Title: Planer flexure mechanisms with two, three or five degrees of freedom

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Introduction and Background

There exist many two-axes planer flexural mechanisms that allow for small translations within the plane of the flexure. Most of these designs incorporate a stacked assembly where one linear stage in mounted perpendicular on a second linear stage resulting in a relatively bulky design. Nevertheless, in this arrangement the two axes are entirely decoupled and the actuation of one axis has no effect on the other. Such an assembly is commonly referred to as a 'serial design' in robotics terminology. In some clever serial designs, the above-mentioned stacking is achieved within a plane.

The disadvantage of serial designs is that the actuator for the second stage has to be mounted on the moving member of the first stage. This not only makes the design unnecessarily complex but also limits the system dynamic performance, for example, speed of response. Ideally, it is desirable to mount the actuators for both the axes on ground, i.e., the fixed base.

Furthermore, if one tries to increase the degrees of freedom of a flexural mechanism using the serial approach, the design becomes increasingly cumbersome and bulky. Therefore, instead of taking this path designers usually develop assemblies that are based on closed-chain parallel designs (as opposed to serial designs). In this kind of designs however, parasitic coupling between the degrees of freedom and cross-sensitivity of actuators become performance limiting factors. These two factors are explained in the next section. There are situations where such parallel designs are used, but accuracy is compromised for economy in size. There are other situations where any degree of errors is unacceptable either at the motion stage itself or at the points of actuator force application. It would be desirable to have mechanisms that have the compactness of parallel designs while axes decoupling and actuator isolation of serial designs.

In this disclosure we shall present a group of flexural mechanisms that are based on parallel elasto-kinematics. It is worthwhile to mention here that the motion of compliant mechanisms is not completely characterized by kinematics; it is strongly dependent on elastic deformations as well. Hence, the study of motion of flexural mechanisms is commonly referred to as elasto-kinematics. The designs presented here make unique use of known flexural units and novel geometric symmetry to minimize or even completely eliminate actuator cross-sensitivity, and parasitic coupling between the two axes. Furthermore, these mechanisms are enhanced to produce out of plane motion, in addition to the in-plane translations. Thus, the resulting planer flexural mechanisms are compact, error-free and can provide multiple decoupled degrees of freedom.

Underlying Design Principle for a Two Axis (2 DOF) Planer Flexure

In the following discussion, X and Y are defined to be the in-plane axes and Z, the out of plane axis. A performance wish list for a two-axis planer flexure design based on parallel elasto-kinematics is presented below:

- No rotation of the motion stage with respect to ground: The flexural mechanism should only allow for pure X and Y translations within the flexure plane
- No parasitic coupling between the two degrees of freedom: The X-axis actuator should produce motion in X direction only, and ideally no motion in the Y direction, and vice versa. Any Y motion produced at the Motion Stage due to an X actuation force is termed as parasitic error or parasitic coupling, and has to be eliminated. The same holds for the other axis.
- Actuator isolation: The point of application of X actuation force should not be affected by any motion of the motion stage in Y direction, and vice versa. This ensures that the actuators do not feel the presence of each other. Furthermore, the point of X actuation itself should be free to move in the X direction only, and should have no error motion in the Y direction, which will cause slippage at the actuation point. This is referred to as actuator cross-sensitivity and should be eliminated.
- Provide a good in-plane structural loop that minimizes out of plane errors

Figure 1 illustrates the rigid and compliant units that the desired flexural mechanism must be composed of, to be able to meet these performance objectives. There are four rigid stages: ground or the fixed stage, the motion stage that is of interest, and two intermediate stages. The motion stage is required to have two translational degrees of freedom with respect to the fixed stage. The intermediate stages are necessary to decouple the motion of the two axes and isolate the actuators that control these two axes. Clearly, the actuator forces cannot be directly applied to the motion stage if both actuators are to be ground mounted. The intermediate stages provide the necessary points for actuator force application. This will become evident in the following paragraphs.

The rigid stages are connected to each other by means of flexure units, which act as frictionless bearings or guides and provide constraints to relative motion. Each of the flexure units A, B, C and D is a single degree of freedom mechanism that only allows translation in the direction shown by the double-sided arrow. The fixed stage is connected to Intermediate Stage 1 by means of Flexure A, which only allows for relative motion along the X direction and constrains all other the other degrees of freedom. This implies that, no matter what the overall configuration of the entire mechanism is, Intermediate Stage 1 will always have a pure X displacement with respect to ground.

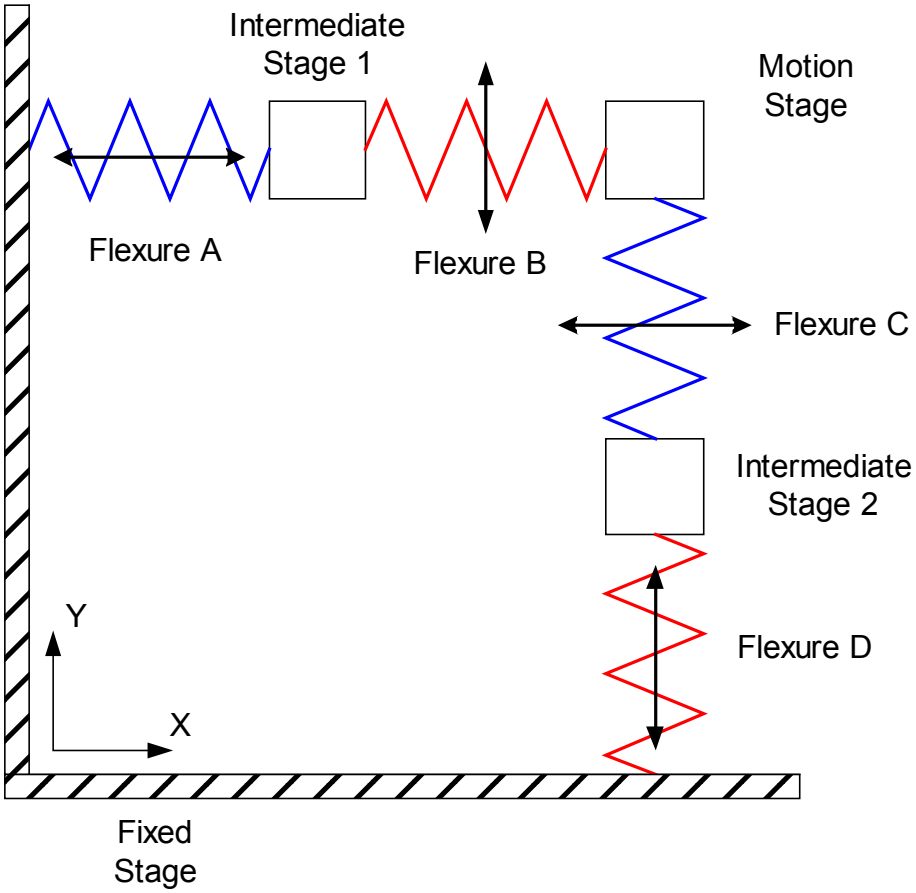


Figure 1

Intermediate Stage 1 and the Motion Stage are connected by means of Flexure B that allows for relative motion in the Y direction only and constrains relative motion along X direction and rotation. This implies that the X motion of Intermediate Stage 1 will be entirely transmitted to the Motion Stage, while any Y motion of the Motion Stage will not be seen by the Intermediate Stage 1 at all. Thus, Intermediate Stage 1 becomes an ideal location for the application of the X actuation force. Flexure A provides the linear guide/bearing for X actuator force. Furthermore, any X force applied at Intermediate Stage 1 is incapable of producing any Y motion of the Motion Stage due to the presence of Flexure B.

On very similar lines, Intermediate Stage 2 is connected to ground by means of Flexure D, which constrains motion in X direction and rotation, but allows for perfect relative motion in the Y direction. Thus, Intermediate Stage 2 can only move along Y and shall have no motion in the X direction or rotation, no matter what the displacement of the Motion Stage is. Intermediate Stage 2 and the Motion Stage are connected via Flexure C, which allows only X motion between the two. Thus, any X motion at the Motion Stage

will not affect the Intermediate Stage 2, which is therefore an ideal location for the application of Y actuation force. At the same time, all the Y motion that is generated at Intermediate Stage 2 due to the Y actuation force will be transmitted to Motion Stage, but is incapable of producing any X motion of the Motion Stage due to the presence of Flexure C.

So in any deformed configuration of the flexural mechanism, Intermediate Stage1 always has a pure X displacement while Intermediate Stage 2 has a pure Y displacement. The Motion Stage inherits the X displacement of Intermediate Stage 1 and the Y displacement of Intermediate Stage 2, thus acquiring two translational degrees of freedom. Since all the connecting flexure units constrain rotation, the rotation of the Motion Stage is constrained with respect to ground.

Units A, B, C and D are idealized single degree of freedom flexure bearings or guides. The only degree of freedom that flexures A and C allow for is relative translation along X, while Flexures B and D are compliant only in relative translation along Y. The arrangement of these flexure units is such that there is no over-constraint in the overall mechanism. With such geometric arrangement, and the idealized flexure units A, B, C and D, we can entirely eliminate any parasitic coupling between the two axes, eliminate rotation of the Motion Stage, and provide complete actuator isolation.

In the above discussion we assumed the flexure units A, B, C and D to be ideal in the sense that they provide error-free motion only in one direction (i.e., zero stiffness in this one direction) and constrain all other degrees of freedom (i.e., infinite stiffness in all the other direction). To design an actual two-axis planer mechanism, we now have to decide what real flexure units come close to these idealizations. These units constitute the key building blocks of the overall flexural mechanism and therefore determine the performance of the mechanism in terms of the desired objectives stated initially.

This is the fundamental underlying principle that can be used in designing high performance planer two-axis error-free flexural mechanisms. Any linear motion flexure unit, which in the designer's opinion comes close to the stated idealizations, can be used as a building block to produce a two DOF planer mechanism. The following sections present designs generated with three types of building blocks: the simple beam flexure, the parallelogram flexure and the compound (or double) parallelogram flexure.

Single Beam Flexure as a building block

Although a simple beam is not a very good single degree of freedom flexure unit, nevertheless due to its simplicity we may use it as a building block in the arrangement discussed in the previous section. From beam bending analysis we know that the beam tip translates (δ) as well as rotates (θ) when it experiences a force. Furthermore, it also exhibits a parasitic error in the X direction (ε).

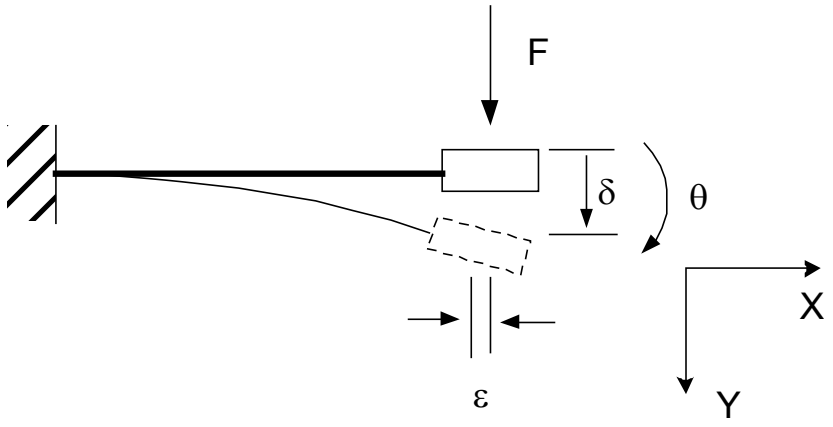


Fig. 2

$$\delta = \frac{FL^3}{3EI} \quad ; \quad \theta = \frac{FL^2}{2EI} \quad \text{and} \quad \varepsilon \sim \frac{\delta^2}{L}$$

where,
L is the length of the beams
E is the Young's modulus of the material
I is second moment of the area of the beam cross-section

Obviously, since the building block itself is not close to the ideal flexure that we desire for units A, B, C and D, we do not expect a very high performance from the mechanisms generated using the simple beam. In subsequent sections we shall employ increasingly more accurate building blocks and thus obtaining better overall flexure performance.

Two-axes planer flexure mechanism designs based on the simple beam flexure

Following the design principle expounded earlier in this document, we come up with a two-axis planer flexure mechanism design, in which the beam flexure is used for Flexure Units A, B, C and D (Figure 3). This is a moderately reasonable design in terms of performance and may be used where accuracy can be compromised but space is at a premium. A quick evaluation of the performance measures is listed here:

- In-plane rotation of the Motion Stage is not completely eliminated
- An X actuation force does produce a parasitic motion of the Motion Stage in Y direction, and vice versa. Although small, parasitic coupling between the two axes is actually present.
- The point of application of X actuation force moves slightly in the Y direction during force application. The same is true for the other axis. Also, an application of X force moves the point of application of the Y force. Hence perfect actuator isolation is not achieved.
- Out of plane stiffness is reasonable but not excellent because the Motion Stage is supported only from one side.

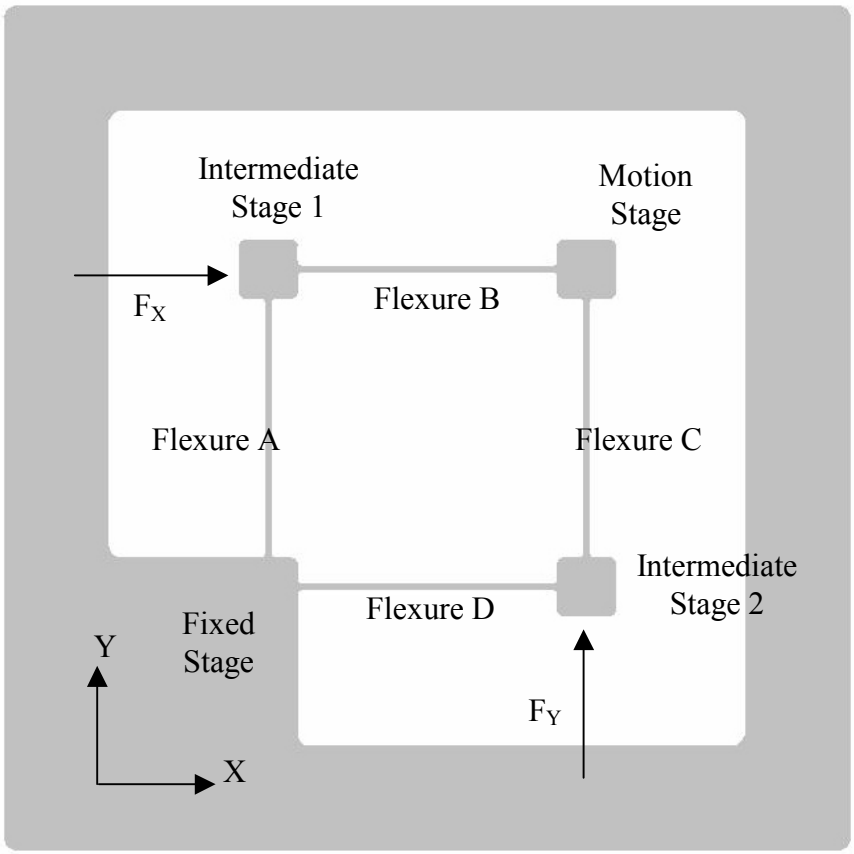


Fig. 3

Many of the above issues are addressed by making novel use of geometric symmetry where the design is mirrored about a diagonal axis, as shown in Fig. 4. The constraint pattern remains the same and the mechanism still has two in-plane translational degrees of freedom. Primes denote the mirrored flexure units. But such symmetry brings about considerable improvements in the performance of the flexural mechanism:

- Rotation of the Motion Stage is further minimized due to the presence of extra rotational constraints arising from the additional parallelogram flexures.
- Parasitic error motion is now reduced. On the application of an X actuation force, the two sides of the mechanism tend to produce parasitic errors in Y direction that oppose each other, and therefore cancel out. Thus, what was lacking due to error-prone building blocks, can be remedied by an innovative use of symmetry.
- Out of plane stiffness is now excellent because of a much better structural loop. The Motion Stage in this design is supported from both sides.
- The design still suffers from actuator cross-sensitivity. Motion in Y direction affects the point of X actuation force application, and vice versa. Furthermore, during the application of an X actuation force, the point of force application moves slightly in the Y-direction as well. Symmetry does not correct this drawback, which may make the design unfit for certain applications.

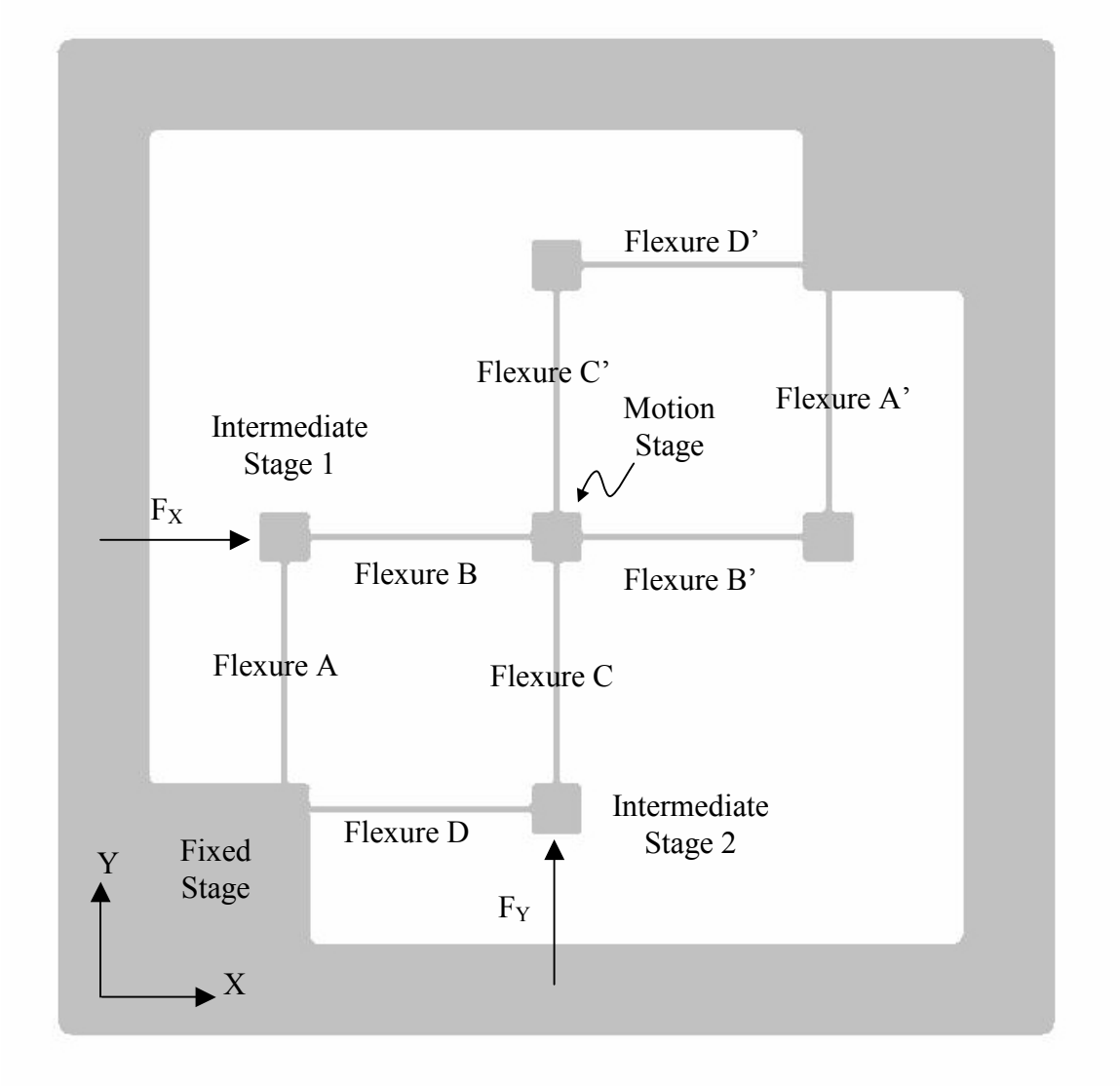


Fig. 4

Although the design illustrated in Fig. 4 is a two-DOF planer mechanism, out of plane motion can also be accomplished by adding horizontal tabs to the beam flexures. This can allow as many as five degrees of freedom: two in-plane translations, and three out-of-plane degrees of freedom. Further details on this idea are presented in subsequent sections, when parallelogram and double parallelogram flexures are used as the building blocks.

Parallelogram Flexure as a building block

The parallelogram flexure unit is a classic design that has been employed in various flexural mechanisms in the past. Fig. 5 provides a schematic of the flexure in its deformed and undeformed configurations. Beam bending analysis can be used to predict the force-deformation characteristics of this flexure. It can be analytically shown that parallelogram flexure offers little resistance to relative motion in Y direction but is very stiff with respect to relative motion in X and rotation. Hence, it a much better approximation for a single DOF flexure as compared to the single beam used in the previous case.

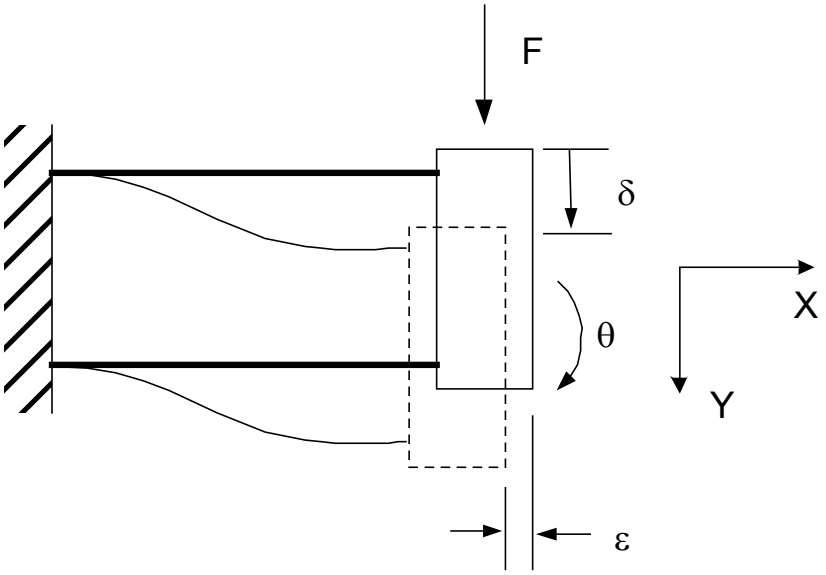


Figure 5

$$\delta = \frac{FL^3}{24EI} \quad ; \quad \theta \approx 2\left(\frac{t}{b}\right)^2 \frac{\delta}{L} \quad \text{and} \quad \varepsilon \approx \frac{3}{5}\frac{\delta^2}{L}$$

where
t is the thickness of the beams
b is the separation between the two beams of the parallelogram
all other quantities are same as defined earlier

As noted in the analytical expressions above, the parallelogram flexure unit does suffer from undesirable parasitic errors. An application of force in the Y direction results in the desired motion δ, in Y direction, and also in undesired motions: ε in the negative X direction, and rotational twist θ. While θ may be eliminated by appropriate placement of the force F, ε is always present. This may or may not be acceptable depending on the application. As of now, we proceed to design a two axes flexural mechanism using this parallelogram unit as a building block. We shall reconsider our choice if the performance of the resulting mechanism is inadequate.

Two-axes planer flexure mechanism designs based on the parallelogram flexure

Following the design principle expounded earlier in this document, we come up with a two-axis planer flexure mechanism design, in which the parallelogram flexure is used for Flexure Units A, B, C and D (Figure 6). This is a better design in terms of performance as compared to the one illustrated in Fig. 3. The accuracy is better but nevertheless errors still exist. This design can used where accuracy can be compromised but space is at a premium. A quick evaluation of the performance measures is listed here:

- In-plane rotation of the Motion Stage in constrained quite well.
- An X actuation force does produce a parasitic motion of the Motion Stage in Y direction, and vice versa. Although small, parasitic coupling between the two axes is actually present.
- The point of application of X actuation force moves slightly in the Y direction during force application. The same is true for the other axis. Also, an application of X force moves the point of application of the Y force. Hence perfect actuator isolation is not achieved.
- Out of plane stiffness is reasonable but not excellent because the Motion Stage is supported only from one side, which is obvious in a side view.

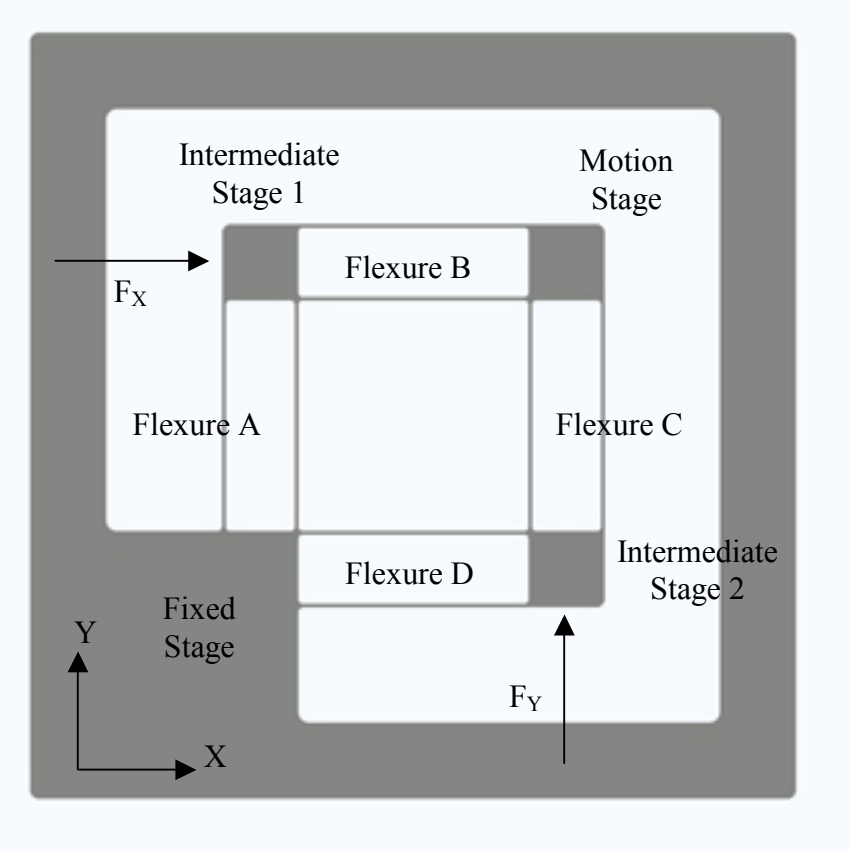


Figure 6

Many of the above issues are addressed by making novel use of geometric symmetry where the design is mirrored about a diagonal axis, as shown in Fig. 7. The constraint pattern remains the same and the mechanism still has two in-plane translational degrees of freedom. Primes denote the mirrored flexure units. But such symmetry brings about considerable improvements in the performance of the flexural mechanism:

- Rotation of the Motion Stage is further minimized due to the presence of extra rotational constraints arising from the additional parallelogram flexures.
- Parasitic error motion is now clearly reduced. On the application of an X actuation force, the two sides of the mechanism tend to produce parasitic errors in Y direction that oppose each other, and therefore cancel out. Thus, what was lacking due of error-prone building blocks can be remedied by an innovative use of symmetry.
- Out of plane stiffness is now excellent because of a much better structural loop. The Motion Stage in this design is supported from both sides.
- The design still suffers from actuator cross-sensitivity. Motion in Y direction affects the point of X actuation force application, and vice versa. Furthermore, during the application of an X actuation force, the point of force application moves slightly in the Y-direction as well. Symmetry does not correct this drawback, which may make the design unfit for certain applications.

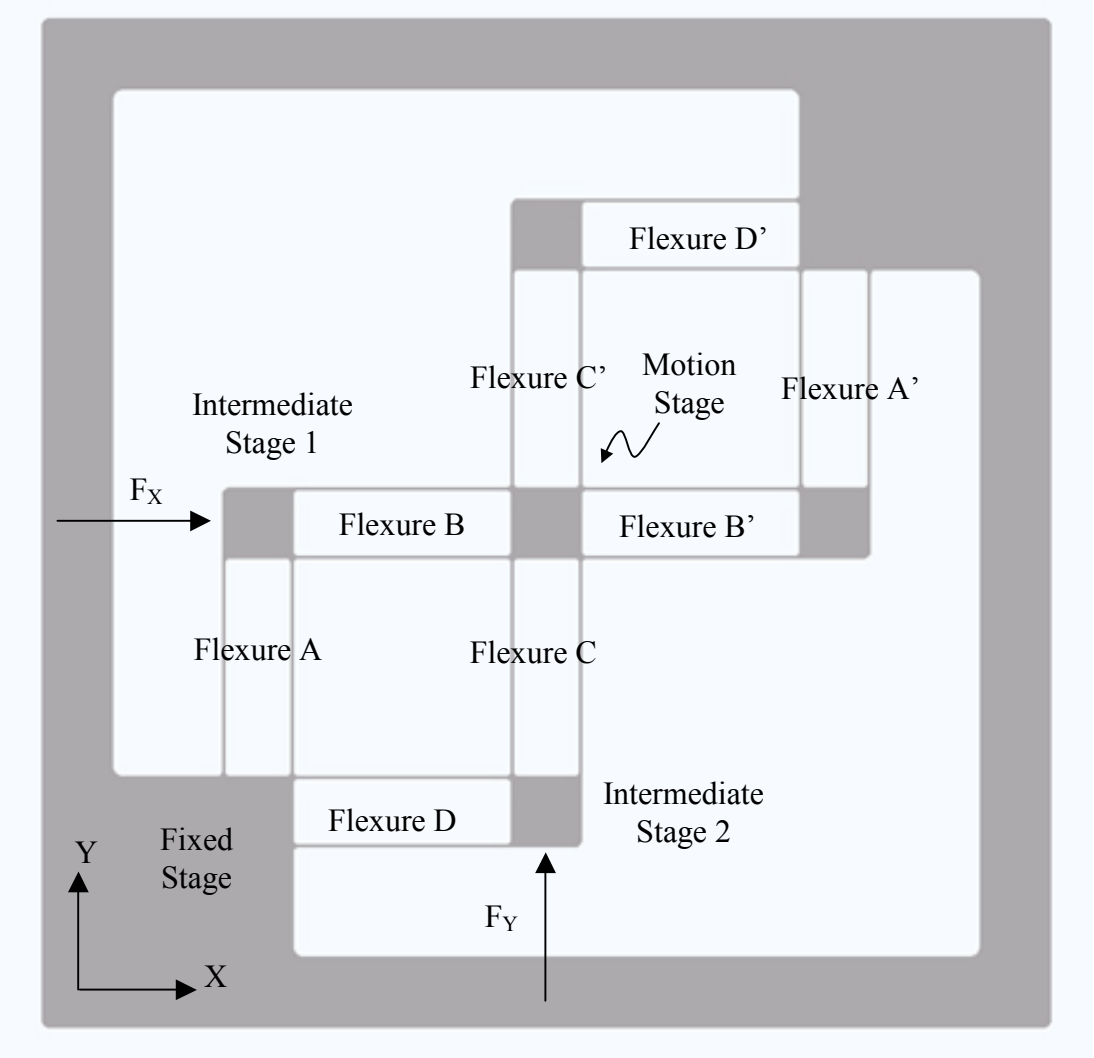


Figure 7

Five degrees of freedom planer flexure mechanism

Using the design principle stated earlier, parallelogram flexures as building blocks, and symmetry to improve the performance, we have been able to generate a two axis planer flexural mechanism that meets the stated objectives quite well.

We now add a new feature to this design which significantly enhances its functionality. Small horizontal blades are introduced in the mechanism in a fashion as shown in Figure 8. The vertical blades belong to the original two-axis flexural mechanism design and the horizontal blades are new additions that now enable out of plane motion as well. By appropriately positioning these horizontal blades each of the intermediate stages, and the Motion Stage can be imparted with a Z degree of freedom.

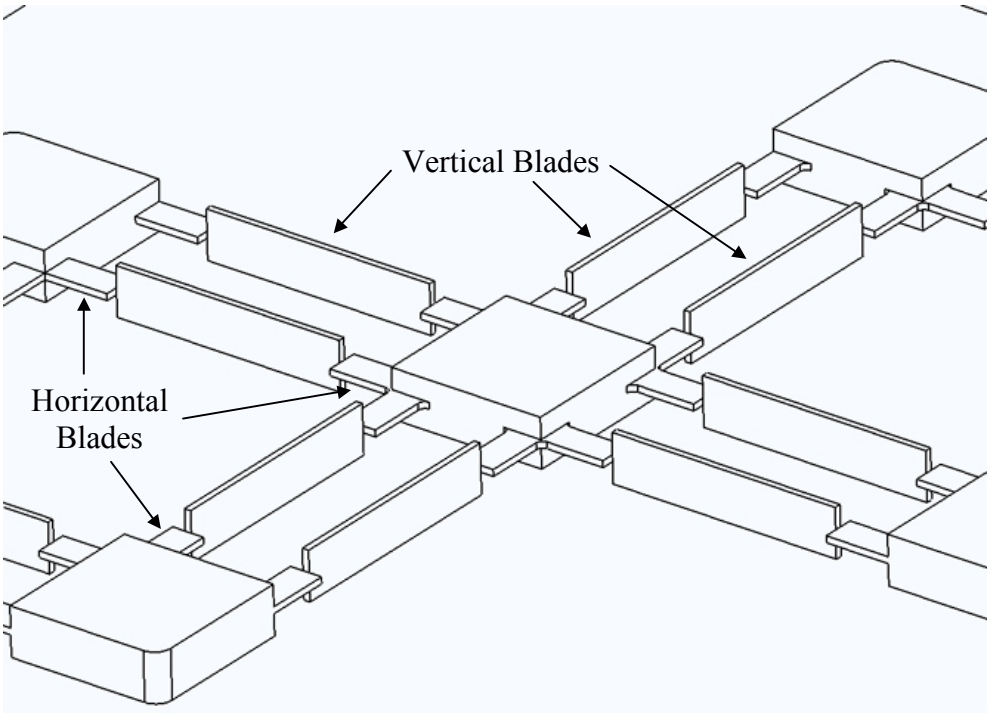


Figure 8

By choosing actuator forces as shown in Figure 9, we can add another three out-of-plane degrees of freedom to the planer mechanism: translation of the Motion Stage along Z, and its rotations about X and Y axes. Since the mechanism now has five degrees of freedom, we call it the 'Pentabot' or 'Pentaflex'. On the other hand if we choose to apply a Z direction force only on the Motion Stage, we obtain a three DOF mechanism with compliance in X, Y and Z directions. In such an embodiment, this mechanism is referred to as 'Tribot' or 'Triflex'. In either case, in plane rotation of the motion stage is completely constrained.

Normal forces F_{xz1} and F_{xz2} acting on opposite intermediate stages result in rotation of the Motion Stage about the X-axis and similarly forces F_{yz1} and F_{yz2} produce a rotation about the Y axis. A combination of these four vertical forces can be used to generate any arbitrary motion along Z and angular twists about X and Y. Clearly, this is not a kinematically exact situation. Four actuation forces for three degrees of freedom implies an over-constraint, but this is not a reason for concern because in compliant mechanisms, elastic averaging eliminates the problems associated with redundant constraints.

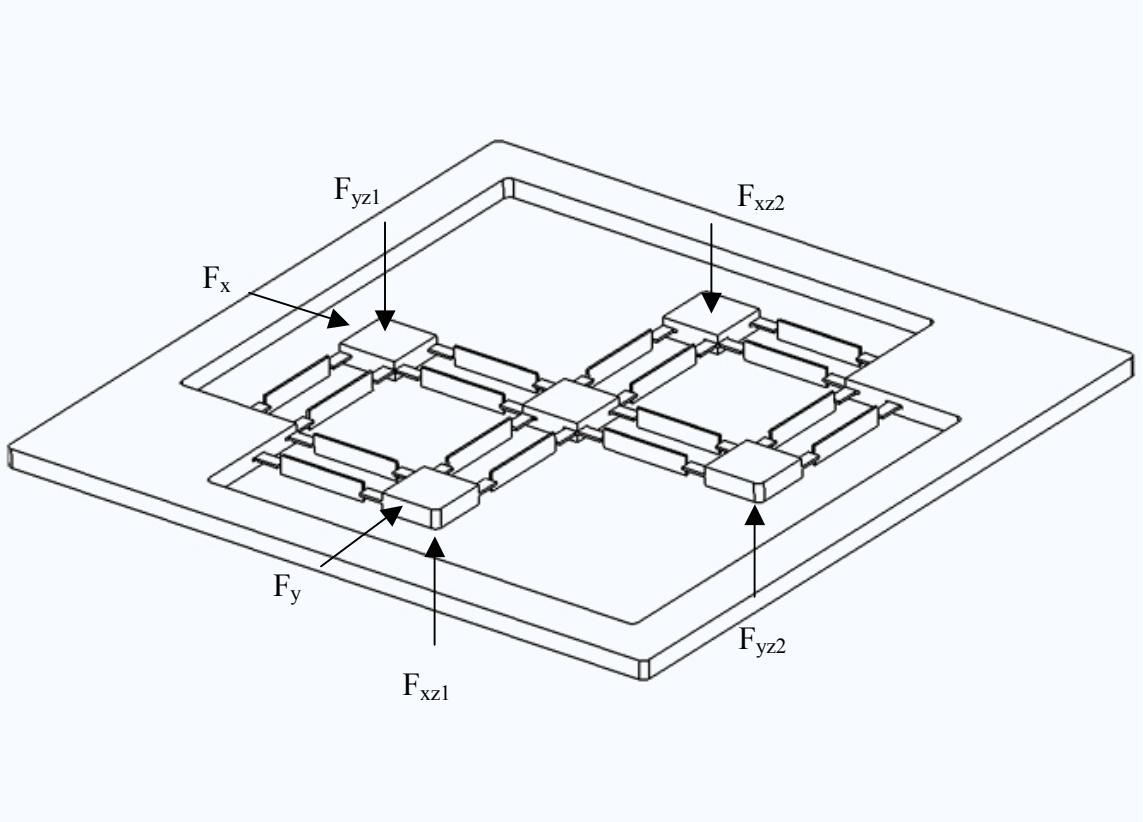


Figure 9

The resulting five degrees of freedom of the Pentabot do have some amount of parasitic coupling. The original two in-plane translations and the out-of plane translation are relatively (not completely, as noted in the previous section) error-free. But the opposing forces F_{xz1} and F_{xz2}, when applied on the intermediate stages, produce not only an X rotation but also a comparable amount of Y rotation. This coupling between the two rotational degrees of freedom can be reduced by shifting the location of the Z-direction forces using extension arms as shown in Fig.10, which presents an actual prototype of this Pentabot design concept.

We can thus get five degrees of freedom from a single planer flexure, which is a novel feature of this design. The degrees of freedom are relatively, but not entirely, decoupled. Minimized by symmetry and appropriate location of actuation forces, parasitic errors are small but do exist. Furthermore, actuator cross-sensitivity exits not only in the original planer degrees of freedom but is also pronounced for the out-of-plane degrees of freedom. Any out-of-plane motion of the intermediate stage causes the point of application of the in-plane forces to move resulting in slippage. Similarly, any in-plane motion will cause the points of application of the normal forces to shift. Some improvement can be obtained by employing non-contact actuation for the out-of-plane motion, for example, use of electrostatic or magnetic forces.

Nevertheless, this design is far from being ideal despite all the fixes. It is prone to parasitic errors between the various degrees of freedom and actuator cross-sensitivity. Most of these anomalies are a consequence of using the parallelogram flexure unit, which is inherently flawed, as our building block. We next propose a design that is constructed from more accurate building blocks.

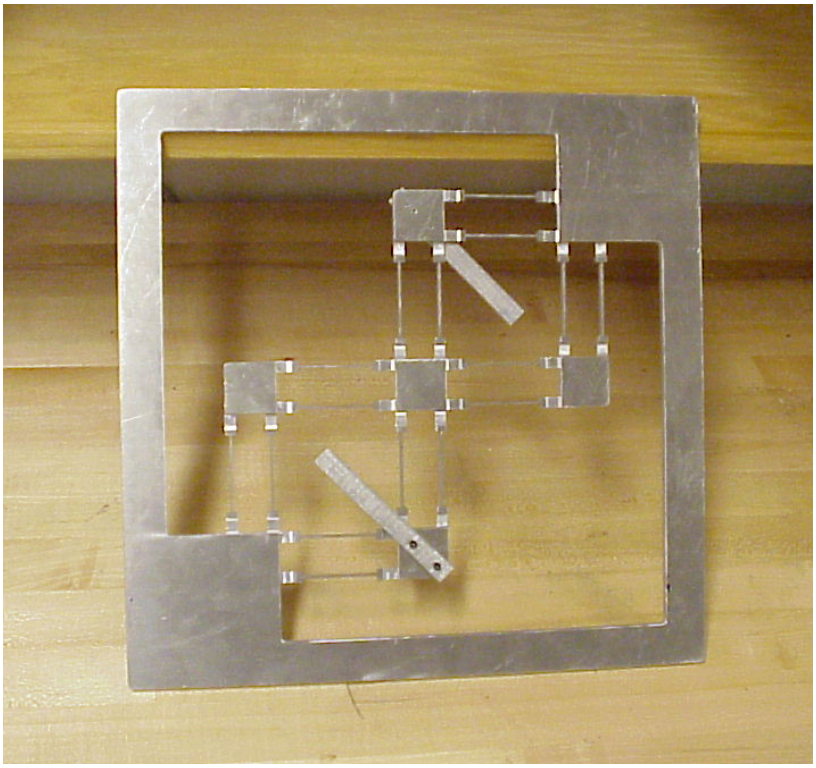


Figure 10

Double Parallelogram Flexure vs. Simple Parallelogram Flexure

As was explained earlier, the simple parallelogram flexure suffers from inherent motion errors due to its geometry. If used as a building block, these errors also appear in the resulting flexural mechanism, thus leading to inadequate performance measures. Instead of using a simple parallelogram flexure, if we use the double parallelogram flexure unit, shown in Fig. 11, as the building block for two-axis planer flexural mechanisms, we can achieve better overall performance. In some of the technical literature this flexure unit is also referred to as the compound parallelogram flexure, folded-beam flexure and crab-leg flexure.

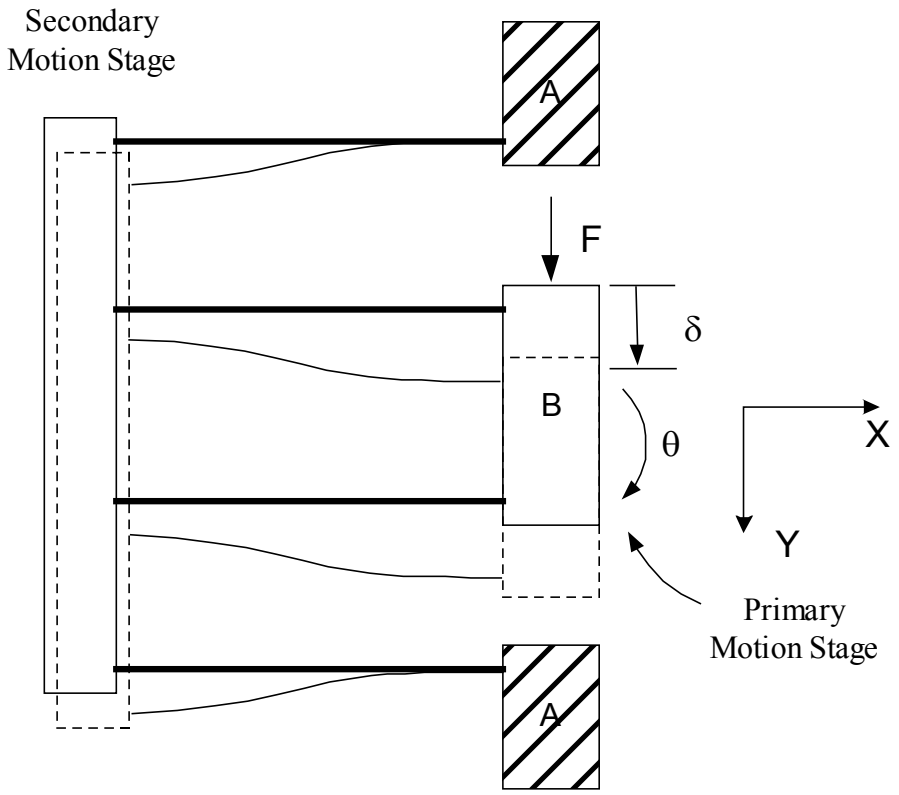


Figure 11

$$\delta = \frac{FL^3}{12EI} \quad ; \quad \theta \approx t^2\left(\frac{1}{b_1^{\,2}} + \frac{1}{b_2^{\,2}}\right)\frac{\delta}{L} \quad \text{and} \quad \varepsilon = 0$$

The double-parallelogram unit is much closer to the idealized flexure units discussed earlier. Analysis shows that the flexure allows relative Y translation between bodies A and B (Fig. 11), but is very stiff in relative X displacement and rotation. The parasitic error ε are theoretically non-existent in this case because any length contraction due to beam deformation is absorbed by a secondary motion stage, the motion of which is inconsequential. There does exist a rotational parasitic motion, which can be eliminated

by appropriate location of the Y direction force. Hence, body A exhibits perfect Y-translation with respect to body B on the application of a Y direction force. This is where this flexure unit scores over the parallelogram flexure.

Two-axis planer flexure mechanism designs based on the double-parallelogram unit

We now proceed to repeat the process of designing a planer two-axis flexure mechanism following the previously stated design guidelines, but this time we shall use the double-parallelogram flexure as the fundamental building block.

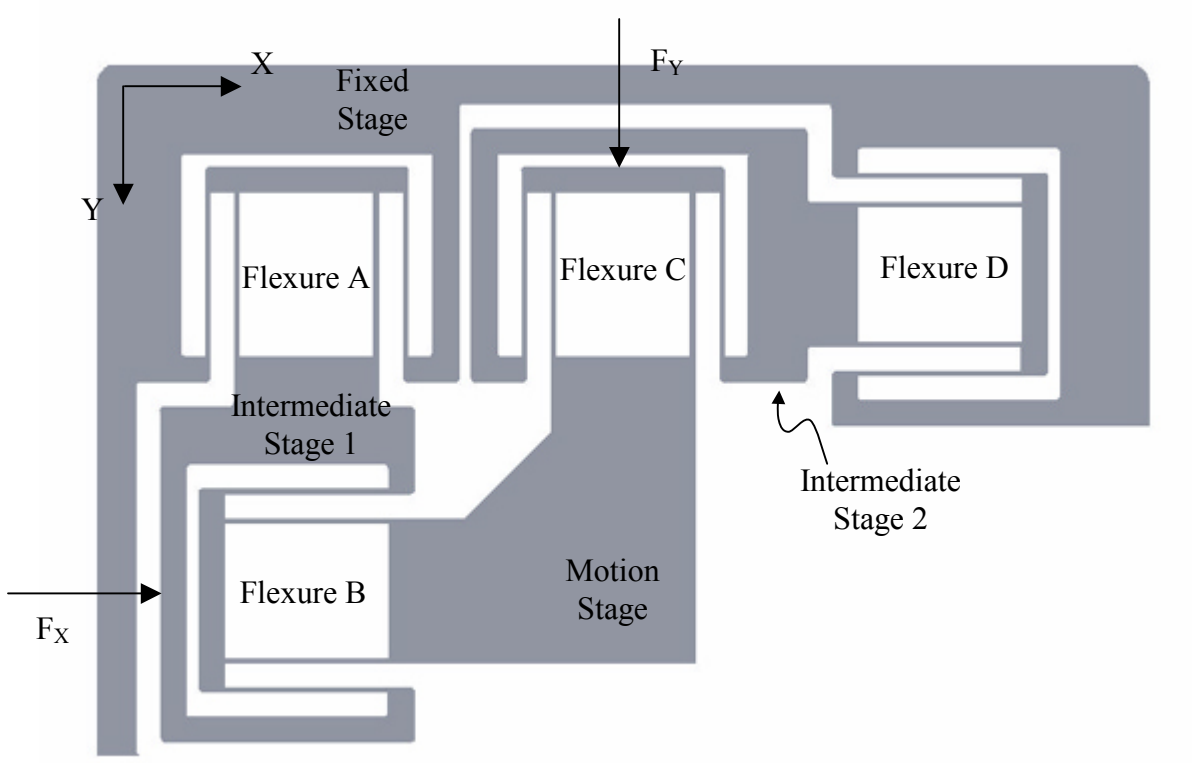


Figure 12

Since the double-parallelogram flexure is a relatively error-free unit, the resulting mechanism design (Fig. 12) is a significant improvement over the previous ones in terms of performance

- In-plane rotation of the Motion Stage is very well constrained
- The two axes are perfectly decoupled with no parasitic errors. X actuation force produces X motion only. The same holds for the other axis
- Actuator Isolation is also achieved in this case. The point of application of X actuation force experiences zero Y displacement as the X force is applied. Furthermore, any Y motion of the Motion Stage has no effect, whatsoever, on the point of application of X actuation force, and vice versa. Thus actuator cross-sensitivity is totally eliminated.

- Out of plane stiffness is reasonable but not outstanding since the Motion Stage is not supported for all sides, and the out-of-plane structural loop is open.

Using the same design principle, but a slightly different arrangement of the four double-parallelogram flexure units, an alternative and more compact embodiment of this planer two-axis flexure mechanism concept has be used by the same inventors in the design of a precision measurement device for the characterization of force-displacement behavior in compliant micro-structures. This is the first instance in which the design concepts described in this document have been reduced to practice.

A further improvement in the out of plane stiffness can be made by resorting to symmetrical mirroring of the double parallelogram flexure layout in Figure 12. This step results in a design embodiment as shown in Figure 13. This design is yet more superior to the previous ones in terms of meeting the desired objectives. Obviously, the out-of-plane stiffness is significantly improved because the Motion Stage is supported from all sides. More importantly, perfect symmetry makes the design more robust against parasitic errors. Symmetry strengthens the existing constraints on the Motion Stage without over-constraining the mechanism.

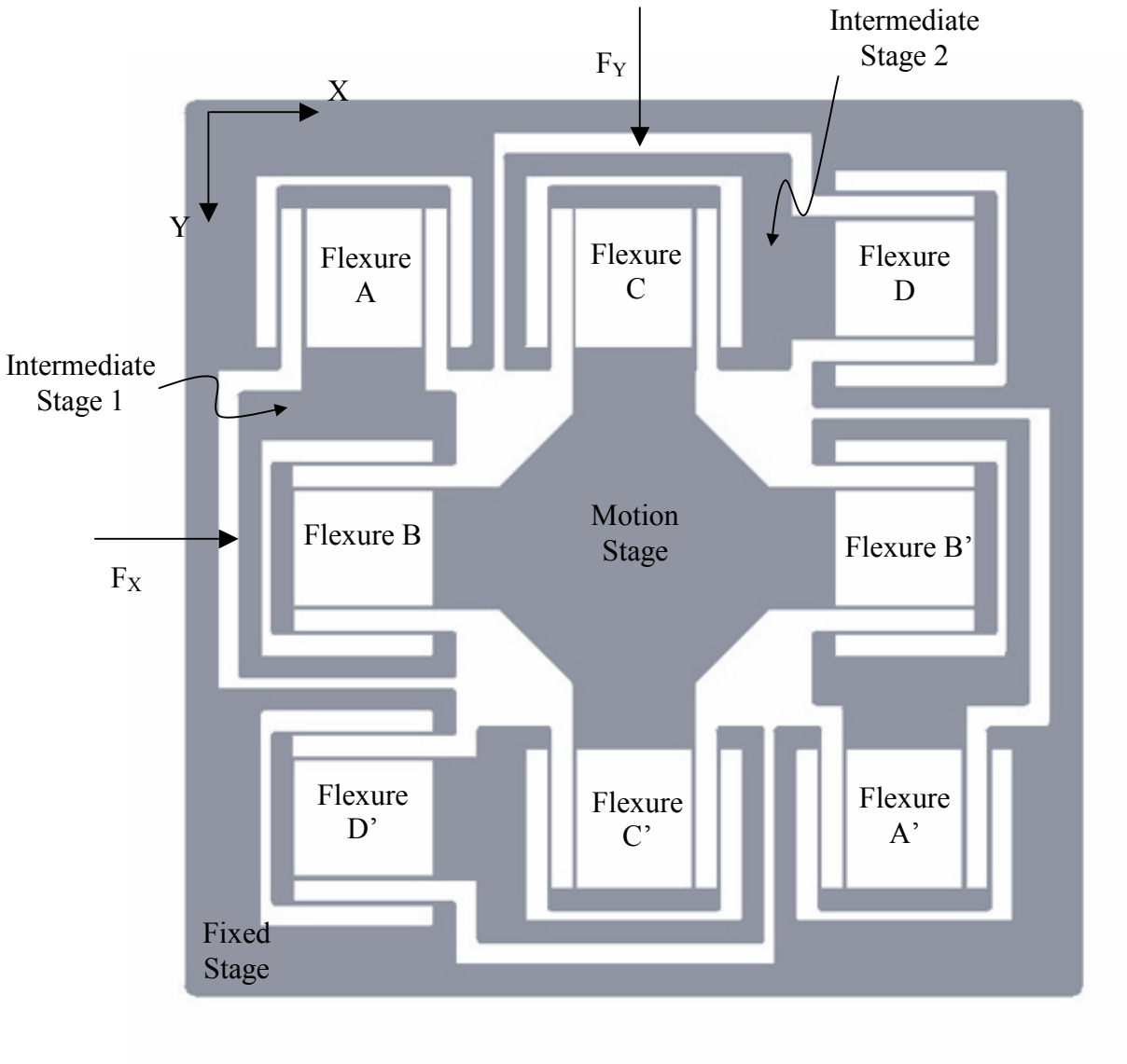


Figure 13

This embodiment is almost ideal as far as performance is concerned, and is the best design that we have come up with so far. Since the constituent building blocks are error free, the entire mechanism itself is free of any errors at both the Motion Stage and the points of actuator force application.

We can now impart out-of-plane motion to the planer mechanism by introducing horizontal blades in some of the double-parallelogram units, as shown in Figure 14. Horizontal blades are added such that the intermediate stages themselves do not have any Z degree of freedom. The secondary motion stages of the double-parallelogram flexures, the motion of which is inconsequential, are the ones that attain a Z degree of freedom. The advantages of this scheme will be elucidated in the following paragraphs.

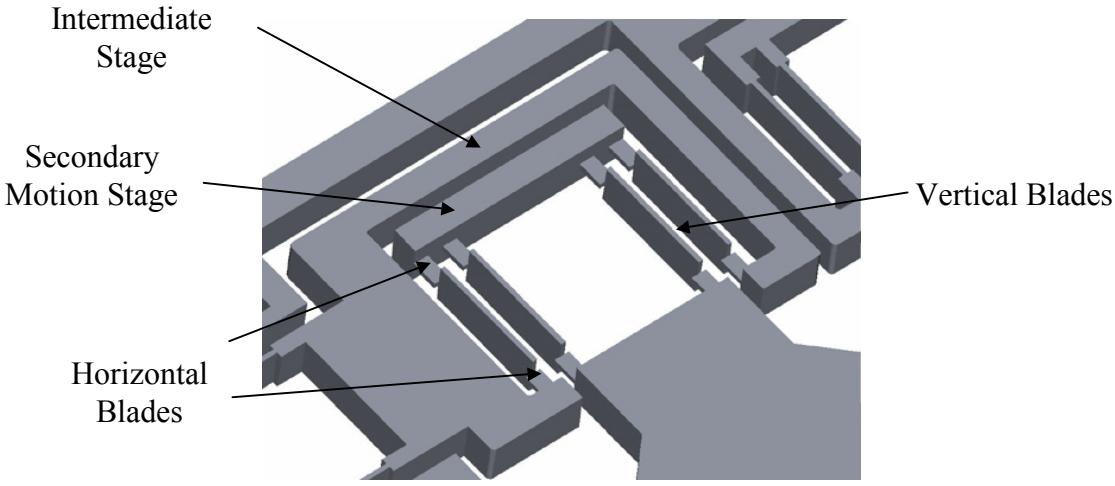


Figure 14

Referring to Figure 15, by applying normal actuator forces, we can add another three out-of-plane degrees of freedom to the planer mechanism: translation of the Motion Stage along Z, and its rotations about X and Y axes. We thus get another version of the Pentabot (or Pentaflex), which is an improvement over its predecessor in many regards. We may also choose to apply a single Z direction force on the Motion Stage, to obtain a three DOF mechanism with compliance in X, Y and Z directions. In either case, in-plane rotation of the Motion Stage is perfectly constrained.

Normal forces F_{xz1} and F_{xz2} result in a rotation of the Motion Stage about the X-axis and similarly forces F_{yz1} and F_{yz2} produce a rotation about the Y-axis. A combination of these four vertical forces can be used to generate any arbitrary motion along Z and angular twists about X and Y. It is very important to mention that the normal forces are not applied on the intermediate stages; rather, they are applied on the sacrificial stages of the four double-parallelogram flexure units that support the Motion Stage. This isolates the normal forces from the in-plane forces. The normal forces no longer affect the points of application of in-plane forces, thus reducing actuator cross-sensitivity. Furthermore, the out-of-plane rotations are much better decoupled as compared to the previous Pentabot design.

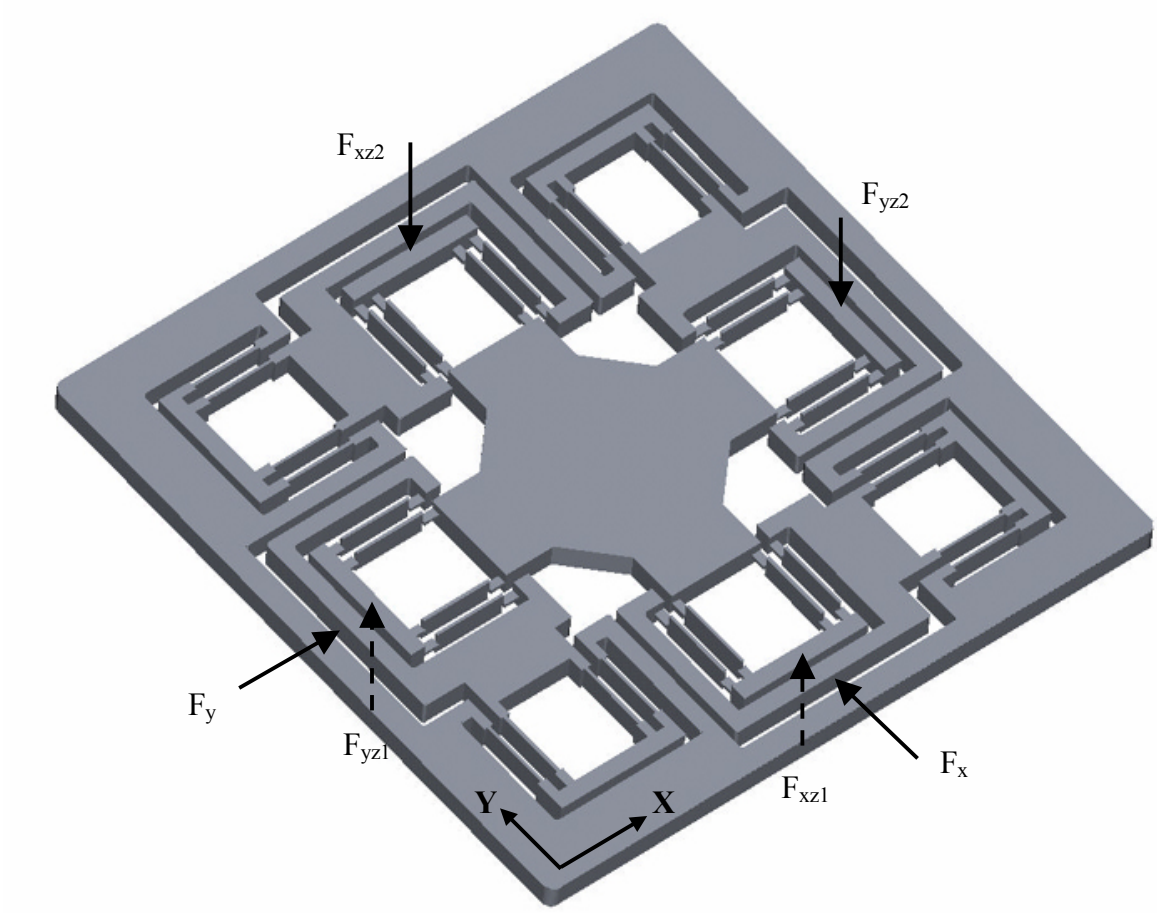


Figure 15

Thus, a planer flexural mechanism has been invented that is compact, because of parallel elasto-kinematics; very accurate, due to error free building blocks, symmetry, and judicious placement of actuators; provides multiple DOF, enabled by horizontal blades; and is highly repeatable since it operates within the elastic limits of the material. All these features make this design invaluable for micro as well as macro precision machines.

Furthermore, the area of the motion stage can be increased by adding another platform above the plane of this Pentaflex, which is attached to the motion stage but does not contact any of the other components of the flexure mechanism. This way, a large working area can be obtained thus increasing the space efficiency of the design. This idea is illustrated in Fig. 16 in a sectional side view. On a micro-scale this can be done by the process of wafer-bonding.

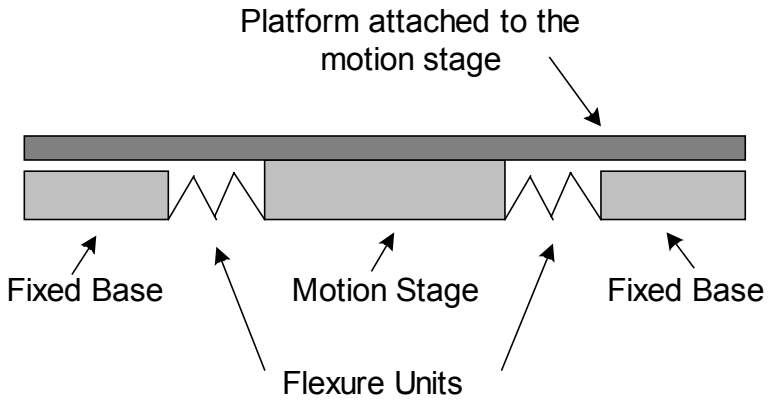


Fig.16

Use of other flexure units as building blocks for planer mechanisms

Any flexure unit that produces error-free single axis translation can be used as a building block for the two-axis planer mechanism. There is no reason why one should be restricted to the use of parallelogram or double-parallelogram flexures. Furthermore, a combination of these flexures units can also be used depending on what the performance specifications are. In one particular embodiment, we use double-parallelogram units for Flexures A and D, and parallelogram units for Flexures B and C. The final choice is always dependent on the actual application, level of performance desired and space constraints.

Conclusions

In all the above designs we have following a common design strategy:

1. Pick a building block that is close to an idealized single DOF flexure unit which provides translation along one direction only.
2. Assemble these building blocks following the guidelines/rules presented in the early part of this document, to generate a planer 2-DOF parallel mechanism
3. Make use of symmetry to significantly enhance the performance of this mechanism
4. Include horizontal tabs appropriately to allow for out-of-plane motions, resulting in a five DOF planer mechanism
5. The location of actuation forces is very critical as this determines the degree of coupling between the various degrees of freedom. Analysis can be done to obtain ideal force locations that would minimize the motion errors in the mechanism.
6. Combination of different building blocks may be used in the same mechanism to optimize space (or size) and performance

Claims

- Parallel elasto-kinematics is used for planer flexure mechanism design that results in simple and compact embodiments.
- A systematic approach for designing planer two and multiple axis flexural mechanisms is presented by elucidating the underlying design principles/rules/guidelines necessary to achieve high performance. This methodology also highlights the importance of using error-free building blocks to construct the planer flexural mechanisms.
- Two-axis (2 DOF) planer mechanism designs based on the above principle and using the simple beam flexure, the parallelogram flexure and the double-parallelogram flexure as building blocks, are presented in Figure 3, 6 and 12 respectively.
- Novel ideas for using symmetry to increase out of plane stiffness and improve performance and robustness in design are presented. Errors due to imperfect building blocks are corrected by use of geometric symmetry. These ideas are implemented in the designs presented in Figures 4,7 and 13.
- An outstanding innovation disclosed in this invention is the use of small horizontal blades (Figures 5 and 11) to produce out-of-plane motion in planer mechanisms.
- Based on the above principles, planer three DOF flexural mechanisms, referred to as Tribot/Triflex (Figure 9), and planer five DOF flexural mechanisms, termed as Pentabot/Pentaflex (Figure 15), are presented. The 5 DOF Pentabot design of Fig. 15 provides highly decoupled translations in X, Y and Z, and rotations about X and Y. In-plane rotation is entirely eliminated. This design embodiment is compact because of parallel elasto-kinematics; very accurate, due to error free building blocks, symmetry, and judicious placement of actuators; provides multiple DOF, enabled by horizontal blades within the plane; and is highly repeatable since it operates within the elastic limits of the material.
- Analysis has been done (not presented in this document because of its mathematical complexity) to compute appropriate placement of actuation forces that would minimize motion errors (i.e. parasitic coupling between axes and actuator cross-sensitivity)
- In any of the designs presented in Figures 4, 7 and 13, the space efficiency (size of the motion stage vs. size of the overall mechanism) can be greatly improved by attaching an additional platform to the motion stage in a plane parallel to the plane of the mechanism (Fig. 16).
- In all the designs presented in this document identical building blocks were used in a given flexure mechanism. But this need not be the case. We can bring about the following variations while constructing a flexure mechanism from building blocks
    - Use different building blocks in the same mechanism, e.g. use a combination of beam flexures, parallelogram flexures, and double parallelogram flexures to construct a planer flexure mechanism
    - Vary the geometric parameters from unit to unit, i.e., the various parallelogram flexure units used in a flexure mechanism need not all be identical. The blade length, thickness, spacing can all be varied from one unit to another.

If we allow for the above variations in design, namely use of different building blocks and variation in the geometric parameter of individual building blocks, and further permit a variation in the actuation force location as well, we can precisely tailor the performance of the overall flexure mechanism.

We may try to minimize parasitic error motions, or on the contrary we made try to produce a desired amount of parasitic motion, for example, there might be an application which does require a certain amount of X motion when a Y force is applied. These deterministic variations vastly increase the design space. By 'deterministic' what we mean is that, analysis can exactly determine the effect that these variations will have on the performance of the overall flexure mechanism.

The capability to deterministically tailor the performance of a flexure mechanism by means of the above mentioned variations is very significant design tool.

Potential Applications:

The designs presented in this document are very fundamental and can be used over a wide range of macro, meso or micro scale precision machines where decoupled multiple degrees of freedom are required. Potential applications can be found in optical instruments, Micro and Nano Electro Mechanical Systems, precision metrology, etc. A few specific applications are mentioned here

- IBM is conducting research on an AFM (Atomic Force Microscope) based data storage concept called the "Millipede" that potentially has an ultrahigh density, terabit capacity, small form factor, and high data rate. The stage that holds the storage media has to be provided with control over the Z motion and planer tilts. It also requires translation capability in the X and Y directions to pan a certain amount of area. The planer Pentabot can be put to use in the implementation of this data-storage concept. As shown in Fig. 3, the polymer storage media (lower most stage) requires five degrees of freedom (x, y, z1, z2 and z3), which are readily and accurately provided by the pentabot. **(IBM is not interested in a collaboration)**

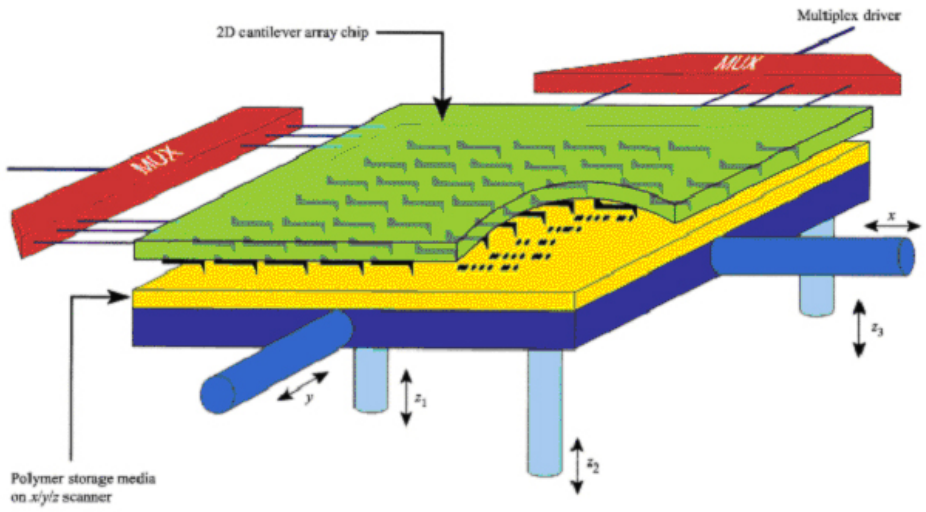


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- High precision two or more axis motion stage: In a certain a high precision microscope that is used for observing the interaction between protein and DNA molecules, the stage that holds the specimen needs to be panned with sub-micron precision. The 2 DOF planer designs presented in this document are ideal candidates for such applications
- Micro-Electro Mechanical (MEM) motion stages for actuators and bearings: These designs are of very significant consequence to MEMS technology where structures need to be etched on Silicon wafers. Planer designs that provide

multiple DOF can have an unprecedented impact on MEMS actuators, bearings and guides.

- Micro-Electro Mechanical (MEM) sensors: A very significant application of the Pentabot design presented in Figure 12, is in the design of multiple axis MEM accelerometers. Being compact and planer, the pentabot is ideally suited for micro-fabrication on Silicon wafers. Because it is free of parasitic errors and cross-sensitivity, it makes the sensor design tractable and ensures high device performance. The capability of producing out of plane motion using horizontal blades makes this invention very promising for the design of two, three or five axis accelerometers.