Designing compact remote centre of compliance devices for assembly robots

Jyotirmoy Ray, Vaibhav Gupta, Sudipto Mukherjee, Jitendra P. Khatait

Abstract

Remote centre of compliance is a critical passive device for successful robotic insertion operations. A variant of the classical device developed in this study is compact and has been manufactured through rapid prototyping. By characterizing the properties of remote compliance through a joint less elastic formulation, the centre location and maximum deflection of the device are derived as functions of geometric dimensions, Youngs modulus, ultimate tensile strength and Poissons ratio. A device is modelled for a round peg-in-hole insertion using a KUKA KR-5 robot. For specified maximum allowable deflection, the theoretical values for stiffness are compared with FE results. The device is then manufactured using rapid prototyping method and the design is validated by testing. The proposed design can be customized for a range of geometrical constraints, centre location and maximum deflection.

Keywords: centre of compliance, compliant mechanism, rapid prototyping

1 Introduction

A common but important task during the assembly process is the insertion of one part into another. During automation of the assembly process, jamming can occur when the parts have very close tolerances. Typically, positioning tolerance needed during insertion is one order of magnitude higher than the clearance when using position control modes. This problem was encountered when trying peg-in-hole insertion using the KUKA KR-5. One way to solve this problem is to have an active feedback system and to compensate for the errors during the process, raising costs in design, operations and maintenance. However, such an active system require force-torque feedback from the joints of the robot for modelling the robot stiffness or high precision imaging and scanning sensors which lead to a costly setup [1]. For example, the KUKA LBR, also

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available in the lab, can be programed for remote compliance but costs ₹90 lakhs as opposed ₹23 lakhs for the KUKA KR-5. An alternative approach to this problem is using a Remote Centre of Compliance fixture (RCC) which passively compensates for the error during the assembly process.

The classical RCC device was developed by Whitney, Drake, etc at Draper Laboratory [2, 3]. The first models used rubber-metal sandwiches called elastomer shear pads to bring compliance to the device [4]. However the results were unpredictable as the analysis model of shear pads were inaccurate. Other models also came up consisting of compliant linkages and elastic joints.

Ciblak & Lipkin [5] presented a new approach to look at the RCC by considering multiple rods being attached to two rigid plates and the analysing the resultant center-of-compliance using elastic beam theory. In this study, the above approach was implemented to develop a joint-less and compact variant of the RCC that can be manufactured through 3D printing and assembled by adhesives.

The aim of the developed device is to compensate for a maximum allowable deflection of $2mm$ and $5°$ for a peg-in-a-hole experimental setup established at PAR Lab, IIT Delhi using a KUKA KR-5 robot.

2 Approach

The device had numerous geometrical dimensions which when coupled with the material properties resulted in a large number of control variables. So, it was decided to reduce the number of control variables.

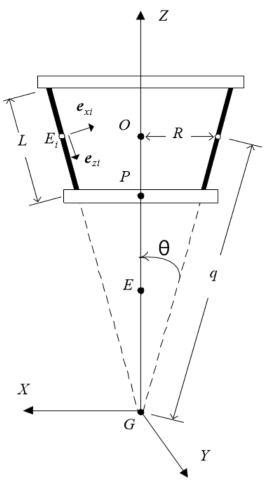

Figure 1: Geometry of the symmetrical beam arrangement design proposed by Ciblak & Lipkin [5]

2.1 Geometrical Properties

The design proposed by Ciblak & Lipkin [5] had presented the centre of compliance's location and stiffness of the model as a function of the geometrical properties of the model. The slenderness ratio of the beams (σ), the angle of slant (θ) of the beams and the projection ratio (p) of the device are closely related to each other. (Eqn. 1).

$$p = \frac{(\sigma^2 - 12)\sin\theta\cos\theta}{12 + 12\cos^2\theta + \sigma^2\sin^2\theta} \tag{1}$$

In eqn. 1, the slenderness ratio is $\sigma = 4\frac{L}{d_o}$ for a beam with circular cross-section of diameter d_o and the projection ratio is a reference to the dimensionless ratio $p = \frac{EO}{R}$ (refer Fig. (1)) and is the position of the remote centre of compliance normalized by the central radius R which serves as a characteristic dimension of the beam arrangement.

As the model was to be tested on an existing setup, the gripper design and length was part of the constraint set, which fixed the position of the compliance centre relative to the robot. To keep the device compact, the projection ratio had to be kept small by manipulating the slenderness ratio of the beam and the slant angle. A flat between these two parameters (refer Fig. (2)) for multiple values of p shows a dead band where possible values of σ is constant for a range of θ. Choosing the parameters from this dead band led to a robust design where, for a constant projection ratio, the required σ remains largely unchanged.

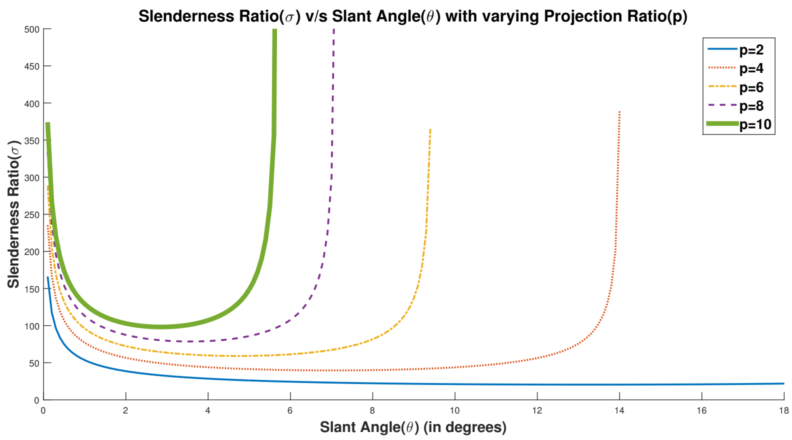

Figure 2: A dead band is observed in the plot. The range of the band increases with the decrease in the projection ratio.

So, the values of slant angle and slenderness ratio became dependent on the value of the projection ratio. The slant angle was chosen to approximately lie in the middle of the dead band.

2.2 Material Properties

After the projection ratio was chosen, the theoretical stiffness matrix was calculated using which the force (F) and moment (M) on the device was found when it is under maximum allowable deflection condition(d_{max} & Θ_{max}). Using this force and moment, maximum stress(σ_{max}) was found and compared to various materials' yield (S_y).

To find the value of maximal stress, we further required the weight($W_{gripper}$) of the gripper and its length($L_{gripper}$).

$$F = k_{XY} \times d_{max} \tag{2}$$

$$M = \kappa_{XY} \times \Theta_{max} \tag{3}$$

$$\sigma_T = \frac{W_{gripper} \times \cos\theta - F \times \sin\theta}{A \times n} \tag{4}$$

$$\sigma_B = \frac{r(F \times L_{gripper} - M)}{I \times n} \tag{5}$$

$$\tau = \frac{F \times \cos\theta + W \times \sin\theta}{A \times n} \tag{6}$$

$$\sigma_{max} = \sqrt{(\sigma_T + \sigma_B)^2 + 3\tau^2} \tag{7}$$

$$\sigma_{max} < S_y \tag{8}$$

Using Eqn. 8, it was ensured that the material does not fail under the desired deflection condition. Using a trial-and-error method, suitable value of the geometric properties and appropriate material was found for the construction of the device.

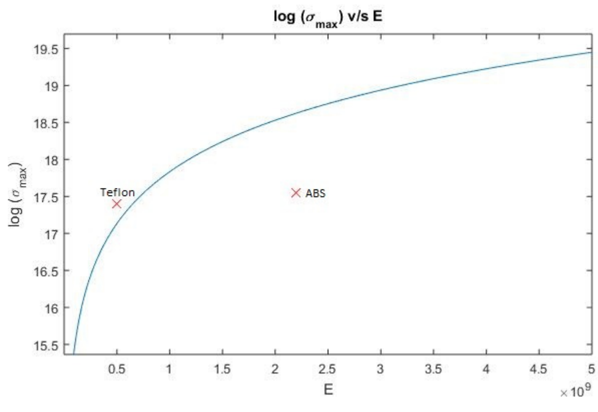

Figure 3: Graph between log of maximum stress and Young's Modulus. As the log of Teflon's Yield Strength lies above the log of maximal stress, Teflon beams are suitable for the design.

3 Initial Design

For the initial design, the simplest possible structure was a three beam model. A plausible design with projection ratio of 8 and slant angle of $3.5°$ was used. Using these values, we got the slenderness ratio of the beams to be 78.63 which when coupled with a beam diameter of $3mm$ resulted in beam length of $59mm$.

An aspect ratio of 1.39 was achieved in the model which is well within the range of typical values found in previously designed RCC devices. The aspect ratio is another dimensionless quantity defined as $\alpha = \frac{L}{2R}$. It is a measure of the device shape analogous to slenderness ratio. Roughly, large α indicates a slender structure and small α a compact one. Typical values in the literature for RCC devices fall within the range from 0.5 to 2.

The designed RCC model could be contained in a cylinder of diameter $23mm$ and height $59mm$. The most appropriate material for the design was found to be Teflon rods which were then to be rigidly attached to the upper and lower discs of the device.

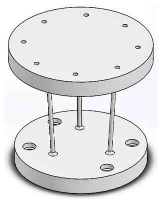

Figure 4: Isometric view of the initial design of the mechanism

3.1 Limitations

When the theoretical values of stiffness were matched with the values calculated using FE methods, a large discrepancy was found. (Table 1)

Table 1: Linear Stiffness from Theory and FE analysis for initial 3 beam design

S. No.	Stiffness Parameter	Theory	FE analysis
1	k_x	$680N/m$	$294N/m$
2	k_y	$680N/m$	$295N/m$
3	k_z	$1.79 \times 10^5 N/m$	$4.15 \times 10^5 N/m$
4	κ_x	$20.6Nm/rad$	$29.9Nm/rad$
5	κ_y	$20.6Nm/rad$	$29.8Nm/rad$
6	κ_z	$0.234Nm/rad$	$0.101Nm/rad$

It was identified that the design proposed was not symmetrical in x-y plane which was one of the main assumptions on which the model was based on. It was found that it was due to small number of beams in the conical arrangement which was corrected by increasing the number of beams to four. This increased the symmetrical nature of the device and the theoretical stiffness of the model was comparable to one calculated using FE methods. (Table 2)

Table 2: Linear Stiffness from Theory and FE analysis for four beam model

S. No.	Stiffness Parameter	Theory	FE analysis
1	k_x	$907N/m$	$893N/m$
2	k_y	$907N/m$	$894N/m$
3	k_z	$2.39 \times 10^5 N/m$	$2.32 \times 10^5 N/m$
4	κ_x	$27.5Nm/rad$	$73.7Nm/rad$
5	κ_y	$27.5Nm/rad$	$73.7Nm/rad$
6	κ_z	$0.312Nm/rad$	$0.283Nm/rad$

Another major problem was the length of the total apparatus. In this design, the gripper mechanism was to be mounted after the RCC model which made the total length to be about $200mm$ ($140mm + 60mm$). To make the device more compact, it was decided that a part of the gripper mechanism would be kept inside the RCC structure which would reduce the overall system height.

4 Final Design

The design which we came up with to account for all the limitations of the previous design was a four beam RCC model with the projection ratio of 2.7 and slant angle of 9°. This resulted in the slenderness ratio 27.3 which gave beams of diameter $3mm$ and length $20.5mm$.

This resulted in more compact design for the device and we got lower base to be large enough to accommodate the gripper mechanism inside the RCC. This reduced the overall length by $\sim 60mm$. When the theoretical stiffness of the device was compared to the one that we got by FE analysis, the errors were found to be within the acceptable range. (Table 3)

5 Results

The analysis was divided into two parts.

The first part was to model the loading on the RCC in such a way that simulates the ideal loading conditions that is proposed in theory[5]. For example, to find the linear stiffness in the x-direction, a simple force is applied at the centre of compliance in the x-direction and the displacement is measured. Refer Table 3.

The linear stiffness are off by 50% with FE values being lower. The angular stiffness in x and y directions from FE analysis show maximum deviation from the theory. They are lower by an order of magnitude. The angular stiffness in z-direction shows opposite deviation in being greater than the theory. We have been unable to identify the cause of these deviations.

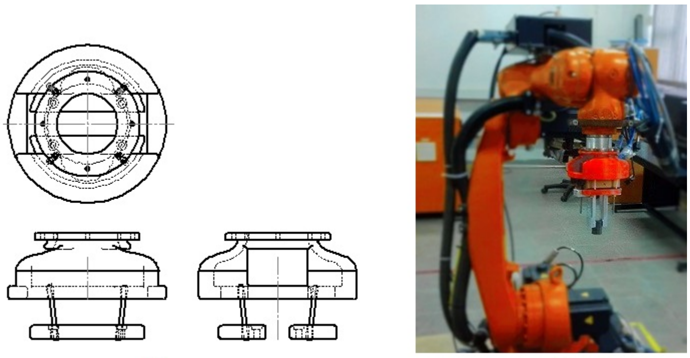

Figure 5: (a) Projection drawings of the final design (b) Final Prototype attached to a KUKA KR-5 robot for testing.

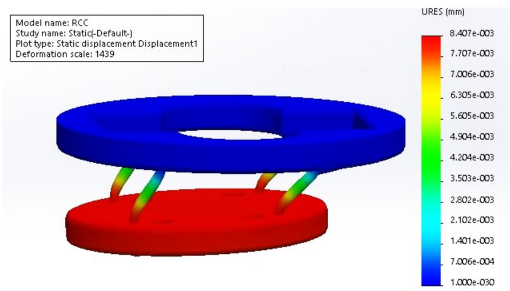

Figure 6: FE analysis of simplified version of final prototype. It was done to find the stiffness values in all six primary directions. Each value was found separately as the stiffness matrix had already been diagonalised.(Displacement scale magnification 1439)

Table 3: Linear Stiffness from Theory and FE analysis for the design.

S. No.	**Stiffness Parameter**	**Theory**	**FE analysis**
1	k_x	$1.87 \times 10^4 N/m$	$1.28 \times 10^4 N/m$
2	k_y	$1.87 \times 10^4 N/m$	$1.28 \times 10^4 N/m$
3	k_z	$6.74 \times 10^5 N/m$	$6.20 \times 10^5 N/m$
4	κ_x	$269 Nm/rad$	$34.4 Nm/rad$
5	κ_y	$269 Nm/rad$	$37.8 Nm/rad$
6	κ_z	$15.0 Nm/rad$	$16.1 Nm/rad$

The prototype was evaluated for stiffness in the plane lateral to the insertion direction by mounting the prototype on a KUKA KR-5 robot (purely as a positioning device with 6 DoF) and recording the readings of a Force-Torque sensor (ground mounted) against which the peg (centre of compliance) interacted. The forces and moments observed for a fixed displacement of $0.5mm$ to the mounting plate is applied to the computer model and the simulated displacement is obtained which is within 10% of the limits estimated by FE analysis (Table 4 and Fig 7).

Table 4: Comparison between experimental data and FE analysis.

Direction	F_x	F_y	F_z	M_x	M_y	M_z	Displacement	
							Observed	FE Analysis
x, y	$0.96N$	$0.40N$	$0.79N$	$0.004Nm$	$0.038Nm$	$0.028Nm$	$0.5mm$	$0.48mm$
z	$0.55N$	$0N$	$8.10N$	$0.56Nm$	$0.20Nm$	$0Nm$	$0.5mm$	$0.37mm$

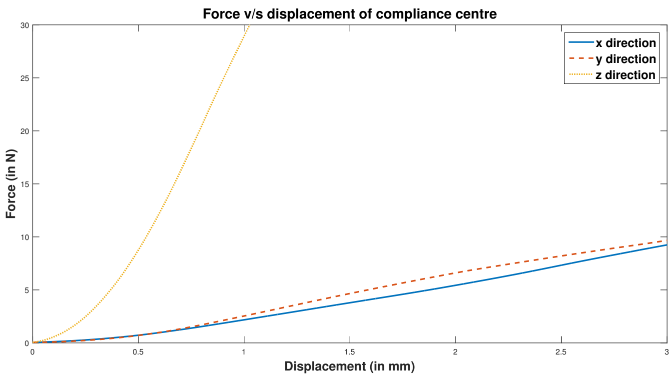

Figure 7: Force v/s displacement of compliant centre during experiment.

6 Conclusion

The device discussed in this paper is a validation of previously presented theoretical results regarding a remote centre of compliance device. This model of the RCC has been developed such that the additional volumetric footprint due to it is small. A jointless design is proposed which can be potentially made through rapid prototyping. The various parameters governing the shape and characteristic stiffness of the device have been optimized systematically according to the maximum allowable deflection conditions while designing. The prototype has been mounted on an industrial robot arm and tested. The linear stiffness of the design obtained by FE analysis are comparable to the theoretical values. Also, the displacement observed in the prototype is compared to the FE analysis result. Deviations observed in the value of the stiffness from theory and FE analysis has to be accounted for.

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Appendix

A Analysis Model by Ciblak & Lipkin [5]

The model developed by Ciblak & Lipkin [5] considered beams of the mechanism as rigid rods and used theory of beam elastic centre as the basis of this model.

A.1 Assumptions

1. Beams have a symmetric cross-section
2. Beams are slender
3. The mechanism have a conical symmetry

A.2 Final Results

The stiffness matrix(K_E) is represented as a diagonal matrix at the elastic centre.

$$K_E = diag\{k_{XY} k_{XY} k_Z \kappa_{XY} \kappa_{XY} \kappa_Z\} \tag{9}$$

$$k_{XY} = \frac{1}{2} n[\lambda_x \cos^2\theta + \lambda_y + \lambda_z \sin^2\theta] \tag{10}$$

$$k_Z = n[\lambda_x \sin^2\theta + \lambda_z \cos^2\theta] \tag{11}$$

$$\kappa_{XY} = \frac{1}{2} n[\mu_x \cos^2\theta + \mu_y + \mu_z \sin^2\theta + R^2 \frac{\lambda_x\lambda_y \sin^2\theta + \lambda_x\lambda_z\lambda_y\lambda_z \cos^2\theta}{\lambda_x \cos^2\theta + \lambda_y + \lambda_z \sin^2\theta}] \tag{12}$$

$$\kappa_Z = n[\mu_x \sin^2\theta + \mu_z \cos^2\theta + \lambda_y R^2] \tag{13}$$