Variable Physical Damping Actuators (VPDAs): Facilitating the Control and Improving the Performance of Compliant Actuation Systems

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Abstract — As recent application domains emerge requiring the employment of robots within unstructured environments, new demands arise requiring more versatile systems which can cope with unpredictable interactions. Recently, compliance has been identified as one of the key features which enables the safe operation of robots interacting with humans and environment. Despite the certain merits gained with the introduction of compliance, this property introduces also some drawbacks as the reduction of the bandwidth achievable in the controlled system and the introduction of oscillatory dynamics which dramatically reduce the stability and accuracy of the system. A solution which can be used to overcome such issues consists in the incorporation of physical damping within the actuator mechatronics. Motivated by the above, this work presents the evolution of such actuators from purely compliant to more complex and performing systems which incorporate variable physical damping as added feature for the development of robust, safe and still well performing robots. The mechatronics of the developed units are analysed. The compact compliant actuator (CompAct™) with variable physical damping is evaluated with experimental trials performed using a prototype unit.

Keywords – Robotics; compliance; variable impedance actuation; variable physical damping; semiactive dampers.

I. INTRODUCTION

Conventional industrial robots make use of stiff1 and non-backdrivable actuation systems controlled by high gain PID controllers to achieve the highest accuracy, speed and repeatability. These machines typically operate within well structured environment to execute repetitive tasks and any interaction with external agents is avoided by means of physical or sensory barriers given their limited ability of interaction, which is due to the high exhibited mechanical impedance. In fact, these robots are unsuited to operate in unstructured environments such as homes and offices as unexpected interactions may result dangerous for the human/environment and for the robot itself [1-2]. Hence, recently emerged application domains which require the direct contact of robots with external agents (such as e.g. service/domestic robotics) set new demands particularly focusing on the mechatronic design of the actuation systems. Taking inspiration from nature, the mentioned safety and robustness issues are solved in mammalians by the intrinsic joint compliance replicated and modulated by muscles and tendons, [3-5]. As a result, several compliant actuators prototypes have been developed to replicate the same physical properties in robots and achieve the mentioned benefits, [6-8].

Although compliance is essential for enabling robots to operate in such unstructured environments, [9], there are also some negative effect as the introduction of typically underdamped dynamics which in fact introduce a limiting factor to the performance achievable in the controlled system at the same time inducing oscillations which dramatically reduce the accuracy of the robot and may ultimately lead to instability. Humans, for instance, would not be able to position a limb rapidly and damp the oscillations achieving high accuracy when an impulsive force is exerted on the limb without joint physical damping [10]. Therefore, from the engineering perspective the incorporation of physical damping within the mechatronics of the compliant joint of the actuation system can in fact constitute a solution for the issues mentioned above, assisting the control of compliant robots and enhancing their accuracy.

This paper presents the mechatronic development of such compliant actuation systems and their evolution towards more complex actuators which additionally incorporate variable physical damping actuators (VPDAs), [11], with the main objective of overcome the drawbacks introduced by compliance to realise safe, robust and still well performing robots and facilitate their control. The paper is structured as follows: Section II introduces the development of a series elastic actuator (SEA), i.e. the CompAct SEA and analyses the main issues in the control of these types of actuators focusing on the improvements which can be gained with the introduction of joint physical damping. Section III presents an overview of possible implementations of damping systems with Sections IV and V focusing on the mechatronic embodiment of variable physical damping in compliant

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1 With the term “stiff” we intend actuators which do not incorporate additional passive compliance. The resulting output stiffness of these systems is typically determined by the structural properties of the transmission system (e.g. gearbox).

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robotic joints, analysing the development, modelling, control and presenting some experimental results obtained with the developed prototype units. Finally, Section VI reports the conclusions and future work.

II. THE COMPACT™ SEA

A. Development

The development of this series elastic actuator was motivated by the need of a compact compliant actuation module with high integration density suitable for integration in multi degree of freedom robots to enable safe and robust interaction of the resulting machine with either human or environment. Its modular structure facilitates the design of full robots while the small dimensions and low mass make it suitable for the development of small scale systems as that presented in [9, 12], Fig. 1c. The high density of integration is due to the series elastic element which is assembled in series to the output of the brushless DC motor – Harmonic Drive gearbox group, Fig. 1. A mechanical assembly involving an input pulley, a three spokes output pulley and six linear compression springs has been developed to guarantee sufficiently high torsion stiffness, guaranteeing at the same time large passive deflection ranges\(^2\). The input pulley is assembled to the output of the reduction drive.

\[ D = 0.25 \text{ Nms/rad} \] (corresponding to a damping ratio \( \zeta = 0.027 \)) due to the use of ball bearings and this is positive from the perspective of robustness to interaction as this maximises the decoupling action between the drive and the link, which is the part usually interacting with external agents, enhancing the versatility to cope with unpredictable disturbance ranging from small uneven terrain variations to unexpected collisions or even accidental falls as shown in [9, 12-13]. On the other hand, such a low damping value makes the control of the actuation system difficult due to the underdamped complex conjugate poles introduced by the elastic joint which create motor side anti-resonance and corresponding link side resonance which are associated to sharp phase variations for both the motor and the link – controlled systems, Fig. 2. This makes the control of the system difficult for frequencies in proximity and above the mechanical resonance especially when considering multi-DOF systems which present compliance at many DOFs, such as the system in Fig. 1c. In fact, compliance acts as a limiting factor which imposes the placement of the bandwidth of the controlled system at frequencies which are well below the mechanical resonance, [14].

![Fig. 1 The prototype of the CompAct™ SEA module, [7]. (a) CAD cross section view, (b) full actuation unit, (c) application to the design of the compliant humanoid cCub, [12].](image)

The bode plot of Fig. 2 depicts the identified voltage to link velocity transfer function of the actuator of Fig. 1a, b, for different joint physical damping levels (the curve with damping ratio \( \zeta = 0.027 \) represents the actual identified model of the CompAct™ SEA) and shows that the magnitude peak at resonance decreases for increasing joint physical damping. This result is quite obvious as this means that the oscillations are reduced when the system is damped and this intrinsically improves the accuracy of the system. In addition, the motor side antiresonance effect, [14], is also attenuated meaning that the actuator is more controllable at this operating frequency. A further effect is that magnitude increases at frequencies above the resonance which means that dynamic performance is improved for systems featuring higher damping values. This can be explained by the fact that for a certain motor with maximum recommended voltage operating at a given frequency above resonance the maximum achievable output

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\( ^2 \) The replicated link impedance can be regulated by means of a standard admittance controller to meet the demands required by the task, [7].
link speed increases. Similarly, assuming that it is required to replicate certain link velocity amplitude the maximum operating frequency increases (this effect can be identified by tracking a constant line in the plot of Fig. 2), reconfirming that joint physical damping enhances the dynamic performance of the actuator. Finally, analysing the phase bode plot of Fig. 2, it can be noticed that an increased level of joint physical damping introduces a substantial phase lead effect which facilitates the control and stabilization of the controlled plant.

The positive effects mentioned above are mainly due to the increased coupling effect between the drive and the outer link. This is beneficial from the stability, dynamic performance and accuracy perspectives however, in contrast to this, it increases the interaction force-torque exchange hence deteriorating the safety, robustness and interaction-related performances. For this reason the optimal solution would consist in a dissipative device which can adjust the joint damping level to adapt the system to the specific task/state to achieve the best interaction-performance tradeoff, similarly as in the fast and soft paradigm which was designed for Variable Stiffness Actuators (VSAs), [15].

III. DAMPING SYSTEMS

Many dissipation devices (dampers) were designed and fabricated over the years as e.g. friction dampers, viscous or viscoelastic dampers and metallic yield dampers and have been applied to different fields of applications ranging from the control of automotive suspension systems, structural response or to transient environmental disturbances in civil structures.

Dampers can be defined as passive, semi-active or active based on the amount of external power required by the damping control system to operate. A passive dissipation system is usually made of an elastic component (which sets the stiffness) and a damper to either load the transmission path of the disturbing vibration or damp the oscillation energy. In these systems the damping level is fixed and tuned to optimally perform within a certain range of frequency. However, as expected this class of systems shows substantial limitations in applications which require operating frequencies outside the range for which the damper is tuned. This motivated the design of active vibration control systems which can regulate the level of damping online. In these systems, an additional force is employed and becomes part of the dissipation system. This additional active force can then be used to control the dissipation system making it more sensitive to the specific nature of disturbances. By using this external actuator active dissipation systems can be controlled to outperform any passive implementation. Active damping systems, [16], can be used for suppressing the oscillations but do not introduce the benefits of the physical property explained above (phase lead and possibility of improving the closed loop bandwidth [17-18]). In addition, usually, active control systems require a substantial amount of energy to produce the required dissipation forces and are ineffective for frequencies above the closed loop bandwidth of the system [17-18].

Differently from active systems which can regulate any component of the mechanical impedance (stiffness, damping or inertia) semi-active devices can control only the amount of physical damping. Differently from active solutions, these devices do not inject mechanical energy into the system and therefore they do not have the potential to destabilize it.

Semi-active systems, [19-20], can be controlled to perform significantly better than passive devices still exhibiting the potential to achieve the majority of the performance of the fully active configuration (e.g. ability to set the damping level over a wide range) [21].

IV. THE VARIABLE PHYSICAL DAMPING ACTUATOR (VPDA)

Basing on the considerations made in the previous sections, a semi-active friction damper to be incorporated into robotics compliant actuators and a related control strategy to enable the regulation of the joint damping level have been designed. It is important to notice that the use of semi-active damping systems placed between the actuator and the load has also been used in robotics in [19-20]. Nevertheless, these works do not make use of mechanical compliance. In fact dampers in these works are employed as clutches and they are hence used to deliver the actuator effort to the load. This means that these systems typically suffer of low energy efficiency when compared to SEAs/VSAs or hybrid compliant-damping solutions like those presented in this paper [11, 17-18].

A. VPDA Mechatronics

Figure 3 shows a conceptual schematic and the design of the VPDA experimental prototype.

![Figure 3](image)

It consists of the variable damping module and a passive torsion spring element coupled in parallel to the damper in order to induce suitable oscillations as required for the evaluation of the damper performance. A link is fixed at the output of the damper/spring network allowing the installation of different masses which will permit the control of the oscillation frequency. Three piezo actuators are used to
generate pressure between the contact surfaces and generate braking torque in parallel to the torsion spring which emulates the flexibility of a possible compliant joint. The part placed in series between the actuators and the ring-shaped loadcell acts as an isolator preventing the generation of lateral forces on the upper tip of the piezos. This behaviour is realised by means of a prismatic guide which makes this part able to slide along the joint axis at the same time constraining its rotation. Another prismatic guide has been designed between the housing of the torsion spring and the central key which is connected to the upper contact surface to permit the precise adjustment of the distance of the latter from the other friction disk by means of a fine pitch screw. This custom part is used to tune the clearance in order to generate contact between the frictional surfaces when the piezo actuators are charged Fig. 3a. This screw is left-hand threaded on the central key side, while a right hand thread is machined on the link side.

Regarding the available sensing, this prototype is able to measure the joint torque by means of a custom-made torque sensor while the force generated by the piezoelectric actuators is monitored by a ring-shaped loadcell. Finally, a 12 bit magnetic encoder measures the link position, Fig. 3a.

B. **Experimental Results**

Experiments were carried out with the purpose of validating the performance of the proposed semi-active damping device. The spring was loaded by moving the link to the end of the available range (-0.35 rad) and then suddenly released to induce oscillations to be damped by the VPDA system. The controller presented in [11] was used for this experiment. Figure 4 shows the VPDA mechatronics is able of replicating different levels of desired damping ratio $\zeta_{des}$.

![Fig. 4 - Experimental evaluation of the VPDA prototype](image)

**V. THE COMPACT™ SEA WITH VARIABLE PHYSICAL DAMPING**

The benefits introduced by compliance in terms of interaction-related performances and the ability of physical damping in coping with the drawbacks introduced by compliance motivated the development of an actuator which exhibits both features by means of the incorporation of the analysed semi-active damping device within the actuator presented in Section II.

A. **Mechatronics**

The mechanical design of this actuator is shown in Fig. 5. The motor-gearbox group and the series compliant element follow the same principle as in the CompAct™ SEA presented in Sec. II. The piezo actuators of the VPDA system have been integrated by incorporating them around the motor-gearbox group, Fig. 5. Thanks to the narrow section of these actuators, the diameter of the full actuator is slightly increased with respect to the pure series elastic actuator version of the actuator, whereas the overall length is not affected. A major improvement offered by the VPDA system incorporated in this actuator with respect to the first prototype presented in the previous section consists in the use of a friction interface based on Kevlar fibre – based material on the contact surface assembled to the link and steel for the contact surface actuated by the piezo stacks. This provides lower stiction (this phenomenon is unwanted as it introduces discontinuities), higher dynamic coefficient of friction and much higher wear and heat resistance with respect to the metallic surfaces used in the damping system shown in Fig. 3. Similarly as in the prototype presented in the previous section, the sliding contact surface, Fig. 5, is rotationally constrained with the compliant joint input pulley but it can slide along the main actuator axis. The available sensing include: joint torque and piezo actuators force thanks to custom-made strain gauge sensors, motor position before and after the gear which are measured by 12-bit optical and magnetic encoders, respectively, and the deflection of the series compliant element by means of a 12-bit magnetic encoder.

![Fig. 5 - The CompAct SEA with variable physical damping, [17]. (a) CAD cross section view, (b) full experimental prototype](image)

B. **Model**

A conceptual schematic showing the mechatronic model of the actuator is shown in Fig. 6. The actuator $A$ regulates the joint braking torque $\tau_j$ by means of the generated normal force $F_a$. The braking torque can be expressed as follows:

$$
\begin{align*}
\tau_j &= \mu_d \cdot G(F_a), \dot{\theta} \neq 0 \\
\tau_f &= \mu_s \cdot G(F_a), \dot{\theta} = 0
\end{align*}
$$

where $\mu_d = 0.4$ and $\mu_s$ are the equivalent dynamic and static friction coefficients of the employed friction interface while $G(F_a)$ takes into account the force generated by the actuator $A$ and the geometry of the contact surfaces.
Referring to Fig. 6, \(\theta\) and \(\theta\), and are the reduced and actual motor angular positions, \(N\) is the gear ratio, \(q\) is the angle of the link, \(J_i\) and \(J_j\) are the moments of inertia of the outer link and of the rotor, whereas \(D_n\), \(D_s\), and \(D_r\) are the viscous damping coefficients at the rotor, in parallel to the compliant module of stiffness \(K_t\) and at the outer link, respectively. The torque generated by the actuator is defined as \(\tau_i\), while the motor torque reflected to the joint side is defined as \(\tau_j\). \(\tau_j\) is the load torque whereas \(\tau_i\) is the braking torque generated by the VPDA system due to the application of the piezo actuators force \(F_a\). The following equations describe the dynamics of the model shown in Fig. 6:

\[
\begin{align*}
J_i \cdot \dot{N}^2 \cdot \dot{\theta} + D_n \cdot \dot{N} \cdot \dot{\theta} - D_s \cdot \dot{N} \cdot (q - \dot{\theta}) - K_t \cdot (q - \theta) - \tau_j \cdot \text{sgn}(q - \dot{\theta}) &= \tau_i \\
J_i \cdot \dot{q} + D_s \cdot 
\end{align*}
\]

(2)

The braking torque \(\tau_j\) can be expressed as a function of the normal force generated by the piezos \(F_a\), the parameters of the ring plate geometry \((D, d)\) and the dynamic friction coefficient \(\mu_d\). Uniform pressure is assumed over the surface of the ring disk.

\[
\tau_j = \mu_d \cdot F_a \cdot D^3 - d^3 \cdot 3 \\
(3)
\]

The discontinuous effect of static friction can be neglected due to the friction property of the friction interface and this permits neglecting the modelling of this effect in (3).

From (3), the linear relationship between torque and force can therefore be expressed with a coefficient \(\mu_{FP}\):

\[
\mu_{FP} = \frac{\tau_j}{F_a} = \frac{\mu_d \cdot D^3 - d^3}{3 \\
(4)
\]

Equation (4) shows that the maximum friction coefficient is achieved by maximizing both the inner and outer ring diameter. On the other hand, the mechanical constraints put an upper bound to the highest feasible diameters; hence the outer diameter was set equal to the diameter of the actuator (93mm), while \(d\) was set to 66mm which was the maximum permitted by the mechanical implementation, resulting in a torque-force coefficient \(\mu_{FP} = 15 \cdot 10^3 \text{ m}\).

C. VPDA Controller

Considering the actuator shown in Fig 1 exhibiting a joint damping value \(D_s\) and identical inertial and stiffness properties results in the following dynamic equations:

\[
\begin{align*}
J_i \cdot \dot{N}^2 \cdot \dot{\theta} + D_r \cdot \dot{N} \cdot \dot{\theta} - D_s \cdot \dot{N} \cdot (q - \dot{\theta}) - K_t \cdot (q - \theta) &= \tau_j \\
J_i \cdot \dot{q} + D_s \cdot (q - \dot{\theta}) + K_t \cdot (q - \theta) &= \tau_i
\end{align*}
\]

(5)

Equalizing the euler equation at the output link of (2) with that of (5), results in the following braking torque \(\tau_j\):

\[
\tau_j \cdot \text{sgn}(q - \dot{\theta}) = (D_j - D_s) \cdot (q - \dot{\theta}) - D_r \cdot \dot{q}
\]

(6)

The same equation can be expressed in terms of desired damping ratio \(\zeta_d\): 2\(D_j \sqrt{K_t/J_i} \leq \zeta_d\)

\[
\tau_j \cdot \text{sgn}(q - \dot{\theta}) = \left(2 \cdot \zeta_d \cdot \sqrt{K_t/J_i} - D_s\right) \cdot (q - \dot{\theta}) - D_r \cdot \dot{q}
\]

(7)

which can be used to replicate the same damping related dynamics in view of possible changes of joint stiffness and load/inertia. By combining (4) and (7), the force to be applied by the piezoelectric actuators \(F_a\) can be obtained:

\[
F_a = \frac{2 \cdot \zeta_d \cdot \sqrt{K_t/J_i} \cdot (q - \theta) - D_s}{\mu_{FP}} \cdot \dot{q} \cdot \text{sgn}(q - \dot{\theta})
\]

(8)

\(F_a\) is affected by the errors in the estimation of the parameters of the system and the deflection velocity state. An outer damping ratio loop has been implemented to compensate for this, which requires the measurement of the damping ratio.

The joint torque measured by the torque sensor, Fig. 5a, is:

\[
\tau_{\text{meas}} = D_r \cdot (q - \dot{\theta}) + K_t \cdot (q - \theta)
\]

(9)

By manipulating (9), the joint physical viscous damping can be obtained:

\[
\begin{align*}
D_s_{\text{meas}} &= \frac{\tau_{\text{meas}} - K_t \cdot (q - \theta)}{(q - \dot{\theta})}, \quad (q - \dot{\theta}) \neq 0 \\
D_r_{\text{meas}} &= D_r, \quad (q - \dot{\theta}) = 0
\end{align*}
\]

Equation (10) requires the measurement returned by the spring deflection encoder and its derivative. When the deflection velocity \((q - \dot{\theta})\) is zero, the damping measurement would return infinite. In order to avoid undesirable behaviour of the system, \(D_s_{\text{meas}}, \zeta_{\text{meas}}\) are set equal to \(D_s, \zeta_s\), respectively, when this condition is verified. The corresponding measured damping ratio is:

\[
\begin{align*}
\zeta_{\text{meas}} &= \frac{\tau_{\text{meas}} - K_t \cdot (q - \theta)}{2 \cdot (q - \dot{\theta}) \cdot \sqrt{K_t/J_i}}, \quad (q - \dot{\theta}) \neq 0 \\
\zeta_{\text{meas}} &= \zeta_s, \quad (q - \dot{\theta}) = 0
\end{align*}
\]

The overall schematic of the VPDA control module is shown in Fig. 7.

D. Experimental Results

Experiments were carried out to validate the capability of the presented mechatronic system in replicating desired values of joint viscous damping. The motor was servo-controlled to
maintain a fixed position and the springs were temporarily removed from the actuator while the link was externally perturbed to make the VPDA system replicate the desired viscous damping coefficient $D_d$. The corresponding joint torque was recorded together with the relative speed between the friction surfaces. Figure 8 shows that the system is capable of replicating different desired viscous damping curves. The properties of the friction interface allows for a stiction-free effect which can be identified in Fig. 8 by the absence of the typical vertical line for relative velocities equal to zero which characterises static friction.

Fig. 8 – Viscous damping regulation of the CompAct SEA with variable physical damping

VI. CONCLUSIONS AND FUTURE WORK

This work presented the analysis, development and control of compliant systems with variable physical damping. Although compliance is a fundamental property which should be incorporated in robots that need to operate in unstructured environments, the introduced joint flexibility introduces some drawbacks as a reduction of the stability margin, control accuracy (due to the induced oscillations) and dynamic performance which can be achievable in the closed loop system [11, 17-18]. The first part of the paper presented the development of a series elastic actuator which has been used for the evaluation of the mentioned effects introduced by compliance. From the analysis it was concluded that a possible solution to the mentioned issues consists in the introduction of variable physical damping in the mechatronics of the transmission. The second part of the paper therefore analysed the incorporation of devices which can adjust this physical parameter in compliant joints and the design, modelling and control of a SEA with variable physical damping. Experimental results validated the mechatronics of the presented systems.

Further work will involve the employment of the CompAct™ SEA with variable physical damping for the development of multi degree of freedom systems.

REFERENCES


