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A Review on the Flexure-Based Displacement Amplification Mechanisms

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ABSTRACT Multifarious flexure-based displacement amplifiers have been proposed and studied in the past decades, showing importance in many industrial fields, such as bioengineering, optical instruments, and semiconductor technology. Displacement amplifiers provide precise motion and large stroke through a simple and low-cost way compared with other piezoelectric actuators. Those merits have opened a door for new and advanced micro devices with unprecedented performance. This paper aims to proffer a comprehensive review on the design, modeling, characteristics, and applications of flexure-based displacement amplifiers, following by pointing out the inherent drawbacks in this research area such as amplification ratio limit, parasitic motion, low lateral stiffness, low natural frequency, and discussing existing solutions and some potential research directions in those topics. Finally, a summary is concluded and the future development perspectives of the displacement amplifiers are discussed. This review contributes to giving a comprehensive understanding of the displacement amplifier, which provides guidance on designing new displacement amplifiers for improving their mechanical output performance. It is also expected to be instrumental for related researchers to understand displacement amplifiers, and to successfully select and design for specific applications.

INDEX TERMS Compliant mechanisms, displacement amplifiers, modeling, applications.

I. INTRODUCTION

Ultra-precision actuation technology plays a vital role in modern cutting-edge precision equipments such as lithography, micromanipulation robot, optical instrument, and space telescope [1]–[9]. Traditional actuators, as DC/AC motor and hydraulic motor etc., perform well in driving force and stroke, exclusive of high positioning precision due to their actuation principles. Piezo-stack actuator (PSA) is capable of fine positioning accuracy, but its short stroke that is about 0.1% of material length restrains the application. Piezo-motor is an alternative precision actuator that is able to achieve high accuracy and large stroke simultaneously [10]–[14]. However, they are not preferred in some special application such as space environment where friction loss is not allowed due to cold welding phenomenon.

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Flexure-based displacement amplification mechanisms (FDAMs) provide another solution for ultra-precision actuation, making a tradeoff among travel range, system complexity, and manufacturing cost. FDAMs enhance the stroke of the PSAs via mechanism amplification effect. The flexure hinge that exhibits merits, such as no friction, zero backlash that maintains the high precision, and the minimal requirement of assembly that reduces the system cost. To date, FDAMs have been employed in many precision engineering fields, for instance, laser scanning, cell micromanipulation, micro-pump, and jet dispensing, etc [15]–[19]. According to the amplification principle, the widely-used FDAMs can be divided into two types, i.e., lever-principle based amplifiers and triangular-principle based amplifiers. This review comprehensively presents the two types amplifiers involving modeling, design, characteristics, and application, among which the inherent drawbacks of the FDAM, such as amplification ratio limit, parasitic motion, low lateral stiffness, and low natural frequency are discussed in detail. The existing solutions and efforts for those issues are also presented. Particularly, some issues such as intrinsic connection between the two types amplifiers and parasitic motion are concluded in new perspectives, and the system stiffness design methodology that helps to improve the working performance of the displacement amplifier is systematically described first.

The review summarizes the classical FDAMs in the past three decades and advanced new works in recent years in order to offer guidance to designers on designing new FDAMs. It is also expected to be helpful for related researchers to understand FDAMs and to successfully select and design for specific applications.

The remainder of this paper is organized as follows: in Section 2, the progress on mathematic modeling of FDAMs is summarized, based on which the intrinsic connection between the amplifiers with different amplification principles is discussed. Section 3 discusses the main approach for amplification ratio increasing, where the system stiffness design methodology is systematically introduced first. In Section 4, the inherent drawbacks of the FDAMs are discussed and the corresponding solutions are concluded. Section 5 shows the applications of the FDAMs. Finally, conclusions are drawn in Section 6.

II. MODELING OF FLEXURE-BASED DISPLACEMENT AMPLIFIERS

Theoretic modeling is a powerful mathematic tool for the mechanism analysis and synthesis, more effective than software based finite element method (FEM) when the optimization design with voluminous calculations is proceeded. Therefore, plenty of works concerning the FDAMs' modeling is reported in the past three decades. This review summarizes those models into three categories, i.e., geometric model, linear model, and nonlinear model. Geometric model focuses on the geometric relation of the mechanism but ignores the properties of the flexure hinge that is replaced by an ideal pivot in the modeling. Thus, geometric model is straightforward but excessively simplified. Nonlinear model improves the accuracy of the model via considering the flexure hinge as a nonlinear elastic element. It performs well in modeling accuracy but the calculation efficiency is compromised by massive calculation resulted from the absence of closed-form solution. Linear model makes a tradeoff between the accuracy and the complexity via simplifying the flexure hinge as a linear elastic element, making it the most popular in research and industries. In this section, three types of models are summarized based on mathematic viewpoints, implementations, and characteristics to help researchers to understand and select a suitable one for a specific case.

A. CLASSIFICATION OF DISPLACEMENT AMPLIFIERS

Before discussing the modeling, the classification of the FDAMs should be introduced. In terms as amplification principle, most of the FDAMs in spite of different structures can be divided into two categories: lever-type amplification mechanism (LAM) and triangular-type amplification mechanism (TAM). Lever amplification principle is a classical and well-known mechanical law. Archimedes, a famous physicist in ancient times, said "Give me a lever long enough and a fulcrum on which to place it, and I shall move the world". The characteristics of the LAM, such as flexible structure and less displacement loss (displacement loss will be systematically introduced in next section), make it easy to construct a multistage amplifier to achieve large amplification ratio. As for the triangular amplification principle, it utilizes the motion of the Scott-Russell at its singular point where the output displacement is larger than the input displacement. Diverse displacement amplifiers are proposed according to this simple principle, among which the bridge-type amplification mechanism (BAM) is the most extensively used due to its compact symmetric structure. BAM will be introduced in detail in the review. Besides, the hybrid-type amplifier that combines two amplification principles can be found in recent research, which exhibits unique performances. FIGURE 1 illustrates the classification of the FDAMs.

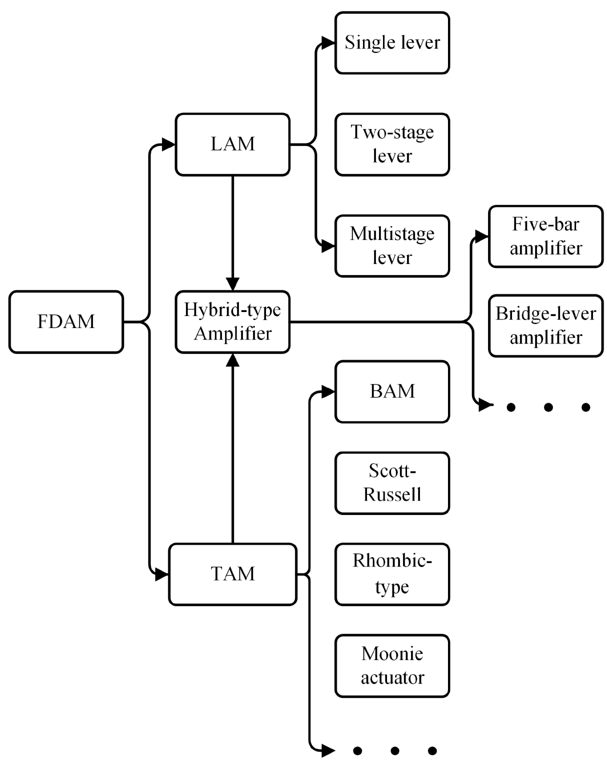

FIGURE 1. Classification of the FDAMs.

B. GEOMETRIC MODEL

Assume that:

1) The flexure hinge provides pure rotation motion around a fixed axis with zero stiffness.
2) Other components are considered as rigid bodies.

The amplification ratio of a LAM can be obtained via lever law, which is the ratio of the long beam to the short beam,

$$Am = \frac{l_b}{l_a}. \tag{1}$$

FIGURE 2 (a) presents the schematic of the LAM.

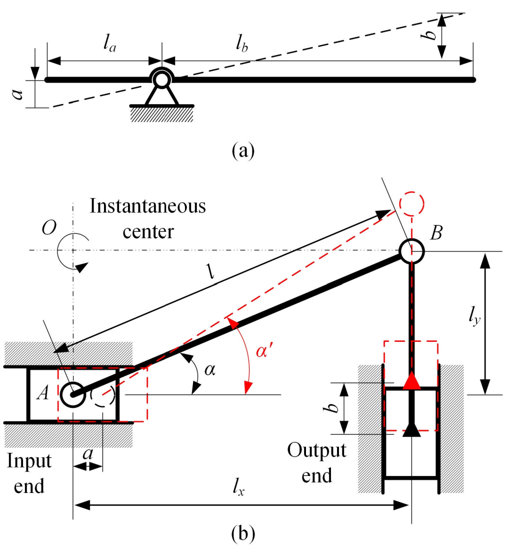

FIGURE 2. Schematic of the amplification principle: (a) Lever amplification principle; (b) Triangular amplification principle.

FIGURE 2 (b) presents a schematic of the TAM. Pokines [20] derived the BAM's geometric model that is related to the bridge arm angle α and α'. Since α' is difficult to be obtained, Lobontiu and Garcia [21] improved the formula via the relations between the projection of link l and displacements a and b. Consequently variable α' is eliminated. The expression demonstrates that the amplification ratio is a nonlinear function of the input a. Ma *et al.* [22] further simplified the equation by applying kinematic theory. A more concise formula is obtained by

$$Am = \frac{l_x}{l_y} = \cot(\alpha). \tag{2}$$

It is interesting that Eq. (1) and Eq. (2) are similar, which implies that LAM and TAM have an intrinsic connection. If the instantaneous center in FIGURE 2 (b) is an actual joint and triangle OAB is a rigid body, the TAM translates to LAM, and vice versa. From this point of view, the LAM can be considered as a special case of the TAM whose instantaneous center is fixed at the link l.

Although the geometric model can't predict the amplification ratio accurately due to the oversimplification, it is still significant. Geometric model provides a straightforward way to estimate the amplification ratio at the initial design phase. Besides, geometric amplification ratio indicates the maximum value that the FDAM can achieve, which is important for the evaluation of amplification performance.

C. LINEAR MODEL

To enhance the model, the property of the flexure hinge should be considered. In general, the deformation of a flexure hinge satisfies the assumption of small deformation, thus the nonlinearity between the force and displacement can be ignored in most cases.
Assume that:

1) The flexure hinge satisfies the assumption of small deformation.
2) Ignore all kinds of nonlinearities including geometry factor and material factor.

The models under the two assumptions are called linear model in the review. In this section, three extensively used linear models are discussed, including Castigliano's second theorem, elastic beam theory, and compliance matrix method. Besides matrix displacement method (linear finite element method) and pseudo-rigid-body-method that are rarely used for FDAM modeling are introduced briefly.

Castigliano's second theorem is an energy-based approach for elastic element modeling. The core idea of this method is that the displacement of the deformed elements equals the first-order differential of the total strain energy with respect to the corresponding external force. The main procedures of this method can be explained by two equations,

$$U = U_a + U_s + U_b \tag{3}$$

$$\delta_x = \frac{\partial U}{\partial F_x}; \quad \delta_y = \frac{\partial U}{\partial F_y}; \quad \theta_z = \frac{\partial U}{\partial M_z} \tag{4}$$

where U_a, U_s, and U_b are the strain energy result from tensile force, shear force, and bending moment respectively. δ and F are the displacement and force. The subscript represents the direction.

Lobontiu and Garcia [21] provided a precedent that formulates the statics of a BAM by Castigliano's second theorem. The amplification ratio, input stiffness, and output stiffness were established and the influence of the structure parameters on the amplification ratio was analyzed. Thenceforth, Castigliano's second theorem draws attention for the displacement amplifier analyzing. Chen *et al.* [23] employed this versatile approach to describe the statics performance of the orthogonal displacement amplifier and Zhu *et al.* [24] utilized it for both statics and dynamics model establishing. Besides, Chen *et al.* [17] performed the analysis of the rhombus-type amplifier via the Castigliano's second theorem. It is easy to find that the modeling process presented in those references is convoluted since numerous inner-forces are required to analyze though the configurations they dealt with are simple. Some researchers noticed this issue and attempted to give their solutions. Ueda [25]–[27] introduced the two-port concept in the Castigliano's modeling process to simplify the modeling procedures, which is well-suited for the nested multistage amplifier. Wu *et al.* [28] integrated the screw theory with the energy method to formulate a general kinetostatic model of compliant mechanisms, which regulates the form of the derived formulas and brings conciseness. Even though, Castigliano's second method is not a mainstream approach of FDAMs' modeling due to its inherent drawback. On the contrary, it plays a more important role in the guiding flexure beams [29]–[31] and notch-type flexure hinges [32]–[34].

Elastic beam theory is a more frequently used theory for FDAMs' modeling [35]–[38]. The main idea of this approach is to establish equations via equilibrium conditions of forces, constraint analysis, or energy conservation law, etc., and solve out the concerned parameters according to the force-deflection relations based on elastic beam theory. Ma *et al.* [22] derived the analytical formula of a BAM for amplification ratio analyzing, by which the extremum phenomenon and its vertex were predicted. Qi *et al.* [39] progressed this model via introducing the analysis of flexure hinge rotation. The form of the eventual formula is simplified as a proper fraction when the stiffness coefficient of the flexure hinge is considered in the model. Liu and Yan [40] expanded the application scope of the model by considering the load force in the model. Although the advancements bring concise formulas for BAM amplification ratio, the usable range is restrained to the traditional BAM with leaf-filleted flexure hinges. Ling *et al.* [41] balanced the conciseness and the application range of the model, and combined the energy conservation law and the elastic beam theory to establish an enhanced model. The rhombus-type and bridge-type mechanisms are discernible based on the distinct force analysis. Besides, the general analytical equations that are applicable to diverse BAMs draw attentions from Wei *et al.* [42], Mottard and St-Amant [43] and Choi *et al.* [44], Those models significantly contribute to the theoretical analysis and structure design of the compliant mechanisms where BAMs are used [45]–[48]. However, convoluted inner-force analysis hinders its further development since it is too individualized to each structure, which is disadvantageous to universal formula. This challenge is crucial when new mechanisms are investigated.

Compliance matrix method is a promising approach specially in ordering formulas and standardizing via matrix operating [49]–[51]. The force-displacement relation of the flexure hinge is obtained via an accumulating operation in the global coordination, where the vectors are transferred from local coordination of the flexible elements. The compliance matrix method can be expressed by a general formula as,

$$\varepsilon_c = \sum_{k}^{l} \sum_{i=1}^{m} \left(\sum_{j}^{n} \left[T_{mj}\right]^T \left[R_{mj}\right]^T C \left[R_{mj}\right] \left[T_{mj}\right] \right) F_m \quad (5)$$

where ε and F_m represent the deformation and the external force of the mechanism; l, m, and n are the number of the chains, the external forces, and the correlative flexible elements.

Nicolae systematically studied the application of the compliance matrix method on the quasi-static response of the planar compliant mechanisms that are divided into serial type [52] and parallel type [53]. The modeling procedures are unified and the formulas are neat no matter what kind of structures are modeled. Wang *et al.* [54] further progressed the matrix model that performs well in the case of arbitrary load applied. Based on those researches, authors [55] analyzed and compared BAMs and LAMs via the compliant matrix method, where some significant conclusions about the two amplifiers are revealed.

In the past decade, screw theory that is a powerful mathematic tool for matrix transforming was frequently used in compliance matrix approach. Rouhani *et al.* [56] employed screw theory to model the inverse kinematics of a micro-hexapod manipulator. Compared with the traditional model that simplifies the flexure hinge as an ideal joint, this model is more accurate since the joint's rotation center's changing is considered. Besides, the screw theory was also used in [57], [58]. Essentially, screw theory based compliance matrix method is fully identical to the traditional one. The screw theory displays its actual superiority in the structural synthesis of parallel robots [59]–[61].

Compliance matrix approach seems a perfect solution for the theoretical modeling of the FDAMs. However, the accuracy can not satisfy every cases. Matrix displacement method is an enhanced matrix-based approach characterized by excellent accuracy [62], [63]. Furthermore, the modeling is appropriate for programming and is easy to realize automatically modeling [64]. Matrix displacement method is a versatile but time-consuming method due to its unfriendliness for new structures [65], [66].

The modeling approaches mentioned above are linear model, used in the small deflection cases. Pseudo-rigid-body model is such an approach that satisfies the linear model characteristic but is applicable to large deflection case. The main idea is to decouple the flexure beam into motion pairs with linear springs and rigid-links, transforming the compliant mechanism model to a traditional rigid model [67]–[69]. It significantly contributes to the modeling of large deflection mechanisms [70], bistable compliant mechanisms [71], [72], and large-deflection flexure hinges [73]. In the past decade, pseudo-rigid-body also employed in small-deflection case, such as displacement amplifier [74], and precision positioning stage [75]–[77].

D. NONLINEAR MODEL

Nonlinear model plays core roles in the analysis of compliant mechanisms with large deflection [78], [79]. Although the flexure hinge of displacement amplifiers is under the small deformation, recently nonlinear approaches are found employed in the FDAMs.

Liu *et al.* [80] applied Timoshenko beam constraint model to predict the deformations of the flexure hinge to investigate the nonlinear stress-stiffening and shear effects of the bridge-type amplification mechanism. Pan *et al.* [81] revealed the nonlinear characteristics of the BAM via a nonlinear closed-form model with load effect considered. The analytical model showed that the amplification ratio that is constant in linear model varies with input force F_{in}. Li *et al.* [82] employed the tensor to accurately describe the kinematic characteristics of the rhombus-type mechanism in an elegant way, where the load-equilibrium equations associated with the deflection of the flexible elements were considered. Friedrich *et al.* [83] carried out the nonlinear modeling of the compliant mechanisms by applying a nonlinear finite beam element approach. The results showed that the nonlinearity can be neglected for single flexure hinge analyzing but not for the system level.

Nonlinear approach indeed proffers high-accurate models for FDAMs. However, the convoluted modeling procedures daunt most designers and the mountainous computations bring troubles in optimization design where calculation will be implemented repetitively. The nonlinear model only serves as a compensation for the linear one in the case where super-high accuracy is required.

III. AMPLIFICATION RATIO

A. AMPLIFICATION RATIO LIMIT PHENOMENON

The main function of the displacement amplifier is to increase the stroke of the PSA. Besides, to increase amplification ratio is advantageous to maintain a compact structure size under the same stroke requirement. Thus, the amplification ratio is a crucial performance index that the researchers and engineers concern.

However, the amplification ratio limit phenomenon brings challenges to a specific design. FIGURE 3 exhibits the relation between amplification ratio and sensitive parameters that are joint's offset distance h for BAM and short beam l_{short} for LAM [84]. According to the amplification principle, the amplification ratio will increase to infinite with structure parameter adjusting, as shown by the geometric model. Nevertheless, a vertex exists in the result curves from linear model and FEM, which implies that the amplification ratio is limited and it is difficult to obtain a large amplification ratio via structure parameters adjusting simply. This phenomenon is called amplification ratio limit in this review.

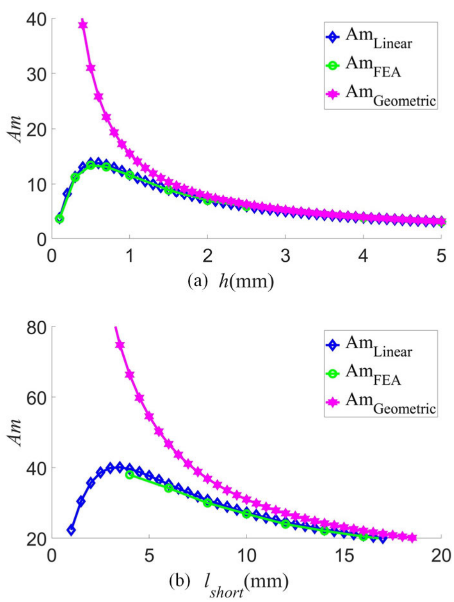

FIGURE 3. Amplification ratio calculated via different methods: (a) bridge-type amplifier; (b) two-stage LAM [84].

To overcome this issue, researchers employ diverse technologies to obtain large amplification ratio and travel range. The efforts for the large amplification ratio can be divided into three categories, multistage amplifier, multi-driving amplifier, and system stiffness analysis methodology. The third approach is systematically introduced first in this review.

B. MULTISTAGE AMPLIFIER

Connecting several amplifiers in series is a straightforward approach to enhance the amplification ratio, which can be calculated via multiplying the amplification ratio of each sub-stage

$$Am = Am_1 \cdot Am_2 \cdots Am_n, \tag{6}$$

where n is the number of the amplifiers.

Kim *et al.* [85] first developed a two-stage BAM that is straightforward connected two BAMs orthogonally, as shown in FIGURE 4 (a). The experiment results showed that the amplification ratio is 10 that yet is far less than the expected value 98. Wu *et al.* [86] applied this structure to drive a fast chopping secondary mirror, whose amplification ratio is about 7, eigenfrequency is 9.7 Hz, and maximum load is 10 Kg. The multi-stage amplifier proposed by Matteo *et al.* [87] consists of two BAMs and a LAM that are arranged in one plane, as shown in FIGURE 4 (b). FEM results exhibited that the amplification ratio is 7.13. Chang *et al.* [88] reported a stage actuated by applying two serially connected Scott–Russell amplifiers. The experiment showed that the stage achieved a vacuum-compatible device with a travel of greater than 100 mm, a resolution finer than 0.04 mm, and an angular deviation smaller than 31.1 mrad.

LAM is more suitable for multistage amplifier development due to its flexible structure. Royson *et al.* [89] presented a lever-type mechanism based monolithic microgripper with two degrees of freedom, as shown in FIGURE 4 (c). FEM shows the amplification ratio is 9.69 and 4.03 for the right and the left arms, which are actuated by a two-stage LAM and by a single LAM respectively. FIGURE 4 (d) exhibits a micro-assembly used microgripper with a two-stage LAM [90]. The amplification ratio tested via prototype experiment is 6. Gao *et al.* [95] developed and analyzed a microgripper with a two-stage LAM whose amplification ratio is 13.13.

Differential lever-type amplification mechanism (DLAM) is a kind of special multistage LAM. FIGURE 5 illustrates the schematic of DLAM. Generally, the DLAM is composed by two stages with the first stage consisting of two LAMs and the second consists of one. At the first stage, the PSA actuates LAM1 and LAM2 simultaneously. The output displacements conflate and then be amplified by actuating the ends of LAM3 in the second stage. According to the lever amplification principle, the amplification ratio of the DLAM

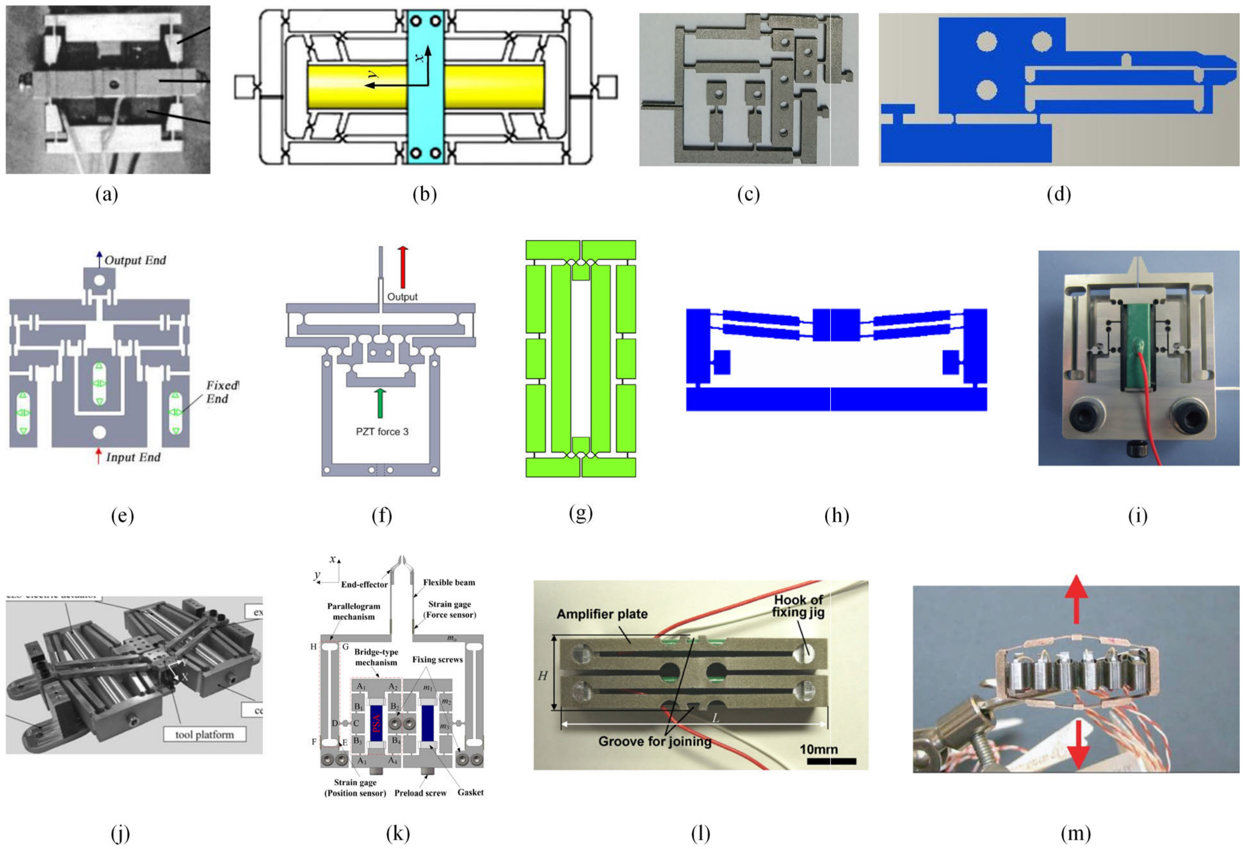

FIGURE 4. Displacement amplifiers with large amplification ratio: (a) Three-dimension bridge-type mechanism [85]; (b) Multistage amplifier [87]; (c) 2-D lever-type microgripper [89]; (d) Two-stage lever-type microgripper [90]; (e) Differential lever amplifier [91]; (f) Micromanipulation for cell [92]; (g) Lever-bridge type amplifier [93]; (h) Five-bar amplifier [94]; (i) SR-lever amplifier [74]; (j) Dual-drive bridge-type amplifier [103], [104]; (k) Dual-drive bridge-lever amplifier [106]; (l) honeycomb link amplifier [112]; (m) Exponential strain amplifier [25], [113].

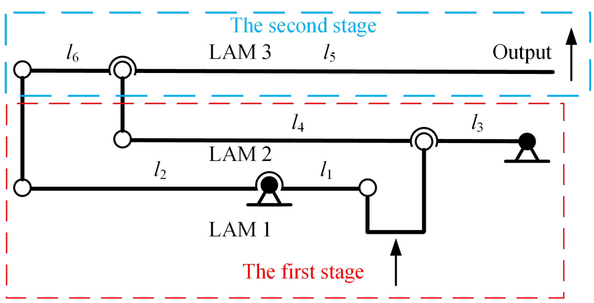

FIGURE 5. Schematic of the differential lever-type amplification mechanism.

can be written as

$$Am = \frac{l_5}{l_6}\left(\frac{l_2}{l_1} + \frac{l_4}{l_3} + 1\right) + \frac{l_4}{l_3} + 1, \tag{7}$$

which is obviously larger than the amplification ratio that LAM1, LAM2, and LAM3 straightforward connected.

Lu *et al.* [91] employed the DLAM with amplification ratio of 7.14 to amplify the stroke of the magnetostrictive actuator, as shown in FIGURE 4 (e). The paper concluded that DLAM can realize larger displacement output with the relatively small structure compared with other amplifiers. Tang *et al.* [92], [96] reported a robotic biomanipulation system where the DLAMs were designed for biological cell capturing and injecting. FIGURE 4 (f) presents the microinjection used mechanism whose amplification ratio is 17.6. Li *et al.* [97] designed a DLAM to actuated a XY micro positioning platform. The amplification ratio tested by the FEM is 12.3 and the natural frequency is 126.3Hz.

Each kind of amplifier possesses advantages and disadvantages. For example, BAM is compact and symmetric, but faces troubles when the multistage structure is configured. LAM is characterized by the flexible structure but the compound motions at the output end restrains its application. Thus, the two amplifiers are always combined to enhance the working performance, which is called hybrid-type amplification mechanism (HAM) in this paper.

Authors [93] proposed a lever-bridge-type amplifier via modifying the input end of the BAM, which features large amplification ratio of 48 and load capacity of 30 N, as shown in FIGURE 4 (g). This configuration helps to reduce the displacement loss that is inevitable in all kinds of FDAMs design. Qian *et al.* [98] progressed this design and proposed a lever-bridge-lever type amplifier to achieve a large displacement at the microgripper tip. FEM results indicated that the amplification ratio is 17.6 and the stroke is 204.5 μm. Ouyang *et al.* [94], [99], [100] systematically studied the symmetric five-bar amplification mechanism that can be considered as a lever-bridge type mechanism, as shown in FIGURE 4 (h). This amplifier achieves a large amplification ratio of 24.2 in a compact size, and it has a high natural frequency of 573 Hz and no lateral displacement. In addition, HAMs are widely used for microgripper actuation due to its exclusive performance [74], [101], [102].

C. MULTI-DRIVE AMPLIFIER

To construct a multistage amplifier is an effective method of amplification ratio increasing. However, according to the law of energy conservation, displacement amplifying implies force reducing. On the contrary, the load at the output end will be sharply amplified at the input end, which results in PSA's displacement loss. Besides, a multistage always results in low stiffness and natural frequency.

Multi-drive amplifier affords another solution for stroke enhancement without paying the cost mentioned above. Kohrs *et al.* [103], [104] serially connected the output ends of two double-sides BAMs in a plane. The displacements at the other output ends are converged and further amplified via a half of BAM arranged on another layer, as shown in FIGURE 4 (j). Ling *et al.* [105] further studied this configuration and theoretically investigated the attenuated displacement of the multistage amplifiers, where the BAM was replaced by the rhombus-type amplifier. Experiment results show that the travel range is 198 μm and the natural frequency is 3280 Hz. FIGURE 4 (k) presents a dual-drive microgripper composed of two BAMs and a LAM, which possesses stroke of 328.2 μm and first natural frequency of 157.5 Hz [106]. Similar designs are widely used in microgripper due to the outstanding mechanism performance [107], [108].

Modular-architecture amplifier is a kind of augmented multi-drive amplifier, where the amplifiers acting as cells are picked very tightly and can be configured in a variety of ways to meet specific performance needs. Some array-type amplifiers composed of bridge-type cells were proposed and investigated two decades ago [109]–[111], which performed well in stroke, energy density, and structure size. Nevertheless, the specific processing piezoelectric materials complicates the manufacturing technique that hinders the industrial applications. Muraoka and Sanada [112] further refined this configuration so that it could equipped with PSAs commercially available, as shown in FIGURE 4 (l). Experiment studies showed that the maximum stroke of a two-cell actuator is 410μm and the amplification ratio of each cell is 7.1. Furthermore, the application of the actuator was studied via XY stage actuation, where a six-cell actuator was designed for both X and Y direction. The measured stroke was approximately 1 mm, and natural frequency is 60 Hz. Ueda's team [25], [113] proposed the nested rhombus actuator that encloses rhombus actuators with a larger amplification structure, as shown in FIGURE 4 (m). The experimental displacement reaches to more than 2 mm with the sacrifice of output stiffness and natural frequency.

D. SYSTEM STIFFNESS ANALYSIS METHODOLOGY

No matter the multistage amplifier or multi-drive amplifier, large stroke implies greater number of layers, which leads to bulky structure and low stiffness. Actually, the amplification ratio limit phenomenon is critical issue for amplification ratio design. According to the analytical results from FIGURE 3, the amplification ratio limit phenomenon can be described by linear model but not by geometric model. Compared with linear model, geometric model pays no regard to the properties of flexure hinge. The flexure hinge is replaced by an ideal rotational pair. Thus, flexure hinge is the source of amplification ratio limit.

Jianying and Qinghua [114] roughly studied the relation between the output displacement and the flexure hinge layout of LAM. Large deviations of the designed FDAMs exist and the displacement loss is ubiquitous in all the layouts. Li's [115], [116] group also noticed the deviation of the amplification ratio between geometric model and FEM results during the LAM design, and concluded that the elastic deformation of the beams, the internal and axial deformation of the flexible hinge will cause the energy and motion loss.

According to Refs mentioned above, displacement loss is ubiquitous in FDAMs, especially in the multistage amplifiers. It is known that parasitic motion deeply affects the performance of the compliant mechanisms [117]–[121], but less notice was taken of the influence of stiffness that is the main reason of the displacement loss. Taking the two-stage amplification ratio as an example, each stage can be considered as a linear spring, as shown in Figure 6. With PSA elongating, the displacement S_{PSA} is amplified to S_1 by stage1 and further to S_2 by stage2 in the case of zero stiffness. In actual case, displacement loss ΔS_1 exists due to the reactive force generated by stage2. At the output end, the displacement loss will be scaled up to ΔS_2 that significantly reduces the amplification ratio.

To quantitatively estimate the impact of the displacement loss, relative displacement rate is defined

$$\tau = \frac{S_2 - \Delta S_2}{S_2} = \frac{S_2'}{S_2}. \tag{8}$$

Under the same input displacement, displacement rate equals amplification ratio rate, so the relative displacement rate can be written in another way

$$\tau = \frac{S_{PSA} \cdot Am'}{S_{PSA} \cdot Am} = \frac{Am'}{Am}, \tag{9}$$

where Am' is the amplification ratio with flexure hinge impact, which can be obtained via static model, FEM, or experiment testing; Am is the amplification ratio without flexure hinge impact, which can be calculated via geometric model. Thus, τ is also called relative amplification rate.

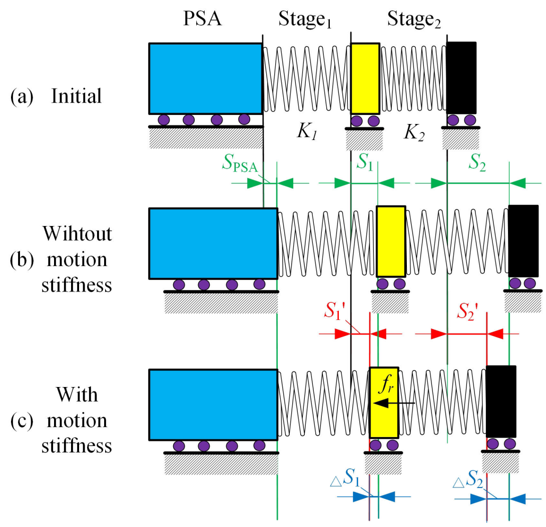

FIGURE 6. Schematic of the system stiffness analysis of the two-stage amplifier.

Figure 7 exhibits the relations of the relative amplification ratio and geometric amplification ratio of the classical multi-stage amplifiers, which is marked by blue stars. It is interesting that almost all the starlike blue markers fall into a belt that lays from up left to low right of the plot. It is called amplification belt in this review. Amplification belt demonstrates that to simultaneously realize high geometric amplification ratio and large relative amplification ratio is difficult (the desirable region is at the up right corner of the plot). It is known that the practice displacement amplifier equals the product of the geometric amplification ratio and the relative amplification ratio. Thus, the experiment results always disappoint the designers compared with their initial ambitious design aim.

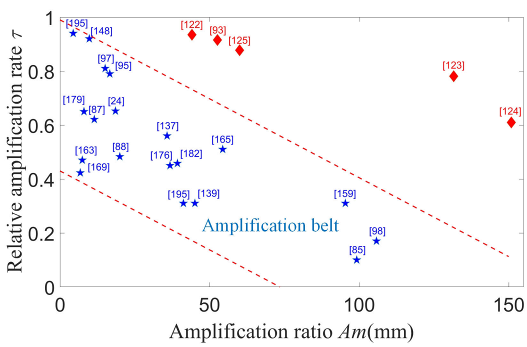

FIGURE 7. Relations between Am and τ of some classical multistage amplifiers.

System stiffness analysis methodology is an effective method to break the boundary of the amplification belt and push the mark point to be closer to the desirable region. According to Hook's law, the reactive force f_r in Figure 6 can be calculated by

$$f_r = K_2 \cdot S_2' \tag{10}$$

where K_2 is the stiffness of stage2, and S_2' is the practice displacement of stage2. The displacement loss ΔS_1 can be obtained by

$$\Delta S_1 = \frac{f_r}{K_1} = \frac{K_2}{K_1} S_2' = \frac{S_2'}{U_2^1}. \tag{11}$$

with

$$U_2^1 = \frac{K_1}{K_2}. \tag{12}$$

Eq.(12) shows that to enhance the stiffness rate U is a method that can reduce the displacement loss. Accordingly, author proposed hybrid flexure hinge method to reconfigure the system stiffness of the three-dimensional BAM [122], as shown in Figure 8(a). Experimental results showed that the designed amplifier achieves an amplification ratio of 41.29 and a relative amplification rate of 0.935. Similarly, bridge-lever-type amplifier was refined by hybrid flexure hinge [93], whose amplification ratio and relative amplification rate were 48.15 and 0.915, respectively. This design was further improved by multi-objective optimization design, where the amplification performance of 102.8 and 0.781 was obtained [123]. The system stiffness analysis in the two articles demonstrates that the LAM is more applicable for the first stage compared to BAM since the force for the LAM is along the axial direction of the flexure hinge while for the BAM is along the motion direction. The former is far larger than the latter. Figure 8(b) presents the experimental setup of the lever-bridge-type amplifier.

Hybrid materials design is another method for system stiffness adjusting. Chen *et al.* [124] further optimized the hybrid flexure hinge based three-dimensional BAM via material selecting and multi-objective optimization design, as shown in Figure 8 (c). To enhance the stiffness rate between the two stages, the first stage is made of spring steel with possession of higher Young's module while the second stage is aluminium alloy whose Young's module is lower. Finally, the amplification ratio reaches up to 91.6, natural frequency is better than 52 Hz, and the maximum load is 50 N and 20 Nm^{-1}.

Flexure hinge individualized design (FHID) method is an effective method for compliant mechanism design, which is developed from hybrid flexure hinge method. FHID required to design each flexure hinge according to the force and motion analysis at the pivot, which helps not only to improve the stiffness but also eliminate the parasitic motion. Figure 8 (d) is the microgripper designed by FHID method [125], [126], which significantly increases the amplification performance and compacts its structure size.

System stiffness analysis methodology provided a new perspective for FDAM design. Hybrid flexure hinge, hybrid materials, FHID, and multi-objective optimization design are the four main tools for stiffness adjusting. According to Figure 7, system stiffness method successfully breaks the boundary of the amplification belt, where the design results (the rhombic red markers) trend to the up right corner. It is an

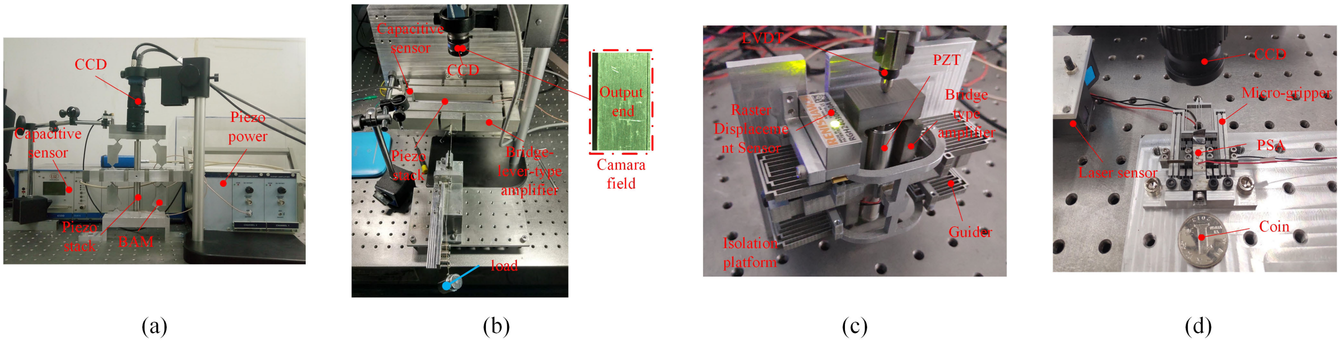

FIGURE 8. **Amplifiers designed via system stiffness analysis method: (a) hybrid flexure hinge based 3-D BAM [122]; (b) bridge-lever-type amplifier [93]; (c) macro/micro 3-D BAM [124]; (d) microgripper [125].**

effective method for super-large amplification ratio achievement. Figure 9 summarizes the flow of the system stiffness analysis methodology.

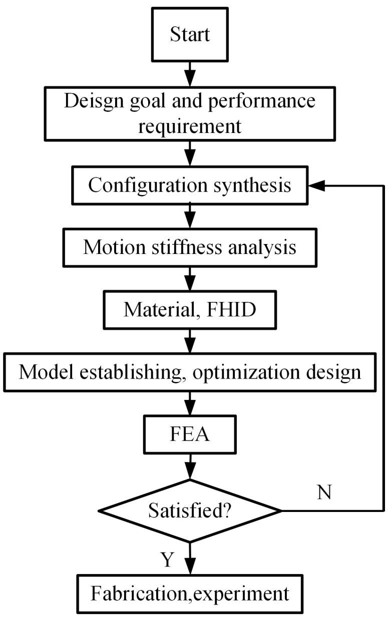

FIGURE 9. **Flow chart of the system stiffness analysis methodology.**

IV. INHERENT DRAWBACK AND THE SOLUTION OF BAM AND LAM

FDAMs feature low-cost, simple structure, and high stability compared with piezo-motors. However, on the opposite side of the coin, some inherent drawbacks brought from the flexure hinge narrow its application, among which the low lateral stiffness of BAM, severe parasitic motion of LAM, poor load capacity and meager natural frequency bring troubles to most engineers who attempt to employ the FDAMs in precision systems. This section discusses those challenges and concludes the existing solutions from previously published papers. It is expected to provide some inspiration for researchers to create better solutions and for engineers to deal with those mentioned issues.

A. ENHANCEMENT OF THE BAM'S LATERAL STIFFNESS

When the output end of a BAM is subjected to the moment of force or the load that is not along the motion direction, the undesired motion will occur at the output end due to the weakness of the lateral stiffness. Most researchers find this issue but less discuss the reason. Chen [126] attemptted to explain it through the degrees of freedom (DOFs) aspect. Figure 10 (a) and (b) illustrate a BAM and its structure sketch. According to Chebyshev formula,

$$L = 3n - 2P_l - P_h, \tag{13}$$

where n is the number of components, P_l is the number of low pair, and P_h is the number of high pair. The DOF of the traditional BAM is 2. With the PSA elongating, the input displacement causes indeterminate movement at the output end, as shown in Figure 10 (c). Complex load aggravates the issue.

Xu and Li [45] addressed this problem via a brilliant simple structure design. Two parallelly aligned bridge arms are used to replace the traditional single bridge arm, based on which the parasitic motion can be eliminated. Figure 10 (d) shows the prototype of the modified BAM that is called compound BAM. According to Eq. (13), the DOF of the half compound BAM is 1, so it implies that the movement of the mechanism is determined when the PSA elongates, as shown in Figure 10 (e). Compound BAM is widely used. Chen *et al.* [132] studied the multimode sensing approach of the compound BAM that is integrated with strain gauges. Wu [133]–[136] designed a compact vertical piezo-actuator and XY parallel nanopositioning stage based on the compound BAM that significantly increases the directionality of the output motion. Liang *et al.* [127] employed the compound BAM to actuate a monolithic 2-DOF pure rotation platform. Thanks for the strong lateral stiffness of the compound BAM, the maximum rotation angles in the X- and Y-axis are 2.04 and 2.12 mrad, and the relative coupling errors are 2.03% and 2.09%, respectively.

To equip the guide mechanism is another effective method to fight off the negative influence of low lateral stiffness. Figure 10 (g) [128] presents a self-guided BAM that consists

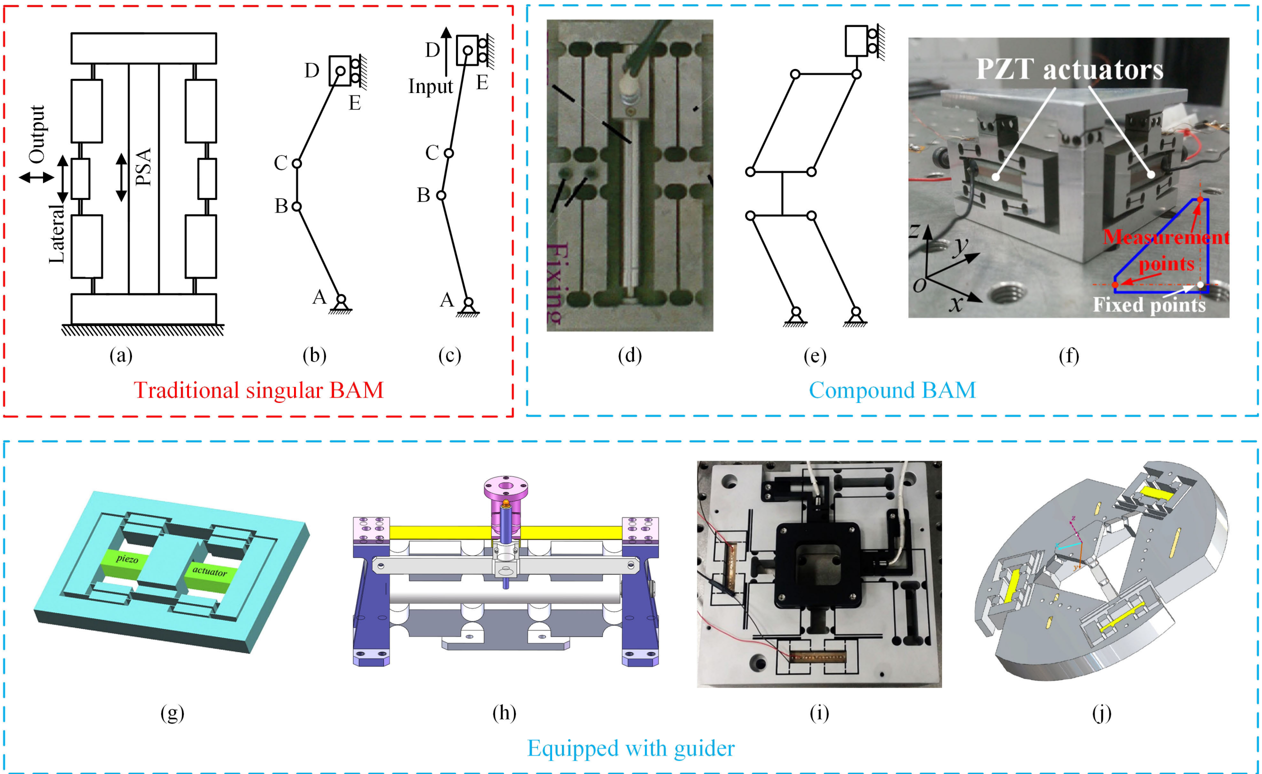

FIGURE 10. Lateral enhancement of the BAM: (a) Traditional BAM and (b) the structure sketch; (c) unexpected motion of the BAM; (d) compound BAM and [45] (e) the structure sketch; (f) 2-DOF rotation platform [127]; (g) self-guided BAM [128]; (h) BAM with singular leaf guider [129]; (i) XY stage actuated by BAM with singular leaf-guider [130]; (j) 3-DOF stage actuated by BAM with double layers leaf-guider [131].

of a normal BAM and a skewed double-compound parallelogram structure acting as a motion guide. The experiment results showed that this mechanism performs well in straightness error and yaw motion error compared with other similar stages. Authors [129] guided the motion of the BAM via a long thin leaf flexure hinge in the 6-DOF positioning system design, as shown in Figure 10 (h). The guide mechanism successfully protects the BAM from influence of gravity. Lee *et al.* [130] introduced a similar guider in the X-Y parallel nano-positioning stage, as shown in Figure 10 (i). FEM studies showed that distortion can be eliminated with the guide mechanism, and the prototype experiment result showed that the cross-coupling is less than 0.65%. Ghafarian *et al.* [131] parallelly arranged two leaf flexure hinges in the XYZ micromanipulator system to enhance the guiding performance, as shown in Figure 10 (j).

B. COMPOUND MOTION IN THE DISPLACEMENT AMPLIFIER

In general case, purely translational motion is expected when FDAMs are used. Nevertheless, compound motion is inevitable, as shown in Figure 11(a) and (b). The trajectory of the output end of LAM is a segment of circle arc and the trajectory of the input end of BAM is a compound motion combined with X and Y direction. The former compound motion causes parasitic deformation and undesired strain for the flexure hinge, and the latter discommodes the development of the multistage amplifier [22], [85], [87].

Symmetrical arrangement is a commonly used method to avert parasitic motion. Figure 11 (c) exhibits a symmetrical two-stage LAM proposed by Choi *et al.* [137]. This LAM is widely used for stage actuating due to the simple neat layout. Wu *et al.* [138] introduced this amplifier to actuate a X-Y nano-manipulator, as shown in Figure 11 (d). The coupled errors between two directions remained at 0.14%. Vu *et al.* [142] employed taguchi method to carry out optimization design of the tensural displacement amplifier that is a classical symmetric multistage LAM. Dang *et al.* [143] developed and optimized a linear compliant mechanism inspired by a beetle animal. A symmetric two-stage LAM was employed to provide active motion, which suppressed the parasitic error down to 0.01515%.

Symmetrical arrangement provides an aesthetic neat layout. However, the structure size is double. Parallel LAMs can balance the parasitic motion and the structure size. Figure 11 (e) illustrates the structure sketch of a parallel LAM. The parasitic error that is the rate of the x component to y

$$P = \frac{1 - \cos(\theta)}{\sin(\theta)}. \tag{14}$$

Accordingly, the parasitic error won't exceed 1% when the motion angle is under 1°.

Qu *et al.* [144] combined parallelograms and LAMs to develop a large-range compliant micro-positioning stage with remote-center-of-motion. Experimental results showed that

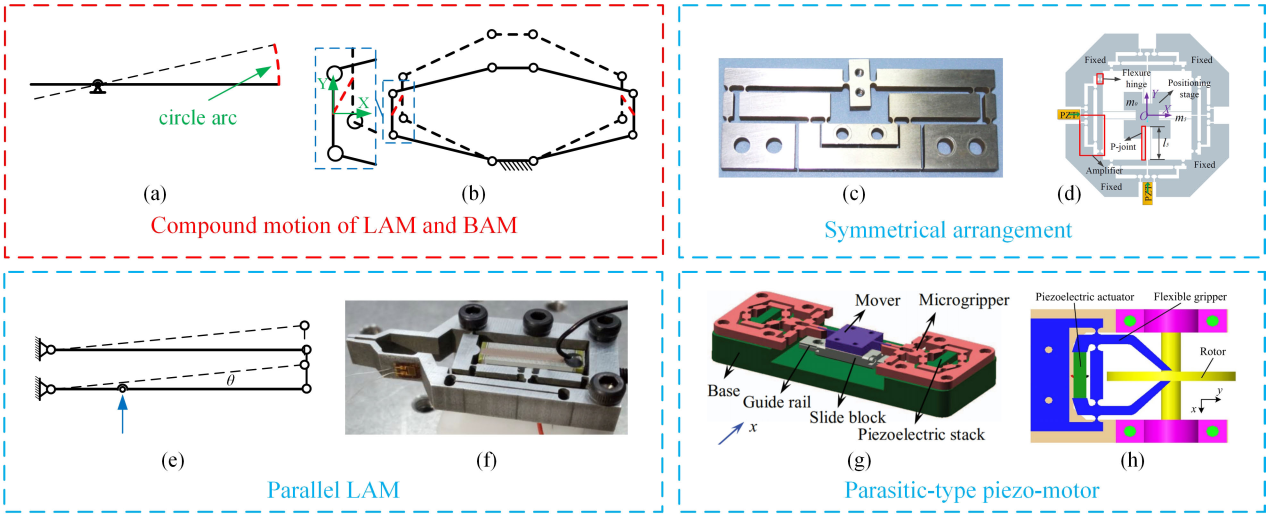

FIGURE 11. **(a) Compound motion of LAM and (b) BAM; (c)-(d) symmetrical arrangement LAM [137], [138]; (e)-(f) parallel LAM [139]; (g)-(h) parasitic-type piezo-motor [140], [141].**

this mechanism possessed a rotational range of 6.96 mrad and the maximum center shift is less than 9.2 μm. Zubir *et al.* [145], [146] studied the grasping mode of a microgripper, and found the angular mode of the gripper-tip deteriorate the stability of the grasping task. To solve this problem, a parallel LAM was employed to replace the traditional LAM in the modified structure that achieved a parallel mode during grasping task. This significant improvement was widely adopted by researchers and engineers. Liang *et al.* [139] reported an asymmetrical microgripper with one active jaw. A parallel LAM is introduced in the mechanism to achieve parallel grasping operation, as shown in Figure 11 (f). Dsouza *et al.* [89] proposed a 2DOF compliant flexural microgripper, where the LAMs were used to offer actuation and parallelograms guaranteed the purely translational motion. Similar microgripper can be found in many other research works [147]–[149].

As a coin has two sides, compound motion also can bring unique advantage in some special applications. Parasitic-type piezoelectric motor is a typical case. Huang *et al.* [140] firstly proposed parasitic-type piezo-motor in 2012 as shown in Figure 11 (g), where a BAM was utilized to offer the major motion and two collaborative LAMs were used to realize pushing and clamping operations via the main and the parasitic motions respectively. To further validate the feasibility of this concept, Huang *et al.* [141] designed a rotary piezo-motor by utilizing the parasitic motions of LAMs, as shown in Figure 11 (h). Based on these investigations, lot of similar motors were developed and analyzed [150]–[157]. Due to the compound motions provided by these fundamental structures, the parasitic-type piezo-motor realizes large stroke, high frequency, and controllability via a simple and easy way.

C. LOAD CAPACITY OF THE DISPLACEMENT AMPLIFIER

Load capacity is important for the FDAMs as it suffers complex loads. Authors [129] designed a space optics alignment used 6-DOF positioning system. Although gravity can be ignored in space, it brought troubles when the experiment was carried out on earth where the gravity applied to the actuator is beyond 30N. Similar issue arises at the high-precision compliant parallel mechanism for large-aperture grating tiling [9].

A general evaluation criterion is crucial for the performance analysis and investigation. Kim *et al.* [158] utilized a concept, blocked force, to evaluate the load capacity of the three-dimensional BAMs, which is defined as the force that makes the output displacement zero when the maximum input voltage is applied. Chen *et al.* [124] introduced this definition to the multi-optimization design of an amplifier based linear actuation system. In [129], authors studied the load characteristic of the BAMs, where the external load is divided into elastic load which is from the reactive force of elastic elements, and the constant load which is the gravity of the system. Analysis results showed that the elastic load will influence the amplification ratio and the constant load will change the original point of the actuator.

Xie *et al.* [159], [163] reported a MEMS-enabled micro-scissor motion amplifier that is formed with diverse materials based on multi-layer structure, as shown in Figure 12 (a). This approach enables amplifier to output displacement of 6.3 μm and provide load capability of 16 mN. Choi *et al.* [160] arranged two BAMs parallelly whose bridge arm angles are invert. The target plate is aligned between the two BAMs and connected to the output ends of the BAMs, as shown in Figure 12 (b). This design is expected to offer more active force for the actuation system. Chen *et al.* [161] proposed a tensural displacement amplifier whose elastic elements are all loaded in tension, as shown in Figure 12 (c). This amplifier is designed to avoid the buckling problem of the flexure hinge, which helps to increase the load capacity of the mechanism. Similar idea is reported in the design of [164], where L-shape of LAM and BAM is combined. Figure 12 (d) presents a displacement amplifier with high load capacity

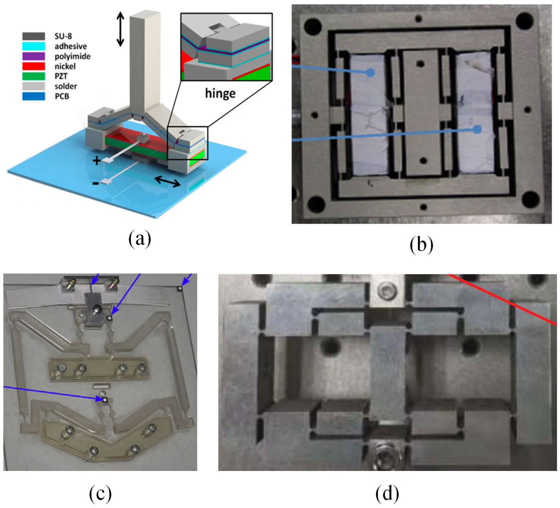

FIGURE 12. Load capacity enhancing amplifiers: (a) Micro Motion Amplifiers [159]; (b) double BAMs [160]; (c) tensural displacement force [161]; (d) high load capacity amplifiers [162].

by arranging the Scott-Russell mechanism in bridge-type configuration [162]. Experimental results showed that the new amplifier is superior in the output stiffness and natural frequency to the traditional design.

According to the system stiffness analysis, the FDAM can be simplified as the linear spring. Thus, an increased stiffness of the spring helps to suppress the displacement loss caused by external force. This is the foundation of the design in paper [162]. Besides, the stress condition of the flexure hinge determines the load limit of the amplifier, which is the reason tensural displacement amplifier is able to load more. Finally, to increase the output force of a PSA can enhance the output force of the amplifier, which is the logic of paper [160]. Overall, although the load capacity is important in some cases, it does not draw enough attention from researchers yet. With the development of the high-end technology and extensive application of precision engineering technology, the demand of the displacement amplifier with large load capacity will sharply increase.

D. NATURAL FREQUENCY OF THE DISPLACEMENT AMPLIFIER

High precision and velocity are both the core requirements in some applications. For example, the working frequency of the displacement amplifier used in the piezo-motor directly impacts the maximum velocity that the motor can achieve [165], [166]; the fast tool servo that is important in ultra-precision machining demands the natural frequency up to hundreds even thousands of hertz [167]–[169]; the precision positioning stages for optical scanning and micro-manipulation need high frequency to guarantee the work quality and efficiency [170], [171].

However, according to the formula of natural frequency, $f = (K/M)^{1/2}/2\pi$, where M and K are the effective mass and stiffness respectively, high flexibility always implies low natural frequency. A review about the high-speed flexure-guided nanopositioning and a guideline on how to design and control these mechanisms can be found in [172] that concluded that the mass of the mechanical amplifier together with the flexible linkages is disadvantageous to the mechanical resonance. Thus, the amplifier is refused in most cases where the high frequency is required [30], [173], [174].

Even so, the efforts on the improvement of the amplifier frequency do not cease. Ling [175]–[177] reported a succession of high-frequency compliant mechanisms with boundary constraint that is able to increase the stiffness of the mechanism system. This method increases the fundamental frequency of the two-stage rhombus-type amplifier to 2.2 kHz and remains the stroke as 0.6 mm. Besides, a piezo-actuated compliant mechanism with frequency of 628 Hz and stroke of 1.44 mm can be found. Zhu [178] enhanced the lateral stiffness and dynamic characteristics of a BAM via combining two symmetric Scott-Russell mechanisms and equipping with the leaf-type double parallelogram at both input and output ends. A natural frequency of 570 Hz and maximum stroke of 31.5 μm are rewarded to the modification. Similarly, Tian's design [179] consisted of L-shape levers and half-bridge mechanism. FEM results indicated that the proposed amplifier is superior in the lateral stiffness and natural frequency compared with the traditional BAM and compound BAM. The output coupling is less than 2% and the first resonance frequency is 362.8 Hz in the experimental studies.

In addition to the boundary adding and structure modifying, optimization design is useful for the dynamic performance improving. Wang *et al.* [180] studied the optimization effects on the BAM's natural frequency by using Taguchi method. The simulation results revealed that the parameters of flexure hinge significant influences the first frequency. Vu *et al.* [142] employed a similar method to optimize the tensural displacement amplifier and increase the first mode shape frequency to 214.06 Hz. Besides, Chen *et al.* [124] and Kim *et al.* [158] optimized the three-dimensional BAM by applying NSGA-II and sequential quadratic programming respectively.

Similar to the load capacity, the main influence factor of the natural frequency is the structure stiffness. The existing methods to enhance the stiffness consist of structure modification and parameters optimization [143]. For the compliant mechanism, stiffness increasing always implies travel range shortening. Thus, a trade-off should be made among stroke, natural frequency, and load capacity via multi-objective optimization design during the amplifier design.

V. APPLICATION

FDAMs show exclusive features in precision engineering, especially when the compact size, large stroke and high resolution are required. Although the piezo-motor that performs well in accuracy and stroke aspects, displacement amplifiers are preferred under some special scenarios. A convictive instance author experienced is the 6-DOF positioning system used in space optics alignment, where the piezo-motor is abandoned for the unreliability result from frictional loss [129].

FDAMs are frequently used in the positioning stage. When the stroke requirement is within submillimeter, FDAM is preferable compared with other actuators, due to its compact size, low cost, and well comprehensive performance. Li *et al.* [49], [97], [115], [181] published a succession of articles about XY positioning stages that are actuated by single-stage BAM or LAM. Yang *et al.* [182] serially connected the LAM and Scott-Russell mechanism to amplify the stroke of a PSA used in the XY positioning stage. Zhu *et al.* [183] proposed a mirror-symmetrically distributed positioning stage incorporating a BAM and a two-stage LAM, achieving large workspace and decoupled motions. Bhagat *et al.* [77] aligned three LAMs in orthometric way to realize in-plane directional motions, among which x-axis motion is decoupled from the other two. Wang *et al.* [184] presented a 3-DOF nanopositioning platform with three symmetrically distributed two-stage LAMs, making a trade-off among high rigidity, large magnification, high-precision tracking, and high-accuracy positioning. For a 3-DOF stage, out-of-plane direction motion is also required. Related researches were reported by Qian *et al.* [185], Zhang *et al.* [186], and Lee *et al.* [187]. Besides, the investigations of higher DOFs positioning stages can be found in [188]–[190]. The main challenges for the displacement amplifier used in the positioning stage is the low lateral stiffness versus complex external loads. The method of lateral stiffness enhancement has been introduced in Section IV part A.

FDAMs also play crucial roles in piezo-actuated microgripper [191]. In the design of an asymmetric flexible micro-gripper mechanism [192], a LAM was utilized for displacement amplification and a parallelogram was for pure motion. Das *et al.* [193] proposed a microgripper based on three-stage amplifiers that consists of a BAM with two-sided output ports serially connected with a two-stage LAM. Noveanu *et al.* [194] studied the influence of the flexure hinge shape on the performance of a microgripper with LAM. Experimental results revealed that the flexure hinge with elliptical profile and stainless steel is the best combination for this design. Xu *et al.* [195] presented a microgripper endowing simplified structure via replacing the existing parallelogram-based jaws to a compliant rotary bearing mechanism. From the investigations mentioned above, one can know that the parallel motion is demanded to microgripper for averting slippage. Thus, parallelogram that performs well in parasitic suppressing is always used as the gripper-tips. Since the motion range of the gripper-tip is large, multistage amplifiers are commonly used.

Displacement amplifier is widely used in some non-resonant piezo-motors that accumulate little steps to achieve large stroke. Compared with PSAs, displacement amplifiers can proffer a larger step and increase the average velocity of the actuator. Belly *et al.* [196], [197] studied the benefits of displacement amplifier in the inertial stepping motor. Three BAMs with different bridge angle replaced a classical direct actuator in the experiment studies that demonstrated that amplifier is advantageous to enlarge the step size but disadvantageous to the dynamic performance. LAM was employed in the linear-rotary inchworm actuator by Sun to amplify the displacement of the clamping mechanism, which addresses the issue from preload and clamping force [198]. Huang *et al.* [166] introduced two sets of LAMs in a linear motor to adjust the contact force and increase the step size respectively. Cai *et al.* [199] adopted the parallelogram mechanism and shear PSA to achieve the motion period of the walking type linear piezo-motor. In addition to these common usages, displacement amplifier can be used oppositely to achieve higher accuracy at the price of lower stroke [200], and utilized in some resonant piezo-motors to generate the standing waves [201].

In addition, displacement amplifiers are used in some special-interest areas. BAM was introduced in the electromagnetic actuator by Nabae to balance the actuated force and motion range [202]. Zhang *et al.* [203] carried out the research on the application of the micro-channel cooling system in the micro-pump, where three types of displacement amplifiers were investigated on modeling, analyzing, and optimizing. Results demonstrated that the hydraulic is preferable compared with BAMs and LAMs. Chen *et al.* [17] focused on a smart transmission mechanism system based on compliant mechanism that assembled using a rhombus-type amplifier, synthesized with flexure hinge and an active rotatable flap. Yu *et al.* [204] proposed a micro-feed mechanism for cell injection with PSA driving. The compound BAM was used to increase the stroke of the mechanism. Arani *et al.* [205] attempted to use the displacement amplifier that is a five-bar symmetric structure made of brass for the application of endorectal prostate magnetic resonance elastography. Besides, [206], [207] reported the applications in optical scanning. More applications of the displacement amplifiers from space to underwater can be found in [208].

Overall, the main function of the FDAMs is to amplify the stroke of the PSA. It proffers an alternative of the actuation way for precision engineering. Furthermore, it is used oppositely to obtain large force or high accuracy in some special cases [200], [209]–[212]. On the other side of the coin, displacement amplifier is undesirable in the case that high frequency and large load are demanded, due to the lower stiffness from the elastic elements.

VI. CONCLUSION

This review summarizes the classical FDAMs and their recent progresses and achievements, emphasizing on four highly concerned topics including the modeling, amplification ratio enhancing, inherent drawbacks as well as its solutions, and the typical applications. The relation of the amplification ratio and the relative amplification rate is illustrated through a two-dimensional plot, where dozens of classical amplifiers are presented. A law in the plot called amplification belt demonstrates that large geometry amplification ratio does not reward commensurate practice amplification ratio compared with their initial ambitious aims. System stiffness analysis method is an effective method to break the amplification belt law and obtain satisfied amplification performance. Concerning the inherent drawbacks, the parasitic motion and lateral stiffness issues are well addressed via structure modifying and guide mechanism equipping; the load capacity and dynamics performance can be improved through structure modifying, boundary adding, and optimization design. However, these methods are essential to increase the structure stiffness that is disadvantageous to the travel range of the compliant mechanism. Thus a trade-off has to be made when these performances are demanded simultaneously in the design process. The FDAM progressively becomes important in precision engineering areas due to its simple structure and satisfying comprehensive performances. In further it will proceed to play crucial role in precision engineering.

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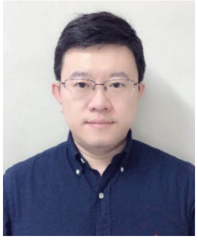

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