WESTERN MICHIGAN UNIVERSITY
Mechanical and Aeronautical Engineering

Compliant Suspension for a Multi-Specimen Test System

by

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Problem Statement

In order to maintain a constant load it is sometimes necessary to have a suspension between the force sensor assembly and the upper specimen holder. Suspensions are spring devices used to compensate for variations in the distance between the force sensors and the surface of the lower specimen when the lower specimen is in motion. Even though the carriage will automatically move up and down to maintain a constant average load, it can only respond to low frequency variations, therefore may require the use of a suspension to make up the difference.

The main problem of the existing suspension lies in the design. It has a cantilever design that causes the tool to vibrate and not wear at the center of the tool tip due to the angle of contact. Since the beam is supported only by one end, there is an overhang that causes significant tool vibration while single point diamond turning. This vibration results in poor surface finish, uneven and accelerated tool wear, higher cutting forces and unstable thrust forces.

The old suspension is also made up of 8 parts that are bolted together. These joints rub against each and cause backlash that reduce the level of precision in the instrument. This rubbing can result in low level vibration within the suspension that could be transferred out to the load cell or the cutting tool resulting in an unstable machining operation.

This problem can be solved by redesigning the entire suspension as a single compliant part. The compliant suspension will have no internal vibration and the symmetry of the part will not cause the tool to cut/contact the workpiece at an angle (as there is no cantilever). The compliant section will also be fixed on all ends to the main structure/frame of the suspension, except along the compliant axis eliminating any movement in the cutting direction.

Advantages of Compliant Mechanisms:

Compliant mechanism is used in a particular application for a variety of reasons. The advantages of a compliant mechanism could be considered in two categories: cost reduction and increased performance. Cost reduction includes aspects such as part-count reduction, reduced assembly time and simplified manufacturing process whereas the increased performance includes increased precision, increased reliability, reduced wear, reduced weight, and reduced maintenance.

One of the main advantages of redesigning the entire suspension as a compliant mechanism is a dramatic reduction in the total number of parts. The total numbers of parts were reduced from eight to just a single part as shown in Figure 1. In general, fewer parts are required for the compliant mechanism than for the rigid mechanism. The reduction in part count may reduce manufacturing and assembly time and cost.

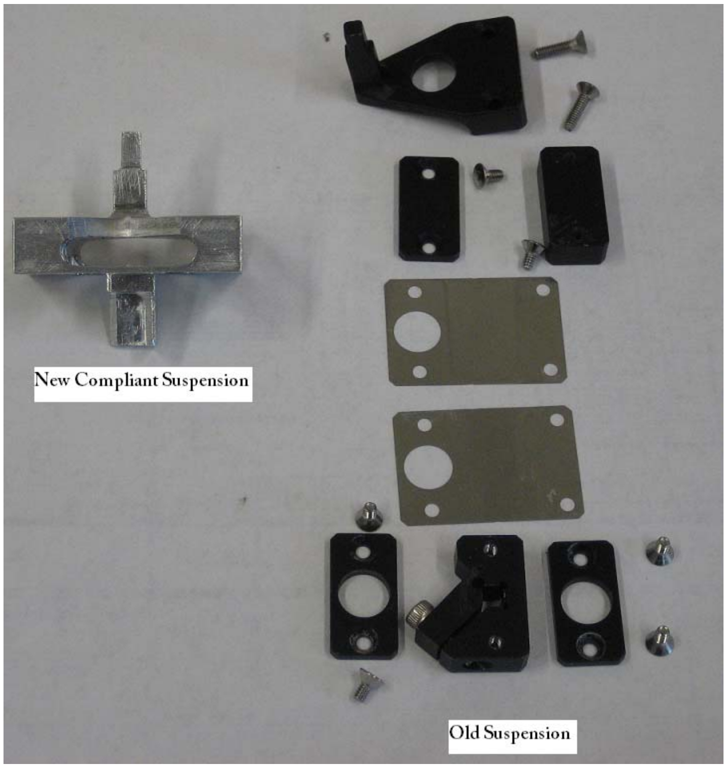


Figure 1: A single piece compliant suspension replacing an eight piece suspension

Compliant mechanisms also have fewer movable joints, such as pin (turning) and sliding joints. This results in reduced wear and need for lubrication. These are valuable characteristics for applications in which the mechanism is not easily accessible, or for operation in harsh environments that may adversely affect joints. Reducing the number of parts will also reduce the number of joints/bolts[1]. This will significantly reduce the vibration within the part which can be crucial in precision machining. Lesser joints can also increase precision, because backlash may be reduced or eliminated. This has been a factor in the design of today's high precision instrumentation. In compliant systems, because the motion is obtained from deflection rather than adjoining parts rubbing against each other, vibration and noise may also be reduced[2].

Because compliant mechanisms rely on the deflection of flexible members, energy is stored in the form of strain energy in the flexible members. This stored energy is similar to the strain energy in a deflective spring, and the effects of the springs maybe integrated into a compliant mechanism's design. In this case, energy is easily stored or transformed to be released at a later time or in a different manner.

It is possible to realize a significant reduction in weight by using compliant mechanisms rather than their rigid-body counterparts. This will further reduce the weight and shipping costs that will benefit companies.

Challenges of compliant mechanisms

Although offering a number of advantages, compliant mechanisms present several challenges and disadvantages in some applications. One of the largest challenges is the relative difficulty in analyzing and designing compliant mechanisms. Knowledge of mechanism analysis and synthesis methods and the deflection of flexible members is required.

Energy stored in flexible elements has been discussed as an advantage, since it can be used to simplify mechanisms that incorporate springs, obtain specified force-deflection relationships, and store energy that is transferred or transformed by the mechanism. However, in some applications, having energy stored in flexible members is a disadvantage. For example, if a mechanism's function is to transfer energy from the input to an output, not all the energy is transferred, but some is stored in the mechanism.

Fatigue analysis is typically a more vital issue for compliant mechanism than for their rigid-body counterparts. Since compliant members are often loaded cyclically when a compliant mechanism is used, it is important to design those members so they have sufficient fatigue life to perform their prescribed tasks. The motion from the deflection of compliant links is also limited by the strength of the deflecting members. Furthermore, a compliant mechanism will not be able to produce a continuous rotational motion such as is possible with a pin joint.

Compliant links that remain under stress for long periods of time or at high temperatures may experience stress relaxation or creep. The good thing about designing compliant systems is that several precautions and design considerations can be made to avoid the bad effects of it.

Design Considerations

Stiffness and Strength

The relationships between stiffness, strength, and deflection are often confused. If a small load causes a relatively large deflection, then the part has low stiffness, but nothing is implied about its strength. The amount of deflection that a load will cause is related to a structure's stiffness or rigidity. Strength is a property of the material that specifies the stress it can withstand before failure. In other words, a structure's stiffness determines how much deflection will occur due to a load, while the strength determines how much stress can occur before failure.

Displacement versus Force Loads

One of the challenges of this compliant suspension is to allow deflections large enough for the mechanism to perform its function, while maintaining stresses below an allowable maximum stress[3]. Once the deflection position of a compliant mechanism is known, the stress analysis is relatively straightforward. The displacement-force load of the old suspension was used as a guideline to maintain a similar ratio for the new design. Figure 2 shows a displacement versus force load plot that was used to design the compliant section of the suspension. The ideal working range shown in the plot is below 490.5mN (50g) where there is a linear relationship between the displacement and force-load. At higher loads (>1700mN), the suspension does not deflect anymore as it is not within the material stiffness ratio. This is intentionally design in such a manner to make sure that the suspension does not bottom out and cause instable forces as a result.

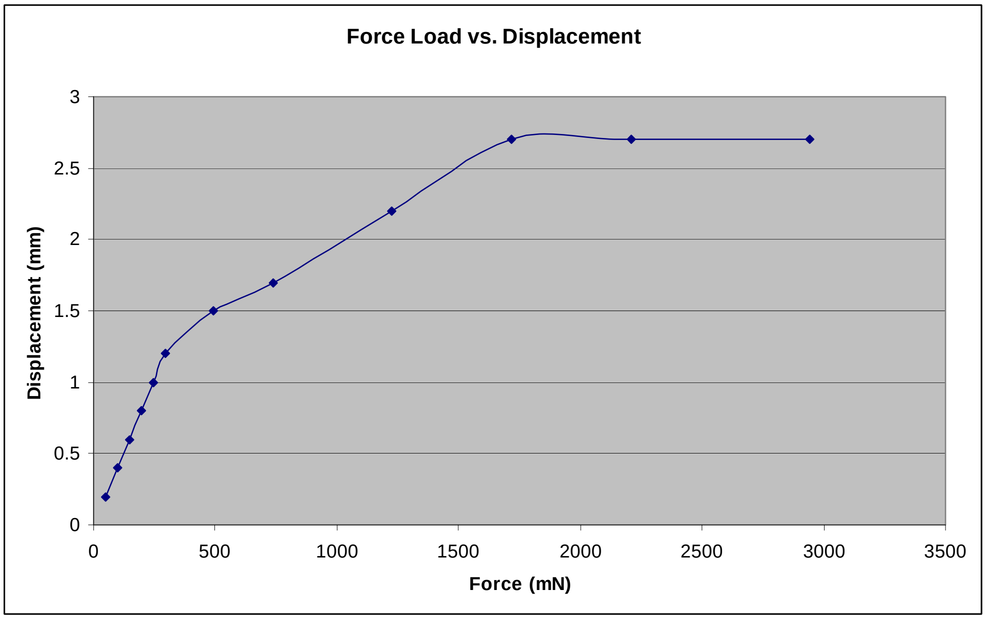


Figure 2: Displacement versus force-load ratio used to design the suspension

Material Considerations

Many different types of materials may be used in compliant mechanism design. Although each application has its own criteria for material selection, the selection process can be guided by principles that can be used in numerous situations[4]. Once again it is important to remember that stiffness and strength are not the same and it is possible to make something both flexible and strong. Ductility and flexibility are also not equivalent, and brittle materials may be used to construct compliant mechanisms if their geometry is made such that they are not overstressed.

The material chosen for the compliant suspension was Aluminum (6061). The reason this material was chosen was for the following reasons:

- Predictable material properties (needed in high-precision instruments)
- Good performance in high-temperature environment
- Biocompatibility

- Low susceptibility to creep
- More predictable fatigue life
- Ability to operate in many harsh environments
- Good machinability
- Low rate of corrosion

The material properties of Al 6061 are as follows:

Density	2.70 g/cc
Hardness, Brinell	95.0
Ultimate Yield Strength	310 MPa
Tensile Yield Strength	276 MPa
Modulus of Elasticity	68.9 GPa
Poissons Ratio	0.330
Fracture Strength	96.5 MPa (@# of cycles 5.00e+8)
Fracture Toughness	29.0 MPa-m1/2
Machinability	50.0%
Thermal Conductivity	167 W/m-K
Melting Point	582 – 651.7 ^{o}C

Table 1: Al 6061 material properties

<u>Other Design Considerations</u>

- Creep and stress relaxation
  - Creep in the form of deformation that occurs in the material under load over time.
  - Stress relaxation is closely related to creep but occurs more often than creep in compliant systems. It occurs when a constant deflection is applied to a structure and the resulting stress decreases with time.

- Linear elastic deflections
  - Mainly using the Bernoulli-Euler equation where the bending moment is proportional to the beam curvature

- Energy storage
    - The energy stored in the compliant section of the system is an advantage in this case because the energy available at the output is less than was provided at the input.

- Stress stiffening
    - Occurs when the stiffness of a structure changes as its deflection changes[5].
    - The stiffness is obtained before calculating the deflection.

Other designs considered:

Several preliminary designs were modeled using SolidWorks. After each model was created, a FEA was carried out using COSMOS. This would help improve the design based on the FEA feedback and interpretation. Some of the designs are as follows:

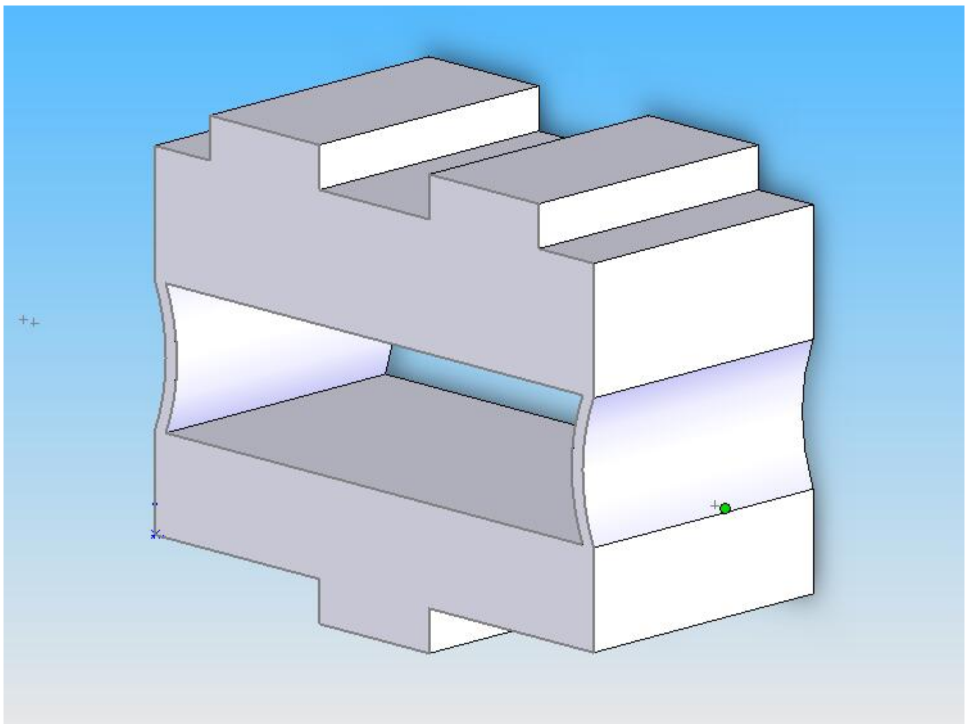


Figure 3: A compliant design with a 1mm thick arched wall that would act as a spring mechanism.

The problem with this design was it was way to stiff and therefore yielded in a deflection in the z-axis of only about 20μm.

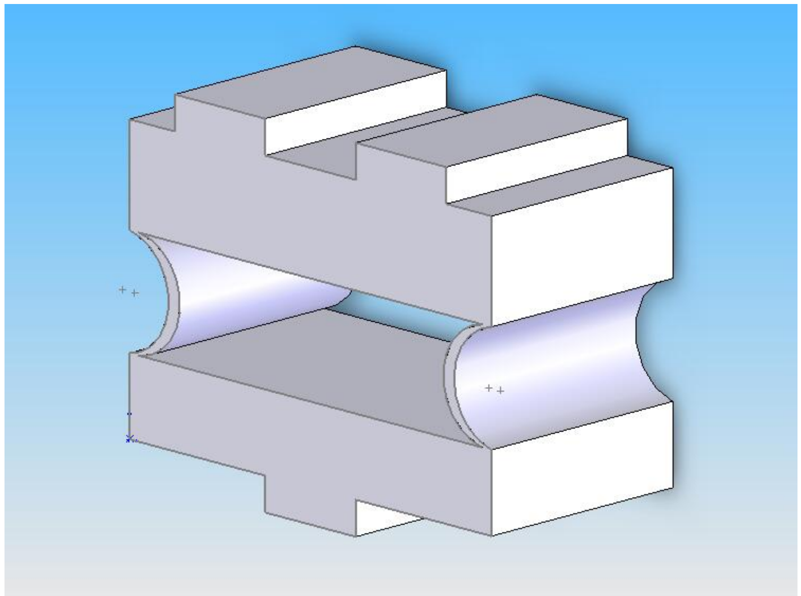


Figure 4: The design is very similar to Figure 3 but the arc has more of a radius added to it.

The problem with this design was it was way to stiff and therefore yielded in a deflection in the z-axis of only about 40µm.

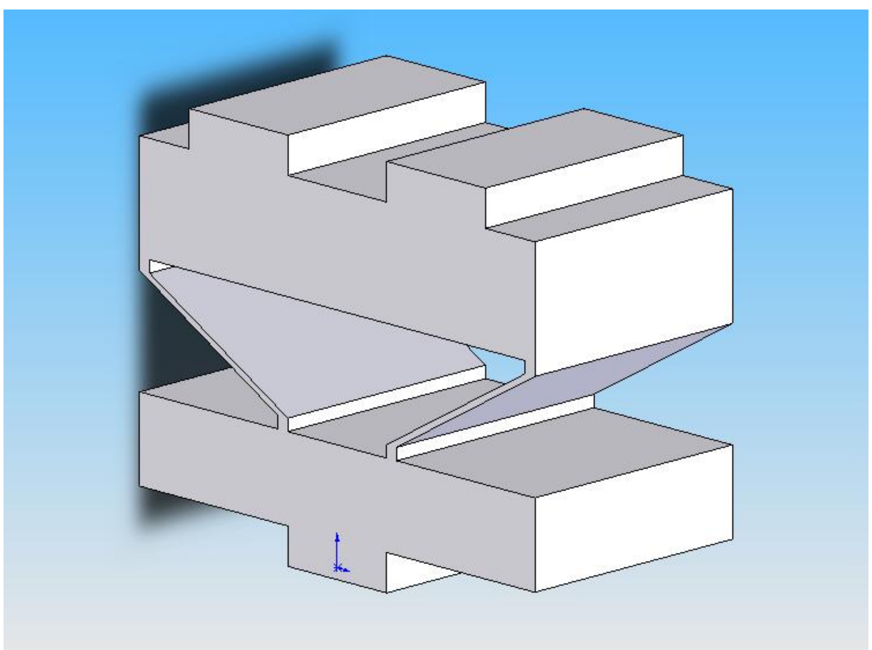


Figure 5: The arched shaped sides were removed and replaced with a straight edge where the starting and ending point lie on a different axis.

The problem with this design was it was way to stiff and therefore yielded in a deflection in the z-axis of only about 60µm.

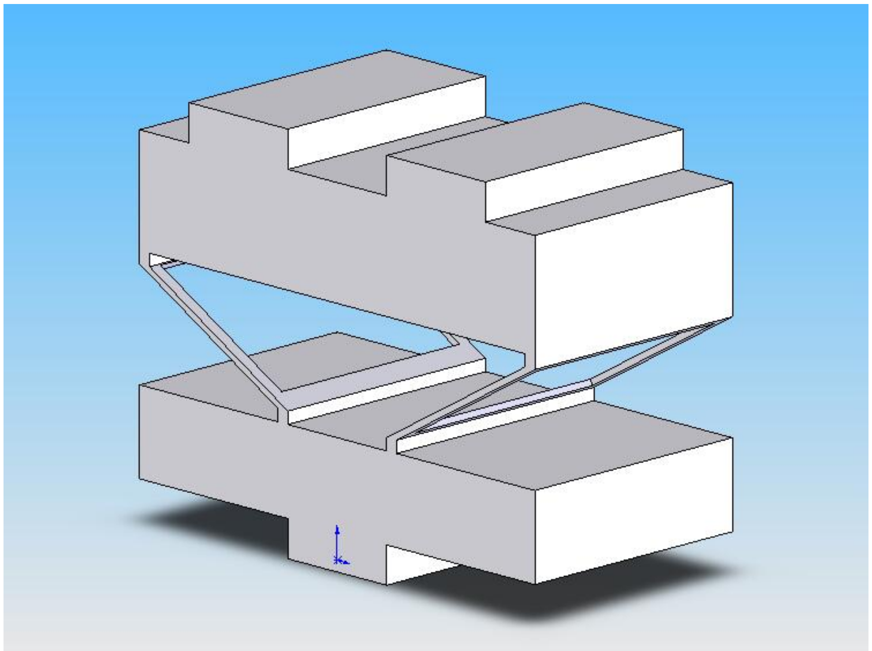


Figure 6: This is basically an improvisation of Figure 5 where now only a frame is along the sides. This means lesser material and technically more deflection. The idea worked but the deflection was still not sufficient (≈ 400μm).

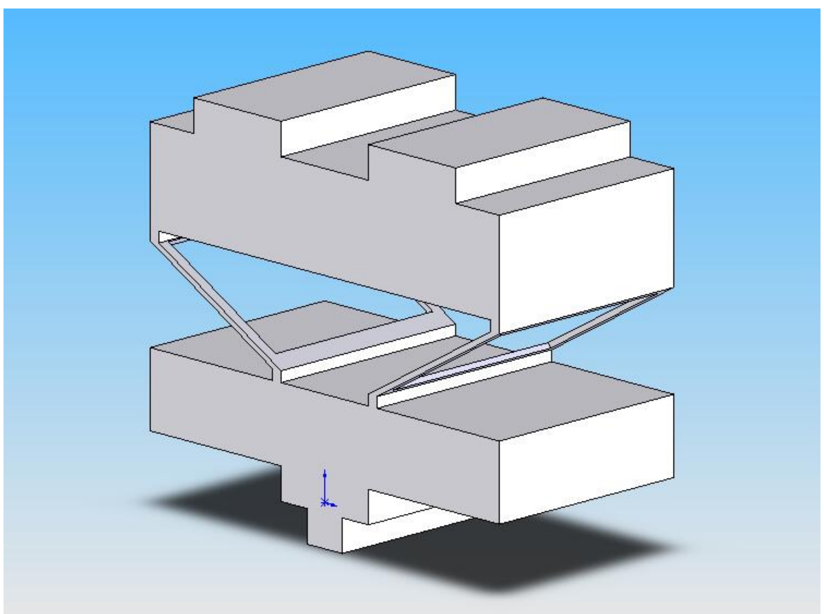


Figure 7: Very similar to Figure 6 except that the extra section was added at the bottom to act as the tool area.

The total area of where the force was applied was reduced therefore the pressure was increased. This however did not make a significant change in the FEA and deflection.

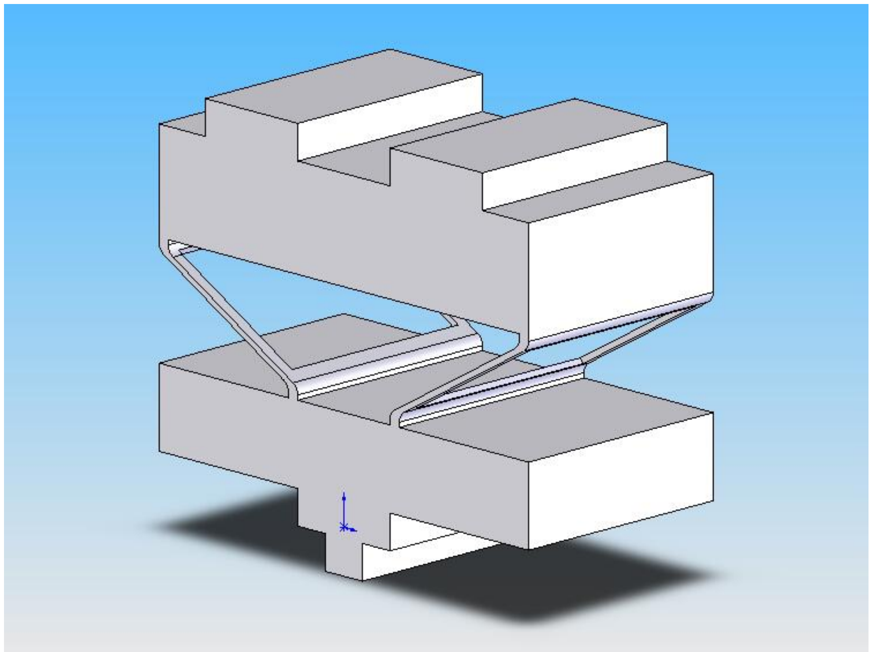


Figure 8: The edges from Figure 7 were rounded to make the system more compliant. The idea worked but the deflection was still insufficient (≈ 500μm).

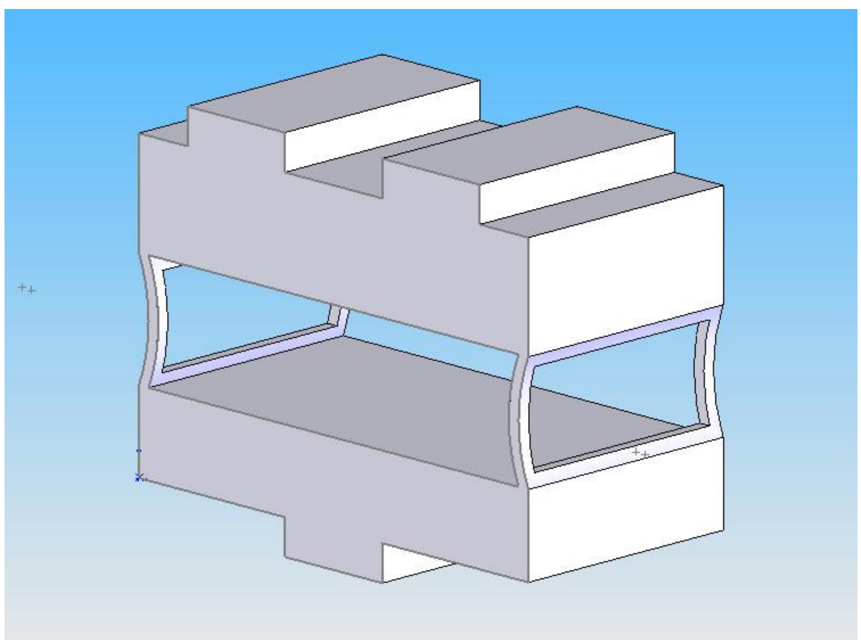


Figure 9: The arched shaped sides were reconsidered but this time with a hollow center.

The results were promising where the total deflection obtained was approximately 4mm. The problem with this design was the side walls of the compliant section were too thin (1mm) and this caused the frame to buckle.

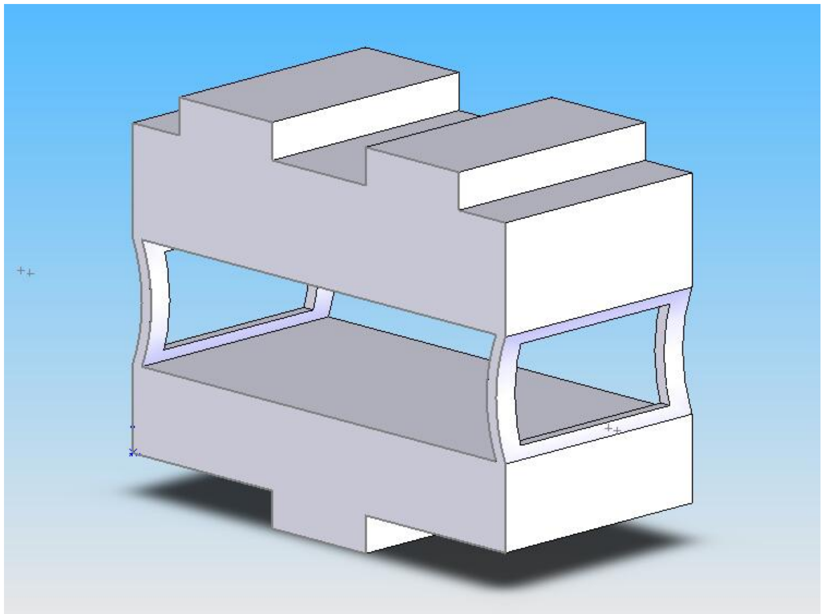


Figure 10: The wall thickness from Figure 9 was increased to 2mm. This yielded in a total deflection of 2.5mm and significantly reduced the buckling of the frame.

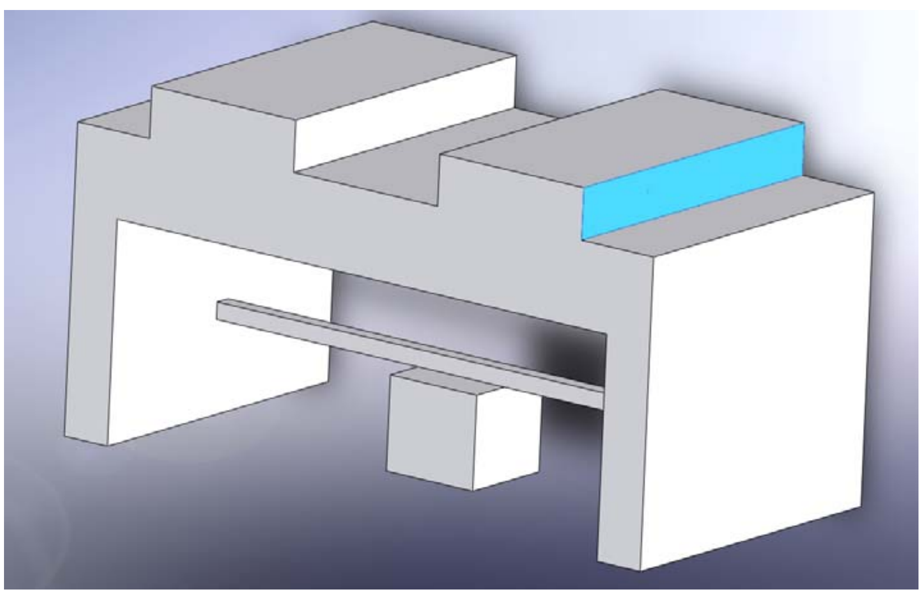


Figure 11: This design was very compliant in the z-direction but was not stiff enough in the x-direction (cutting direction). This would cause vibration of the tool and a rotational motion of the beam can cause the tool to wear away from the cutting edge radius.

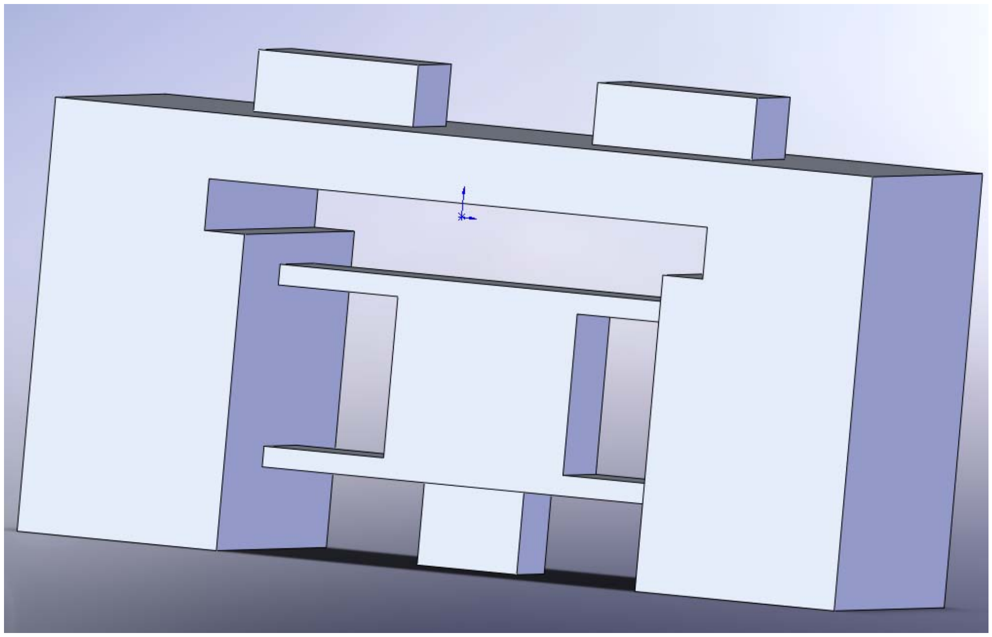


Figure 12: The design was not compliant enough in the z-direction. It only yielded a maximum deflection of 20µm for a load of 1700mN. If the compliant sections were to be any thinner than the above shown, this would yield in rotation in the x-axis causing the tool to vibrate.

Final Design

<u>CAD Drawings:</u>

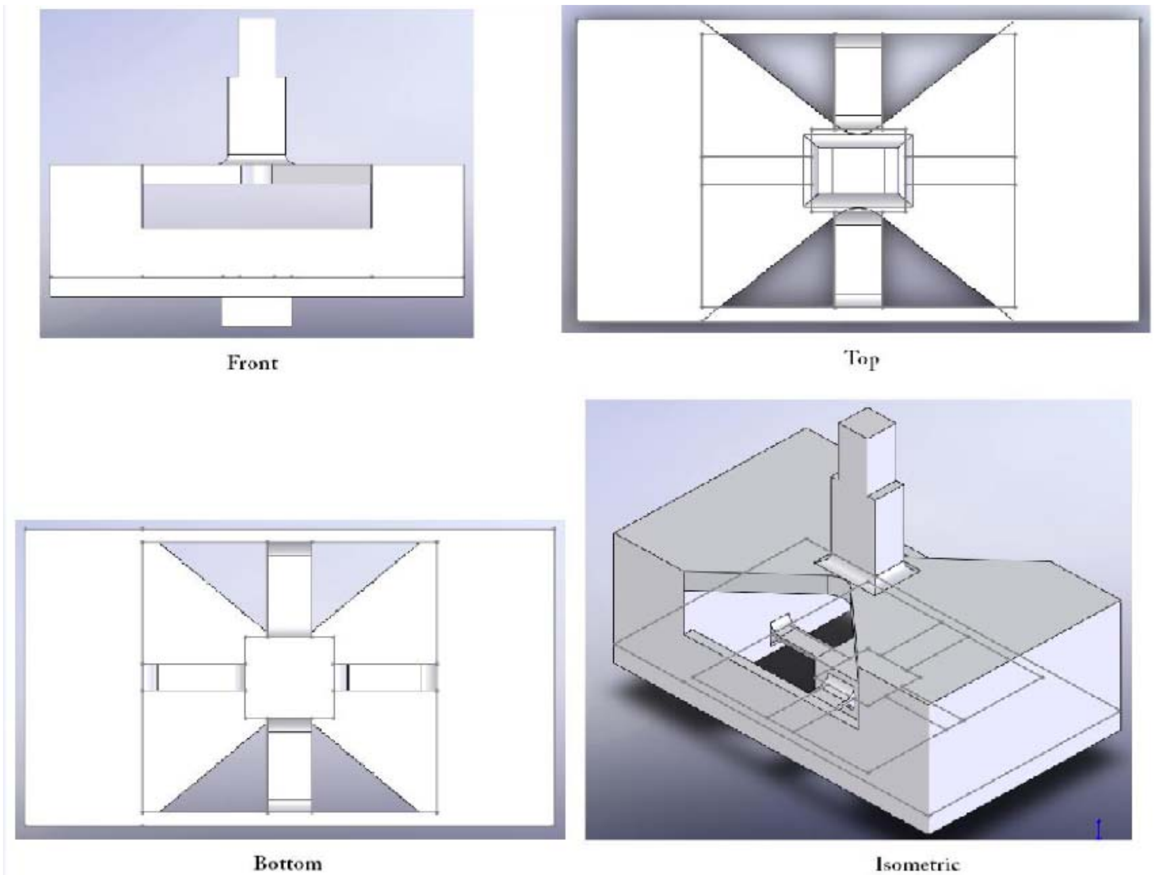


Figure 13: Shows the CAD drawing of four different views of the final compliant suspension design.

e-Drawings:

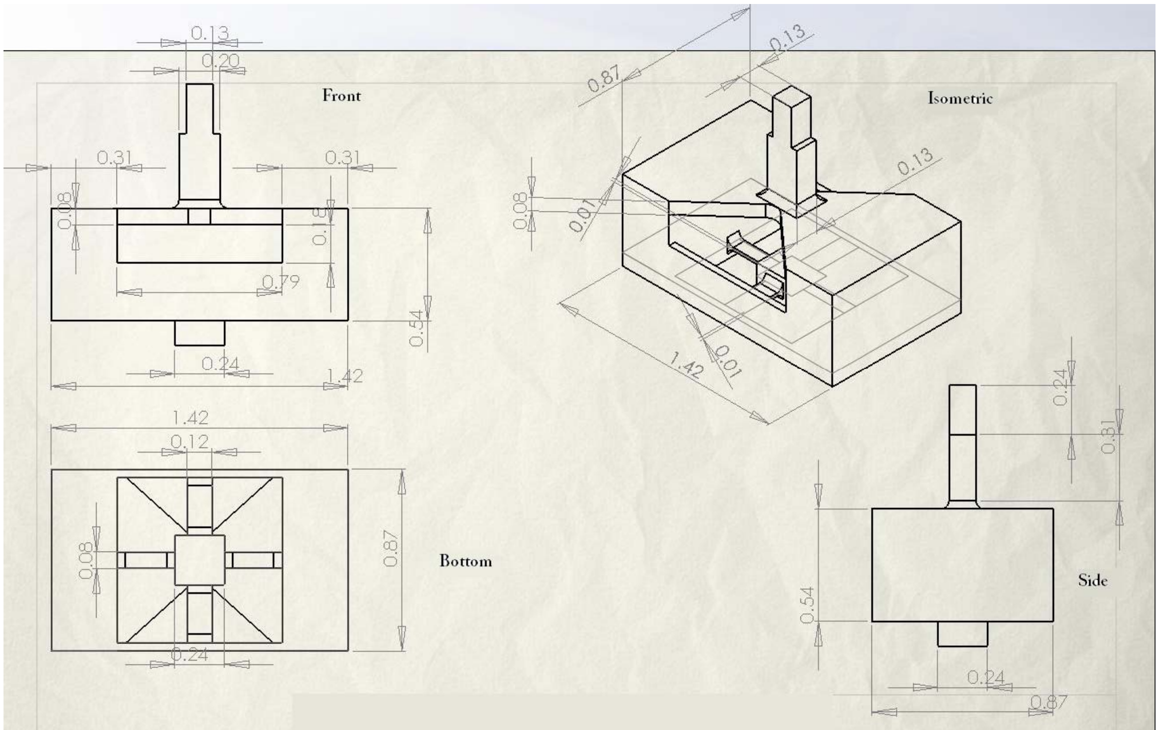
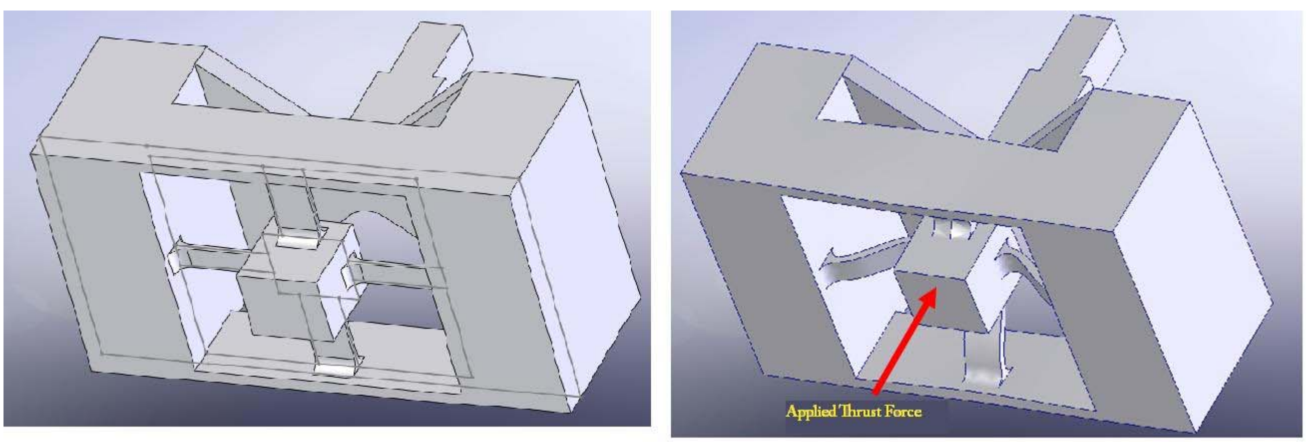


Figure 14: Shows the e-drawing of four different views of the final compliant suspension design with their corresponding dimensions in inches.

Deflection of Compliant Section:



Before Loading   After Loading

Figure 15: Compares the suspension before and after loading.

The deflection shown after loading is a result of an applied force of 1.7N which is the maximum recommended force for this design.

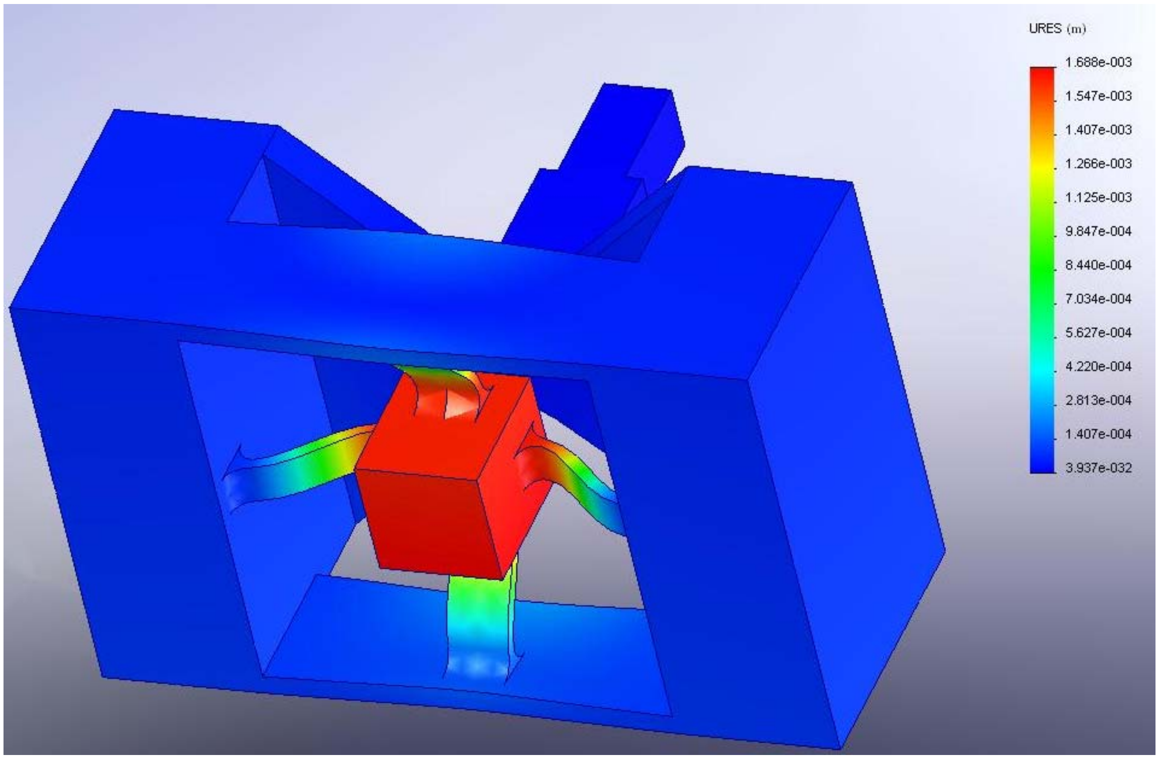


Figure 16: FEA showing the deflection in the compliant section of the suspension.

The maximum deflection is about 1.7mm for a maximum recommended load of 1.7N. The load-deflection ratio is similar to the old suspension provided by CETR. The FEA also shows that the new compliant suspension does not bottom out at any point under applied load. The compliant section is stiff in the cutting direction and this will eliminate tool vibration. An FEA was carried out in the cutting direction and no significant deflection was obtained.

Stresses:

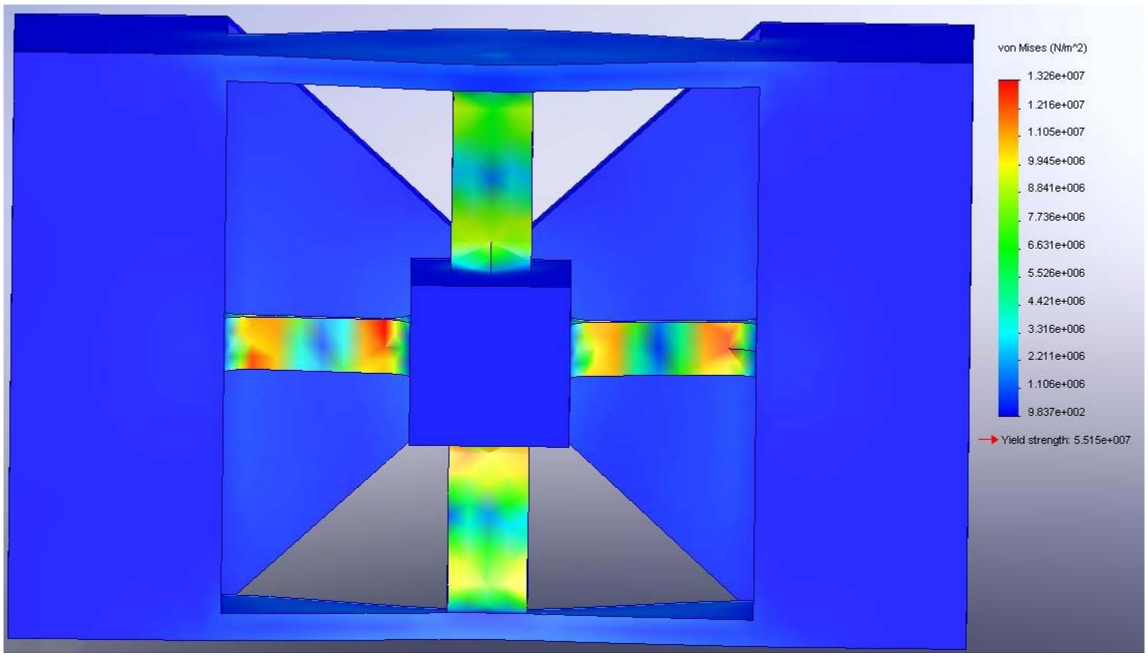


Figure 17: FEA showing the von Mises stress distribution in the compliant section of the new suspension.

The maximum stress obtained from the FEA is 1.33e7 and the material has yield strength of 5.52e7. The compliant suspension is designed based on a factor of safety of 4.16.

Assembly Pattern:

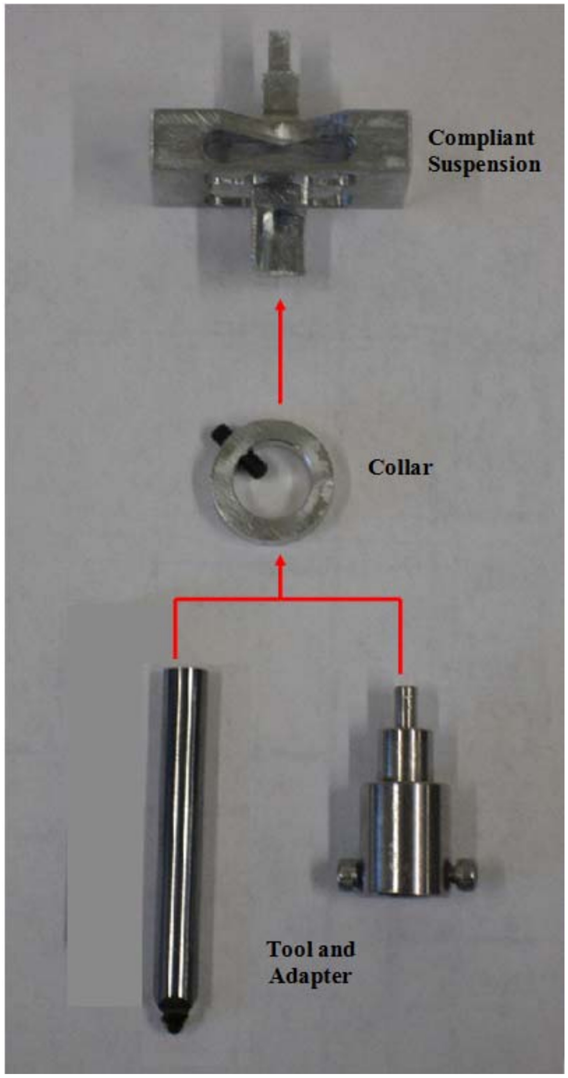


Figure 18: The image shows the assembly diagram for the compliant suspension.

The tool/any adapter sits in a v-groove and is held by a round collar. The v-groove is capable of housing shanks with variable diameters (round and square).

Figure 19: Image shows a diamond tool clamped on the compliant suspension with a collar.

Figure 20: Image shows the suspension in the load cell on the UMT.

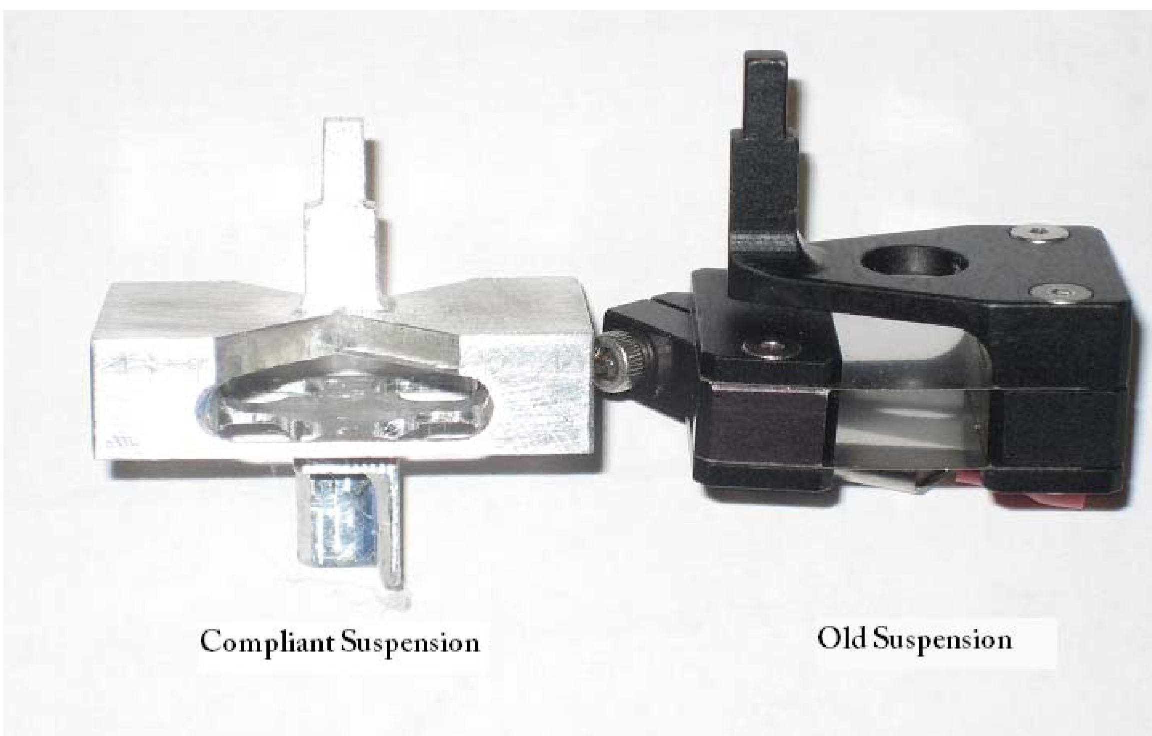


Figure 21: Image compares the new compliant suspension and the old suspension by CETR. The new suspension is slightly more compact in size.

Conclusion

The overall compliant suspension design was a success as is also shown in the finite element analysis. The total number of parts was reduced to a single part (from 8 parts) with no joints, elimination any form of vibration within the suspension. The compliant section in the suspension is also fixed on all four sides to the main suspension frame making it stiff in all directions (especially the cutting direction) except along the compliant axis (the z-direction). Since the entire suspension is symmetrical, there is no overhang (unlike the cantilever design), enabling the tool to cut perfectly straight and wear at the center of the cutting tool tip. If the complaint suspension is loaded within the recommended force-load (1.7N), it is expected to have an infinite life cycle.

Further testing of the compliant suspension will be carried out over time to compare the old and new design. Several experiments (diamond turning and scratches) will be conducted to compare the vibration on the suspension structure, vibration on the tool, stability of cutting and thrust forces, machined surface finish of the workpiece and tool wear pattern.

References

[1] Motsinger, R.N., 1964, "Flexural Devices in Measurement Systems," Chapter 11 in Measurement Engineering by P.K. Stein, Stein Engineering Services, Phoenix, AZ.

[2] Tottle, S.B., 1967, "Semifixed Flexural Mechanism," Chapter 8 in Mechanisms for Engineering Design, Wiley, New York.

[3] Howell, L.L., and Midha, A., 1994, "The Development of Force-Deflection Relationships for Compliant Mechanisms," Machine Elements and Machine Dynamics: Proceedings of the 1994 ASME Mechanism Conference, DE-Vol. 71, pp. 501-508.

[4] Dowling, N.E., 1993, Mechanical Behavior of Materials, Prentice Hall, Upper Saddle River, NJ.

[5] Pilkey, W.D., 1997, Peterson's Stress Concentration Factor, Wiley, New York.