Recent Developments in the Design and Optimization of Constant Force Electrical Contacts

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Abstract
This paper contains a summary of our recent work in the field of constant-force electrical contacts. These mechanisms are promising solutions to problems such as contact fretting and high contact resistance. The work that is summarized here focuses on the design and optimization of these mechanisms, while reducing the sensitivity to factors such as manufacturing uncertainty and friction. Using optimization based methods we can identify manufacturable contacts that provide near constant force (98% constant) over a large deflection range (0.64 mm). Also, our design and optimization approach identifies geometric and mating conditions that lower the standard deviation in our percentage of constant force output from 2.46% to 0.76% which implies a significant decrease in manufacturing tolerance sensitivity. Frictional effects have also been studied and can be reduced through optimizing geometric and mating conditions.

Keywords: Constant Force Mechanism, Compliant Mechanism, Electrical Contact, Friction, Multiobjective Optimization, Robust Design Optimization, Surrogate Modeling.

1. Introduction
Constant-force mechanisms are part of a developing technology that has already been used in several mechanical systems where a variable force over a displacement/deflection range is undesirable. In the case of electrical connectors, too much force can cause fretting and wear, while too little force results in high contact resistance. Many connectors rely on deflected cantilever-type or pogo-type contacts to generate the contact normal force. As such, the magnitude of the force will have a linear relationship with the deflection of the contact. Ideally the deflection of the contact can be controlled to keep the contact normal force at a desirable magnitude, but smaller scale devices, manufacturing uncertainties, and possible dynamic operating conditions make this control difficult in practice. The problem is worsened when connector systems with an array of contacts are considered – variations in planarity of the contact tips can adversely affect the performance of adjacent contacts. This problem becomes more and more significant, the smaller the connector designs become.

To solve this problem, a constant-force mechanism has been proposed to keep the magnitude of the contact normal force nearly constant at a desirable value for large range of possible deflection motion. A prototype of this design is shown in Fig. 1a. This design makes use of a compliant current-carrying member similar to other cantilever-type designs, but the free end of the compliant member makes contact with, and slides along, a cam as the input deflection is applied. This cam is shaped such that as the applied deflection is increased the component of force it transmits to the compliant member (opposing the direction of the applied deflection) is decreased. This decreasing resistance to deflection is balanced with the increasing resistance from the strain of the compliant member to maintain a nearly-constant net reaction force, as shown in Fig. 2a. A detailed description of the mechanical behavior of these devices and their behavior is provided in a publication by Weight et. al. As reported in their publication the first constant force contact prototype design maintains a 73.20% constant force over a 0.64 mm deflection range when evaluated using finite element methods. During physical testing of the handmade prototypes, however, variations in constant-force performance were observed that were attributed to manufacturing uncertainties and frictional effects.

In recent publications summarized in this paper, we have focused on improving the design method to increase the nominal performance and decrease the variation in performance for these mechanisms. Specifically we formulate several design optimization problems that have more design freedom than the formulation used to design the prototypes. These formulations include methods for reducing sensitivity to mating uncertainty, manufacturing uncertainty, frictional effects, and cam geometry. The next section presents the non-linear finite element model for the mechanism that we use in each of our optimization formulations.

Figure 1: Sample geometry shown in physical prototype (a) with critical features defined and finite element model (b) with initial and final displacements shown.

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2. Model Development

In this section we present a finite element model and the basic optimization formulation that we use for all the other formulations that will be addressed later in the paper. The finite element model is illustrated in Fig. 1b. As shown in the figure, each node in the model is located using x and y Cartesian coordinates. The nodes and elements define the two-dimensional profile of the compliant member. One node at the beginning of the compliant member has fixed boundary conditions, and another at some predetermined location has an applied deflection load (Δ). The locations of these two nodes are specified as part of the desired application for the contact, and are held constant throughout the design process.

The free end of the compliant member makes contact with a rigid cam surface whose profile may be arbitrary. In the cases we present, however, the cam is constrained to be circular, with the exception of the friction reduction formulation where we allow elliptical profiles for increased design freedom. This simplifies the optimization by requiring only three variables to define a circular profile, or five to define an elliptical profile.

To create a more realistic model we include various possible mating conditions for the cam and the free end of the compliant member. If there is interference between the two mating parts we force the compliant end to deflect to remove the interference. This models what will occur during the assembly of the two parts. If there is a gap between the mating parts no interaction forces are placed on the free end of the compliant part until the gap is closed. When contact is made an interaction force develops at the cam interface. This force is comprised of a component normal to the cam surface (\( F_N \)) and, in the case of a frictional analysis, a component tangent to the cam surface (\( F_f \)). The magnitude of the normal component is determined by the constraint that the free end must follow the cam profile. The magnitude of the tangential, or frictional, component is determined by Coulomb's theory for friction:

\[
F_f = \mu F_N \tag{1}
\]

where \( \mu \) is the coefficient of friction determined from experimental data. We use a Newton-Raphson procedure to iteratively solve the large deflection equations and boundary constraints at several substeps of the applied deflection. From this we estimate the force deflection curve, and the internal stresses in the compliant member. As can be seen in Fig. 2a the output force varies slightly within the operational deflection range. By comparing the maximum force (\( F_{\text{max}} \)) and the minimum force (\( F_{\text{min}} \)) that are developed in this range we can characterize the performance of the constant-force mechanism. We define the constant force percentage (\( C \)) as:

\[
C = 100 \frac{F_{\text{max}}}{F_{\text{min}}} \tag{2}
\]

A value of one in the ratio could only occur if there were no variation within the operational deflection range, and the mechanism would be 100% constant. This value represents an objective for the design optimization. A general form of the design optimization is:

\[
\max_v C(v, p) \tag{3}
\]

subject to:

\[
g(v, p) < B \tag{4}
\]

where \( v \) is a vector of design variables, \( p \) is a vector of design parameters, g is a vector of constrained values, and \( B \) is a vector of limits for the constraints. In the case of our electrical contact model we constrain the reaction force to be within a desirable range, and the maximum stress to be lower than the yield stress for the material. We also constrain the design domain area in which our profile geometry must fit. Manufacturing capabilities are considered by constraining the minimum element length and minimum bend angle. Finally, we constrain the spring geometry to prevent any elements from passing through the cam, or aligning the final element perfectly with the normal direction at the interaction point, since these situations can lead to inaccurate results or non-converging solution routines. Complete formulations of the constraint functions can be found in our previous work. 2–4

When we apply this optimization routine without considering varying mating conditions, uncertainty, or friction we identify a design that is 97.50% constant. This is a significant improvement when compared to the 73.20% constant design of the prototype that came from earlier design methods.

To determine the sensitivity of the design to uncertainties we perform a Monte Carlo simulation where each of the design variables and parameters is randomly selected from a normal distribution about the nominal value. Using this method we find a standard deviation of 2.46% for our optimized design and 1.68% for the benchmark (prototype) design. To put physical meaning to the standard deviations found, it was found that approximately 30% of the benchmark’s prototypes were functional as manufactured. The optimization model that did not take uncertainty into account has a standard deviation of 2.46% which is much higher than the benchmark model implying that manufacturing the part to the specified tolerances would result in even more rejected parts. The next three sections discuss variations of our optimization formulation that we use to design robust mechanisms, and reduce frictional effects. A more detailed...
description of this non-linear finite element approach is provided in another publication.5

3. Robust Mating Condition
In this section we look specifically at the mating condition between the compliant member and the cam, and create an optimization formulation to find a robust solution, following a procedure developed in our earlier work.2 The aggregate objective function (J) can be evaluated as:

\[ J = W_1 C_L (v, p) + W_2 C_N (v, p) + W_3 C_U (v, p) \]  (5)

where \( C_L \) is the constant force percentage evaluated at the lower extreme tolerance on the mating condition (smallest gap or largest interference), \( C_N \) is the percentage evaluated at the nominal condition, and \( C_U \) is the percentage evaluated at the upper extreme tolerance on the mating condition (largest gap or smallest interference). The weighing factors \( W_1, W_2, \) and \( W_3 \) can be adjusted as needed to find a suitable design. Essentially this objective finds designs that perform well at three specific mating conditions within our tolerance window. We must, however use a Monte Carlo simulation after we find the optimal design to verify that mating conditions falling somewhere between the extremes do not significantly reduce the mechanism’s performance. This Monte Carlo simulation also shows how much other uncertain factors, such as material properties, affect the final performance.

When performing the optimization we apply the same constraints as in the general case, except we increase the restrictions on each constrained value, to account for uncertainty in that value. Doing this we find the optimal design to be 98.20% constant with a standard deviation of 0.76%. It is important to note the degree in which the standard deviation has improved over the benchmark design signifying that the new design will perform much better when manufactured within the same tolerances. One reason for the decreased in standard deviation for the optimal design is an interference mating condition which helps improve the overall performance.

4. Surrogate Approach to Robust Design
The formulation from the previous section has some limitations that need to be discussed. First, the formulation is based on various mating conditions without supplying evidence that the mating conditions cause significant variations in performance. In addition to this limitation, the previous formulation does not try to decrease the effect that other uncertainty factors, such as cross sectional dimensions and material properties, may have on the performance. Finally the previous formulation explores only three points within the tolerance window to minimize performance sensitivity throughout the window.

In this section we present another formulation, developed in detail in our earlier publication7 that handles some of these limitations. This formulation builds a polynomial function to approximate \( C \) using a ‘least squares regression’ fit to data obtained from a simulated statistical experiment. This polynomial, or surrogate model, makes it possible to efficiently explore numerous locations within the tolerance window. Building the surrogate model to include all the variables and parameters, however, is computationally expensive and can not be done within the optimization routine. In order to update our surrogate model with every iteration of the optimization process we must limit the number of variables and parameters we include. To determine which variables to include in our optimization we build a single use, simplified polynomial approximation. The terms having the highest coefficients are the terms that have the largest effect in the calculation of \( C \).

In the case of the design we found using our deterministic approach (97.50% constant), this screening model identifies the most significant factors as the location of the cam center, the cam radius, and the location of the compliant member’s free end. These factors are also the only factors that determine the mating condition we used in the previous section. This adds some support to the approach we used in the previous section. Now, however, we develop an aggregate objective function with a more complex polynomial function to more thoroughly explore the tolerance window.

\[ J = W_1 \hat{C} (u) + W_2 \frac{\hat{C}_{\text{min}} (u)}{\hat{C}_{\text{max}} (u)} \]  (6)

In this equation we use \( \hat{C} \) to signify an approximation of \( C \) from our polynomial model. The functions \( \hat{C}_{\text{min}} (u) \) and \( \hat{C}_{\text{max}} (u) \) are the maximum and minimum approximations we find from a thorough exploration of the tolerance window. The variable \( u \) is a set of the most significant variables and parameters we identified in the screening model we created. Again we use weights \( W_1 \) and \( W_2 \) that can be adjusted before the optimization to find a suitable design. The ratio in this objective function is similar to the one in our definition of \( C \). A value of one can only occur if every design we sample in the tolerance window has the same performance, which means we have a robust design. The constraints for this formulation, like the constraints in the last section, have been shifted to account for uncertainty in the constrained values.

Using this optimization we can identify a design that has a nominal value of 93.61% for \( C \). Using our Monte Carlo simulation on this optimum we find that it has a standard deviation of 0.77%. While this design has a much smaller standard deviation than the deterministic solution, it is higher than the solution we found using only various mating conditions. This illustrates one of the difficulties of using surrogate models within optimization routines. Because the model is only approximate the results may not be as good, and we must always work to develop better surrogate models. This approach has resulted, however, in a better understanding of the role that the mating condition plays in the performance of the mechanism through the results of the screening model.
5. Reduction of Frictional Effects

Until now all of our optimization formulations have neglected frictional effects. Friction can have a severe effect on the constant force percentage of a mechanism, especially for small mechanisms such as electrical contacts. Since friction acts in the direction opposing motion it can either add to the output reaction force, or subtract from it depending on whether the compliant member is sliding up or down the cam. This results in a difference in the force-deflection curve as shown in Fig. 2b, where the top curve represents a compression motion, and the bottom represents an expansion motion. The area between these curves represents the energy lost to friction as a completely reversed motion is applied.

We can estimate the effect friction has on a mechanism using the model outlined in Section 2. When we add the frictional forces to our finite element model we can estimate the difference in the output force for compression and expansion.

In order to minimize the difference between the output force of the two motions we evaluate and minimize the energy lost to friction. By holding our displacement range constant and minimizing this energy loss we reduce the difference in output force that occurs within a reversing motion. Our optimization formulation is:

\[
\min _{v} J = -W_C(v, p) + W_F E_F(v, p) \tag{7}
\]

subject to the constraints from our deterministic optimization. In this equation \(E_F\) is the energy loss due to friction, while all other variables are as previously defined. As a starting design for this optimization we use our design from Section 3 that is 98.20% constant. Applying friction to this design we find that 0.61 mJ are lost during the full range of motion. After running the optimization routine we identify a design that loses only 0.39 mJ to friction, a reduction of 36.01% of the starting design. The percentage of constant force for this optimum is lower, however, at 79.57%. This shows a trade off between a high non-frictional performance and a high frictional performance.

6. Conclusion

In this paper we have presented several different optimization based design methods for designing constant force contacts and reducing the sensitivity to various factors. Table 1 compares the results of each of these optimizations.

We have shown that one can create a constant force spring that is 98.20% constant and is relatively insensitive to uncertainties such as variations in input variables, parameters as well as manufacturing and mating uncertainties. We have also shown that by using shape optimization we can reduce the effect of friction on a mechanism. Importantly we have shown how to reduce the energy lost due to friction by 0.22 mJ, 36.01% of the starting design.

7. References


Table 1. Table of results for the four different optimization formulations presented in this paper.

<table>
<thead>
<tr>
<th>Design</th>
<th>Constant Force</th>
<th>Avg. Constant Force</th>
<th>Std. Deviation</th>
<th>Frictional Loss</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benchmark Design</td>
<td>73.20%</td>
<td>70.24%</td>
<td>1.66%</td>
<td>NA</td>
</tr>
<tr>
<td>Deterministic Opt.</td>
<td>97.50%</td>
<td>94.82%</td>
<td>2.46%</td>
<td>NA</td>
</tr>
<tr>
<td>Mating Opt.</td>
<td>98.20%</td>
<td>97.64%</td>
<td>0.76%</td>
<td>0.61 mJ</td>
</tr>
<tr>
<td>Surrogate Opt.</td>
<td>93.61%</td>
<td>92.99%</td>
<td>0.77%</td>
<td>NA</td>
</tr>
<tr>
<td>Friction Opt.</td>
<td>79.57%</td>
<td>NA</td>
<td>NA</td>
<td>0.39 mJ</td>
</tr>
</tbody>
</table>
8. Pictures of Authors

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He received his BS and MS in Mechanical Engineering from Brigham Young University and his PhD in Mechanical Engineering from Rensselaer Polytechnic Institute. Prior to joining BYU, Prof. Mattson was the Global Director of Engineering Design and Research at ATL Technology and a member of the company’s executive committee. He established and managed ATL’s Silicon Valley office (1999-2000), and ATL’s twenty-five person Engineering Design Center in mainland China (2004-2006). He has designed or led the design of various products, which have been used by over 15 million people around the world. He has over 20 articles published in journals and conference proceedings, has been an invited lecturer at two universities in China, and holds two patents. He is a member of ASME, AIAA, and Sigma Xi. Prof. Mattson has served as a member of the AIAA Multidisciplinary Design Optimization Technical Committee since 2003.

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He obtained a BS in Mechanical Engineering from Brigham Young University in 2008 and is continuing on to obtain his MS and PhD also in Mechanical Engineering. His technical interests include numerical optimization and product development with an emphasis on creating products that are resistant to reverse engineering.