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A novel variable stiffness mechanism with linear spring characteristic for machining operations

Ngoc-Dung Vuong†, Renjun Li‡, Chee-Meng Chew‡, Amir Jafari§ and Joseph Polden†*

†*SIMTech, 71 Nanyang Drive, 638075, Singapore. E-mail: ndvuong.work@gmail.com*
‡*National University of Singapore, 9 Engineering Drive 1, 117576, Singapore. E-mail: lirenjun, chewcm@nus.edu.sg*
§*Department of Mechanical Engineering, University of Texas at San Antonio, San Antonio, Texas, USA E-mail: amir.jafari@utsa.edu*

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SUMMARY
Variable stiffness mechanisms are able to mechanically reconfigure themselves in order to adjust their system stiffness. It is generally accepted that only antagonistic designs, featuring quadratic springs, can produce linear spring-like behaviour (i.e., a linear relationship between the displacement and its resultant force). However, these antagonistic designs typically are not as energy efficient as series-based designs. In this work, we propose a novel variable stiffness mechanism that can achieve both linear-spring behaviour whilst maintaining an energy efficient characteristic. This paper will present the working principle, mechanical design and characterization of the joints stiffness properties (verified via experimental procedure). The pros and cons of this novel design with reference to the other Variable Stiffness Actuator (VSA) designs will be discussed based on experimental results and in the context of general machining tasks.

1. Introduction

To expand the capability of industrial robots into manufacturing domains outside of well-established motion-based applications, force control capability was introduced.[1,2] By doing this, positioning error can be compensated by force control as the robot is approaching the work-piece, as illustrated in Fig. 1. The absolute positioning accuracy of the tool-center-point (TCP) with respect to the work-piece in the force controlled direction is no longer critical, as the robot can now search for the location of the work-piece before carrying out the machining processes.

Since most industrial robots available in the marketplace today do not feature force control capability, there is a need for an end-effector module that can introduce this functionality in these conventional systems. In searching for a suitable mechanism for this end-effector, an interesting concept that involves the deliberate use of a compliance element in between the actuator and the output port,[3,4] called series elastic actuation, was found. The development of the Series Elastic Actuator (SEA) has enabled a new way of designing robotic devices for contact operations. In particular, when SEA mechanisms are employed in robot joints, it has been demonstrated that the introduced compliance improves disturbance rejection capability,[4,5] especially to high frequency disturbances such as impact and machining force.

However, these SEA actuators feature fixed levels of compliance, which limits their usability to a certain extent. For example, when a compliant robot is used for machining, high stiffness is preferred when the end-effector is travelling in a free space before going near the work-piece, whilst lower stiffness is preferred when searching for the contact point. This is not readily achievable for a robot with fixed joint stiffness. Furthermore, logic dictates to us that a joint optimized for one task will not

* Corresponding author. E-mail: poldenjw@SIMTech.a-star.edu.sg

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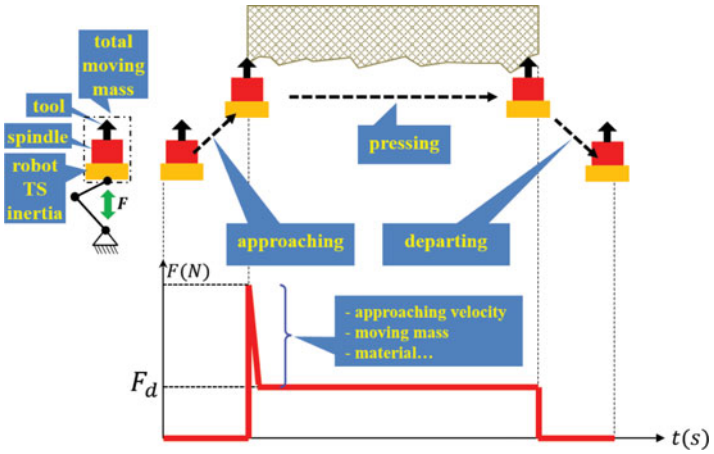


Fig. 1. Force control for machining process.

be able to provide optimal performance across a range of different tasks. Therefore, as an extension to the fixed compliance joint mechanism, many variable stiffness actuators (VSA) have been developed to adapt to different applications, or applications that require an adjustment of stiffness in the process.

It appears that this relatively new concept, which has been demonstrated in human-robot interaction applications,[5,6] could be a well suited for profiling operations (such as machining or deburring) because of it's ability to adjust system stiffness on-the-fly. This is particularly useful in machining operations, as stiffness can be dynamically altered in order to reduce vibration experienced by the robot and also improve robot accuracy across the entire process.[2]

2. Categorization of Variable Stiffness Mechanisms

In general, most of variable stiffness designs can be categorized into five groups: antagonistic approaches (C1–C3) and series-based approaches (C4–C5).

- *C1: Unidirectional* configurations are the simplest class of antagonistic variable stiffness mechanism. As shown in Fig. 2(a), each actuation unit is connected to the output link through a non-linear unidirectional spring. Unidirectional springs are only able to apply force in a single direction. As a result, in this configuration only one actuator is able to apply force to output link at any given time. Examples of these types of unidirectional actuator are: Biologically inspired joint stiffness control,[7] Actuator with Mechanically Adjustable Series Compliance (AMASC),[8] and Plated Pneumatic Artificial Muscles (PPAM).[9]
- *C2: Cross-coupling* actuators (Fig. 2(b)) introduce an additional non-linear spring between the two actuation units, which we will term the cross-coupling spring. The cross-coupling spring grants each individual actuation unit a full steering capacity of the link. As a result, each actuation unit will require only half the force output capability of their counterparts in an equivalent unidirectional mechanism. By counter-rotating each actuation unit relative to one another, the stiffness properties of the overall joint are adjusted. The VSA[10] is a prominent example of an actuator utilising this cross-coupling design.
- *C3: Bidirectional* mechanisms (Fig. 2(c)) each actuation unit is connected to the output link through a pair of non-linear springs. Each spring pair consists of two non-linear unidirectional springs arranged in a bidirectional configuration, which gives each actuation unit the ability to both push and pull the output link. Examples of actuators in this class include: VSA-II,[11] VSA-CUBE,[12] Bidirectional Antagonism with Variable Stiffness (BAVS).[13]
- *C4: Pretension based* actuators are categorised as series-based actuators. In these designs, the first actuation unit is connected to the output link via springs which control the link position, whereas the second one regulates the pretension of the spring as shown in Fig. 3(a). Since this spring is non-linear, it's stiffness can be altered by changing the pretension. Some examples of the actuators in this class are: Mechanically Adjustable and Controllable Compliance Equilibrium Position Actuator (MACCEPA),[14] (MACCEPA 2.0),[15] Variable Stiffness Joint (VS-joint),[16] Safe

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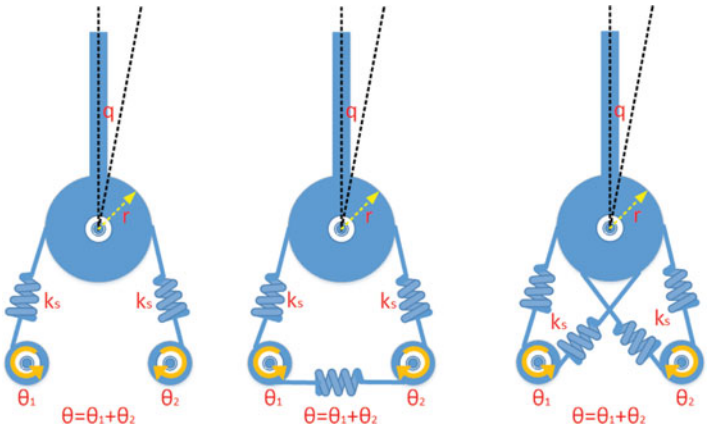


Fig. 2. Antagonistic-type variable stiffness designs: (a, left) Unidirectional (b, centre) Cross-coupling (c, right) Bidirectional.

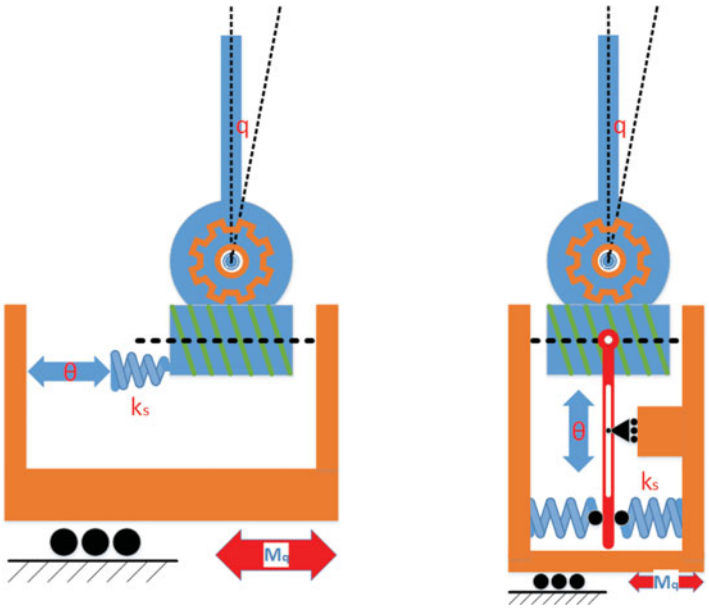


Fig. 3. Series-type variable stiffness designs: (a, left) Pretension based (b, right) Position based.

Joint Mechanism (SJM I)[17] and (SJM II).[18] It is worth noting that in these examples linear springs are employed. However, as the stiffness adjustment mechanisms used provide a non-linear force-deflection profile, these springs can be considered as non-linear.

- *C5: Position based* actuators are able to alter their transmission ratio between the input force and the spring deflection through the variation of a lever mechanism. This lever mechanism has three principal points: the pivot, i.e., the point around which the lever can rotate; the spring attachment point, i.e., the point at which springs are located; the force point, i.e., the point at which the force is applied to the lever. By changing the position of one of these principal points, the lever stiffness can be varied. Some more recent position based VSAs are: Actuator with Adjustable Stiffness (AwAS-I),[19] AwAS-II,[20] Compact Variable Stiffness Actuator (ComPact-VSA),[21] Energy Efficient Variable Stiffness Actuator[22] and Variable Stiffness Actuator University of Twente.[23]

Although the purpose of this paper is not to review these designs in detail, we would like to highlight the following observations of note:

- In general, the size of C1–C3 designs are larger than C4–C5 due to employing two motors of same size. Interested readers may refer to www.viactors.org for more details.

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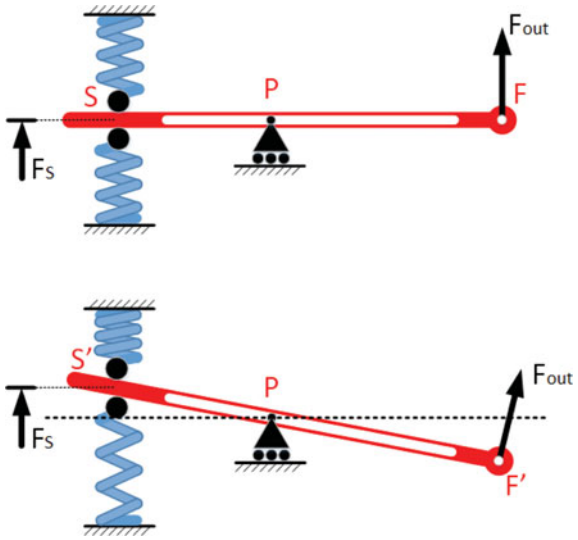


Fig. 4. Underlying stiffness adjustment mechanism of AwAS-II[20] and CompACT VSA[21] VSAs. (a, top) equilibrium point, (b, bottom) delected state.

- If we assume that the springs k_s in C1–C3 can be represented in this form $k_s = k_0 x^n$, where $n \geq 0$ is an integer representing the non-linearity order of the spring, θ is a generic variable to regulate the mechanism stiffness (K). The relationship between the displacement versus resultant force/torque in these antagonistic systems is:

$$T = -rk_0 A \begin{bmatrix} (p + rq - \theta)^n \\ (p - rq - \theta)^n \\ (p + rq + \theta)^n \\ (p - rq + \theta)^n \end{bmatrix}, \tag{1}$$

  where A is [0 0 −1 1] for C1 and C2 and [−1 1 −1 1] for C3, r is the pulley radius, q is the joint angle, θ is the stiffness controlled variable and p is the spring pretension. The differential of T with respect to q will give us the system stiffness. As is seen, the system's stiffness can only be variable if $K = dT/d\theta$ is a function of θ i.e., $n \geq 2$. In other words, variable stiffness is possible if, *and only if*, the mechanical spring elements k_s are non-linear.
- Spring-like characteristics of the system can only be achieved if K is NOT a function of q (joint displacement). From the above equations, it can be seen that $n = 2$ is the only solution for antagonistic approaches.
- C5's is the only VSA which can vary it's system stiffness from zero to infinity (in theory).
- The energy required to change the system stiffness from k_{min} to k_s is higher for antagonistic approaches ([24]).

From these observations, we can see that the C5 VSA is a good candidate for machining operations. First, it can adjust its stiffness over the full range (from zero to infinity) in an energy efficient manner; something which the other actuators intrinsically fail to achieve both with respect to energy efficiency and stiffness range. In addition, it can feature a more compact size than the other actuators with the same output power. The major drawback, however, of the C5 class is that it's system stiffness is not linear with respect to the joint's passive deflection.

The work presented in this paper proposes a new C5 actuator design that is able to maintain many of these beneficial characteristics, whilst also removing this non-linear stiffness relationship with deflection. The details of this design are presented over the next few sections of the paper, but before this we will explain this non-linearity in C5 actuator responses. And so why one would want to remove this effect from a VSA designed for manufacturing applications.

To better understand the causes of non-linearty, stiffness adjustment mechanism in two actuators under this class: AwAS-II[20] and CompACT VSA[21] is shown in Fig. 4. In Fig. 4, F_{out} is applied to the

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"force point" (F) of the lever, the lever rotates around the "pivot point" (P) and a spring force F_s will be induced at the "spring point" (S) of the lever at the other end. The aforementioned non-linearity comes from two aspects of this mechanical structure. First, when the lever deflects, F_{out} will no longer be parallel to F_s (Remember F_{out} is always perpendicular to the lever while F_s is aligned with the fixed spring). Second, although the distance between force point to the pivot point is constant, i.e., $PF = PF'$, the length of the lever is not fixed in these actuators, i.e., $FS \neq F'S'$. Once the lever deflects, PS is no longer equal to PS', which implies that the lever ratio (PF/PS) does not remain fixed during deflection. This type of lever mechanism forms a key component of the C5 actuator, and this changing lever ratio which occurs during deflection will induce a non-linear stiffness response in the system.

This non-linearity between deflection and system stiffness is undesirable, as single frequency disturbances (such as vibration) which are input to the system will generate multiple frequency resultant forces. For example, a typical system force response to deflection of a C5 actuator can be represented by

$$T = \theta^2 k_0 \sin(2q)[24]. \tag{2}$$

If a single frequency disturbance

$$q_{dist} = q_0 \sin(w_0 t), \tag{3}$$

enters the system, the Taylor expansion of T around a point $2q = a$ to the 3rd order gives:

$$T = \theta^2 k_0 (\sin(a) + \cos(a)(2q - a) - \frac{1}{2}\sin(a)(2q - a)^2 - \frac{1}{6}\cos(a)(2q - a)^3 + O[2q - a]^4). \tag{4}$$

Substituting Eq. (3) into Eq. (4) and simplifying, T can be written in the form

$$T = \theta^2 k_0 \left(\alpha_0 + \alpha_1 \sin(w_0 t) + \alpha_2 \sin^2(w_0 t) + \alpha_3 \sin^3(w_0 t)\right), \tag{5}$$

where α_i is a function of a and q_0. As seen, the Fourier transform of $\sin^n(w_0 t)$ will in fact have additional frequency components at nw_0. In other words, the introduction of a single frequency disturbance (w_0) will generate a force/torque disturbance featuring multiple frequency components.

From a controller design point of view, it makes sense to ensure that the effect of disturbances are minimised during the mechanism design stage. To overcome this, we will propose an enhanced variable stiffness mechanism based on the C5 design which possesses the following properties:

- linear-spring characteristics (i.e., linear relationship between displacement and system stiffness)
- energy efficient in stiffness adjustment
- wide range of stiffness adjustment (0 to inf)

We have now presented the requirements for our design. In the remaining sections of this paper, design of the new variable stiffness joint will be presented. The paper is organized as follows: Section 3 will present the working principle and proposed mechanical structure. The resultant force-properties of this novel design are then discussed. In Section 4, experiments are performed to verify the force-stiffness characteristics of a prototype joint, and a discussion of these results is carried out. Finally, conclusions are drawn in Section 5.

3. Mechanical Structure

3.1. Working principle

Based on the requirements presented in Section 2, a novel mechanism to control joint stiffness is now outlined. On a fundamental level, the working principle of the joint relies on a cantilever arm. As shown in Fig. 5, a linear spring with stiffness k_0 is connected to one end, whilst the output force F_F is

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exerted at the other. Note that, in this paper, we assume that the spring is perfectly linear. Non-linearity associated with the spring constant will not be considered here. This assumption is common for all VSA designs mentioned in Section 2. Also from Fig. 5, the cantilever arm's motion is constrained at each end by two linear guides. Sliders on these linear guides are used to define the force acting point. The output load T is transmitted to the cantilever through a rack and pinion mechanism as shown in Fig. 6. The pinion pushes the slider on the linear guide down by q_f when the output shaft turns by angle q. The lever rotates about its axis P by angle α, moving the slider on the other side the lever by q_s. In other words, the spring will be effectively compressed by q_s due to the output torque T. It should be noted that the springs are pre-compressed with pretension p to ensure both will always be in compression. The position of the pivot point L_1 is controlled by a small additional motor, which we will term the stiffness motor, via a lead screw.

It can be seen that both the output force (F_F) and the spring force (F_S) will always act along the linear guides that are used to constrain the lever mechanism's motion. The linear guides are oriented in a manner to ensure that these forces will then always act in a direction parallel, but opposite, to one another. In addition, the two triangles $\triangle SPS'$ and $\triangle FPF'$ will always be similar to each other, i.e., $\triangle SPS' \sim \triangle FPF'$ always holds, regardless of the lever rotation angle α:

$$\frac{q_s}{q_f} = \frac{L_1}{L_0 - L_1} = -\frac{F_F}{F_S}. \tag{6}$$

This implies that the ratio of two sides of the lever arm will be independent from the deflection angle of the joint. In other words, the lever ratio (FP/PS) will remain constant regardless of deflection. This ensures the relationship between load and joint stiffness is kept linear, which satisfies the key design requirement outlined in Section 2.

If the output torque is T and radius of the pinion is R, and assuming the joint deflection angle due to the load T is q, the rotational stiffness will then be:

$$K = \frac{T}{q}. \tag{7}$$

It can be calculated by substituting the following equations and expression (6) into (7):

$$q_f = Rq, \tag{8}$$

$$T = F_F R. \tag{9}$$

The rotational stiffness is obtained as follows:

$$K = k_0 R^2 \gamma^2, \tag{10}$$

where $\gamma = \frac{L_1}{L_0 - L_1}$, k_0 is the stiffness of the linear-spring. Note that when deriving K, no assumption has been made on the magnitude of the rational angle. Thus, this equation will always hold, regardless of the joint deflection angle q. In other words, the system stiffness is independent of q, and therefore has the same characteristic of a physical linear spring. In addition, the system stiffness K can be regulated by adjusting L_1 or equivalently to modify γ (k_0, R and L_0 are all constant).

Figure 7 shows the mechanical structure of the novel variable stiffness joint prototype. As shown in this figure, the joint motor and the *stiffness motor* are connected to the input shaft and the lead screw, respectively. Here, the term *stiffness motor* is coined to imply that this motor is meant for changing the stiffness of the mechanism. Four linear guides are used to form a lever mechanism. The output load is transmitted to the lever through a rack and pinion mechanism.

3.2. Characteristics of the novel design

In this section, the theoretical characteristics of the proposed design are analysed and compared with the design requirements presented in Section 2.

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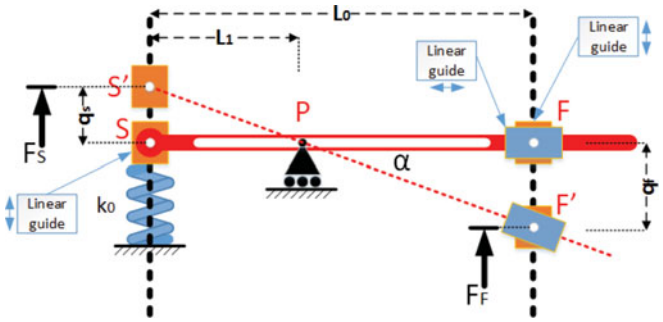


Fig. 5. Schematic diagram of the cantilever arm of the proposed variable stiffness joint.

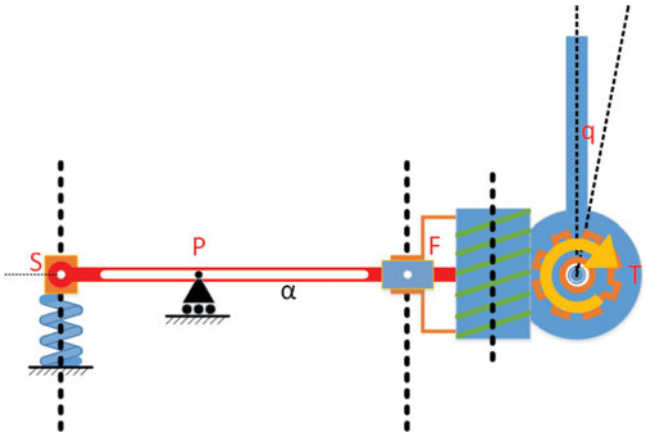


Fig. 6. Schematic diagram of the proposed variable stiffness joint.

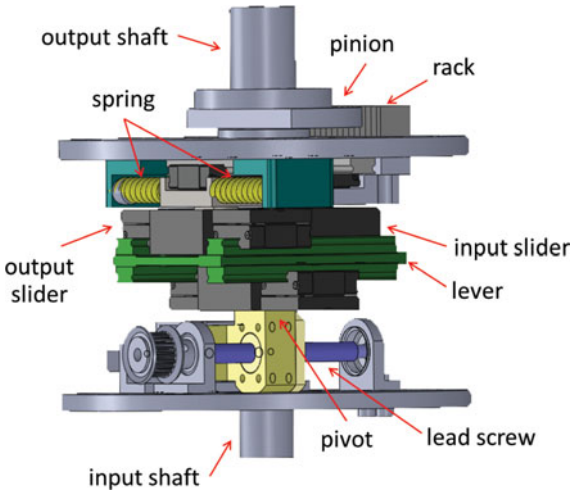


Fig. 7. Design of the variable stiffness joint. Lever mechanism is constructed using multiple linear guides. Its pivot point is controlled through a lead screw.

3.2.1. Key parameters. The key parameters of the joint are presented in 3.2.1. The input inertia refers to the total combined inertia of the parts that are rigidly connected to the driving motor, such as the casing frame, the lead screw and the stiffness motor, etc. The output inertia refers to the total inertia of the moving parts, such as the lever arm and the sliders that are connected to the lever arm. The input and output inertias are approximated based on inertia values of each of the individual parts, which were calculated with CAD software. Minimal output inertia is obtained by moving the pivot as

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Table I. Key parameters of the joint.

Description	Value
Energy storage capacity (J)	0.02
Apparent inertia inertia (gmm^2)	$2.92 \times 10^5 \sim 3.85 \times 10^5$
Time to change stiffness from 0 to ∞ (s)	3
Maximum link deflection (rad)	1.5
Output torque (Nm)	1
Size ($m \times m \times m$)	$0.2 \times 0.2 \times 0.4$

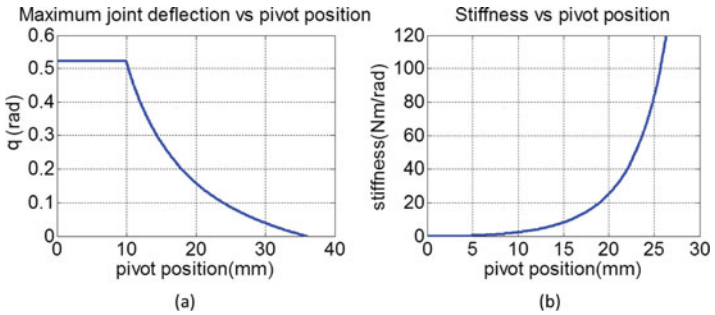


Fig. 8. Theoretical characteristic of the variable stiffness joint: (a). maximum allowable joint deflection(θ) vs. pivot position; (b). stiffness vs. pivot position.

close to the centre of the lead screw as possible. This is because a beam has minimum inertia about its centre of mass.

3.2.2. Joint deflection range. In our design, motion is constrained to a certain range at both ends of the cantilever arm. Therefore, maximum allowable deflection in the joint changes with the pivot position. For example, spring limits will be reached when the pivot is far away from the spring. Figure 8(a) shows the maximum allowable deflection with respect to the pivot position. The joint is designed in such a way that when the pivot is approximately within 10 mm distance from the spring, the maximum allowable internal joint deflection is $\pm 30^{\circ}(\pm 0.525$ rad).

3.2.3. Stiffness characteristics. Figure 8(b) depicts the simulated stiffness characteristic curve with respect to the pivot position. It can be seen that the stiffness ranges from zero to infinity. As the pivot moves further away from the spring, the slope of the curve increases. Therefore, the stiffness resolution will be higher when the pivot is closer to the spring. In practice, the spring stiffness can be selected based on a given application, allowing a higher resolution of stiffness adjustment to be obtained for the given operating range.

The above analysis demonstrates that the proposed mechanism features the key properties required to perform robotic machining. In the next section, a prototype of the proposed design will be shown and tested. The characteristics of the prototype will also be identified through experimental procedure.

4. Performance Evaluation

To evaluate the performance of the our variable stiffness joint mechanism, experimental trials with a physical prototype, shown in Fig. 9, were conducted. The key characteristic we are interested in is the relationship between the input load and the resultant internal joint displacement, the goal is to verify a linear relationship between the two.

4.1. Experiment procedures

The experimental setup is shown in Figs. 9 and 10. The input shaft of the variable stiffness joint is connected to a position-controlled motor and the output shaft is connected to a rigid link. The link is fixed while identifying the load and internal joint displacement relationship. The motor which controls the location of the pivot point of the cantilever, and hence the corresponding stiffness of the joint is connected to the lead screw through a belt and pulley mechanism. PID (proportional–integral–

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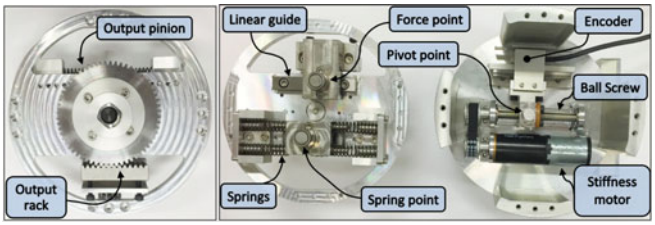


Fig. 9. (left) Output and springs; (centre) Lever Mechanism; (right) Stiffness adjustment mechanism.

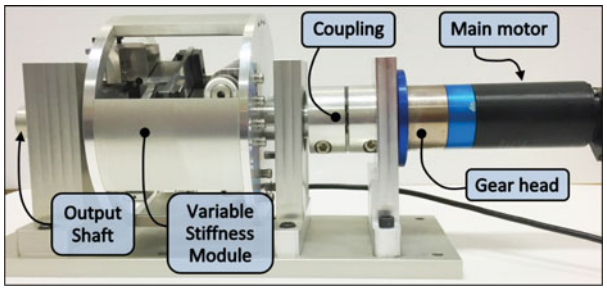


Fig. 10. The novel variable stiffness joint: Test-bed of the joint.

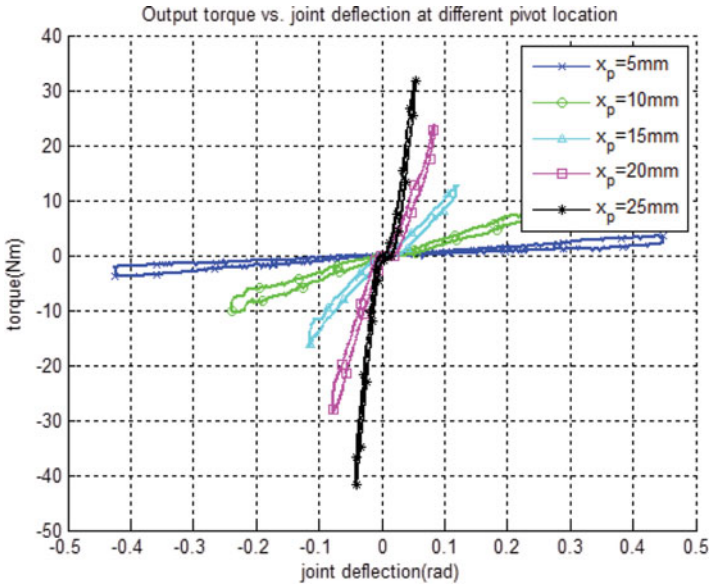


Fig. 11. Torque vs. Deflection with different pivot locations.

derivative) controllers were designed and tuned for both motors in order to accurately control their position. A force/torque sensor is connected to the output shaft to measure the output load. The input position, output position and the pivot position are measured by digital encoders.

In the experiment, the joint motor's desired position is altered from zero to the maximum allowable joint deflection. This is done at a slow rate, so that at each moment, the system's state can be regarded as quasistatic. The rotation of the motor which controls the joint stiffness remains stationary when the joint motor moves. During the experiment, the real joint deflection and the torque are recorded. The experiment is performed for five different pivot positions, and results averaged over 10 trials.

4.2. Results and discussion

Figure 11 shows the result of the linearity test of the variable stiffness joint, which indicate good linearity for all five pivot positions, ranging from low to high levels of stiffness; satisfying the design requirement.

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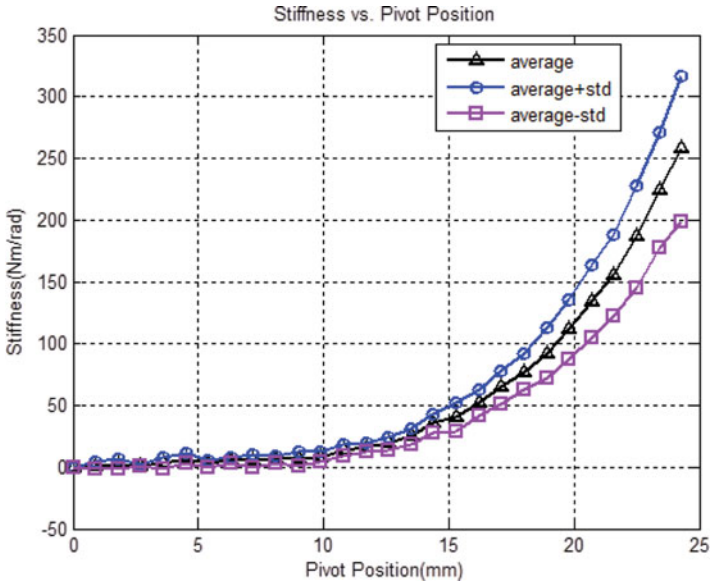


Fig. 12. Stiffness vs. pivot position.

In Fig. 12, the average joint stiffness, calculated over 10 experimental trials, is presented. The "average+std" and "average-std" represent the average stiffness summed with plus and minus the standard deviation, respectively, representing the error in all 10 trials. Figure 12 also shows that a fine stiffness resolution is evident when the pivot is positioned close to the spring. Conversely, the stiffness is observed to increase rapidly as the pivot moves away from the spring, which implies that the second and third design requirements, presented in Section 2, have been met. Small deviations between simulated and experimental results were observed in these load-displacement curves. These errors were attributed to imperfections present in the mechanical components.

From these experiments, a number of key observations are made. First, the relationship between output deflection and the input force is linear as predicted during the design phase. Second, near the equilibrium ($q = 0$) point, some discontinuities in joint deflection were observed. These discontinuities can be attributed to mechanical backlash from the rack and pinion gear used (see Fig. 9). In addition, a small degree of hysteresis was observed at lower stiffness ranges, as shown in Fig. 11, which most likely occur due to mechanical friction between components in the joint, which become more dominant in the lower stiffness ranges. Figure 12 also shows us the achievable stiffness range of the joint. In theory, this range should be infinite. However, due to the mechanical arrangement which limits the range of the pivot point of the prototype, this range is also has limited bounds.

5. Conclusions

In this paper, a novel variable stiffness joint designed specifically for robotic machining applications was presented. Design requirements such as linear load-stiffness relationship, zero to infinity stiffness range and high stiffness resolution in low stiffness ranges were proposed for machining processes. These requirements were met by the developed variable stiffness mechanism which is comprised of a cantilever arm featuring a constrained path of motion at each end. In Section 3, the working principle of the proposed joint was presented and discussed in detail. Experimental procedures conducted with a working prototype confirmed that the characteristic stiffness properties of the joint followed closely with the required design specifications. For future work, a new prototype will be built with higher specification in order to resolve some the mechanical issues, such as backlash and hysterisis, observed during experimental procedures carried out in this work. Additional controllers will be developed to allow the new prototype to achieve good force control capabilities that can be applied to real world machining applications.

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