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COMPUTER MODELLING AND SIMULATION OF CONTROLLABLE COMPLIANCE JOINT*

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Abstract. Controllable compliance of robot joint is simulated by software, using two actuators – one for position control and other for compliance control, including a leaf spring. The joint is simulated when changing spring's dimensions. Results are shown for impact interaction and fast positioning.
Key words: robot, compliance, joint, simulation.

1. Introduction

The robots seem that not in the far future will leave the closed cells in industry and will enter the human environment and will work within it after a series of studies in recent years [1], [2], [3]. These new human-oriented robots will look very different from today's industrial robots. They will have a rich sensory information, light structure and compliant actuators to achieve adequate performance and security in the interaction with people in an unknown environment.

The first studies that achieve the necessary performance and security in the interaction with people are based on an active approach – only by sensors, software and feedback control [1]. The main problem of this approach is a low security for the human and the robot, when there is an unexpected collision.

Passive approach is developed to overcome this problem – providing genuine security by introducing the physical flexibility in the joints of the robot. Velocity and torques in the joints of the robot comparable to those of humans and new joint drives with controllable compliance are developed [2],

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[3], in order to achieve the values of compliance. Controllable compliance joints are designed using two actuators – one for position control and the other for compliance control. There are different approaches to managing the physical compliance [4], for example, the so-called structural controlled compliance [5]. Effective structure of the physical spring is manipulated in this approach. A special mechanism is used for the compliance adjustment that changes one of the structural parameters of spring – its length, as it is shown in Fig. 1[6].

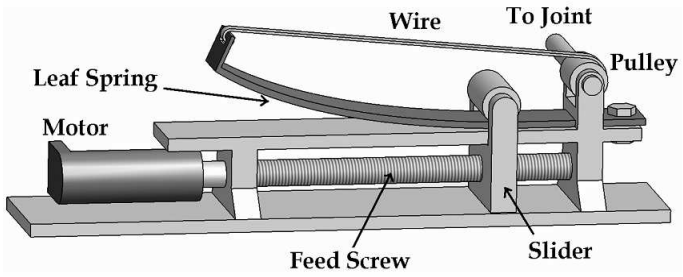


Fig. 1. Passive compliance adjuster

Advantage of this mechanism is that it is easy to build. The position control and the compliance control are independent. It allows all intermediate positions. A disadvantage of this solution is that the load changes at a large deformation of the spring due to a change in the configuration of the mechanism.

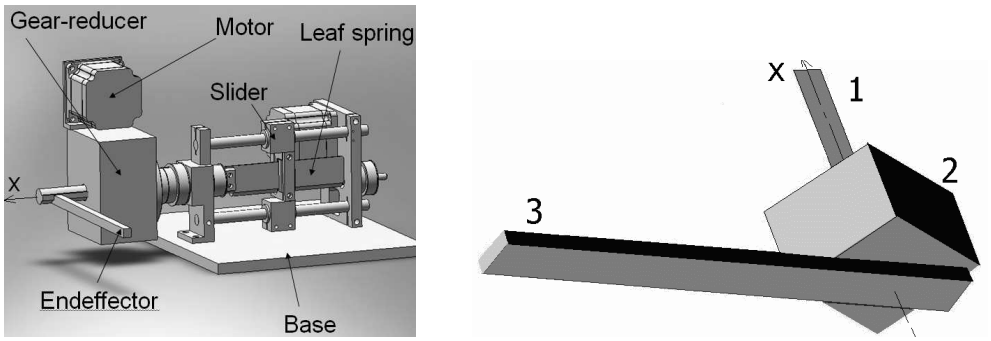
Analytical models and finite elements based techniques are commonly used to predict the behaviour of flexible systems as compliant robots. For example, in [7] novel generalized Newton-Euler equations and relations for inertia forces in nodes of flexible elements are used, or in [8] alternative methodology using advantages of analytical and numerical techniques is utilized.

The objective of the work presented in this paper is to create a model of compliant joint and simulations to be carried out, using suitable program software. The behaviour of the joint to be evaluated when changing based mechanical characteristics. The obtained results for those parameters must be studied with respect to fast positioning and impact interaction.

2. Modelling of compliant joint with controllable stiffness

The study is applied on a joint with "Structure-controlled compliance" (Fig. 2 (a), (b)) that consists of: a leaf spring – 1, which is loaded on its longitudinal axis with torsion; motor-gear – 2 for positioning the arm through one rotating joint; arm – 3. Compliance is controlled by altering the length

of the spring. Motor-gear is linked to the leaf spring by second rotating joint. Compliant joint is a system with two degrees of freedom, which is driven by two engines, one for stiffness and one for position. Modelling and simulation of compliant joint is performed with software package "SOLID DYNAMICS'04", as shown in Fig. 2.



(a) (b)

Fig. 2. Model of compliant joint with motor-gear and executive unit

Since in that decision spring is twisted and isn't bended, the axis of deformation and loads coincide and the linear relation between them retains. The external disturbance is held by the spring which creates torque. Deformations are expressed by φ_x [9], measured by turning the x-axis section over the end section of the spring:

$$\varphi_x = \frac{M.l}{G.Jt}, \tag{2.1}$$

where: M is torsion torque in x-axis;

G – angle deformation module ($G = 7.60\text{E+}10$ N/m^2);

Jt – inertia characteristics of cross-section.

The stiffness is expressed by the torque and the angular deflection of the spring:

$$K_\varphi = \frac{M}{\varphi_x} = \frac{G.Jt}{l}. \tag{2.2}$$

Stiffness control is performed by modulation of the spring length [10]. We calculate stiffness for the length within $l = [0.0006 - 0.09]$ [m], width $h = 0.018$ [m] and thickness $b = 0.001$ [m]. The result stiffness is variated from 733 to 4.89 Nm/rad.

3. Dynamical Simulations on compliant joint

Software package "SOLID DYNAMICS'04" is used. The leaf spring is modelled as compliance in the rotational joint with variable stiffness K_φ. One body is the motor-gear and other – the arm. Desired law of motion $\varphi(t)$ is given in the joint motor-arm.

3.1. Dynamical Simulations on compliant joint at positioning

The movement of the arm is analysed at $\varphi = 0.1$ [rad] at velocity $\dot{\varphi} = 2$ [rad/s]. The masses of the gear are $m_1 = 1.6$ [kg] and the arm $m_2 = 0.2$ [kg].

Resulting moves towards the axis of the joint are shown in Fig. 3, where 1 is the programmed movement of the arm at the positioning joint, 2 is the resulting movement of the arm, and 3 – movement in the compliant joint. The desired position can not be reached at low stiffness, in terms of gravity. Furthermore, the resultant position is reached after a period of continuous oscillation. This problem is reduced when increasing the stiffness in the joint – Fig. 3 (b), (c).

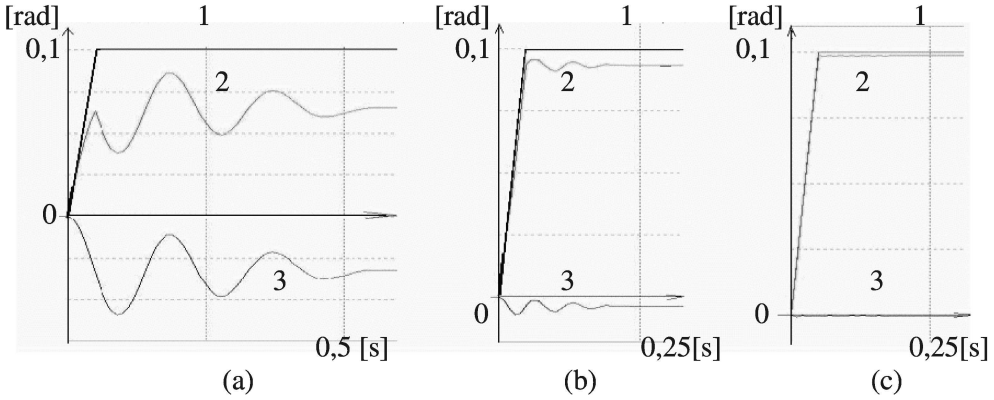


Fig. 3. Positioning movement in the joint at stiffness:
(a) $K_\varphi = 11$ [Nm/rad], (b) $K_\varphi = 88$ [Nm/rad], (c) $K_\varphi = 733$ [Nm/rad]

The stiffness must be high in the phases of acceleration at starting and stopping, but at the phase of constant high velocity the stiffness must be low. This reduces the risk in case of collision. A law of stiffness variation is shown in Fig. 4. The stiffness reaches its maximum $K_\varphi = 733$ [Nm/rad] in the initial and final phase, and in the middle – minimal value $K_\varphi = 4.89$ [Nm/rad]. Movements at $\varphi = 1$ [rad] with velocity $\dot{\varphi} = 2$ [rad/s] are shown in: Fig. 5(a) at constant stiffness $K_\varphi = 4.89$ [Nm/rad]; and (b) at variable law from Fig 4. The positioning is exact and without fluctuations at the end phase, and the

movement is very fluctuating in the middle phase.

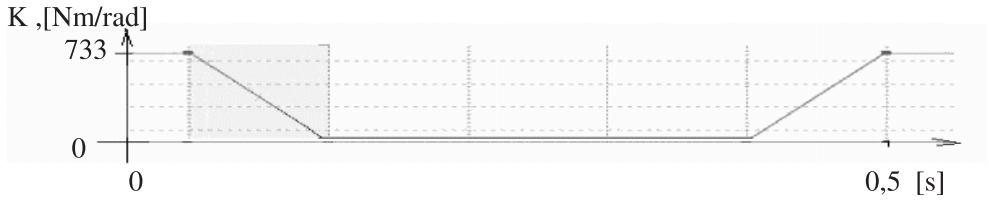


Fig. 4. Input Law of stiffness variation

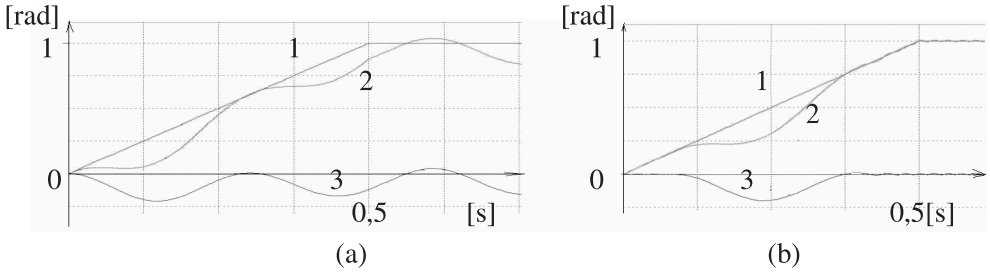


Fig. 5. Positioning movement in the joint at:
(a) $K_\varphi = 4.89$ [Nm/rad]; and (b) at changing stiffness from Fig. 4

3.2. Simulation of a collision

Moments and forces should be evaluated at collision. Simulation conditions are as in the previous case, but the joint has constant compliance and the arm does not fulfil the task, because of collision with an obstacle. The motor in the positioning joint performs the entire task, resulting in deformed spring and resistive moment in the joint. The admissible moment of the spring M is determined by the admissible torsion tension τ [9], shown below:

$$M = Wt.\tau, \tag{3.3}$$

where: Wt is resistant moment of the spring.

Admissible torque tension [9] for spring steel CT85 is $\tau = 49.10^7$ [Nm]. Admissible moment calculated from (3.3) for the chosen spring [10] is $M = 2.8371$ [Nm]. The admissible torsion moment M gives admissible angle of the spring deformation with a given stiffness:

$$\varphi = \frac{M}{K}. \tag{3.4}$$

The simulations are carried out for two cases – with stiffness $K_\varphi^1 = 4.89$ [Nm/rad] and $K_\varphi^2 = 11$ [Nm/rad] with admissible deformations from (3.4):

$$\varphi^1 = 0.581 \text{ [rad]} \quad \text{and} \quad \varphi^2 = 0.258 \text{ [rad]}. \tag{3.5}$$

Input movement in the joint is $\varphi = 2$ [rad] at velocity $\dot{\varphi} = 2$ [rad/s]. Results from the simulation for $K_\varphi^1 = 4.89$ [Nm/rad] are shown in Fig. 6 (a), (b).

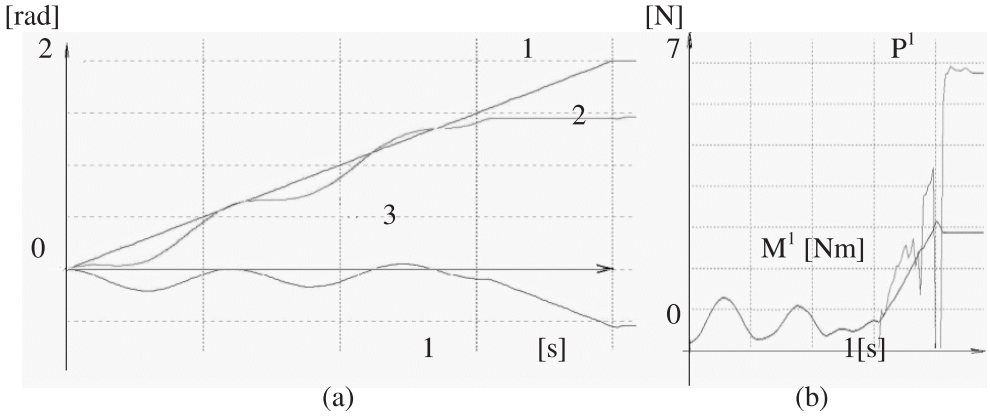


Fig. 6. Compliant joint with stiffness $K_\varphi^1 = 4.89$ [Nm/rad]: (a) movement in the joint; (b) resistive moment M^1 and contact force P^1

The resulting position 2 of the arm is different from the given position 1 as seen in Fig. 6 (a), which is compensated with the deformation of the spring – 3. The deformation here $\varphi^1 = 0.516$ [rad] is less than maximum allowed $\varphi^1 = 0.581$ [rad], so the resulting resistive moment shown in Fig. 6 (b) is $M^1 = 2.8$ [Nm] $\leq M = 2.8371$ [Nm], and the contact force is $P^1 = 6.7$ [N].

The deformation of the spring $\varphi^2 = 0.516$ [rad] exceeds the allowable $\varphi^2 = 0.258$ [rad] at $K = 11$ Nm/rad, so the resulting resistive moment is $M^2 = 6.9$ [Nm] and exceeds the admissible $M = 2.8371$ [Nm], and the contact force is $P^2 = 16.7$ [N].

Increasing stiffness reduces the security by shortening the compliant length of the spring. On the other hand, if the admissible deflection of the spring is exceeded, the resistive moment and the contact force increase. This can be overcome by actively limiting the moment of the motor at passive obstacles.

3.3. Simulation with an impact with external force

External impact force $P = 7$ [N] is applied on a non-moving arm. In Fig. 7 the variation of the force P and the resistive moment M in the joint are shown at given stiffness: (a) $K_\varphi^1 = 4.89$ [Nm/rad] and (b) $K_\varphi^2 = 11$ [Nm/rad].

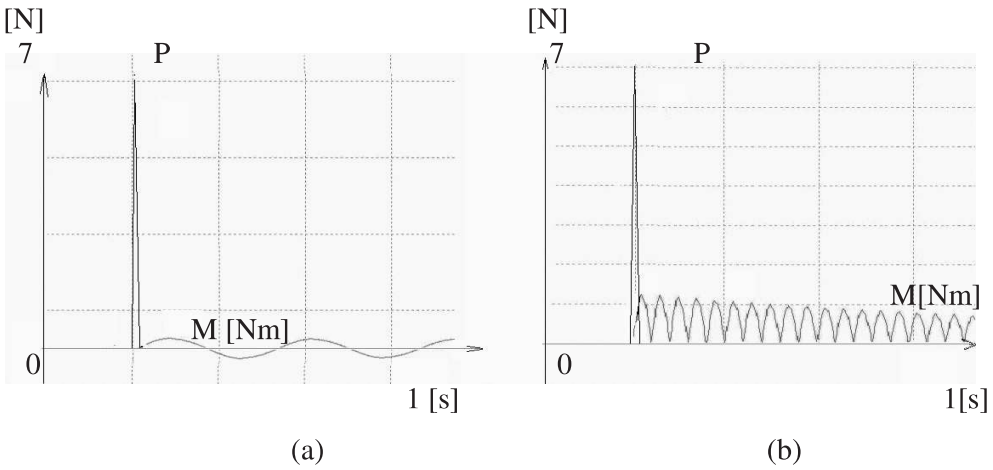


Fig. 7. Force P and resistive moment M in the joint at given stiffness: (a) $K_\varphi^1 = 4.89$ [Nm/rad] and (b) $K_\varphi^2 = 11$ [Nm/rad]

Figure 7 shows that the force is reduced by compliant joint leading to higher security. Fluctuations occur in the resistive moment and therefore, fluctuations in the position of the joint. The amplitude of these fluctuations and their frequency increases with increasing of the stiffness. Amplitude of the resistive moment exceeds the limit values of the torque of spring and may lead to destruction at higher stiffness. It is appropriate to use not-selfstopping gearing, and torque control in positioning joint to reduce the risk of breakage and for increasing security in collision.

4. Conclusion

The simulations show that lowering stiffness results in fluctuations and lowering the precision in positioning. The requirements for security and precision are combined by a variable stiffness. The stiffness variation law is useful in case of long motions controlled from point to point. It is difficult to control the stiffness in cases of short motions.

Lower stiffness increases security providing time to stop and smooth growing contact force in a collision with a barrier. However, its residual value is determined by the torque of the engine in the positioning joint. The low

frequency increases the safety in a collision and reduces the force, but at a continuous force excitation the contact force can be reduced only by active control of the driving torque. The last one means implementation of the active compliance in the positioning joint in addition to the passive joint compliance for best result.

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