell [40], i.e., the duality principle of freedom and constraint (a system that is added to the number of non-redundant constraints and its number of degrees of freedom is six). Blanding [29, 30] has adopted a constraint-based design to synthesize the mechanism. Especially, Blanding proposed that the duality criterion should be followed between the degree-of-freedom (DOF) lines and constraint lines (referred to as Blanding rules). That is, all free lines in the system should be intersected with constraint lines. For example, as shown in Figure **3** [29], the rigid body is constrained by 5 lines (solid line) according to the Blanding rules. It is easy to find a shaft position (dashed line) which allows the rigid body to move. Constraint-based design determines all constrained position and orientation of the mechanism first. Then, it uses the duality criterion of DOF and constraint to determine the motion of the mechanism. Its advantage is that the motion of mechanism can be visualized. Therefore, it is more suitable at the early design stage of flexible mechanism. However, this method is lack of a systematic design process. It is not easy to get an optimal design, and it also requires considerable accumulation of knowledge and experience.

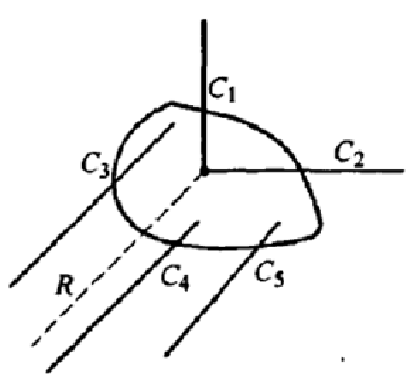


Figure 3: DOF analysis by Blanding rules [29].

2.4. Freedom and Constraint Topology Synthesis Based on Screw Theory

Freedom and constraint topology synthesis based on screw theory [41, 42] is proposed by Hopkins, who was inspired by the Blanding rules. He combined it with the screw theory organically. Then, by the gradual improvement from Su [34] and Yu [35], it has been developed into a systematic flexible mechanism configuration and integrated approach. According to the screw theory, the allowed movement of the object brings a series of specific dimensions of freedom space (FS) and constraint space (CS). FS represents the allowed motion of objects in the space. On the other hand, CS represents the limited allowed motion of objects in the space which is the constraint of the object. These spaces, which are represented geometrically, provide more visual concept. Actually, the relationship between FS and CS can be presented by reciprocal screw system, and the Blanding rules can be seen as a special case of reciprocal screw system. Freedom and constraint topologies can realize the multi-degree of freedom comprehensive configuration for the flexible mechanism. A typical synthesis process is shown in Figure **4** [32]. Under the guidance of screw theory, the resulting configuration also has a certain degree of completeness. However, this method is currently used along with thin rod or plate as a flexible constraint. So, the mechanism may exhibit a larger parasitic movement.

2.5. Building Block Approach

Building block approach [36-39] is to use a flexible element or body, which is known as a module, to achieve the required performance according to certain rules about serial and parallel connection. The modular design concept is firstly described (including performance indicators as required by movement) and

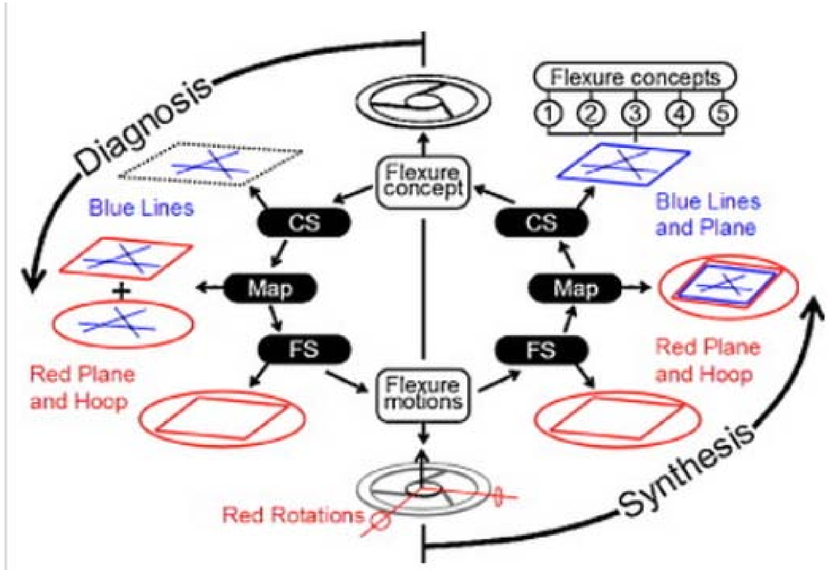


Figure 4: The flowchart for flexible mechanism compounding with FACT method [32].

abstracted into a mathematical or geometric representation (such as freedom of space). Then, it is matched with the modules according to the requirements of the performance of each module in the module library. If it meets the requirements of better performance modules, then it is the end of the process and this module will be the final version. If there is no proper module in module library, then the problem is disassembled, and sub-problems are matched with module library, and so on, until all the modules are found to meet all the sub-problems. Finally, according to certain combination methods, the modules are combined to form the final mechanism. The premise of the method is to first create a module library. The library module must be the flexible element which is aware of the characteristics. As a conceptual design method, using a modular approach can generate better configuration. But this may not converge to optimization process and result in a comprehensive and unsuitable mechanism. The flowchart of flexible mechanism compounding with building block approach is shown in Figure **5** [39].

2.6. Structural Optimization

The method of structure optimization design [11] is a kind of comprehensive method based on integrated configuration and size. Its main idea is based on the required constraints and movement of the flexible

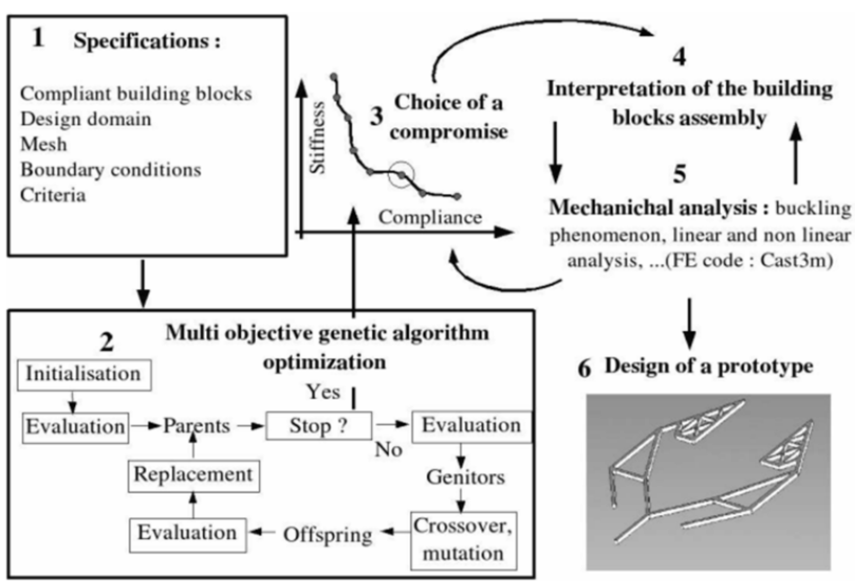


Figure 5: The flowchart for flexible mechanism compounding with building block approach [39].

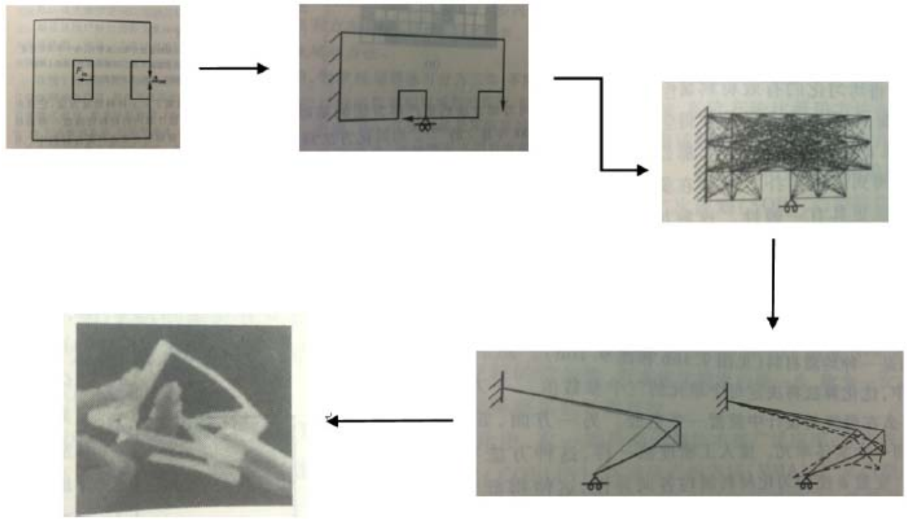


Figure 6: Optimization process [11].

mechanism to design the objective function. Then, under the given boundary conditions, optimization algorithm is used to generate a reasonable mechanism. The optimization process is shown in Figure **6**. Based on the use of optimization algorithms, this method often involves a structure to selectively remove the finite-element mesh to generate the design. Although this method is a systematic integrated approach and the original design can be generated directly from the requirements, it is heavily dependent on the objective function and the optimization algorithm. The objective function may produce invalid result if it is an unreasonable design, and sometimes the generated design is very difficult to manufacture. As a result of numerical algorithms, the experience of designer cannot be used, and the physical meaning is not clear. The method is generally used to design the flat flexible mechanism. It is difficult to achieve multi-axis or spatial movement. Figure **6** [11] shows a typical optimization process.

3. DEVELOPMENT OF LARGE-STROKE COMPLIANT MECHANISM

Owing to its fine performance, compliant mechanism has been widely applied in recent decades. Many researchers designed flexible mechanisms with different properties according to the requirements. This section introduces some representative large-stroke compliant mechanisms.

3.1. Large-Stroke Compliant Mechanism with Heavy Coupling

Based on graph theory, Lin [16, 43] put forward a comprehensive flexible structure of a large-stroke compliant mechanism as shown in Figure **7a**. It is a flexible mechanism based on pseudo-symmetrical structure. Since there is no compensation for the coupling displacement, the structure produces a large amount of displacement in the non-working direction. According to simulation analysis, its working range is 2.34 mm. However, the amount of coupling displacement in non-working direction is almost 390 μm. In addition, it has a low stiffness along the working direction, as shown in Figure **7b**.

3.2. Serial Large-Stroke Flexure Mechanism

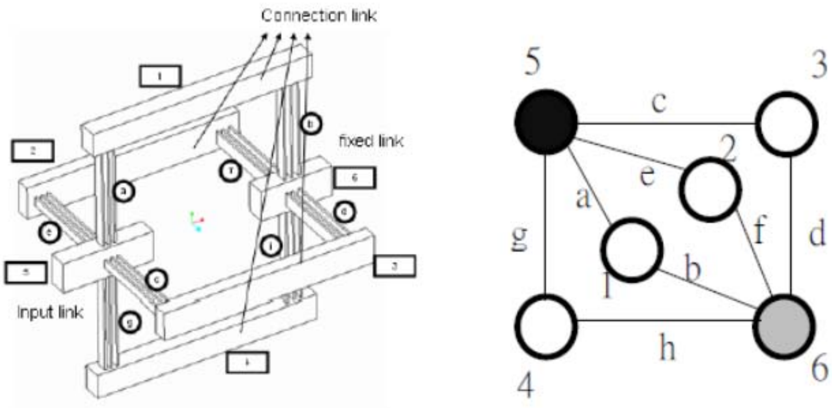
Shan [44] designed a novel serial flexure mechanism based on common right-circular flexure hinge, as shown in Figure **8a**. The pseudo-rigid-body model approach and the theory of mechanics of materials were applied to establish the theoretical model of stiffness and maximum stress (see Figure **8b**). In the case of a series of 6 layers as shown in Figure **8c**, its working stroke can reach 5 mm, and its positioning accuracy in the working range is better than 60 nm.

3.3. Parallel Large-Stroke Flexure Mechanism

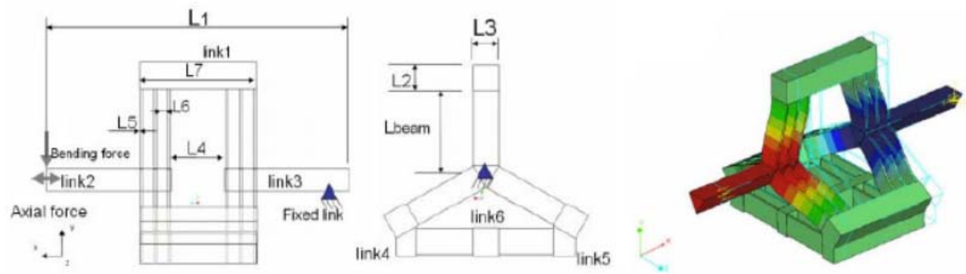
According to the building block approach, Zong [23] designed a displacement stage based on dual-Roberts mechanism working in parallel as shown in Figure **9** [23]. Its working stroke can reach ±2 mm, and the positioning accuracy in the working stroke is better than 1 μm.

3.4. Displacement Amplifier

Wang [45] used uniaxial arc-shaped flexible hinge and amplification principle to achieve a mechanism

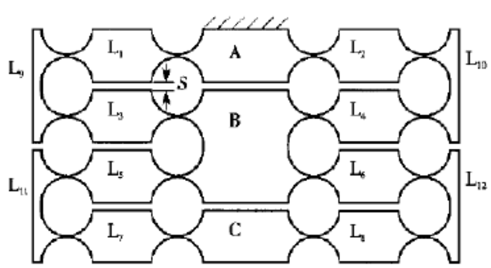


(a) Graph theory

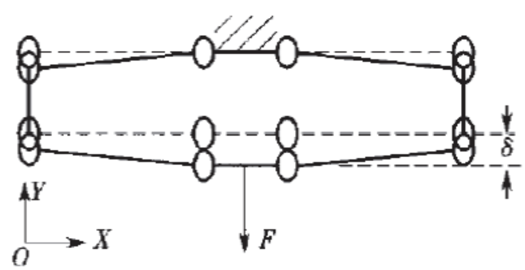


(b) Compliant mechanism

Figure 7: Early large-stroke compliant mechanism [16].



(a) Common right circular flexure hinge hinge



(b) Pseudo-rigid-body model of the flexure

(c) Flexure hinge with a series of 6 layers

Figure 8: Novel series large stroke flexure mechanism [44].

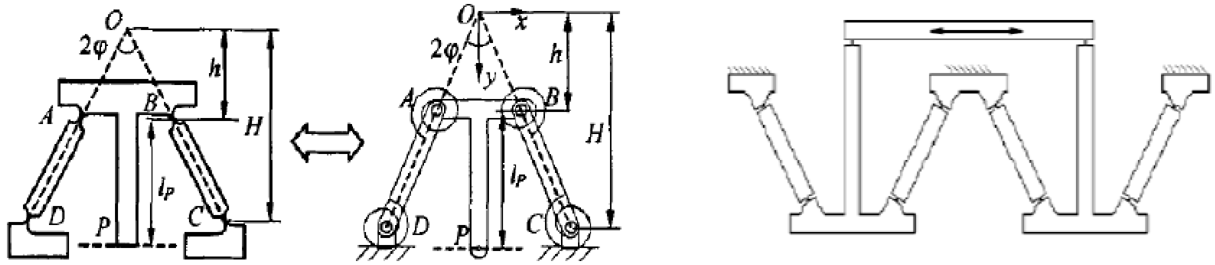


Figure 9: Parallel large stroke flexure mechanism [23].

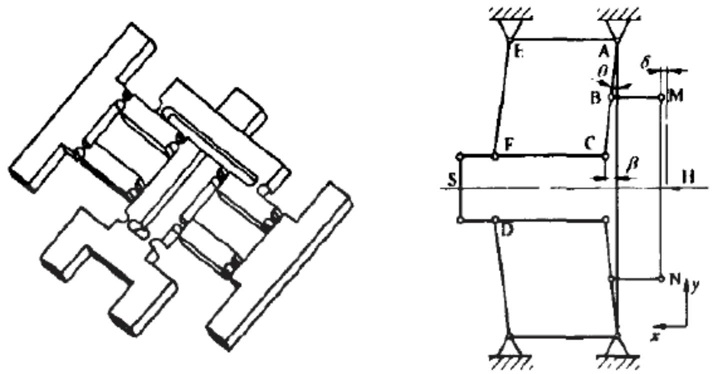


Figure 10: Displacement amplifier [45].

with output displacement amplification, as shown in Figure **10**. It employs an input displacement amplifier which increases an output displacement by 1.8 times. In addition, it eliminates the unnecessary displacement to achieve a precision displacement and positioning.

3.5. Large-Stroke Rotational Compliant Mechanism

In the design of large-stroke rotational compliant mechanism, Zhao [46] designed a new annulus-shaped large-deflection flexure pivot as shown in Figure **11a**. It employs the distributed leafy rotary flexure pivot, which is obtained by distributing curved flexure elements (see Figure **11b**) around a point in plane. It can reach a rotation stroke up to 7.4-degree. The position error is less than 2%, and the rotation error does not exceed 9%.

4. ADVANCED LARGE-STROKE COMPLIANT MICROPOSITIONING STAGE

The demand of compliant micropositioning stage with large motion range and high precision is increasing. This part introduces several kind of advanced large-stroke compliant stage as developed in recent years.

4.1. XY Planar Micropositioning Stage

Many scholars have conducted research on flexible XY compliant stage. Most of the stages are based on planar parallel or double parallel four-bar parallel flexible modules. Flexible unit is used in both centralized compliance model. Two kinds of large-stroke XY compliant stages are introduced below. Both of them have very large work range (20 mm x 20 mm) and relatively compact size, and they exhibits different design methods.

In order to achieve a large pure translational motion in one direction, Xu [47] proposed a new concept of multistage compound parallelogram flexures (MCPF) as shown in Figure **12b**, which is superior to the compound parallelogram flexures (CPF) as shown in Figure **12a**. This concept has been widely used in [48, 49].

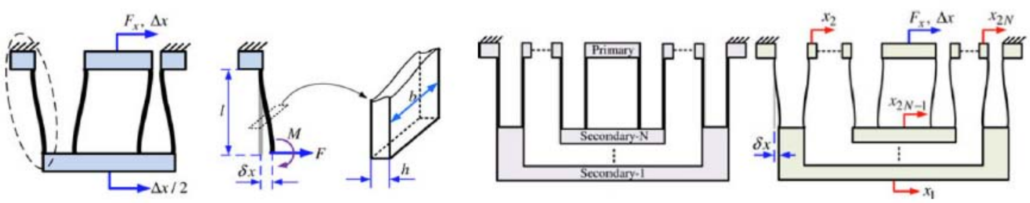
According to MCPF concept, N = 2 is selected to design an XY stage. Then, Xu designed two configurations of the two-stage MCPF constructed with two modules, as shown in Figure **13**.

To avoid the buckling/bending effect as shown in Figure **14a** [47], Xu has added connecting bars to avoid the buckling/bending, as shown in Figure **14b** [47].

Previous work shows that it is challenging to design an XY stage with both a large workspace and a compact physical dimension. To overcome this problem, a more compact stage is invented by Xu [50], as shown in Figure **15**.

(a)Annulus-shaped large-deflection flexure pivot (b) Curved flexure elements

Figure 11: Large stroke of rotational compliant mechanism [46].



(a) CPF (b) MCPF

Figure 12: Multistage compound parallelogram flexures [47].

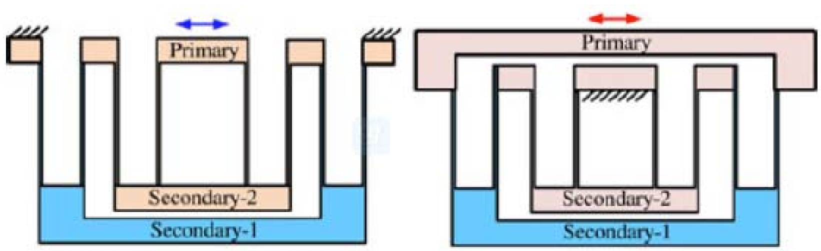
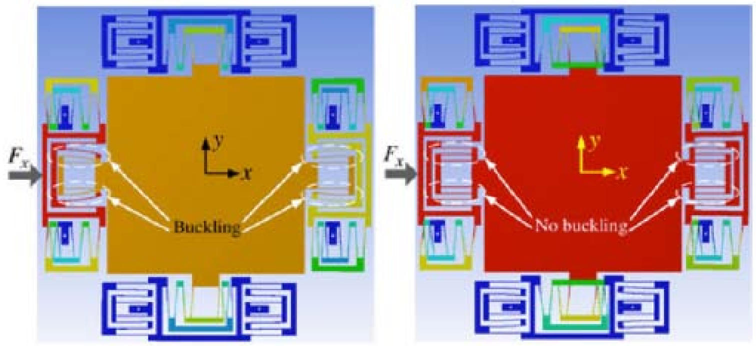


Figure 13: Two-stage MCPF [47].

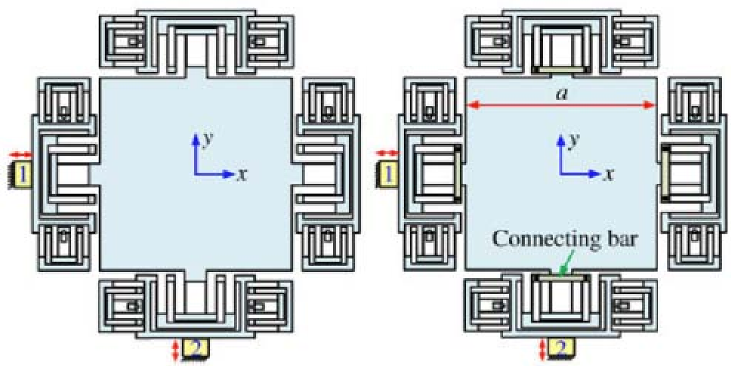
With this design, the FEA simulation results show that a reachable workspace of 20 mm × 20 mm is obtained, and the dimension of the stage is just 120 mm x 120 mm. Moreover, Xu proposed an index of area ratio, i.e., workspace size to planar dimension of the stage, to quantify the compactness of the mechanism. The specific performances of the stage are shown in Table **1**.

In the work [50], the major contribution lies in that a reachable workspace of 20 mm X 20 mm is obtained, which indicates a high area-ratio of 2.7778%. Due to the hardware constraint, a workspace range of 11.75 mm x 11.66 mm is generated. It corresponds to an area ratio of 0.9514%.

Another design is made by Li [51]. At first, he built a kinematics model which called 4-PP&1-E (see Figure **16a**) based on [52]. Based on the configuration of the adopted 4-PP kinematics, its redundant constraint unit E (see Figure **16g**) improves the rigidity of non-functional direction of movement. Then, according to the design in [48] (see Figure **16b** and **16c**), he replaced the gap-type flexible hinge with the distributed flexible reed (see Figure **16d** and **16e**), and got the flexible unit PI and PII. PI and PII were replaced with the kinematic pair outside and inside the kinematic model, and one of the two adjacent PI is selected for drive. The PI and PII have consistent stroke. So, the reed needs the same thickness T and length L. At the same time, this design also allows the driven unit PII to have a smaller stiffness than the active unit PI. At last, the kinematic pairs PI and PII and the planar kinematic pair E (which come from the design of Hopkins and Culpepper [53] based on freedom and constraint topology synthesis, as shown in Figure **16f**) are put into the 4-PP&1-E kinematic model. After that, compliance matrix model is established, and the parameter of the mechanism is determined to get the whole stage (see Figure **16h**).



(a) Buckling/Bending effect



(b) Connecting bars

Figure 14: Connecting bars to avoid buckling [47].

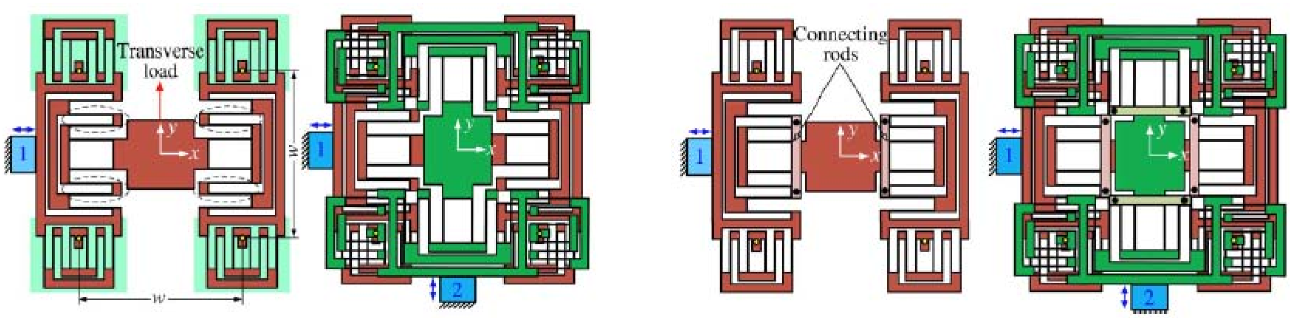


Figure 15: Compact XY stage [47].

Table 1: Specific Performance of the XY Stage in [50]

	Performance	
Parameter	**X**	**Y**
Dimension	120 mm	120 mm
Resonant frequency	29.3 Hz	29.6 Hz
Motion range	11.75 mm	11.66 mm
Accuracy	338 nm	328 nm

Although this design achieves a large work range in two direction (20mm x 20mm), but the area ratio ($\frac{20mm \times 20mm}{432mm \times 432mm} = 0.214\%$) is not as good as the design in [50]. The specific performances of the stage are shown in Table **2**.

In the work [51], the designer did not use the pseudo-rigid-body model for modeling, but established a compliance matrix to model the flexible micro-positioning stage, and used it to determine the parameters of the XY stage.

4.2. Large-Stroke Multi-DOF Compliant Micropositioning Stage

In the design [55], the authors used a compliant amplifier to amplify the work range and achieved

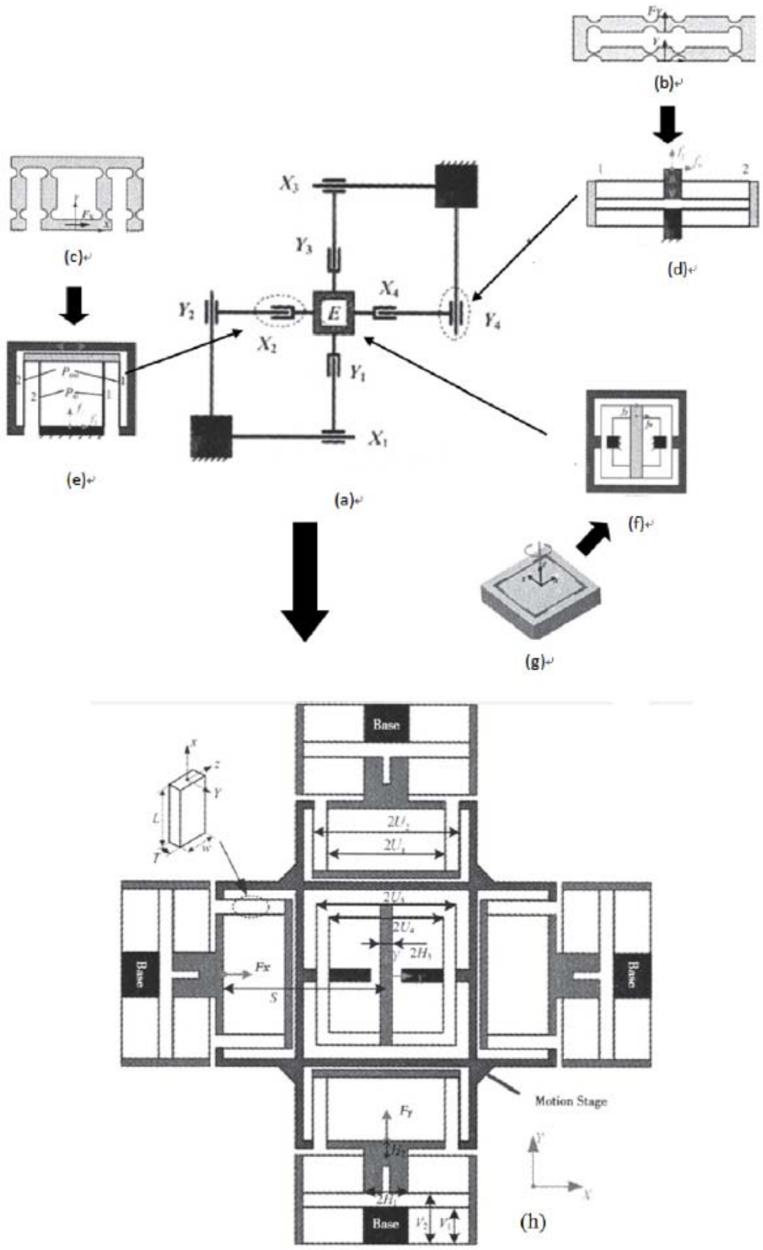


Figure 16: Flow diagram of the stage configuration [51].

Table 2: Specific Performance of the XY Stage in [51]

	Performance	
Parameter	**X**	**Y**
Dimension	432mm	120mm
Resonant frequency	-	-
Motion range	20mm	20mm
Accuracy	0.12%	

3-DOF (X- and Y-axes and rotation about the Z-axis) movement. It also has a small and compact size (142 mm x 110 mm). In this design, a new kind of hinge which displays excellent loading behavior and lowered stress concentrations [55-57] was used. In addition, the mechanism design employs the primary use of cantilever mechanism to achieve the 3-DOF motion. The cantilever sections allow the motion of the TCP on the central platform in the X-direction to be decoupled from motion in the Y- and θ-directions. Cantilever sections transfer motions orthogonal to their axis while resisting elongation along their axis. This decouples the motion in the X-direction from those in Y/θ-directions. The symmetry of the mechanism will help in further reducing elongation, that could result in the reduction of cross-axis coupling. As a result, the mechanism topology is designed such that the mechanism

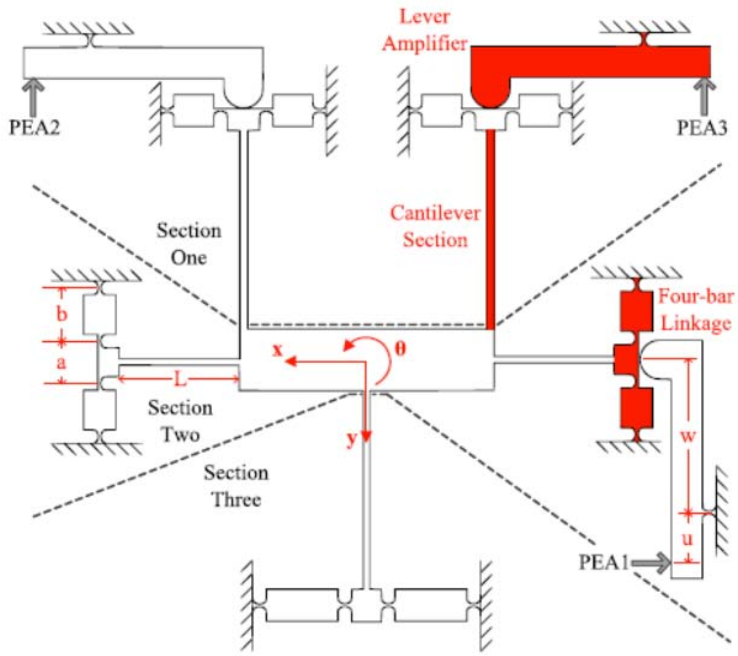


Figure 17: The construction of a stage [55].

kinematics is a simple linear function of piezoelectric actuator positions, and no linearization of Jacobian is required.

As seen in Figure **17**, the mechanism consists of two distinct components: a mainframe, and three actuators, each of which connects to an amplification lever. Piezoelectric actuators (PEAs) are the preferred choice because they provide continuous motion with infinite resolution, and they are free of backlash and stick-slip friction [58, 59]. Furthermore, PEAs have favorable dynamic performance due to their high stiffness and high resonant frequency. Hence, three piezoelectric actuators (PEAs) are employed to produce the required input displacements. Each PEA is capable of 11.6-μm displacement under continuous operation. PEA1 achieves displacement in the X-direction, while PEA2 and PEA3 achieve the motion in Y-direction and rotation about Z-axis, as shown in Figure **18**. All the used hinges in the mechanism have

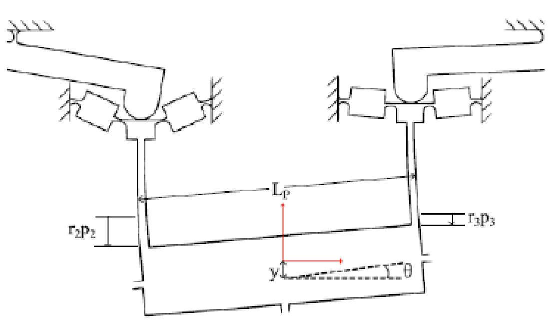


Figure 18: The rotation of z-axis [55].

been designed utilizing filleted leaf flexure geometries. The filleted leaf flexures are highly compliant and induce low stress, which is desired while at the cost of decreased rotational precision [60]. However, this reduction is mitigated because of the symmetric design of the mechanism. The specific performances of the stage are shown in Table **3**.

Table 3: Specific Performance of the 3-DOF Stage in [55]

	Performance		
Parameter	**X**	**Y**	**θ_Z**
Dimension	100 mm	100 mm	-
Resonant frequency	745.6 Hz	1161.9 Hz	1494 Hz
Motion range	44.6 μm	35.6 μm	0.88 mrad
Accuracy	0.5%	0.32%	-

The main point of the work [55] is that, although the PZT is applied to actuate stage, amplifications is applied to enhance working range, so the amplification make sure that the stage still have a relatively big working range. In addition, by controlling the three amplifiers simultaneously, this design can get a rotational DOF around Z-axis.

Another design comes from Tang and Chen [48], a 3-PPP structure with three decoupled translational motions and a workspace of more than 1 mm x 1 mm x 1 mm is designed as an example. Dimension optimization based on pseudo-rigid-body model is conducted. According to the screw theory, for a parallel manipulator, the end-effector is constrained by several limbs, i.e., its DOF is determined by the intersection of all the twists. Thus, the motion of the end-effector is the intersection of the three identical limbs, which is expected to be the three translational motions decoupled each another. The possible structures of the limbs are shown in Table **4** and Figure **19**.

Table 4: Possible Structures of the Limbs [48]

Limb type	Geometric conditions
PPP	Every two axes of the prismatic joints are perpendicular each other.
PRRR	The axes of the three revolute joints are parallel and not in a plane. The axis of the prismatic joint is parallel with those of the revolute joints.
PPRRR PRRRP	The axes of the three revolute joints are parallel and not in a plane. The axis of one prismatic joint is parallel with those of the revolute joints. The axes of the two prismatic joints are perpendicular.

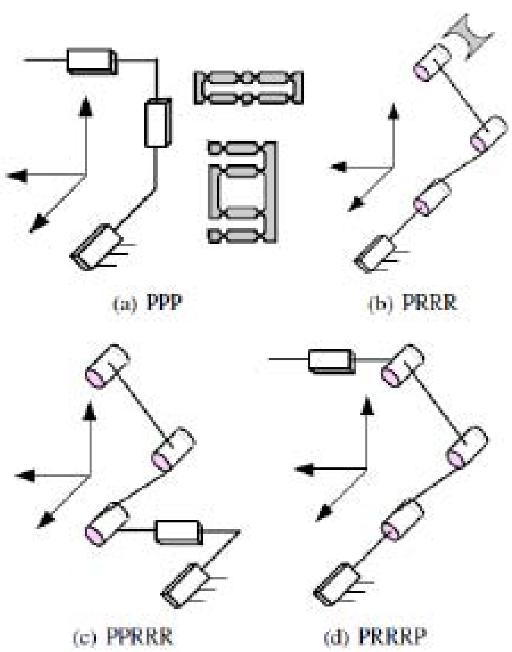


Figure 19: Models of different limbs [48].

The DOFs of 3-PPRRR and 3-PRRRP structures are determined by the following expression.

$$(XYZ\theta_X) \cap (XYZ\theta_Y) \cap (XYZ\theta_Z) = (XYZ)$$

In the work [48], a 3-PPP structure is developed as an example of decoupled large-displacement XYZ flexure parallel mechanism (FPM). The XYZ FPM is desired to achieve a displacement more than 1 mm along three axes, and the three-axis translational motions are required to be decoupled. Therefore, large-displacement prismatic joints are used to configure the XYZ stage. XYZ FPM is designed to be a parallel combination of three orthogonal limbs. Each limb has three DOFs along the X, Y and Z-axes, and it has a serial structure combining two large-displacement prismatic joints (see Figure **20**). The 3D diagram of XYZ FPM is shown in Figure **21**. In Figure **21**, U and W represent two large-displacement prismatic joints and Vi represents the large-displacement prismatic joint.

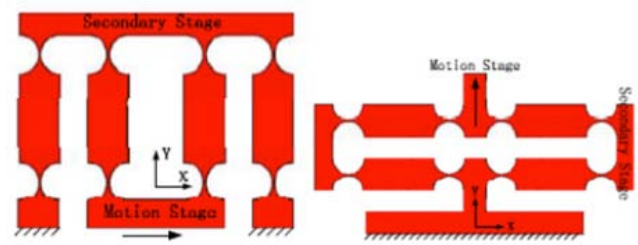


Figure 20: A new large-displacement prismatic joint [48].

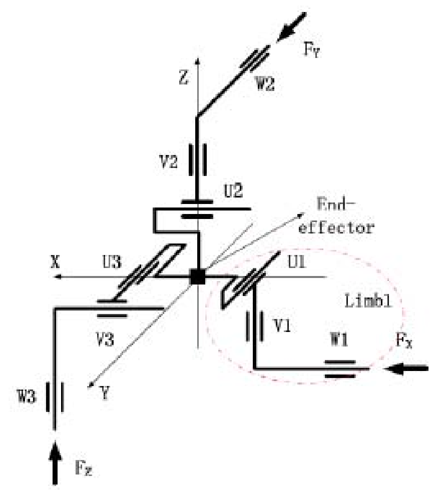


Figure 21: 3D diagram of XYZ FPM [48].

The working principle can be explained based on Figure **21**. When the force FX is directly applied on W1 joint, W1 joint deforms to generate the X-axis motion and transfers the force to U1 and V1 joints. Theoretically, U1 joint and V1 joint move along X-axis without deformation, and directly transfer the force to U2 and V3 joints through the end-effector. At the same time, U2 and V3 joints deflect to create the X-axis motion. Therefore, the end-effector can move along X-axis. Briefly, when FX is exerted, W1, U2 and V3 joints deflect to generate the X-axis motion, U1 and V1 joints move the same displacement as the end-effector without deformation, and other prismatic joints remain stationary. The motions along the Y- and Z-axes are identical. The prototype of the XYZ-FPM has been manufactured as shown in Figure **22**. The specific performances of the XYZ stage are shown in Table **5**.

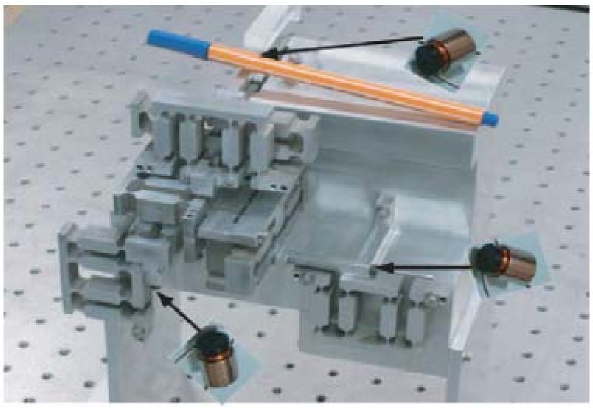


Figure 22: The prototype of the XYZ-FPM [48].

Table 5: Specific Performance of the XYZ Stage in [48]

	Performance		
Parameter	**X**	**Y**	**Z**
Dimension	150mm	120mm	120mm
Resonant frequency	56Hz	56Hz	56Hz
Motion range	1mm	1mm	1mm
Accuracy	1.9%		

The major point of work [48] is that, in order to overcome the stiffening phenomenon which can reduce the motion range significantly and is prone to occur in the prismatic joint, a new kind of large-displacement prismatic joint (shown in Figure **20**) based on screw theory is designed. Moreover, this is one of the few 3-DOF compliant micropositioning stages which consist of prismatic joint and drive by VCM motor directly.

In the design of Dong [54], a new type of spherical kinematic pair flexible hinge was used to provide a large motion range. The design process of this kind of flexible hinge design is shown in Figure **23**. The common spherical kinematic-pair flexible hinge is elongated. For the convenience of machining and establishing the stiffness model of a flexible hinge, the rounded corners instead of right angle on both ends are used. After the above treatment, in fact, the elongate shaft is still difficult to machining. So, assembly method was used. After the design process as described above, such large-stroke flexible hinge design not only can achieve a high precision movement, but also can provide a millimeter-level motion stroke.

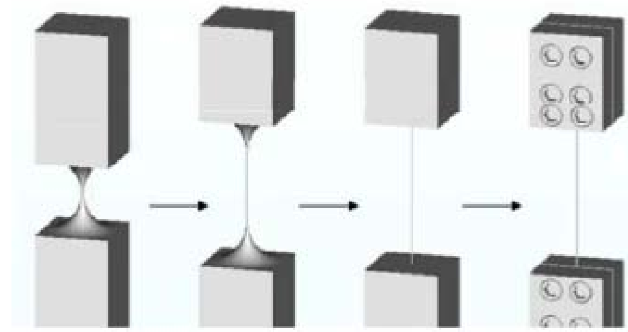


Figure 23: The process of the flexible hinge design [54].

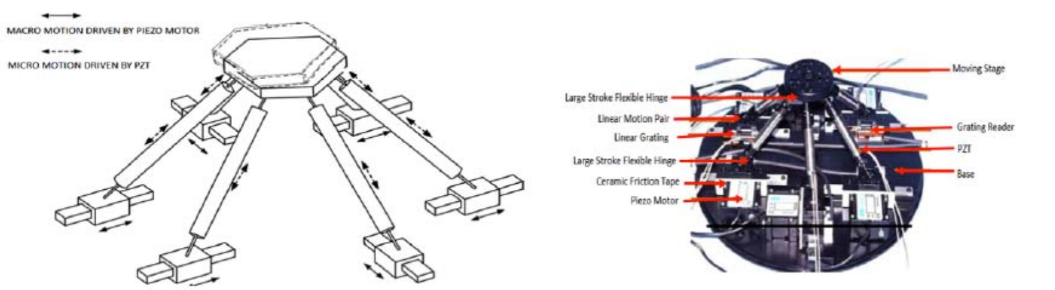
Based on the design of large flexible hinge, Dong proposed a 6-PSS large stroke flexible hinge parallel structure as shown Figure **24b**. The structure consists of six chains. Each chain is assembled as shown in Figure **24a**, which includes large-stroke flexible hinge, rigid rod and fastening device. The translation pairs P (rail slider group) is fixed on the base as active joints, and it is driven by a linear motor. The spherical kinematic pairs S (large stroke flexible hinge) act as passive joints, which eliminate the gap error of the spherical kinematic pair in parallel structure. As compared with the general micro-motion stage, it can provide a larger workspace.

The aforementioned parallel robot system based on large-stroke flexure hinge can be considered a 6-PSS structure. However, only with a single piezo motor drive, the performance of the stage cannot make further improvement. So, the PZT was embedded in the chain and the whole stage becomes a parallel 6-SPS structure. The movement of the stage can be further adjusted. The overall structure employs the macro and micro parallel dual-drive, which consists of 6-PSS structure and 6-SPS structure (see Figure **25a**). The hardware components of the system are shown in Figure **25b**. The specific performances of the stage are shown in Table **6**.

In this work [54], the most important feature is that, this compliant micropositioning stage has six DOFs. Furthermore, for purpose of enhancing the work range, the macro and micro parallel dual-drive is applied to drive the stage. In this new driving style, the linear displacement in each direction of the system can reach 10 mm, the rotation in each direction can reach 6°, and the resolution of the system is up to 3 nm.

(a) Large stroke flexible hinge Hinge (b) 6-PSS Large Stroke Flexible

Figure 24: 6-PSS large stroke flexible hinge parallel structure [54].



(a) 6-PSS structure and 6-SPS structure (b) Hardware components of the system

Figure 25: Construction and hardware components of the system [54].

Table 6: Specific Performance of the 6-DOF Stage in [54]

	Performance					
Parameter	**X**	**Y**	**Z**	**X(Θ)**	**Y(Θ)**	**Z(Θ)**
Dimension	90 mm	90 mm	61 mm			
Resonant frequency	-					
Motion range	10 mm	10 mm	10 mm	6°	6°	6°
Accuray	9.9 nm	8.2 nm	6.9 nm	0.18μrad	0.18μrad	0.18μrad

5. CONCLUSION AND FUTURE WORK

Compliant mechanism can act as a high-precision micropositioning stage. As compared with rigid-body mechanism, compliant mechanism has the merits of easy manufacturing, light weight, and simple assembly. Flexible displacement mechanism is an important branch of flexible mechanism design, and it is valuable in precision positioning fields. In order to cater for the needs of industry, the large-stroke compliant micropositioning stage becomes a hot research topic in the field of precision machine design. This article introduces the current development status of large-stroke compliant micropositioning stage from different aspects.

Through the review, it is found that a great progress in the research on large-stroke compliant micropositioning stage have been achieved by researchers. However, there are still some issues which require further research. These issues are listed below.

1. To optimize the design of flexible mechanism such as light weight design and miniaturization design, so as to get a more optimized and compact structure on the premise of meeting the performance requirements.

2. Synthesis theory of flexible mechanism needs a more in-depth research to find a better method

for combining the flexible mechanism design and rigid-body design, so as to provide a calculation method for the combined design.

3. Since most of the compliant mechanism design is an integrated process, new design method to avoid the parasitic motion and achieve a decoupled design is another promising research direction.

4. Because most design methods of compliant mechanism are complicated, a more standardized and efficient design method, which just likes the design method as used in traditional mechanical design is needed.

5. In order to ensure practicality and stability of flexible bodies, more fatigue test for flexible mechanism, and the service life study need to be conducted.

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