CONTROLLABLE COMPLIANCE JOINT FOR HUMAN ORIENTED ROBOTS

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ABSTRACT. In the paper, different approaches for compliance control for human oriented robots are revealed. The approaches based on the non-antagonistic and antagonistic actuation are compared. In addition, an approach is investigated in this work for the compliance and the position control in the joint by means of antagonistic actuation. It is based on the capability of the joint with torsion leaf springs to adjust its stiffness. Models of joint stiffness are presented in this paper with antagonistic and non-antagonistic influence of the spring forces on the joint motion. The stiffness and the position control possibilities are investigated and the opportunity for their decoupling as well. Some results of numerical experiments are presented in the paper too.

KEY WORDS: Human oriented robots, antagonistic, redundant, actuator, stiffness, compliance.

1. Introduction

Robots are going to fulfil not only industrial tasks in the future, but also jobs related with the mutual interaction between robot and human being. Such problems arise in the rehabilitation process, in the prosthesis, surgery, also in the entertainment and the show industry as well as in the office and the domestic environment. Thus, the next robot generation must be designed with the possibility to fulfil contact problems with the requirement for preserving the human security. The basic problem is the compliant motion realization for the creation of such kind of human friendly robots.

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The human ability to vary the joint stiffness is the main reason for the successful interaction with the surrounding environment and for the interaction security as well. The first known solution for the compliance and the stiffness control is the impedance control scheme [1] in the industrial robots. It significantly simplifies the manipulation tasks related with the kinematical restrictions. The relation “force – displacement of the end effector” is controlled in this scheme, with the use of controllers for position control, introducing an external feedback chain by a force sensor. This is an approach of active compliance, which guarantees a wide range of stiffness control. It does not ensure high level of safety due to a low resolution or noise of the sensors, long calculation time and the servo system instability.

The passive compliance implementation is reasonable for the increase of safety in the human oriented robots, which is independent on the servo responses. So, in the human oriented robots the so called “serial elastic actuation” is used [2]. The high impedance values are limited to the values of the stiffness in the elastic link by means of an elastic element implementation with a constant stiffness between the driver output and the driven limb. This actuation is a combination of an active and a passive approach for impedance control, because except of a passive compliant element, a feedback for regulation of the output torque in the drive and the spring system is used.

The question for compliance adjustment towards the surrounding environment arises [3] with the passive compliance implementation. This is the reason for the use of auxiliary actuators. Let us note that the robots with adjustable compliance are redundant ones.

On the other hand, the approaches of redundant actuation for compliance control can be divided into two basic types - antagonistic and non-antagonistic, accordingly whether additional actuators can work in opposition against each other or not. At the antagonistic approach, the robot possesses parallel structure, while at the non-antagonistic one, the implemented structure is usually a serial one. Below, well known approaches for the non-antagonistic passive compliance control are shown.

One type for passive compliance realization is the structure controlled stiffness [4]. Here, the stiffness control can be achieved through manipulation of the effective structure of a spring. In each joint, except an actuator for position control, an additional actuator for passive stiffness control is added.

Solutions of an actuator, based on the structure controlled stiffness concept are the following. “Jack Spring Actuator” [5] where a helical spring is used as a compliant element. The active coil length can be changed by inserting an axis into the spring, with a surface that can slide between the rings of
the spring. In the solution presented in [6], the spring structure consists of many layered sheets. Thus, the idea is to increase the stiffness of this element by increasing friction, pressing the sheets together. A similar solution named “Mechanical Impedance Adjuster” is also known [7]. In the joint a compliant element – a bending leaf spring is mounted. The joint stiffness is controlled by a slider mechanism displacement. The so called “Mechanically Adjustable Compliance and Controllable Equilibrium Position Actuator” or MACCEPA [8] can be related to the non-antagonistic approaches. The control of the compliance is based on the control of the pre-tension of a spring in a lever mechanism.

All the approaches for passive compliance based on the structure controlled stiffness are characterized by a wide adaptable range of the compliance, including all possible intermediate steps between a compliant and a very stiff one. Both compliance and equilibrium position are independently controllable.

The antagonistic approach [9] for the passive compliance control uses antagonistic actuation of two compliant elements, such as springs, in order to vary physically the effective stiffness of the joints. This approach is close to the human muscular system, where the co-activation of the antagonistic muscles defines an equilibrium condition of the joint. Limb deviation from this equilibrium mode results in the generation of a restoration torque, which is a function of the muscle mechanical properties and is independent on any other feedback. This approach creates a natural security and is appropriate in the interaction with human.

In this approach, to obtain independent stiffness and position control, a non-linear springs are often used (force is proportional to the power of two of the deflection) [10]. The joint stiffness is a function of the preliminary pressure of the two actuators, by means of which it is controlled. Problems arise when designing a convenient quadratic spring. This is the reason many solutions to be evoked [11], [12] and [13], in which to design the non-linear springs, the linear characteristic of the spring is transferred into a quadratic one, by using special mechanisms, with a great number of pulleys, cables or shaped pieces.

The antagonistic scheme of a passive compliance control can be related to the use of different kinds of artificial muscles, like shape memory alloys, electro rheological fluids [14], electro-strictive and magneto-strictive materials and electro active polymers. The usage of these new materials as compliant actuators is not very commonly spread, because their velocity is very low and the displacements are very small.

Different types of pneumatic artificial muscles are most widely distributed [15], [16]. They are actuators with a high power to weight ratio and can be directly coupled to the joint, without a heavy and complex gearing mechanism.
The force-deflection characteristic of pneumatic muscles is strongly non-linear. Thus, they can be used to achieve compliance control by means of an antagonistic setup. The drawbacks of the pneumatic muscles are the presence of hysteresis, making difficult precise position control and the necessity of pressurized air.

All approaches of the antagonistic passive compliance control possess one basic disadvantage – very difficult realization of the actuators with ideal non-linear characteristics for achieving a decoupled stiffness and position joint control. The possibility for combination of the structurally controlled stiffness scheme with the antagonistic scheme is studied in this paper in order to improve the cited above disadvantages.

Objective of this work is to present a solution of a structurally controlled compliance joint and to examine this solution in antagonistic scheme for stiffness control. Objective is to bring out the main characteristics of the joint, to make numerical experiments and evaluations.

2. Controlled compliance joint with torsion leaf spring

An approach for passive compliance with structure controlled stiffness based on torsion loaded leaf spring is developed. A solution of controlled compliance joint [17] based on this approach is presented in Fig. 1. Compliant joint consists of leaf spring L1 with thin rectangular cross-section, which can be loaded on its longitudinal axis with torsion. The compliance is controlled by altering the length of the spring by moving a slider S1 through motor d1b and screw-shaft. Position of the executive unit (arm) “A” is controlled using motor-gear d1a-R1. Motor-gear is bonded to other shaft, bonded to one end of the spring. Shaft is restricted by a rotation bearing joint to the base. This compliant joint is a system with two degrees of freedom, which is driven by two motors, one for stiffness and the other one for positioning.

Deformations of the spring are expressed by the angle of torsion $\varphi$ [18], measured by rotation of the motor d1a over the zero-rotation section of the spring at the slider, given by expression:

\[
\varphi = \frac{M.L}{G.J},
\]

where:
- $M$ is spring torque;
- $G$ – modulus of angular deformation (for steel $G = 7.60E+10$ N/m$^2$);
- $J$ – inertia characteristics of the intersection;
Fig. 1. A solution of a compliance controlled joint.

$L$ – active length of the spring, between the slider, and the shaft linked to the motor.

The stiffness is expressed by the torque $M$ over the angular deflection of the spring (1):

(2) \[ K = \frac{M}{\varphi} = \frac{G.J}{L}, \]

and the spring compliance is expressed by the inverse of (2) or:

(3) \[ B = \frac{L}{G.J}. \]

Stiffness control by structural approach is performed through control of the geometric parameter – length of the spring – $L$. Influence of $L$ is presented [19] by numerical experiments for 20 positions of $L$ within $[0.0045 – 0.09]$ [m], with thickness $b = 0.001$ [m] and width $h = 0.018$ [m]. When the spring is rectangular and $h/b > 4$ then

(4) \[ J = \frac{1}{3}(h.b^3 - 0.63.b^4). \]

The results of calculations as a function of L according to (2), (3) and (4) are shown in Table 1, and a 3D graph of the results is shown in Fig. 2.

Joint with a leaf spring must be compliant but strong enough to carry the entire load. The simultaneous satisfaction of these requirements makes the leaf spring design difficult.
Table 1. Stiffness $K$ and compliance $B$ over different lengths $L$ [m]

<table>
<thead>
<tr>
<th>No</th>
<th>1</th>
<th>3</th>
<th>5</th>
<th>7</th>
<th>9</th>
<th>11</th>
<th>13</th>
<th>15</th>
<th>17</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L$ [m]</td>
<td>0.0045</td>
<td>0.0135</td>
<td>0.0225</td>
<td>0.0315</td>
<td>0.0405</td>
<td>0.0495</td>
<td>0.0585</td>
<td>0.0675</td>
<td>0.0765</td>
<td>0.09</td>
</tr>
<tr>
<td>$B$ [rad/Nm]</td>
<td>0.0102</td>
<td>0.0307</td>
<td>0.0511</td>
<td>0.0716</td>
<td>0.092</td>
<td>0.1125</td>
<td>0.1329</td>
<td>0.1534</td>
<td>0.1738</td>
<td>0.2045</td>
</tr>
</tbody>
</table>

Fig. 2. Spring stiffness and compliance as a function of $L$

The admissible torsion tensions $\tau^a$ determine the maximum load of the spring $M^a$:

\begin{equation}
M^a = W \cdot \tau^a,
\end{equation}

where $W$ is moment of resistance of the spring. When ratio width/thickness $(h/b) > 4$, it is:

\begin{equation}
W = \frac{1}{3} (h \cdot b^2 - 0.63 \cdot b^3).
\end{equation}

Assuming $\tau^a$ to be half of the admissible bending stress [18], for stainless steel X3CrNiMo13-4 – $\tau^a = 49.10^7$ [N/m²]. The admissible torque (5) for a spring with $b = 0.001$ [m], $h = 0.018$ [m] according (5) and (6) is:

\begin{equation}
M^a = 2.84 \text{ Nm}.
\end{equation}

The admissible torque $M^a$ according to (1) determines the admissible deflection angle of the spring. The chosen compliance control approach allows a wide range of compliance control mainly determined by the geometry. This range
varies from semi-hard to very hard, theoretically to infinity hard at zero length of the spring. On the other hand, the position motor allows unlimited control in the position of the joint, independent of the stiffness.

3. Antagonistic adaptable compliance joint with torsion leaf springs

Another approach for passive compliance control based on torsion loaded leaf spring is developed. The possibility for combination between the structure controlled compliance with the antagonistic compliance is studied.

In this approach, two leaf spring compliant joints are used, set parallel to the arm. Compliance joints type is as shown in Fig. 1, but without arm-positioning motors, as shown in the simulation in Fig. 3. The stiffness of the two springs determines the resulting stiffness of the joint. They are defined by the positions of the two sliders S1, S2 by the two motors d1, d2. The resulting position of the arm is the equilibrium position determined by the tensions of the two antagonistic springs. Thus, by two motors, like in the above case, both the compliance of the joint and its position are controlled. The parallel placement of the two springs increases the safety of the joint.

Equilibrium position is a function of the springs stiffnesses $K_1$ and $K_2$ determined by the positions of the sliders (the active lengths $L_1$ and $L_2$) and of the springs pre-tensioned constant angles values $\varphi_1$ and $\varphi_2$. The angles of pre-tension, determined at the sliders, remain constant at all slider positions within
each separate experiment, but the sliders can slide along the springs, changing
the two different $L_1$ and $L_2$. Figure 4 shows a section towards the torsion axis
of the pre-tensioned springs for angles $\varphi_1$, $\varphi_2$ and also the equilibrium position
$\varphi$ of the arm.

As a result of the spring’s tension, a resistive moment occurs:

\begin{align}
M_1 &= (\varphi_1 - \varphi)K_1, \\
M_2 &= (\varphi_2 - \varphi)K_2.
\end{align}

They establish the arm equilibrium:

\begin{equation}
M_1 + M_2 = 0.
\end{equation}

Equations (8), (9) and (10) allow finding the equilibrium position:

\begin{equation}
\varphi = \frac{\varphi_1 K_1 + \varphi_2 K_2}{K_1 + K_2}.
\end{equation}

Considering equation (2) for each spring, the dependence of the equi-
librium position on the lengths of the springs is found:

\begin{equation}
\varphi = \frac{\varphi_1 L_2 + \varphi_2 L_1}{L_1 + L_2}.
\end{equation}

The equilibrium position depends as on the spring’s lengths as well as
on the value of their pre-set tension angles $\varphi_1$ and $\varphi_2$, limited by the strength
characteristics of the springs.
Assuming, that the equilibrium torque in the joint (8) or (9) is equal to the admissible torque (5):

\[ M_1 = -M_2 = M^a, \]

equations (8) and (9) give the restriction to the pre-tensioning, determined by the admissible torque:

\[ \varphi_2 - \varphi_1 \leq M^a \left( \frac{1}{K_2} + \frac{1}{K_1} \right). \]

If the variable nature of stinesses (2) is considered in equation (14), the mutual limit of the lengths \( L_1 + L_2 \) is found, defined by pre-set tension and by admissible torsion torque:

\[ L_1 + L_2 \geq \frac{\varphi_2 - \varphi_1}{M^a} GJ. \]

The control in the joint position by control of the \( L_1 \) and \( L_2 \) according to (12) can work only according to restriction (15).

Antagonistic stiffness in the joint with parallel springs is equal to the sum of the stiffness in both springs:

\[ K = K_1 + K_2. \]

Taking into account the variable nature (2) of the stiffness \( K_1 \) and \( K_2 \), the above equation gives the effective stiffness as a function of lengths \( L_1 \) and \( L_2 \):

\[ K = GJ \left( \frac{1}{L_1} + \frac{1}{L_2} \right). \]

The stiffness control performed by control of the lengths \( L_1 \) and \( L_2 \) according to (17), depends on the strength characteristics of the springs, so restriction (15) must be kept.

4. Numerical experiments

Numerical experiments were carried out on the antagonistic compliance joint with two equal springs as it’s given in Section 2 with an admissible torque (7). The length of each spring can vary \( L = [0.0045 - 0.09] \) [m] and calculations were made in 20 points shown in Table 1. For those springs, according (14), the sum of admissible angles of pre-tensioning at minimal stiffness \( K_1 = K_2 = \)

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4.8893 [Nm/rad] is \((\varphi_2 - \varphi_1) < 66.5^\circ\). At first, values selected for pre-tensioning of the springs are \(\varphi_1 = -12^\circ = -0.209667\) rad and \(\varphi_2 = 12^\circ = 0.209667\) rad.

For these values, a 3D graphic is built for the position of the arm \(\varphi\) according to (12) as a function of lengths \(L_1\) and \(L_2\), which is shown in Fig. 5(a). When the strength constraint (15) of lengths \(L_1\) and \(L_2\) is taken into account, the graph is like in Fig. 5(b).

![Position control: (a) without lengths \(L_1, L_2\) restriction; (b) with \(L_1, L_2\) restriction](image)

As is shown in Fig. 5, when simultaneously changing the lengths of the springs - increasing one and decreasing the other, a joint motion occurs, which is limited within the pre-tensioned springs angles \((-0.209667/0.209667)\) [rad]. For some values of lengths, this can not be obtained – for example, while shortening the spring’s lengths sum below the calculated limit (15).

Within the limits of the springs, lengths from Table 1, according to (17) a 3D graphic was built for control of the effective stiffness of the joint, which is shown in Fig. 6(a). The same, but with added torsion restriction (15) is shown in Fig. 6(b).

Theoretically, the control in the spring’s length in the selected range allows modification of stiffness in the joint within \((9.77/195.57)\) [Nm/rad] (Fig. 6(a)). Assuming the tension limit (15), real stiffness can be \((9.77/104.77)\) [Nm/rad] (Fig. 6(b)). This limit is shown as an inadmissible area due to strength restrictions, where the sum of both springs lengths are below the limit (15).
The study of decoupled stiffness and position joint control by means of a position control of two spring lengths is of concern. At first, the possibility of control the position (12) is researched, while there is a constant value of stiffness in the joint (17). Equations (12) and (17) are calculated for 20 different values of stiffness K in one joint within (9.77/104.77) [Nm/rad], assuming the restriction (15). Specified stiffness defines by (17) correlation between the lengths $L_1$ and $L_2$. So, the control of position is given as a function of stiffness $K$ and the length of one of the springs $L_2$. The results are shown in Fig. 7. The experiment shows that the range of position control depends on the value of the chosen stiffness – the range is more limited at higher stiffness. Also, control of the position is limited by the ratio of the two lengths (15).

In the next experiment, the possibility of controlling the joint stiffness (17) is researched, while there is a constant value of the position (12). Equations (12) and (17) are calculated for 20 different values (S1-S20) of the joint position $\varphi$ within ($-0.209667/0.209667$) [rad] assuming the restriction (15). For a permanent position, a correlation (12) between the lengths $L_1$ and $L_2$ exists. So, the stiffness $K$ is derived as a function of position $\varphi$ and of one of the lengths $L_2$. The results are shown in Fig. 8. The experiment shows small changes of stiffness when the arm is in the middle position. When the arm reach to the max deviation positions $\varphi = -0.209667$ [rad] (S1) and $\varphi = 0.209667$ [rad] (S20), stiffness is maximal (Fig. 8) and can reach values from Fig. 6(b). This
is due to the strength constraints defining the relationship between the lengths $L_1$ and $L_2$.

Fig. 7. Effective joint position for 20 constant values of stiffness and $L_2$

Fig. 8. Effective joint stiffness for 20 constant values of position and $L_2$

The experiments carried out show that by antagonistic approach the compliant joint allows control of the position besides stiffness control. The range of this control is determined by the value of the initial tensioning. Larger values $??$ extend positioning range, but reduce the range of stiffness control, and vice versa.

The concerned antagonistic approach allows decoupled stiffness control via the position in some limits (Fig. 8). Maximal joint stiffness can not be fully achieved for some positions. The position decoupled adjusting via the stiffness is also possible but its range is limited to the set stiffness value (Fig. 7).

5. Conclusions

In the work presented, some approaches for the compliance control for human oriented robots are revealed. An approach based on the structure controlled stiffness and non-antagonistic scheme is presented at first. In addition, an approach is developed for the compliance and the position control in the joint by means of antagonistic actuation. It is based on the capability of the joint with torsion leaf springs to adjust its stiffness. Models of joint stiffness are derived with antagonistic and non-antagonistic influence of the spring forces on the joint motion. The stiffness and the position control possibilities are investigated using numerical experiments. This leads to the next conclusions.
A position control of the joint is possible to occur by simultaneously changing the lengths of the springs, which is limited within the pre-tensioned spring's values. Also, controlling the spring's length allows stiffness control in the joint. It depends on strength restrictions of the springs, expressed as an inadmissible area of spring's lengths.

The considered antagonistic scheme allows decoupled stiffness and position joint control. The range of position control depends on the value of the chosen stiffness – the range is more limited at higher stiffness. On the other hand, the range of stiffness control depends on the value of the joint position – when the arm reaches the maximal deviation, stiffness is maximal.

The stiffness deviation and the position according to this scheme are limited, but the parallel scheme of spring increases the safety of the joint compared to non-antagonistic scheme.

Approaches for compliance control and derived conclusions given in the present work will be tested by means of physical experiment in the future. Evaluation criteria such as: the range of stiffness change and joint position control; parameter influence and linearity change, etc. will be formulated.

The results of this study may be useful for creating rehabilitation devices, prosthesis, robots etc., where the control range of the position and the stiffness are limited in a narrow range.

REFERENCES


