Development of antagonistic wire-driven joint mechanism capable of rapid motion and variable stiffness

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Abstract

The advancement of digital transformation in manufacturing is expected to demand novel tasks for industrial machinery unlike before. When adding mechanisms or actuators to enhance machine functionality, concerns arise about potential system enlargement and increased complexity. The design concept of integrating multiple functions into mechanisms using a few actuators and components serves as one means to address these issues. We aim to elucidate the structural arrangement capable of achieving three functions, including normal motion, rapid motion, and variable stiffness, using two electric actuators. In this paper, we introduce a newly devised antagonistic wire-driven joint mechanism aimed at improving the variable stiffness function within the articulated mechanism we have developed, enabling three functionalities.

Keywords: Rapid motion, Variable stiffness, Link mechanism

1. Introduction

Digital transformation is being promoted through the use of robots and digital technologies, such as the Internet of Things and artificial intelligence, with the aim of increasing productivity and manpower saving. In response, robots will be required to perform tasks that have not been done before, and it will be necessary to promote research and development of robots that can perform movements and functions that have not yet been generalized. One area is soft robotics, which can perform rapid motion and variable stiffness. For example, in the food industry, in order to realize the automation of the serving operation of prepared food, it is necessary for the robot to grasp the hardness of the food and to pick and place it according to the hardness of the food so that the soft food does not fall apart. Mochizuki et al. are conducting research on non-contact acoustic impedance estimation using ultrasonic waves before picking to optimize the gripping rigidity in order to reduce damage caused by food picking operations using robot hands [1]. To achieve automation, this system needs to be supplemented with a hand that can change the stiffness of the fingertips. In addition, percussion testing is being researched to predict the firmness of tomatoes to determine their expiration date, to estimate the sugar content of fruit, and to determine whether canned goods are defective [2], [3]. To automate percussion testing, an appropriate hammering mechanism is required in addition to a sensing system.

Adding additional mechanisms and actuators to integrate multiple functions into a robot can increase system complexity and size. Therefore, a design concept that allows mechanisms to have multiple functions with fewer actuators and parts is one way to address these issues.

Aiming to realize three functions with a small number of actuators, we developed a joint mechanism that realizes normal motion, rapid motion, and variable stiffness with two motors [4], [5]. Our research results have shown that these functions can be realized. However, in the variable stiffness function, an end-effector does not rotate unless the load torque applied to the end-effector exceeds the torque applied to the end-effector by the pre-compressed spring to increase stiffness. In this paper, we propose a new antagonistic wire-driven joint mechanism that can rotate the end-effector at the moment load torque is applied to the end-effector, even in a stiff state, in order to improve the functionality related to variable stiffness of joint mechanisms that can realize the three functions.

2. Antagonistic wire-driven joint mechanism

Fig. 1 shows the proposed mechanism. The proposed mechanism consists of the end-effector, a main joint, two wires, two springs, two linear guides, and two cam units. Assume that the positions of the linear guide and main joint are fixed in space. Additionally, the linear guide and slider are a prismatic pair. The internal structure of the cam unit consists of a slider, a passive pulley, an input pulley, a cam follower, a cam, a worm wheel, a worm, and a motor.The function of the cam unit is to extend and retract the wire and release the energy stored in the spring. Rotation of the motor causes the cam to rotate through the worm gear mechanism. Since the cam follower and the input pulley are connected, when the input pulley rotates, the cam follower performs an orbital motion. Therefore, the input pulley rotates driven by the rotation of the cam. For ease of understanding, we will now discuss the structure using the simplified proposed mechanism shown in Fig. 2.

The main difference between Fig. 1 and Fig. 2 is that the cam unit is represented by an electric actuator. As shown in Fig. 3, during normal motion, the end-effector rotates counterclockwise with actuator 1 in the active state and actuator 2 in the passive state. For rapid motion, as shown in Fig. 4, when the two actuators are active, the two sliders move toward the main joint and energy is stored in the two compression springs. Now, when actuator 1 is in the braking state and actuator 2 is in the passive state, the forces on the wires are out of balance and the two sliders are moving away from the main joint. Since actuator 1 is in the braking state, the compression spring force acting on the sliders acts on the end-effector through the wires, and the end-effector performs a rapid motion. Next, consider the variable stiffness function. As shown in Fig. 5, first assume a state in which the two springs are compressed by ε_{sp} and the two actuators are braked. Then assume a state in which a load torque is

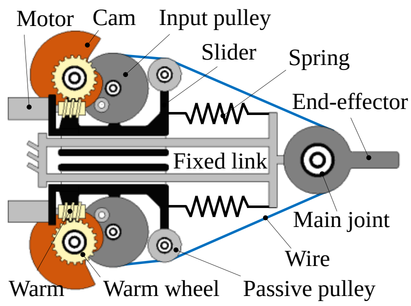

Fig. 1 Conceptual model

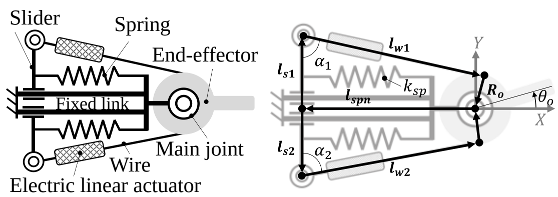

Fig. 2 Simple model

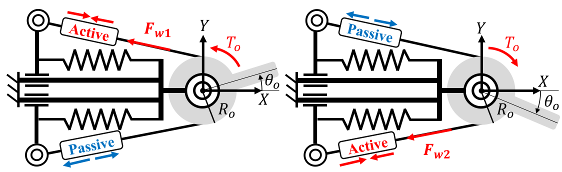

Fig. 3 Normal motion

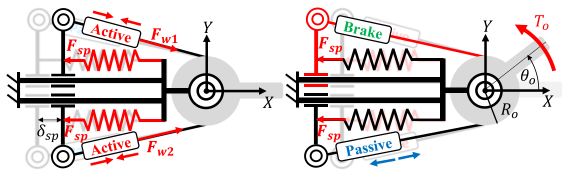

Fig. 4 Rapid motion

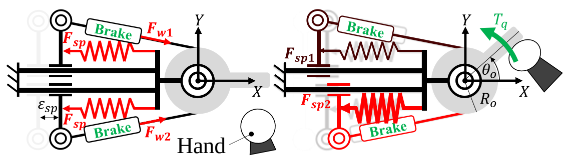

Fig. 5 Variable-stiffness function

applied to the end effector and the end effector is rotated counterclockwise to θ_o. At this time, wire 2 is wrapped around the end effector and slider 2 makes a translational movement toward the main joint, increasing the compressive force acting on spring 2. Therefore, the torque due to the compression force of spring 2 acts on the end-effector in the direction of resisting the load torque. On the other hand, since the wire 1 originally wound around the end-effector is unwound, the slider 1 makes a translational movement in the direction away from the main joint, and the compression force originally acting on the spring 1 decreases. In other words, the torque due to the compression force of spring 1 acts on the end effector in the direction to assist against the load torque. By using this load torque support mechanism, the end-effector can be rotated at the moment when a load torque is applied to the end-effector. In addition, by adjusting the value of ε_{sp}, the compression force of the spring can be changed, and thus the rotational rigidity of the joint can be varied. The relation between the load torque T_q and the internal force acting inside the mechanism is expressed as follows based on their geometric relation:

$$T_q = k_{sp} R_o \left(\frac{\delta_{sp1}}{\sin \alpha_1} - \frac{\delta_{sp2}}{\sin \alpha_2} \right) \quad (1)$$

In Eq. (1), α_* is expressed as follows:

$$\alpha_* = \tan^{-1} \frac{l_{sp*} l_{w*} + l_s R_o}{l_s l_{w*} - l_{sp*} R_o} \quad (2)$$

The rotational stiffness K of the end-effector is expressed using the following equation by differentiating the load torque T_q exerted on the end-effector by the angle θ_o of the output link:

$$K = \frac{dT_q}{d\theta_o} = K_{b2} - K_{b1} \quad (3)$$

In Eq. (3), K_{b*} is expressed as follows:

$$K_{b*} = \frac{R_o^2 k_{sp} l_{w*}}{l_{sp*} \sin \alpha_*} + \frac{\left[\frac{{R_o}^2 k_{sp} \delta_{sp*} l_{w*} l_s}{l_{sp*} (l_{sp*}^2 + l_s^2)} - \frac{R_o^3 k_{sp} \delta_{sp*}}{(R_o^2 + l_{w*}^2)} \right]}{\sin \alpha_* \tan \alpha_*} \quad (4)$$

In Eq. (4), with $\varepsilon_{sp} = 0$, $K_{b1} = 0$ for counterclockwise rotation of the output link and $K_{b2} = 0$ for clockwise rotation.

3. Analysis of output characteristics

Using the equations derived in Chapter 2, we analyzed the output characteristics with respect to variable stiffness. Fig. 6 shows the profile of the load torque T_q and angle θ_o of the end-effector at initial spring displacements ε_{sp} of 0, 5, 10, and 15 mm (Eq. (1)). Further, Table 1 shows the values of each parameter used in the analysis. The analysis was performed assuming that a load torque was applied to the end-effector and the angle θ_o was rotated from 0[deg] (initial posture) to +40[deg] and then from 0[deg] to -40[deg]. The analysis results confirm that the load torque increases as the angle θ_o increases. Furthermore, as the value of ε_{sp} increases, the load torque required to rotate the joint increases. Even when ε_{sp} was set, the load torque applied to the end-effector did not have to exceed the torque applied to the end-effector by the pre-compressed spring. On the other hand, it can be observed that the maximum displacement angle decreases as the value of ε_{sp} increases. This is because as ε_{sp} increases, the distance between the slider and the main joint becomes shorter and the range of motion of the slider becomes smaller. Fig. 7 shows the profile of the rotational stiffness K and angle θ_o of the end-effector when the initial spring displacement ε_{sp} is 0, 5, 10, and 15 mm (Eq. (3)). It can be observed that as ε_{sp} increases, the rotational stiffness of the main joint increases. It can also be observed that the value of each stiffness increases rapidly near the end point of the motion. This is because as the angle θ_o increases, the distance between the slider and the main joint becomes shorter, and the tension generated in the wire increases as α_* becomes smaller.

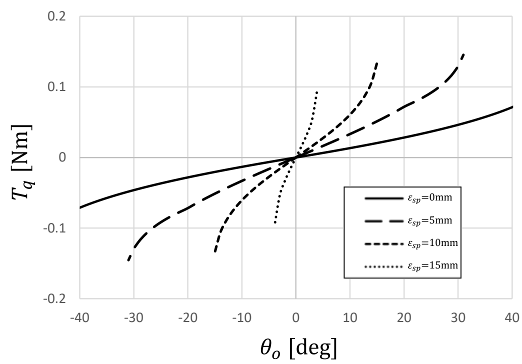

Fig. 6 Relationship between the rotational angle of end effector and load torque

Table 1. Design parameters

k_{sp}	Spring constant	500 N/m
R	Radius of end-effector	0.01m
l_s	Length of slider	0.02 m
l_{spn}	Natural length of spring	0.02 m
$max(\theta_o)$	Maximum movement range	+40°
$min(\theta_o)$	Minimum movement range	−40°

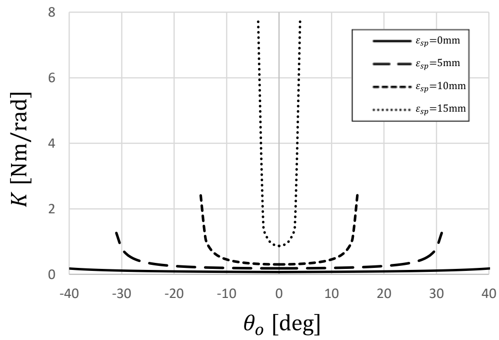

Fig. 7 Relationship between the rotational angle of end effector and rotational stiffness

4. Conclusion

In this study, we describes the structure of the new mechanism and demonstrated a method for realizing normal motion, rapid motion, and variable stiffness. Additionally, we derived expressions for variable stiffness and analyzed. In the previously developed joint mechanism, the end-effector does not rotate unless the load torque exceeds the torque applied to the end-effector by the spring preloaded to increase the rigidity. As a result of the analysis, it was theoretically clarified that the newly proposed joint mechanism can solve this problem and rotate the end-effector at the moment the load torque is applied. Future issues include the search for a more optimal link ratio and the need to verify the effectiveness of the system by fabricating an actual machine and evaluating its performance.

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