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Abstract

I. INTRODUCTION

The beneficial effects of compliance on specific aspects of the robotic system, including interaction, safety and energy efficiency, have been studied and validated in many works [1], [2], mainly by employing actuators with preset level of compliance integrated within their motor and gear transmission systems. As an extension to these fixed compliance units, more recently, variable stiffness actuators (VSAs) have been introduced targeting to further maximize these compliance benefits at the robot body-ware level with respect to safety during interaction [3] as well as to further improve the energy efficiency of the robotic system particularly during locomotion [4]. VSAs typically employ two actuator units in combination with passive elastic elements to control, independently, the compliance and the equilibrium position of the actuated joint. However, the introduction of an extra actuator to achieve the functionality of the stiffness adjustment ultimately increases the complexity, size and the total weight of the actuation system. To move forward from single dof VSA implementations towards multi-dof VSA systems, requires careful consideration in all stages of the actuator development from the selection of the functional stiffness regulation principle, to the optimization of the mechanical assembly, and the integration of all mechatronic components. As a consequence, the implementation of VSAs in multi-dof robotic structures such as robotic arm, legs, or full humanoids still remains a challenging task requiring the realization of more compact and modular variable stiffness units.

The realization of a VSA design able to fulfill the mechanical requirements such as compact size and reduced weight, while maintaining high performance levels (high torque capacity, wide range of stiffness and short time in stiffness regulation) is not a trivial task. Usually, the size/weight reduction of most of the existing VSAs is obtained by reducing the range of stiffness and torque capacity. The specifications of some the most recently developed actuators can be found in [5]. In general, most of the variable compliant actuation systems presented in literature are implemented through antagonistic [6], [7], [8], [9], [10], quasi antagonistic [11] or serial configurations [12], [13]. The variable stiffness actuators in antagonistic configuration employ two compliant elements to provide power to the joint. This design is biologically-inspired, since mammalian anatomy follows the same concept, i.e. a joint actuated by two muscles arranged in an antagonistic manner. Alternatively, the VSAs in serial configuration, are characterized by the mechanical series motor-gear-compliant element and they can be designed both for linear and rotational implementations. The vast majority of the variable stiffness actuators adjust the stiffness by varying the pretension of the elastic element [14], [15]. However, this requires considerably high amount of energy since the actuator which regulates the stiffness needs to work against the spring forces. In [16], [17] and [13] a design approach for adjusting the stiffness based on a variable lever arm mechanism has been introduced. In these implementations, the stiffness at the elastic transmission mostly depends on the length of the lever arm and on the actual stiffness of the springs inserted with fixed pretension in the mechanism. However, to achieve a wide stiffness variation, this concept
may require a lever arm with a reasonable length, which in turns increases the size of the overall VSA’s structure.

This paper presents a new compact variable stiffness actuator, the CompAct-VSA. The actuator uses a cam-based lever arm with variable pivot for adjusting the stiffness (Fig. 1). The mechanism employed permits to substantially increase the range of the achievable stiffness requiring a shorter lever arm and smaller springs compared to [16], [17], which effectively reduce the size of the Compact-VSA with respect to previous VSA lever arm based designs. The use of shorter lever arm has the important consequence of reducing the time needed for the stiffness regulation, since the lever travel distance to regulate the stiffness from minimum to maximum stiffness is shorter.

The paper is structured as follows. Section II describes the concept implemented for adjusting the stiffness. Section III presents the mechanics of the actuator while Section IV discusses the actuator modelling. Section V presents the evaluation analysis of the proposed actuator, while Section VI discusses the preliminary experimental results. The work concludes in Section VII.

II. FUNCTIONAL PRINCIPLE OF COMPACT-VSA

The functional principle implemented for the CompAct-VSA is based on the use of a lever arm mechanism with a variable pivot axis. A simple schematic is shown in Fig. 2. In this figure, \( F_E \) represents the force generated by the springs when the lever arm rotates around the pivot point \( P \) due to the application of the external load \( F_L \) at point \( A \). \( F_A \) is the elastic force generated at point \( A \) in the opposite direction of the load force \( F_L \) as a result of the spring force \( F_S \). The stiffness regulation is achieved by moving the position of the pivot point \( P \) of the lever arm with respect to the center of rotation of the joint (B). When the pivot moves, the amplification ratio of the lever changes. This is defined as the ratio between \( \delta_1 \) and \( \delta_2 \) considered as the distances \( (BP) \) and \( (PA) \), respectively. In detail, the amplification ratio is infinite when the pivot reaches \( A \), while it reduces to zero, when the pivot is placed in \( B \).

III. MECHANICS OF COMPACT-VSA

For the mechanical realization of CompAct-VSA unit particular attention was paid to optimize the size, weight and modularity of the mechanical assembly in order to allow the future integration of the actuation unit into multi-dof VSA robotic systems. The high density of the integration is due to the novel mechanical implementation of the variable stiffness module. To minimize dimensions while achieving wide stiffness range, high torque capacity and fast stiffness regulation a mechanical arrangement involving a cam based profile lever arm mechanism with a variable pivot point was implemented.

![Fig. 2. Schematic of the lever arm mechanism with variable pivot point.](image)

![Fig. 3. CAD assembly of the CompAct-VSA variable stiffness module showing the lever arm and pivot kinematics.](image)
The stiffness module is connected in series with the actuator for the joint positioning (\( M_1 \)). It consists of a brushless frameless DC motor [Emoteq:HT-02300] with peak torque of 2.35 Nm and a harmonic drive gearbox (CSD25) with a reduction ratio of 50:1 and a peak torque of 127Nm. A cross section of the overall Compact-VSA actuator assembly is presented in Fig. 5.

The platform is equipped with four position sensors and one torque sensor, allowing for full state feedback control of the Compact-VSA. In detail, an incremental optical encoder (MicroE Optical encoder with 12 bit of resolution) monitors the position of \( M_1 \) after the reduction drive for calibration purposes and one additional optical incremental encoder (MicroE Optical encoder with 14 bit of resolution) measures the deflection angle of the link. Finally an optical encoder (MicroE Optical encoder with 12 bit of resolution) is mounted at the motor \( M_1 \), to regulate the position of the pivot axis.

IV. MODELLING OF COMPACT-VSA

The modelling of the CompAct-VSA is based on the detailed schematic of the cam shaped lever arm mechanism used in CompAct-VSA presented in Fig. 7. When an external torque \( \tau_{\text{ext}} \) is applied at the output link, it rotates the cam with an angle \( \phi \) with respect to the center of rotation of the joint. This in turns displaces the springs with stiffness \( k_s \) generating a force \( F_s \) applied in (\( \Gamma \)) given by

\[
F_s = 2k_s \overrightarrow{PO} \sin \phi \cos \phi. \tag{1}
\]

Due to the lever mechanism, a force

\[
F'_E = F_s \frac{\overrightarrow{TP}}{\overrightarrow{AP}} \tag{2}
\]

is therefore applied in (\( A \)), where \( \frac{\overrightarrow{TP}}{\overrightarrow{AP}} \) is the amplification ratio and

\[
\overrightarrow{AP} = \sqrt{\Delta^2 + \delta_1^2 - 2\delta_1 \Delta \cos(\theta_s)}. \tag{3}
\]

with

\[
\Delta = \delta_1 + \delta_2 \tag{4}
\]

where \( \delta_1 \) is the distance from the pivot (\( P \)) to the center of rotation of the joint (\( B \)) and \( \theta_s = q - \theta_1 \) is the deflection of the elastic transmission defined as the difference between the generalized coordinates of the link \( q \) and the motor \( M_1 \) position \( \theta_1 \), respectively. By replacing (1) in (2) the force \( F'_E \) can be reformulated such as

\[
F'_E = 2k_s \alpha \delta_1 \sin(\phi) \tag{5}
\]

and

\[
\alpha = \frac{\overrightarrow{PO}}{\overrightarrow{AP}} = \frac{\Delta - \overrightarrow{AP}}{\overrightarrow{AP}}. \tag{6}
\]

Since \( \sin \phi = \frac{\overrightarrow{AP}}{\overrightarrow{AB}} \sin \theta_s \) the (5) becomes

\[
F'_E = 2k_s \alpha \delta_1 \Delta \sin(\theta_s) \frac{\overrightarrow{AP}}{\overrightarrow{AB}} \tag{7}
\]

and the projection \( F_E = \frac{F'_E}{\cos(\phi - \theta_s)} \) along the perpendicular to \( \overrightarrow{AB} \) is

\[
F_E = \frac{2k_s \alpha \delta_1 \Delta \sin(\theta_s)}{\overrightarrow{AB} \cos(\phi - \theta_s)}. \tag{8}
\]

As a consequence, the elastic torque \( \tau_E = F_E \Delta \) which, at the equilibrium, counterbalances the external torque is obtained such as

\[
\tau_E = \frac{2k_s \alpha \delta_1 \Delta^2 \sin(\theta_s)}{\overrightarrow{AB} \cos(\phi - \theta_s)}. \tag{9}
\]

Assuming the deflection of the elastic transmission \( \theta_s \) to be small, also due to the mechanical locks discussed in Section
III, the formulation of the elastic torque in (9) simplifies such as
\[ \tau_E = \frac{2k_s \delta_s^2 \Delta^2 \theta_s}{(\Delta - \delta_1)^2} \]  
(10)
and the stiffness of the elastic mechanism \( K = \frac{\partial \tau_E}{\partial \theta_s} \) is therefore formulated such as
\[ K = \frac{2k_s \delta_s^2 \Delta^2}{(\Delta - \delta_1)^2}. \]  
(11)

Given the spring rate \( k_s \), the stiffness \( K \) in (11) mainly depends on the relative position of the pivot point with respect to the center of rotation, \( \delta_1 \). This is adjusted through a rack and pinion transmission driven by the motor \( M_2 \) such that
\[ \delta_1 = n \theta_2 \]  
(12)
where \( \theta_2 \) is the angular position for the motor (\( M_2 \)) and \( n \) is the transmission ratio between the rack and the pinion. Based on (10), it is also possible to formulate the energy stored in the elastic mechanism such as
\[ U_E = \frac{k_s \delta_s^2 \Delta^2 \theta_s^2}{(\Delta - \delta_1)^2}. \]  
(13)

Finally, the dynamics of the CompAct-VSA, neglecting the gravity contribution, are described as
\[ I \ddot{\theta} + N \dot{\theta} + \tau_E = \tau_{ext} \]
\[ B_1 \theta_1 + \phi_1 \theta_1 - \tau_E = u_1 \]
\[ B_2 \dot{\theta}_2 + \phi_2 \theta_2 + \tau_R = u_2 \]  
(14)
with \( \phi_i = \nu_i^2 D_i + \nu_i^2 K_{el} K_{el} / R_{mi} \), \( u_i = \nu_i K_{el} V_i / R_{mi} \) and \( V_i \) being the input voltage with \( i \in [1, 2] \). The resistant torque \( \tau_R = \frac{2u_i \nu_i^2}{\theta_2} \) applied at the motor \( M_2 \) due to the elastic coupling is given by
\[ \tau_R = \frac{2k_s \nu_i^2 \theta_2 \Delta^2 \dot{\theta}_2^2}{(\Delta - n \theta_2)^3}. \]  
(15)

The nominal values in (14) are presented in Table I. Note that, to simplify the notation the motor inertias (\( B_i \)) are already scaled by the transmission ratios.

V. PERFORMANCE ANALYSIS OF COMPACT-VSA

Analysis of the CompAct-VSA performance was conducted through simulation on the basis of the model presented in Section IV. Initially, the behavior of the maximum deflection \( \theta_s \) achievable by the elastic transmission has been analyzed. As mentioned in Section III, the maximum deflection of the elastic transmission \( \theta_s \) is constrained to be \( \pm 0.35 \) [rad] by a pair of mechanical locks. However this value changes on the basis of the pivot position \( \delta_1 \) due to the maximum displacement which can be achieved by the linear springs with respect to their initial level of pre-compression. This is clarified in Fig. 8, here it is possible to note that the maximum deflection \( \theta_s \) substantially decreases for increasing values of the pivot position.

Following this, the maximum value of the elastic energy \( U_E \), defined as in (13), that can be theoretically stored in the elastic mechanism is obtained as a function of the pivot position \( \delta_1 \) and the deflection \( \theta_s \) such as \( U_E = 0.35 \) J. The behavior of the elastic torque \( \tau_E \), formulated as in (10), is further investigated through simulation and also for this case the variation with respect to \( \delta_1 \) and \( \theta_s \) is analyzed. Figure 9(a) shows the ability of this VSA to achieve substantially high elastic torque values when the pivot position \( \delta_1 \) travels up to the maximum distance with respect to the center of rotation of the joint. The resistant torque as in (15) applied to the motor \( M_2 \) for different pivot positions and elastic deflections is shown in Fig. 9(b). Finally, the behavior of
the stiffness $K$, which is calculated as in (11), is analyzed with respect to the variations of the pivot position, $\delta_1$. In Fig. 9(c) it is possible to notice that for a spring rate $k_s = 10000$ [N/m] and a lever arm with total length of $\Delta = 0.015$ [m] the stiffness value can change substantially for extremely small variations of the pivot position, when it reaches the end of the travel distance. To summarize the conducted analysis, Table II presents the main specifications of the CompAct-VSA.

VI. PRELIMINARY EXPERIMENTAL TRIALS WITH THE COMPACT-VSA

Experiments were conducted with the CompAct-VSA prototype shown in Fig. 12 in order to evaluate the performance of the actuator. For this purpose, proportional derivative controllers (PD) were implemented for both motors $M_1$ ($K_P = 1000$ [V/rad], $K_D = 10$ [Vs/rad]) and $M_2$ ($K_P = 200$ [V/rad], $K_D = 3$ [Vs/rad]) on a custom DSP-board. The trials were carried out to validate the range stiffness and the ability of the actuator to rapidly regulate the stiffness level. For the implementation of the first prototype two linear springs made of 0.0012 [m] C85 Carbon Steel wire were used. The springs are characterized by a stiffness of 10 [kN/m], free length of 0.03 [m] and a maximum deflection of 0.012 [m]. Initially, sinusoidal waves at the frequency of 1 Hz were sent as reference signals to the motor $M_2$ to regulate the pivot position reference on the basis of the (12). Figure 10(a) presents simulation and experimental data of the motor $M_2$, coupled through the rack pinion transmission while regulating the pivot position on the basis of the sinusoidal reference signal. The same performance can be observed in the stiffness tracking since, given the formulation in (11), it depends on the tracking performance of the pivot point (see Fig. 10(b)). The deviation observed between the simulated and experimental trends are due to uncertainties in the parameters of the system model as well as to the un-modelled dynamics related to friction (e.g the friction between the sliding surfaces of the rack and the guiding groove has not been considered).

Finally, the step response of the stiffness regulation was assessed by feeding the motor $M_2$ with step references of different amplitude. The response was recorded to show the ability of the stiffness module to track rapid stiffness changes. The performance of the system for a low stiffness steps is shown in Fig. 11(a) and 11(c), and for larger steps in Fig. 11(b) and 11(d). In both cases the results demonstrate the capability of the stiffness module to respond fast and therefore successfully follow abrupt stiffness changes.

VII. CONCLUSION

In this paper the design, implementation and modelling of a new compact variable stiffness actuator (CompAct-VSA) were presented. The unit is intended to drive multi degrees of freedom VSA robotic systems. The modularity of the high performance variable stiffness unit was achieved by realizing the variable stiffness module with the use of a cam shaped lever arm mechanism with a variable pivot point. The lever arm with the variable pivot axis allows the overall assembly size to be reduced without affecting the output stiffness range. Due to the moving pivot principle the stiffness amplification ratio can theoretically change from zero to infinite and therefore the level of stiffness can be set from very soft (almost 0) to rigid. The model of the actuator was presented accompanied by performance measures indicating the capabilities of the proposed implementation. Experimental trials were performed which confirmed the fast
stiffness response of the actuator and the wide range of achievable stiffness.

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REFERENCES


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