# Design of Compliant Mechanisms for Practical Applications using Selection Maps

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## Abstract

We present an interactive map-based technique for designing compliant mechanisms that meet the requirements of a practical application. Our map juxtaposes user-specifications with the attributes of real compliant mechanisms stored in a database so that not only the practical feasibility of specifications can be discerned quickly but also modifications can be done intuitively to the existing compliant mechanisms.

The upper and lower bounds on the specification variables entered by a user are used to show a part of the feasible region on a 2D map. The basis to draw the feasible region is the spring-leverage (SL) model, which is used to characterize the kinematic and elastic behavior of single-input-single-output compliant mechanisms for the purpose of static analysis. The SL model consists of three constants: (i) the input spring stiffness, (ii) the output spring stiffness, and (iii) the inherent geometric amplification factor. The three constants are computed a priori for the compliant mechanisms in the database by using the finite element analysis of their beam-based 2D meshed models. This enables us to show compliant mechanisms on the map. If any of them lie within the feasible region and has the appropriate geometric amplification factor, a solution is readily found. If not, a solution is found by modifying the mechanism. The meshed models can be interactively modified to change their attributes such as: (i) overall size in x and y directions, separately and together, (ii) uniform re-sizing of the in-plane widths of all the beam elements, (iii) uniform re-sizing of the out-of-plane thicknesses of the beam elements, and (iv) material. A case study that describes the design procedure in detail is also presented while additional case studies are posted on the web-site: www.mecheng.iisc.ernet.in/~hegde/paper.

**Keywords:** Compliant mechanisms, spring-leverage model, selection map

## 1 Introduction

Conventional mechanisms use rigid links connected by pin joints, sliding joints and others to transfer or transform input force and input motion. Compliant mechanisms, on the other hand, use elastic deformation of their members. In this paper, we present a design technique for compliant mechanisms in view of practical applications. Our method is based on selection from an existing set of compliant mechanisms and modifying them to suit the practical requirements of a new problem. The compliant mechanisms in the database are already manufactured and some are used in real applications.

Predicting the movement of the output point of a compliant mechanism for applied forces and boundary conditions, in general, is not intuitive. Thus, if one uses pure intuition, the design of a compliant mechanism can be a very difficult task, especially for a novice designer. Developing systematic design techniques has been an active area of research in the field of compliant mechanisms. Two such design techniques are the pseudo-rigid body approach ([1,2]) and the topology optimization approach ([3,4] and references therein). The pseudo-rigid body approach uses the rigid-body linkage analysis methods with torsional and translational springs in the analysis of compliant mechanisms. This approach gives a preliminary design which may be further refined to meet the design goals by using a detailed elastic deformation analysis. Topology and the configuration of linkage, i.e., the number of links and their connectivity, are usually assumed in this method. It is also difficult to automate this method. In contrast, the topology optimization method uses the elastic deformation analysis intrinsically coupled with optimization algorithms to arrive at the optimal topology of the mechanism with minimal user intervention. Two open-access software programs (TopOpt [5] and YinSyn [6]) that use the topology optimization method are available.

Both the methods are fairly sophisticated and require good understanding of kinematics, elastic deformation analysis, and optimization in order to implement and use them. Usually, the solutions obtained with these need some manual refinements in order to meet the practical requirements after a basic functional intent is met. Efforts will continue to make these methods automatic and practically useful. Because of the above two categories of methods, numerous compliant mechanism designs exist today for different applications and functionalities. It is plausible then that one or more of these designs might suit a new practical application. Systematic exploration of this premise is the focus of this paper.

In this work, we take a different approach towards design of compliant mechanisms which is suitable even for a novice or a user who is not familiar with the intricacies of kinematics, elastic deformation mechanics and optimization. It also involves the user in the process to design a compliant mechanism that meets the functional and the practical requirements specified by the user. Our approach is based on selection among known designs, most of which were produced by topology optimization but some were conceived intuitively. We do this for two reasons: (i) the number of compliant mechanisms in use for practical applications is steadily growing and there is a big enough set of these mechanisms to choose from, and (ii) all practical requirements cannot be easily incorporated into the topology optimization or pseudo rigid-body approach at the current stage of their development.

Our approach allows the representation of practical requirements as well as the behavior of the real compliant mechanisms in the database, simultaneously on a 2D map. Thus, the methodology presented here allows the user to see on a 2D map how the different existing compliant mechanisms fare against the requirements of a new practical application. The map then aids in selecting a suitable mechanism for that application. It is possible that none of the mechanisms in the database is able to satisfy the requirements at first. The same map is then used to modify the selected mechanism in conjunction with a graphical user interface (GUI), developed in Matlab, so that the practical requirements are met by the modified mechanism. The interactive procedure is rapid and yields mechanisms that satisfy the requirements of the problem.

The basis for the simultaneous representation of the practical requirements and the suitability of the existing mechanisms on a 2D map is the characterization of a compliant mechanism by only three lumped spring-leverage (SL) model constants (see [7] and [8]). The SL model is explained in Section 2. These lumped-model SL constants are parameterized as a function of the size, aspect ratio, material, beam width and the overall thickness of the compliant mechanisms, and are stored a priori or calculated in real time. The design process, which consists of the representation of the practical requirements as a feasible region on the 2D selection map and the modification of the selected mechanism on the same map to match the user-specified requirements, is explained in Section 3. A graphical user interface (GUI) developed in Matlab will be used to present a case study that illustrates the use of our approach on a real practical problem. This is done in section 4. Additional case studies are posted on a web-site for immediate reference of the readers (www.mecheng.iisc.ernet.in/~hegde/paper). Main points of the paper are noted in section 5.

## 2 Spring Leverage (SL) model

The kinematics and the elastic deformation of the compliant mechanisms can be expressed by the SL model to account for its displacement-amplifying and force-transferring features [8]. The concept of the SL model is similar to representing the static behavior of an elastic structure by a single spring stiffness.

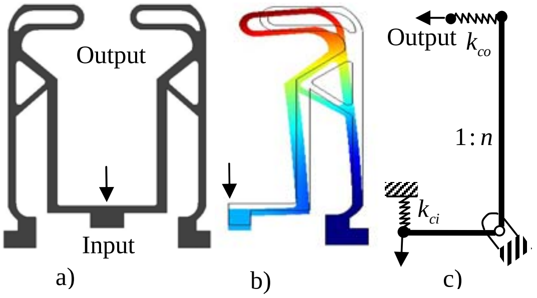


**Fig. 1:** Shown are a) a compliant mechanism, b) its symmetric half used in analysis, and c) its representation as a lever with geometric amplification factor, $n$, the input side spring stiffness, $k_{ci}$, and the output side spring stiffness, $k_{co}$.

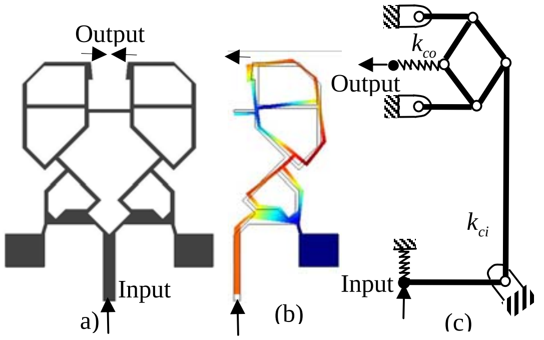


**Fig. 2:** Shown are a) a compliant mechanism, b) its symmetric half used in analysis, and c) the need of a leveraging mechanism to revert the direction of motion from input to output in the SL model.

However, the compliant mechanism under consideration here, have two ports of significance: the input port and the output port. The input port is where the input force or input displacement is applied and the output port is where the output force or the output displacement is desired. Consequently, two springs are introduced at the input and the output ports, the input port spring stiffness, $k_{ci}$, and the output spring stiffness, $k_{co}$. In order to account for the amplification between the input and the output, a lever is introduced between the two springs (see Fig. 1). Sometimes, as shown in Fig. 2, an inverting mechanism may be needed to reverse the direction of the lever on the output side to match the specified output direction.

The lever in Fig. 1 and the leverage mechanism of Fig. 2 are only symbolic but they can also be designed to match $n$ of the compliant mechanism. The important feature is the amplification of the lever, called the inherent geometric amplification factor, $n$, of the mechanism. Thus, as shown in Figs.1 and 2, the compliant mechanism is replaced by a lever or a leverage that has finite stiffnesses at the input side as well as at the output side. It can be seen that this lumped model can also explain the following fact observed in compliant mechanisms: the de-amplification factor of the motion from the output to the input is not the reciprocal of amplification factor from the input to output. For a given compliant mechanism, the SL constants can be determined with the help of two load cases. When the SL constants of two compliant mechanisms are the same, they both behave in exactly the same way as far as the single-input-single-output static application is concerned. The method of computing the SL constants is explained next.

### 2.1 Determining the SL constants

The SL constants, as explained above, describe the abstracted input-output terminal behavior of a compliant mechanism. In order to compute them, we need to do two deformation analyses of the compliant mechanism under two different conditions of applied loads.

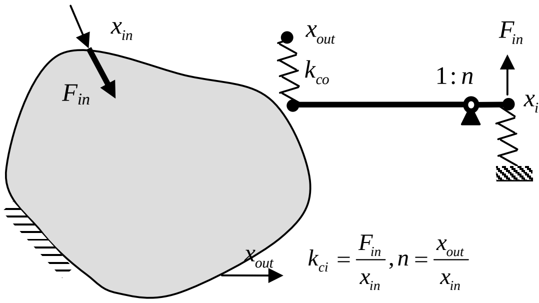


**Fig. 3:** The solution of the first load case is found out by finite element analysis. This is used to find the input side stiffness, $k_{ci}$, and the inherent geometric amplification factor, $n$.

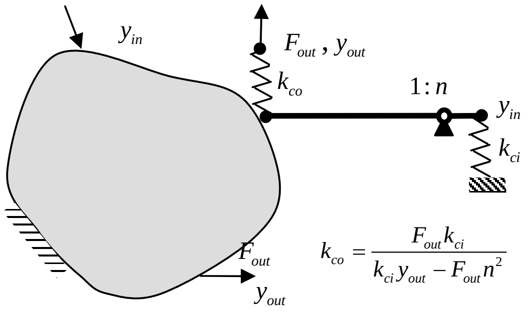


**Fig 4:** The solution of the second load case is found out by finite element analysis. The solution is then used to find the output side stiffness, $k_{co}$.

In the first load case, we apply an input force $F_{in}$ only and measure the displacement $x_{in}$ at the input (see Fig. 3). With these, as in an ordinary lumped spring modeling of an elastic structure, we compute the input-side stiffness, $k_{ci}$.

$$k_{ci} = \frac{F_{in}}{x_{in}} \tag{1}$$

By also measuring the output displacement, $x_{out}$, we compute the inherent amplification, $n$.

$$n = \frac{x_{out}}{x_{in}} \tag{2}$$

Next, we consider a different loading condition as shown in Fig. 4, to get $k_{co}$. Here, the force $F_{out}$ is applied only at the output to measure the resulting displacement at the input, $y_{in}$, and at the output, $y_{out}$. By considering the SL model of this situation shown in Fig. 4, we write the potential energy, $PE$, as follows.

$$PE = -F_{out} y_{out} + \frac{1}{2} k_{co} (y_{out} - n y_{in})^2 + \frac{1}{2} k_{ci} {y_{in}}^2 \tag{3}$$

Note that the potential energy is the sum of the strain energy and the negative of the work done by the external forces. By differentiating $PE$ with respect to $y_{in}$ and $y_{out}$, and equating them to zero for static equilibrium, we get two equations that can be solved to get $y_{in}$ and $y_{out}$. The resulting expression for $y_{out}$ is used to solve for $k_{co}$ as follows.

$$k_{co} = \frac{F_{out}}{y_{out} - n y_{in}} \tag{4}$$

Thus, the lumped SL constants for a given compliant mechanism can be found out by doing two finite element analyses of the mechanism under two load cases. It is important to note at this point that these SL constants for a mechanism will depend on its attributes such as the in-plane width, thickness, etc., of the members.

## 3 Selection Methodology

The selection and re-design methodology consists of three steps. The first one involves showing the requirements of the practical application on the 2D map. The second step consists of representing the existing compliant mechanisms in the database on the same map, and the selection of the mechanism(s) that is(are) closest to the feasible region. The third step consists of the process of re-design of the selected mechanism so that the modified mechanism satisfies the practical requirements of the problem. These three steps are explained next.

### 3.1 Plotting the requirements on the map

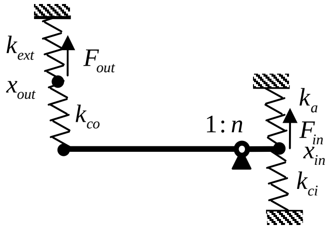


**Fig 5:** The variables involved in a single-input-single output compliant mechanism. Note that the compliant mechanism is represented by $k_{ci}$, $k_{co}$ and $n$.

The variables involved in specifying the function of a single-input-single-output compliant mechanism are listed below.

i) Input force = $F_{in}$

ii) Input displacement = $x_{in}$

iii) Output load = $F_{out}$

iv) Output displacement = $x_{out}$

v) Actuator's stiffness (assumed to be linear) at the input = $k_a$

vi) External stiffness (assumed to be linear) at the output = $k_{ext}$

The user specifies upper and lower bounds (maximum and minimum values) for the specification variables. We want to first find out the ranges of values of the SL constants that can meet these specifications. For that, the SL model shown in Fig. 5 is used to get the following expressions for $k_{ci}$ and $k_{co}$.

$$k_{ci} = \frac{\left(F_{in} + nk_{co}x_{out}\right)}{x_{in}} - n^2 k_{co} - k_a \qquad (5)$$

$$k_{co} = \frac{\left(F_{out} - k_{ext}x_{out}\right)}{x_{out} - nx_{in}} \qquad (6)$$

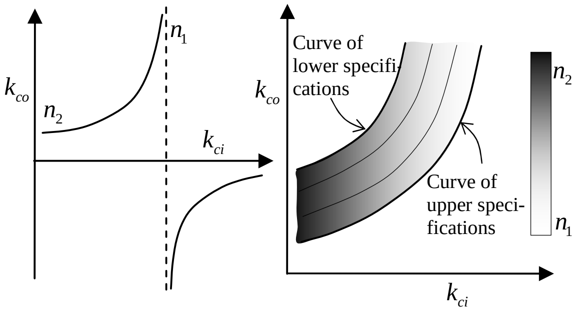


**Fig. 6:** The feasible region is bounded by the upper and lower specification variables curves. The region is filled with a gray color scale with white indicating $n_1$ and black indicating $n_2$.

These expressions help us draw the curve of lower specifications on a $k_{ci}$-$k_{co}$ map by varying $n$ while fixing all the specification variables to their lower specification values. The significance of this curve is that $k_{co}$, $k_{ci}$ and $n$ corresponding to a point belonging to this curve satisfy the lower bounds of the specification variables. Similarly, the curve of upper specifications can be drawn using the upper bounds on the specification variables and varying $n$. The value of $n$ that makes the $k_{co}$ of Eq. 6 negative, forms the lower limit of $n$. The upper limit of $n$ is taken to be 50 or the $n$ that leads to a negative $k_{ci}$ in Eq. 6, whichever is lower. Now, several such curves can be drawn which lie in between the lower specifications curve and the upper specifications curve by varying the six specified variables uniformly between the lower and the upper bounds (see Fig. 6). We call the area bounded by the upper and lower specifications curve as the feasible region though the actual feasible region will be much larger than this. Thus, we only consider a subset of the total feasible region. We do this for identifying each point ($k_{ci}$, $k_{co}$) within the region with a specific value associated with every specification variable and hence with a unique $n$, which would have been otherwise not possible. Thus, some valid area is forgone but the consideration of only the subset greatly simplifies the latter stage of the design process.

The map, being in two dimensions, can show only $k_{ci}$ and $k_{co}$ explicitly. Therefore, when the $k_{ci}$ and $k_{co}$ values corresponding to an existing compliant mechanism are used to plot a dot on this map and if the dot lies within the feasible region, it does not automatically imply that that the mechanism would fulfill the functional requirements of the problem. This is because the $n$ value of the mechanism (denoted by $n_m$ from here onwards) might not be the same as the $n$ (denoted by $n_s$ from here onwards) value associated with that point on the map. Therefore, we not only color the feasible region on a gray scale corresponding to $n_s$ but also show the filling color of the dot on a gray scale corresponding to $n_m$. Thus, the objective of interactive user-driven design of the selected compliant mechanism amounts to matching $n_s$ and $n_m$ within the feasible region.

The choice of the values for the specification variables rests with the user, and will have a bearing in the selection of the final mechanism. In some cases, the values for the specification variables are fixed and the user does not have the freedom to put bounds on some of the specification variables. For example, the input stiffness of an actuator whose value is given to the user cannot be changed. Although there are lower bounds and the upper bounds on some of the specification variables entered by the user, it is important to note that the final mechanism will represent a point on the $k_{ci}$-$k_{co}$ map and hence single values for each of the specification variables. Thus the final mechanism will represent a single set ($F_{in}$, $F_{out}$, $u_{in}$, $u_{out}$, $k_a$, $k_{ext}$). However, it is ensured that the values of $F_{in}$, $F_{out}$, $u_{in}$, $u_{out}$, $k_a$ and $k_{ext}$ will be in the range of their respective lower and upper bounds entered by the user. This could be better explained with an example.

If an application requires a force of 10 N, and the user does not mind the corresponding input displacement value to be any value between 3 mm and 4 mm and an output displacement value between 5 mm and 6 mm, the method could finally yield a mechanism with specifications of output force of 10 N, input displacement of 3.2 mm and the input displacement of 5.6 mm. This is entirely different from a constant force application where the output force remains constant at 10 N over the range of input displacements from 3 mm to 4 mm, and over the range of output displacements from 5 mm to 6 mm.

The interactive re-design can be done in six different ways as explained ahead.

## 3.2 Selecting the mechanism for re-design

As noted earlier, each compliant mechanism is represented on the $k_{ci}$-$k_{co}$ map as a dot: the location of the dot indicates the values of $k_{ci}$ and $k_{co}$, while the gray scale value of the fill color of the dot represents the $n$ of the mechanism (see Fig. 7). If any dot of matching $n$ color lies within the feasible region, it means that the existing compliant mechanism can meet the specifications; and there is no need for further modifications. The user can stop and select the mechanism. But this happy situation happens rarely. Hence, the user needs to re-design some likely candidates. The mechanism whose position is closest to the feasible region on the map is chosen for designing. The process of re-designing is explained next.

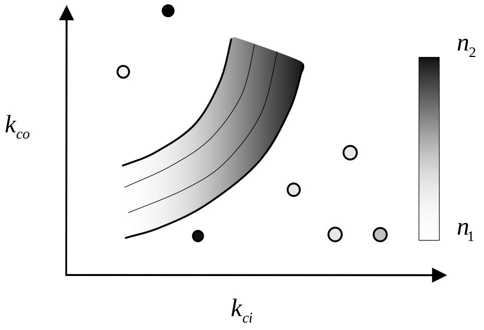


**Fig. 7:** Each dots represent a compliant mechanism in the database. The position of adot represents $k_{ci}$ and $k_{co}$, while its fill color represents $n$ of the mechanism.

### 3.3 The parameter curves on the same map

The selected mechanism is modified so that its function meets the design requirements. The modification is done by changing the parameters of the mechanism, which are a set of size and shape variables as well as the material of the mechanism. The size of the mechanism is changed by uniform increase or decrease of the in-plane width or the out-of-plane thickness of the beam members. In other words, the ratio of the in-plane widths of the different members of the selected mechanism always remains the same even after its modification. The shape of the mechanism is changed in the following three ways: (a) the coordinates of the nodes of the meshed model are uniformly changed only in the *x*-direction, (b) the coordinates of the nodes of the meshed model are uniformly changed only in the *y*-direction, and (c) the coordinates of the nodes are changed uniformly in both the *x* and *y* directions. It is quite possible that within the space considerations, using only the shape and size variables does not yield a mechanism which meets the problem specifications. Thus, in addition to the size and shape variables, the user has the flexibility of changing the material of the selected mechanism. The parameters of the compliant mechanism that can be changed are listed below.

i. Re-sizing the mechanism only in the x-direction
ii. Re-sizing the mechanism only in the y-direction
iii. Re-sizing the mechanism in both the directions at a chosen aspect ratio
iv. Uniformly changing the in-plane widths of all the beam elements
v. Uniformly changing the out-of-plane thicknesses of all the beam elements
vi. Changing the material of the mechanism

Each parameter curve indicates how $k_{ci}$ and $k_{co}$ of the mechanism change as the particular parameter is increased or decreased from its current value.

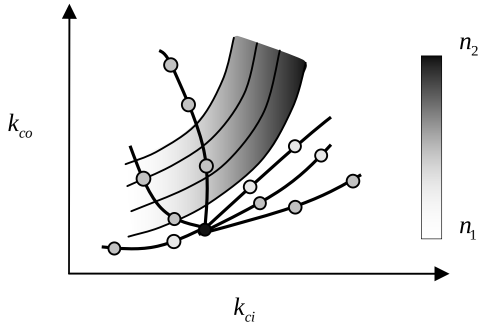


**Fig. 8:** The selected mechanism is the mechanism closest to the feasible region. The six curves indicate that the $k_{ci}$ and $k_{co}$ of the mechanism are a function of the six attributes. The color of the dot along the curve represents the changing $n$ of the mechanism.

In re-designing the chosen mechanism, any of the six re-design variables can be chosen to move the current dot into the feasible region (see Fig. 8). Note that the gray scale fill color of $n_m$ should match that of $n_s$. If that happens (see Fig. 9), the user saves that mechanism into a checkout-bin of feasible mechanisms for selection at a later stage.

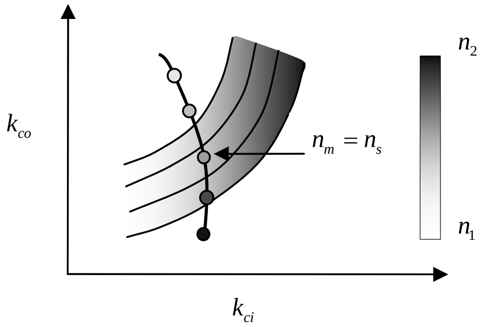


**Fig. 9:** One of the curves is selected for design. Note that there is a matching of $n_m$ with $n_s$ along this curve. This happy situation though happens rarely.

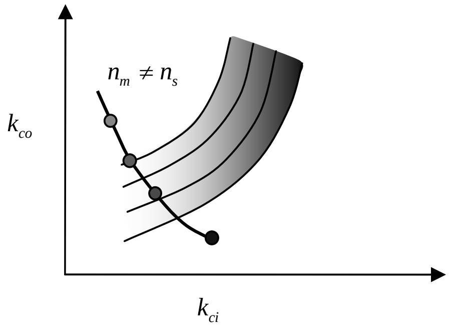


**Fig. 10:** One of the curves is selected for design. Note that there is no matching of $n_m$ with $n_s$ along this curve.

It is quite possible that if we use only one design variable as the option for modifying the mechanism, there is no match between $n_s$ and $n_m$ in the feasible region (see Fig. 10). When this happens, the other design variables have to be looked into. If none of the six redesign variables can give feasible mechanism, the other alternative is to get into the feasible region or close to the feasible region using one of the design variables, and then use the other design variable to get a matching of $n_m$ and $n_s$ within the feasible region (see Fig. 11). This designed mechanism is also saved in the 'check-out bin' of feasible mechanisms.

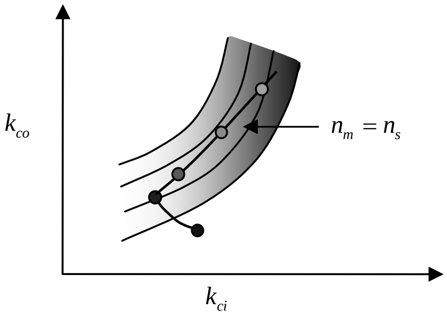


**Fig. 11:** The matching of $n$ is achieved in two stages.

So far, we have designed the same mechanism in two ways. If the user has a concern about the manufacturability of the designed mechanism or any other aspect, the user may choose the next mechanism closest to the feasible region. It is again designed in the same way to get a matching between $n_m$ and $n_s$ in the feasible region. This interactive procedure is fast. The user stops after sufficient number of feasible mechanisms are found. This number may be just one or many—usually many.

The next section describes a case study which shows how an existing compliant mechanism can be selected and further designed to suit the practical requirements of a new problem.

# 4 GUI Features and a Case Study

Since the design process requires an interaction from the user, a GUI (Fig. 12) is developed in Matlab to aid this process. The GUI is made up of a number of panels and a plot area. In the *Specifications* panel, the user enters the values for the lower and upper bounds for the six specification variables. The *Show and Select* panel allows for the visualization of the feasible region in the plot area and also the various compliant mechanisms in the database. Depending on the number of mechanisms the user wishes to see, the GUI allows that many mechanisms that are closest to the feasible region to be visible to the user. The *Draw Parameter Curves* panel can be used to select one, many or all of the parameter curves for the purpose of drawing them on the map. The *data-cursor* icon allows the user to select a parameter curve to move the current dot into the feasible region. As the user is following the curve, he/she can see the difference between $n_m$ and $n_s$ in real-time. The GUI automatically places a green dot when there is a matching between $n_m$ and $n_s$. The *Current State* panel shows the current attributes of the selected mechanism. These values change in real time during the process of redesign. Apart from the attributes of the mechanism, this panel also shows the maximum stress in the mechanism. The GUI has a provision to save the modified mechanism into a checkout-bin which could be used by the user for the final selection of the mechanism.

We used the GUI to solve a number of practical problems of interest to us. We discuss in detail only one of them here due to limited space. Additional case studies are available, for interested readers, at www.mecheng.iisc.ernet.in/~hegde/paper.

## 4.1 Amplifying Mechanism for a Piezoelectric Actuator—a case study

Piezoelectric actuators are known to produce large forces but have very small strains. The design problem (see Fig. 13) here is to amplify the motion of a piezoelectric actuator. The characteristics of the piezoelectric actuator are as follows. It has a stiffness of 7.5 N/$\mu m$. The piezoelectric actuator has a blocking force of 800 N. It has a stroke of 100 $\mu m$. The requirement of the ap plication is that the output of the mechanism has to move by a distance of 300 $\mu m$, an amplification of about three times. Since, the output stroke of the actuator is 100 $\mu m$, the compliant mechanism will be designed for a input displacement which is less than 100 $\mu m$. The input displacement that the compliant mechanism will be designed for is 90 $\mu m$. The given load that is acting at the output is 200 N.

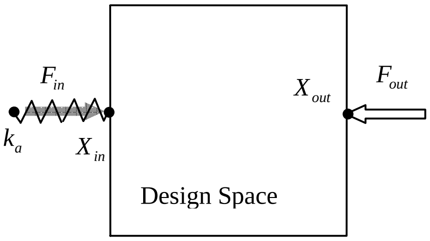
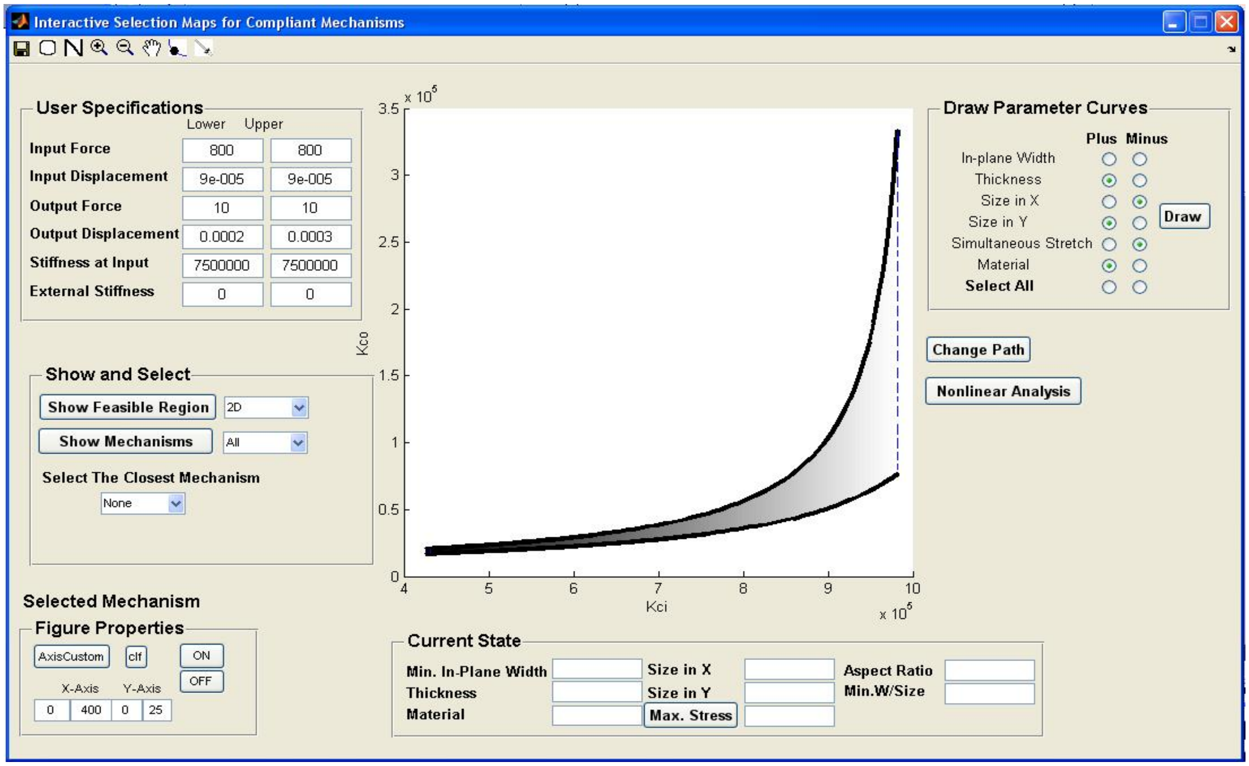


**Fig. 13:** Design problem to design a compliant mechanism to amplify the motion of a piezoelectric actuator

**Table-2: Specifications entered by user**

| Specification variables: | Min | Max |
|---|---|---|
| $F_{in}$ (N) | 800 | 800 |
| $x_{in}$ (m) | $90e-6$ | $90e-6$ |
| $F_{out}$ (N) | 200 | 200 |
| $x_{out}$ (m) | $290e-6$ | $310e-6$ |
| $k_a$ (N/m) | 7.5$e$6 | 7.5$e$6 |
| $k_{ext}$ (N/m) | 0 | 0 |



Interactive Selection Maps for Compliant Mechanisms

User Specifications

| | Lower | Upper |
|---|---|---|
| Input Force | 800 | 800 |
| Input Displacement | 9e-005 | 9e-005 |
| Output Force | 10 | 10 |
| Output Displacement | 0.0002 | 0.0003 |
| Stiffness at Input | 7500000 | 7500000 |
| External Stiffness | 0 | 0 |

Show and Select

Show Feasible Region: 2D

Show Mechanisms: All

Select The Closest Mechanism: None

Selected Mechanism

Figure Properties

| X-Axis | | Y-Axis | |
|---|---|---|---|
| 0 | 400 | 0 | 25 |

Draw Parameter Curves

| | Plus | Minus |
|---|---|---|
| In-plane Width | ○ | ○ |
| Thickness | ● | ○ |
| Size in X | ○ | ● |
| Size in Y | ● | ○ |
| Simultaneous Stretch | ○ | ● |
| Material | ● | ○ |
| Select All | ○ | ○ |

Current State

| Min. In-Plane Width | | Size in X | | Aspect Ratio | |
|---|---|---|---|---|---|
| Thickness | | Size in Y | | Min.W/Size | |
| Material | | Max. Stress | | | |

**Fig. 12:** The graphical user interface, developed using Matlab, aids in the selection and design.

**4.1.1 Feasible region:**

The feasible region is obtained using the specifications of Table 2 (see Fig. 14). All of the area of the feasible region lies on the negative side of the $k_{ci}$ axis. This suggests that the input spring in the SL model of the mechanism should have a negative stiffness, and this is not practicable. This is a case where, before going for selection of a mechanism, we can conclude that the requirements of the user are not possible to be met by any mechanism. The requirements are too stringent. Since the characteristics of the piezoelectric stack cannot be changed, we have to relax the other requirements of the user, namely the output load or the output displacement. The requirement of the output displacement has to come down or the requirement of the output load that the mechanism should act against should come down. Now there are two scenarios, a) the output load is reduced and, b) the output displacement is reduced.

Let us consider the first option, where the output load requirement is relaxed and the output load is taken as 100 N. This too is not enough. It is found that feasible region does not lie in the positive quadrant of $k_{ci}$ - $k_{co}$ axes. When the output load is 10 N, the feasible region lies in the positive quadrant of $k_{ci}$ - $k_{co}$ axes which is shown in Fig. 15.

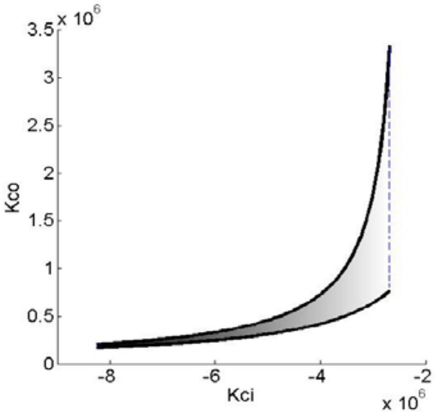


**Fig. 14:** The feasible region is bound by the curves representing the lower and upper bounds on specification variables

The feasible region of Fig. 15 is bound by the curves of the lower and upper bounds on the specification variables. The feasible region is bound by two solid lines and two dashed lines. One of the two solid curves, the one towards left corresponds to the lower bounds on the specification variables, while the other on the right corresponds to the upper bounds on the specification variables. There is a range of values of $n$, the third SL constant, within the feasible region. The range of $n$ within the feasible region is shown by the gray-scale color bar.

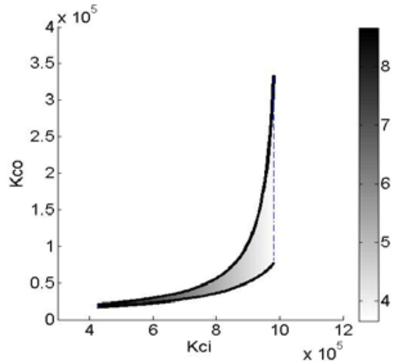


**Fig. 15:** The new feasible region obtained with the new specifications.

### 4.1.2 All mechanisms on the same map and the selection of the closest mechanism:

All the mechanisms are plotted on the same map (see Fig. 16). Most of the mechanisms are very far from the feasible region; hence, only the closest mechanism is visible (grey colored dot) in the figure. The mechanism which is closest to the feasible region is selected for design. In this case, this mechanism happens to be the amplifying frame mechanism (see figure 16 and Fig. 17). Now, the next task is to modify this mechanism. The modification amounts to moving the dot on $k_{ci}$ - $k_{co}$ map, which is currently outside the feasible region, into the feasible region.

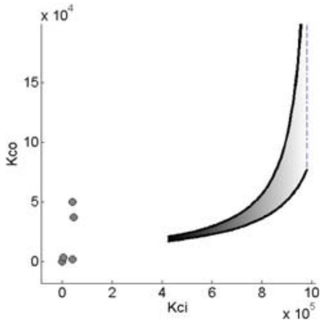


**Fig. 16:** Only the closest mechanism is shown, as the other mechanisms are too far from the feasible region.

### 4.1.3 Parameter curves from the present state:

Figure 17 shows the curves emanating from the point representing the selected mechanism. The red curves indicate the changes in $k_{ci}$ and $k_{co}$ of the designed mechanism when the parameters are modified with their values increased from the present value. The black curves indicate changes in $k_{ci}$ and $k_{co}$ of the designed mechanism when the parameters are modified with their values decreased from the present value. The legend in the figure shows the name of the curve. As an example, the red-colored width line, the *Width+* line will indicate how $k_{ci}$ and $k_{co}$ change when the in-plane width is increased progressively from the current width, with all the other parameters of the mechanism remaining the same. In a similar way, the other curves show how $k_{ci}$ and $k_{co}$ of the designed mechanism change when the concerned parameter is changed, with all other parameters kept fixed at their current values.

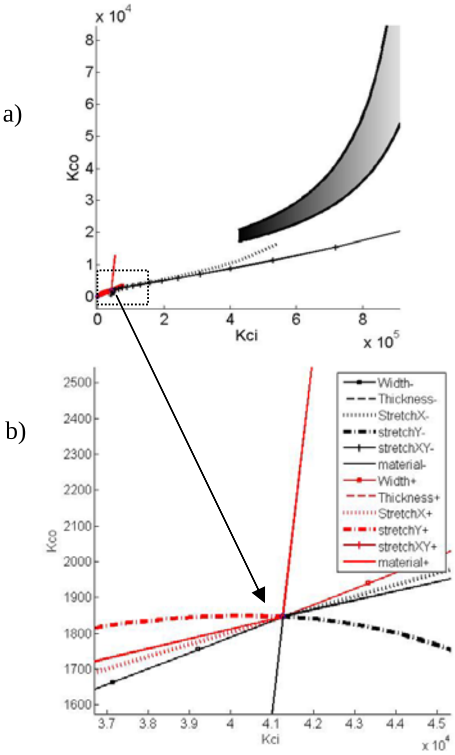


**Fig. 17:** The six curves represent the parameter curves from the present state.

### 4.1.4 Moving the current dot into the feasible region:

The tast in the design-modification step is to move the current dot on the map into the feasible region using one of the six parameter curves. This is achieved in a few steps, which is discussed next.

At the outset, none of the curves pass through the feasible region (see Fig. 17). With a view to move into the feasible region, *stretchXY*-curve is selected and is followed only up to a certain distance indicated by the green line (see Fig. 18. a). At the end of the green line, a *Change Path* option is exercised. The curves emanating from this new point of design are shown in Fig. 18. b. All the curves except one curve can lead the design into the feasible region. This parameter curve is the *material+* curve. Along the curve *material+* (see Fig. 18. c), there is a match of $n_m$ and $n_s$ values within the feasible region, which is indicated by a green dot. This indicates that the modified mechanism satisfies the user specifications. The parameters of the modified mechanism are saved in the check-out bin and can also be seen in the *Current State* panel.

As the mechanism is being modified, the modified values of the attributes of the mechanism are continuously displayed. This allows the user to watch for any other constraints, e.g., to see if the mechanism is exceeding the space restrictions, the width of the mechanism is becoming too narrow, the maximum stress constraint etc. More importantly, the GUI displays the *n* value of the mechanism and the *n* of the background. The GUI shows the final parameters of the modified mechanism that meets the requirements (Fig. 19) in the *Current State* panel. Figure 20 shows the solid models of the original mechanism in the database and the modified mechanism. While the overall size of the original mechanism (E = 1.5E9) is 180 mm × 120 mm, the size of the modified mechanism (E = 6.7E9) is 49 mm × 32 mm.

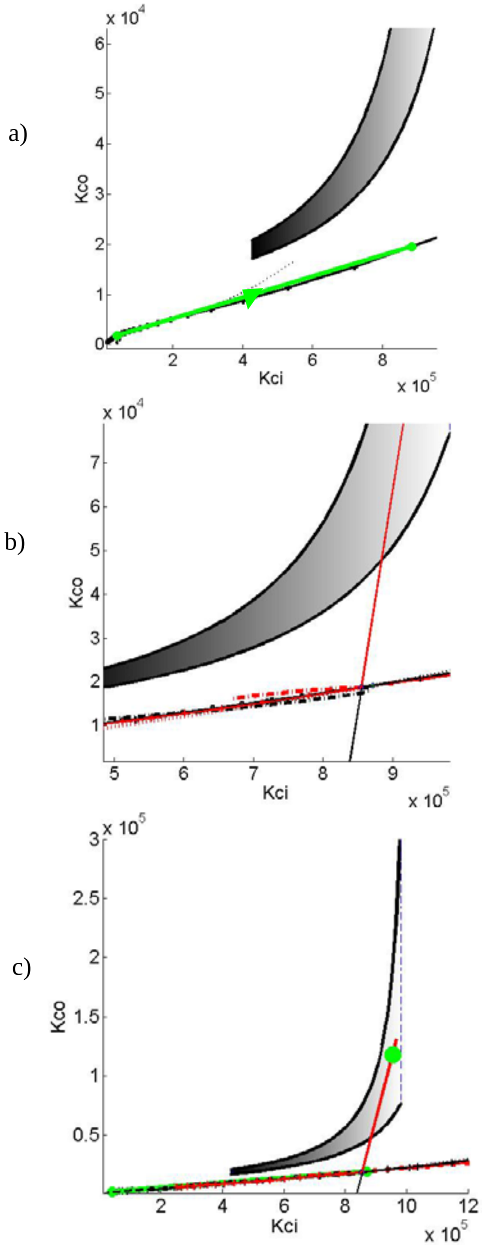
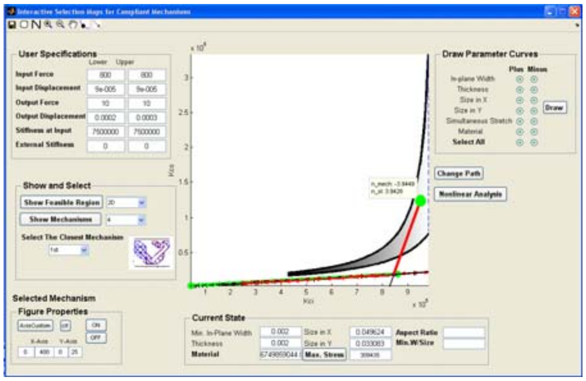


**Fig. 18:** a) The *stretchXY*- line is followed up to a certain distance shown by green line, at the end of which the change of path option is exercised, b) the different parameter curves emanating from the new point of design, c) the green dot indicates that there is a matching of *n* values inside the feasible region



User Specifications

| | Lower | Upper |
|---|---|---|
| Input Force | 800 | 800 |
| Input Displacement | 9e-005 | 9e-005 |
| Output Force | 10 | 10 |
| Output Displacement | 0.0002 | 0.0003 |
| Stiffness at Input | 7500000 | 7500000 |
| External Stiffness | 0 | 0 |

Show and Select

Show Feasible Region: 20

Show Mechanisms: 4

Select The Closest Mechanism: 1st

Draw Parameter Curves

| | Plus | Minus |
|---|---|---|
| In-plane Width | ○ | ○ |
| Thickness | ○ | ○ |
| Size in X | ○ | ○ |
| Size in Y | ○ | ○ |
| Simultaneous Stretch | ○ | ○ |
| Material | ○ | ○ |
| Select All | ○ | ○ |

Selected Mechanism

Figure Properties

X-Axis Y-Axis
0 400 0 25

Current State

| | | | | | |
|---|---|---|---|---|---|
| Min. In-Plane Width | 0.002 | Size in X | 0.049624 | Aspect Ratio | |
| Thickness | 0.002 | Size in Y | 0.033083 | Min.W/Size | |
| Material | 6749859044 | Max. Stress | 309435 | | |

**Fig. 19:** The GUI at the final stage of design; the *Current State* panel on the right hand side indicates the final attributes of the mechanism

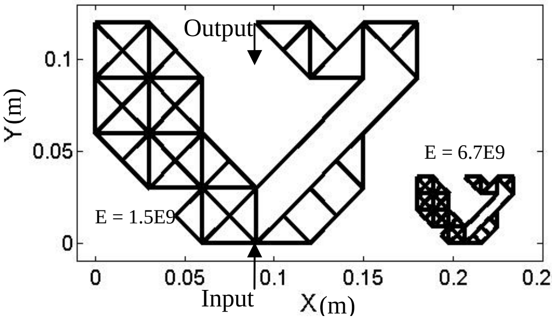


**Fig. 20:** The left hand side part shows the original mechanism in the database while the right hand side part shows the modified design.

## 5 Closure

We introduced an alternate design methodology for compliant mechanisms in view of practical applications. The design is based on selection among a known set of existing compliant mechanisms. The design process is done with the help of a 2D map in conjunction with a graphical user interface developed in Matlab. The design process is very rapid and interactive, allowing the user to see the mechanism obeying or violating the space constraints, manufacturing constraints, and maximum stress constraints. The method is used on a number of practical problems to design mechanisms. A case study is presented where the user specifications are met by selecting the mechanism from the database and then modifying it. Other case studies are posted on a website: www.mecheng.iisc.ernet.in/~hegde/paper.

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