Safe link mechanism based on nonlinear stiffness for collision safety

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Abstract

A safe robot arm can be achieved by either a passive or active compliance system. A passive compliance system composed of purely mechanical elements often provide faster and more reliable responses for dynamic collision than an active one involving sensors and actuators. Since both positioning accuracy and collision safety are important, a robot arm should exhibit very low stiffness when subjected to a collision force greater than the one causing human injury, but maintain very high stiffness otherwise. To implement these requirements, a novel safe link mechanism (SLM), which consists of linear springs, a double-slider mechanism and shock-absorbing modules, is proposed in this research. The SLM has the advantages of variable stiffness which can be achieved only by passive mechanical elements. Various experiments of static and dynamic collisions showed the high stiffness of the SLM against an external force of less than the critical impact force, but an abrupt drop in the stiffness when the external force exceeds the critical force, thus guaranteeing collision safety. Furthermore, the critical impact force can be set to any value depending on the application.

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1. Introduction

For industrial robots, safe human–robot coexistence is not as important as the fast and precise manipulation. However, service robots often interact directly with humans for various tasks. For this reason, safety has become one of the most important issues in service robotics. Therefore, several types of compliant joints and flexible links of a manipulator have been proposed for safety.

A safe robot arm can be achieved by either a passive or active compliance system. In the actively compliant arm, collision is detected by various types of sensors, and the stiffness of the arm is properly controlled. The active compliance-based approach suffers from the relatively low bandwidth because it
involves sensing and actuation in a response to dynamic collision. This rather slow response can be improved slightly when non-contact sensors such as proximate sensors are employed. Furthermore, the installation of the sensor and actuator in the robot arm often lead to high cost, an increase in system size and weight, possible sensor noise, and actuator malfunction. To cope with these drawbacks, a collision detection method that used only proprioceptive sensors and provided the information on the direction of the robot reaction after collision was proposed [1]. However, it also suffered from some problems with active compliance approaches.

On the other hand, the robot arm based on passive compliance is usually composed of the mechanical components such as a spring and a damper, which absorb the excessive collision force. Since this approach does not utilize any sensor or actuator, it can provide fast and reliable responses even for dynamic collision. Various safety mechanisms based on passive compliance have been suggested so far. The programmable passive impedance component using an antagonistic nonlinear spring and a binary damper was proposed to mimic the human muscles [2]. The mechanical impedance adjuster with a variable spring and an electromagnetic brake was developed [3]. The programmable, passive compliance-based shoulder mechanism using an elastic link was proposed [4]. A passive compliance joint with rotary springs and a MR damper was suggested for the safe arm of a service robot [5]. A variable stiffness actuator with the nonlinear torque transmitting system composed of a spring and a belt was developed [6]. The compliance method in the drive system so as to mechanically decouple the heavy actuator inertia from the link inertia was also introduced [7].

Most passive compliance-based devices use linear springs. However, one drawback to the use of a linear spring is positioning inaccuracy due to the continual operation of the spring even for small external forces that do not require any shock absorption and due to undesirable oscillations caused by the elastic behavior of the spring. To cope with this problem, some systems adopt the active compliance approach by incorporating extra sensors and actuators such as electric dampers, motors or brakes, which significantly impair the advantages of a passive system. In this research, therefore, a novel passive compliance-based safety mechanism that can overcome the above problems is proposed.

Some tradeoffs are required between positioning accuracy and safety in the design of a manipulator because high stiffness is beneficial to positioning accuracy whereas low stiffness is advantageous to collision safety performance. Therefore, the manipulator should exhibit very low stiffness when subjected to collision force greater than the one that causes injury to humans, but should maintain very high stiffness otherwise. Of course, this ideal feature can be achieved by the active compliance approach, but this approach often causes the several shortcomings mentioned above.

In this research, this ideal feature is realized by a novel design of the safe link mechanism (SLM) which is based on the passive compliance. SLM is composed of the passive mechanical elements such as linear springs, a double-slider mechanism, and shock-absorbing modules. The springs and shock-absorbing modules are used to absorb the high collision force for safety, while the double-slider mechanism determines the safety or nonsafety of the external force so that the SLM operates only in case of an emergency. The main contribution of this proposed device is the variable stiffness capability implemented only by use of passive mechanical elements. Without compromising positioning accuracy for safety, both features can be achieved simultaneously with the SLM.

The rest of the paper is organized as follows. The operating principle of the SLM is discussed in detail in Section 2. Section 3 presents further explanation about its operation based on simulations. Various experimental results for both static and dynamic collisions are provided in Section 4. Finally, Section 5 presents conclusions and future work.

2. Construction of Safe link mechanism

The passive safety mechanism proposed in this research is composed of a spring, a double-slider mechanism and a shock-absorbing module-wire system. Section 2.1 presents the concept of the transmission angle of the double-slider mechanism and the characteristics of the double-slider mechanism in combination with the spring. Section 2.2 deals with the construction of the shock-absorbing system.
2.1. Double-slider mechanism

Springs have been widely used for a variety of safety mechanisms because of their excellent shock-absorbing property. Since the displacement of a linear spring is proportional to the external force, the robot arm exhibits deflection due to its own weight and/or payloads when a spring is installed at the manipulator joint. This characteristic is beneficial to a safe robot arm, but has an adverse effect on positioning accuracy. To cope with this problem, it is desirable to develop a spring whose stiffness remains very high when an external force acting on the end-effector is within the range of the normal operation, but becomes very low when it exceeds a certain level of force due to collision with the object. However, no such springs with this ideal feature exist. In this research, the power transmission characteristics of the 4-bar linkage are exploited to achieve this nonlinear spring feature.

Consider a 4-bar linkage mechanism shown in Fig. 1. When an external force \( F_E \) is exerted on point B of the input link in the \( x \)-axis direction, an appropriate resisting force \( F_R \) acting in the \( y \)-axis direction can prevent the movement of the output link. In the 4-bar linkage, the transmission angle is defined as the angle between the floating and the output link. The power transmission efficiency from the input to output varies depending on this transmission angle. If the transmission angle \( \gamma \) is less than 45° or greater than 135°, a large force is required at the input link to move the output link. That is, only a small \( F_R \) is sufficient to prevent the output link from moving for a given \( F_E \) in this case. However, as the transmission angle approaches 90°, the power transmission efficiency improves, thus leading to easy movement of the output link of a 4-bar linkage [8].

The 4-bar linkage can be converted into a double-slider mechanism shown in Fig. 2. If the output link (link 3) is assumed to be infinite and located in the \( x \)-axis direction, then revolute joint A between link 2 and link 3 can move rectilinearly only in the \( y \)-axis direction. Likewise, if the input link (link 1) is assumed to be infinite and located in the \( y \)-axis direction, then joint B can move only in the \( x \)-axis direction. In this case, the 4-bar linkage can be regarded as a double-slider mechanism. Note that the transmission angle of a double-slider mechanism can be also defined as the angle between the floating link (link 2) and the output link. The force balance of the forces acting on sliders 1 and 2 can be given by

\[
F_R = -F_E \tan \gamma
\]  

Note that the value of \( \tan \gamma \) is always negative because \( \gamma \) is in the range of 90°–180° in Fig. 2, thus requiring the minus sign in Eq. (1). In Eq. (1), for the same external force, the resisting force changes as a function of \( \gamma \).

If the pre-compressed spring is installed between points C and D in Fig. 3, the spring force \( F_S \) can offer the resisting force \( F_R \), which resists the movement of slider 1 caused by the external force \( F_E \). When the external force is balanced against the spring force, the external force can be described in terms of the transmission angle and the other geometric parameters as follows:

\[
F_E = -k(s_0 - c + d + l \sin \gamma) \cot \gamma
\]  

Fig. 1. 4-bar linkage.
where $k$ is the spring constant, $s_0$ the initial length of the spring, $l$ the length of link 2, $y_0$ the initial length of $AO_1$ and $y$ the displacement of slider 1. Although $y$ does not explicitly appear in Eq. (2), it is directly related to $\gamma$ by the relation of $y = y_0 + y = l \cos(\gamma - 90^\circ)$. For example, when $k = 0.8 \text{ kN/m}$, $l = 19 \text{ mm}$, $s_0 = 34 \text{ mm}$, $c = 36 \text{ mm}$ and $d = 6.5 \text{ mm}$, the external force for the static force balance can be plotted as a function of $\gamma$ in Fig. 4. The spring force does not need to be specified for static balance because it is automatically determined for a given $\gamma$. As shown in the figure, the external force diverges rapidly to positive infinity as $\gamma$ approaches $180^\circ$, so even a very small spring force can make this mechanism statically balanced against a very large external force. In this research, the transmission angle in the range of $160^\circ$–$170^\circ$ is mainly used in consideration of the mechanical strength of the mechanism.

In this proposed mechanism, the external force required to balance with the spring force is defined as the critical impact force. For a given $\gamma$, a static balance is maintained when the external force equals the critical impact force, as shown in Fig. 4, but the spring is rapidly compressed once the external force greater than this critical value acts on this mechanism. The detailed explanation about the motion of this double-slider mechanism combined with a spring is given below.

Since slider 2 is a large portion of the proposed SLM, its mass is much larger than those of slider 1 and link 2 in Fig. 3. It is therefore assumed that only the mass of slider 2 is considered. In this case, the motion of the
double-slider mechanism combined with a spring can be simply modeled as a 1 DOF mass–spring system, as shown in Fig. 5. The spring resisting force acting on slider 1 in Fig. 3 is transmitted to slider 2 via link 2 whose transmission angle affects the force transmission ratio. Therefore, the equivalent spring force can be given by

\[ F_s' = -F_s \cot \gamma = -k \gamma \cot \gamma \]  

(3)

Since the displacement \( y \) of slider 1 and the displacement \( x \) of slider 2 are related by \( y = \sqrt{l^2 - (l \cos \gamma(0) - x)^2} - l \sin \gamma(0) \), the equivalent stiffness \( k' \) of the spring attached to slider 2 can be described by

\[ k' = \frac{k}{x} \left[ l \cos \gamma(0) \left( 1 + \tan \gamma(0) \sqrt{\frac{(l \cos \gamma(0) + x)^2}{l^2 - (l \cos \gamma(0) - x)^2}} \right) - x \right] \]  

(4)

where \( \gamma(0) \) is the initial transmission angle, and \( l \) and \( k \) are the same parameters with Eq. (2).

Fig. 6a shows the equivalent stiffness curves as a function of displacement of slider 2. In this analysis, the spring constant \( k \) is set to 0.8 kN, the length \( l \) of link 2 to 19 mm and the initial transmission angle \( \gamma(0) \) to 165°. The equivalent stiffness is maintained very high for a small displacement of slider 2, but it quickly drops as the displacement increases. Hence this nonlinear stiffness can be realized by the double-slider mechanism.
As the external force acting on slider 2 increases linearly up to 40 N during 1 sec, the displacement is changed as shown in Fig. 6b. Since the critical impact force was set to 28 N in this simulation, the transmission angle for the static equilibrium becomes $165^\circ$ from Eq. (2). When the external force increases from 0 to 28 N, which is below the critical impact force, slider 2 does not move. As the external force $F_{E}(t)$ increases above the critical impact force, the static equilibrium cannot be maintained and the slider starts moving. Since the stiffness of the spring rapidly decreases, as shown in Fig. 6a, slider 2 moves left rapidly. In summary, the SLM stiffness remains very high like a rigid link while the external force is below 28 N. However, as the external force becomes larger than 28 N, the stiffness abruptly diminishes, thus causing the SLM to behave as a flexible link.

2.2. Shock-absorbing system

A rigid-plastic material such as a crash panel and an automobile bumper does not deform under normal operation, but must deform plastically and absorb the shock in case of a large impact accident [9]. It deforms when subjected to the stress of more than the critical stress and absorbs the shock during this plastic deformation. This feature is well suited for the safety mechanism, but the rigid-plastic material cannot be restored to its original shape after deformation. To overcome this disadvantage, a shock-absorbing wire–module device is proposed below.

As shown in Fig. 7a, small cylindrical modules are connected in series by a wire. Under normal operation, these modules remain tightly connected together by the tensioned wire. One end of the wire is attached to the left of slider 2 in Fig. 3. Suppose a large impact greater than the critical impact force is applied to one end of...
the system, as shown in Fig. 7b. This large external force causes slider 2 to move slightly to the left because of the wire tension. This movement then breaks the static balance and moves slider 2 abruptly to the left, and the wire becomes loosened. This loose wire makes disconnects the modules to absorbs the impact force effectively, as shown in Fig. 7b. This phenomenon will be explained in more detail in the next section.

3. Safe link mechanism model

3.1. Prototype modeling

The mechanisms introduced conceptually in the previous section are now integrated into the Safe Link Mechanism (SLM), which suggests a new concept of safe robot arm. SLM consists of a double-slider mechanism, a linear spring and a module-wire system, as shown in Fig. 8. As shown in Fig. 8, the moving plate (link 4 in Fig. 3) can slide relative to the fixed plate (slider 2 in Fig. 3) along the prismatic joint (P-joint) composed of a linear busing guide. Note that a combination of fixed link 4 and an assembly of slide 1 and the spring, which could not move in the x-axis direction in Fig. 3, are now allowed to move, whereas moving slider 2 in Fig. 3 now functions as a fixed plate in Fig. 8.

In this prototype, the collision force acting on the end-effector is amplified according to the geometric relations of the modules and the robot arm, and is transmitted to the moving plate by a wire. Therefore, the external force exerted on SLM is proportional to the collision force. The relationship between both forces will be described in more detail in Section 3.3.

If the external force exceeding the critical impact force is applied to the moving plate shown in Fig. 8b, then the moving plate is pulled toward the fixed plate by a wire connected to the base plate. Then, slider 1 linked to link 2 is forced to move up the guide shaft to compress the spring. This movement of slider 1 reduces the transmission angle, so maintaining the static balance requires a greater resisting force for the same external force. However, the increased spring force due to its compression is not large enough to sustain the balance. This unbalanced state causes slider 1 to rapidly slide up, thus bringing the moving plate further toward the fixed plate. As a result, the wire becomes loose and the modules are disconnected, which absorbs the collision force,
as explained before. However, if the external force amplified from the collision force is less than the critical impact force, the end-effector does not move at all, and the modules remain tightly connected together, thus providing high stiffness to the SLM.

3.2. Compliance analysis

The 1-DOF robot arm with the SLM is modeled in Fig. 9 for analyzing the change of its compliance. If a large collision force is applied to the end-effector, then the shock-absorbing modules of SLM separate and rotate about the contact point of both modules, as shown in Fig. 9b. In this case, the 1-DOF arm with SLM can be regarded as the 2-link manipulator with two links (proximal and distal links) and joints (joint 1 and joint 2).

The stiffness of joint 1, $k_1$, can be adjusted by controlling the joint actuator. The stiffness of joint 2, $k_2$, can be expressed by the derivative of the torque of joint 2, $\tau_2$, with respect to an angular displacement of joint 2, $\theta_2$.

$$k_2 = -\frac{d\tau_2}{d\theta_2}$$  \hspace{1cm} (5)

Note that the above derivative is always negative because the torque reduces as the angular displacement increases due to the nature of SLM. Therefore, the minus sign is required to make the stiffness positive in Eq. (5).

As shown in Fig. 10a, the joint torque $\tau_2$ can be given by

$$\tau_2 = hF_C = rF_W$$  \hspace{1cm} (6)

where $h$ is the distance between the rotation center and the end-effector, $F_C$ the collision force acting on the end-effector, $r$ the radius of the module, and $F_W$ the tension of the wire. Since the wire tension $F_W$ is equal to the external force $F_E$ acting on slider 2 in Fig. 5, the joint torque can be given by

$$\tau_2 = -rk\left[l\cos\gamma(0)\left(1 + \tan\gamma(0)\sqrt{\frac{(l\cos\gamma(0)+x)^2}{l^2-(l\cos\gamma(0)-x)^2}}\right) - x\right]$$  \hspace{1cm} (7)
Because the displacement $x$ of the moving plate is equal to the distance $M_1M_2 = 2r \sin(\theta_2/2)$, the stiffness of joint 2 can be obtained as follows:

$$k_2 = rk \left( r \cos \left( \frac{\theta_2}{2} \right) + \frac{l \cos \gamma(0) \tan \gamma(0)}{2 \sqrt{\left( s_0 - 2r \sin \left( \frac{\theta_2}{2} \right) \right)^2 + \left( l \cos \gamma(0) - 2r \sin \left( \frac{\theta_2}{2} \right) \right)^2}} \right) \frac{2r \cos \left( \frac{\theta_2}{2} \right) \left( l \cos \gamma(0) - 2r \sin \left( \frac{\theta_2}{2} \right) \right)}{l^2 - \left( l \cos \gamma(0) - 2r \sin \left( \frac{\theta_2}{2} \right) \right)^2} + \frac{2r \cos \left( \frac{\theta_2}{2} \right) \left( l \cos \gamma(0) - 2r \sin \left( \frac{\theta_2}{2} \right) \right)^3}{\left( l^2 - \left( l \cos \gamma(0) - 2r \sin \left( \frac{\theta_2}{2} \right) \right)^2 \right)^2} \right) \right]$$

(8)

Since the stiffness of each joint is independent of each other, the stiffness matrix is given by

$$K_q = \begin{bmatrix} k_1 & 0 \\ 0 & k_2 \end{bmatrix}$$

(9)

The compliance matrix in the Cartesian coordinate can be defined as:

$$K^{-1} = J K_q^{-1} J^T$$

(10)

where $J$ is the Jacobian matrix. Solving the eigenvalue problem, the compliance ellipsoid can be drawn from the eigenvalues and eigenvectors. The compliance ellipsoid is useful for the safety analysis of a robot arm. The length of the major axis of the ellipsoid represents the maximum compliance, while the length of the minor axis determines the minimum compliance. Therefore, the longer the length of the major axis, the more safety is provided to the system.

Fig. 11 shows some compliance ellipsoids according to the transmission angle of the SLM when the angle of joint 1 is fixed at $45^\circ$ and the collision force acts on the end-effector. In this analysis, the spring constant $k$ is set to 10 kN, the length $l$ of link 2 to 19 mm, the initial transmission angle $\gamma(0)$ to $165^\circ$, the radius of the module $r$ to 12 mm and the stiffness of joint 1 $k_1$ to 52.4 N m/deg (=$3$ kN m/rad).

When the external force amplified from the collision force is less than the critical impact force, the stiffness of joint 2 is very high, like that of a rigid joint because of the characteristics of the SLM. For this reason, the
length of the major axis of the ellipsoid becomes so short, and consequently, high positioning accuracy of the robot arm can be achieved in normal operation, as shown in Fig. 11a.

However, when the external force exceeds the critical impact force, the angular displacement of joint 2 occurs and its stiffness rapidly drops owing to the operation of the SLM. As shown in Fig. 11b–d, the transmission angle decreases, and thus the length of the major axis of the compliance ellipsoid increases. Therefore, collision safety can be ensured by exploiting the SLM.

3.3. Safety criterion and analysis

The safety criterion can be divided into static and dynamic collisions. The static collision means that the collision speed between the robot arm and a human is very low (e.g., below 0.6 m/s). The human pain tolerance for static collision can be expressed by

$$ F \leq F_{\text{limit}} $$

where $F_{\text{limit}}$ is the injury criterion value which has been suggested as 50 N by several experimental researches [10].

In the case of dynamic collision, both the collision force and the collision speed are important. To represent human safety associated with the dynamic collision of the SLM, the head injury criterion (HIC) used to quantitatively measure head injury risk in car crash situations is adopted in this research [11].

$$ \text{HIC} = T \left( \frac{1}{T} \int_0^T a(t) \, dt \right)^{2.5} $$

where $T$ is the final time of impact and $a(t)$ is the acceleration in the unit of gravitational acceleration $g$. An HIC value of 1000 or greater is typically associated with extremely severe head injury, and a value of 100 can be considered suitable to normal operation of a machine physically interacting with humans.

Fig. 11. Compliance ellipsoid of a manipulator with safe link mechanism.
Some studies have been conducted on collision safety between the human and the robot [12,13]. In this research, the stiffness model of a human neck was added to the models proposed by [12,13]. Furthermore, initial conditions and parameters of the collision model were realistically modified, as shown in Fig. 12.

It is assumed that the robot arm rotates at an initial velocity and a constant joint torque is provided by the actuator. In Fig. 12c and d, the robot arm without SLM can be simply modeled as a 2 DOF mass–damper–spring system and that with SLM as a 3 DOF system because the arm can be regarded as two links. Since the collision of the robot arm without SLM was already studied [12,13], the collision analysis of the robot arm with the SLM will be explained in this research.

The equation of motion is given by

\[ M \ddot{x} + C \dot{x} + Kx = F(t) \]  

(13)

where \( M, C, K, x(t) \) and \( F(t) \) are the mass matrix, the damping coefficient matrix, the stiffness matrix, the displacement vector and the force vector, respectively. The mass and stiffness matrices are given by

\[
M = \begin{bmatrix}
    m_{l1} & 0 & 0 \\
    0 & m_{l2} & 0 \\
    0 & 0 & m_{\text{head}}
\end{bmatrix},
\]

\[
K = \begin{bmatrix}
    k_{\text{SLM}} & 0 & 0 \\
    0 & k_c & 0 \\
    0 & 0 & k_n
\end{bmatrix}
\]

(14)

where \( m_{l1} \) is the mass of link 1, \( m_{l2} \) the mass of link 2, \( m_{\text{head}} \) the mass of a human head, \( k_c \) the stiffness of the covering, \( k_n \) the stiffness of a human neck, and \( k_{\text{SLM}} \) the stiffness of the SLM. It is assumed that the damping coefficient matrix is proportional to the mass and stiffness matrices.

The solution of Eq. (13) can be obtained by the modal analysis. The acceleration of each mass can be given by

\[
\ddot{x} = M^{-\frac{1}{2}} P \ddot{f} = M^{-\frac{1}{2}} P \begin{bmatrix}
    \ddot{r}_1 \\
    \ddot{r}_2 \\
    \vdots \\
    \ddot{r}_i
\end{bmatrix},
\]

where \( \ddot{r}_i(t) = e^{-i\omega_i t}[A_i \sin(\omega_i t + \phi_i + \psi_i) + B_i \sin(\omega_i t + \phi_i)] \)  

(15)

![Fig. 12. Collision model between human head and robot arm (a) without SLM and (b) with SLM, and simplified collision model between human head and robot arm (c) without SLM and (d) with SLM.](image-url)
where
\[ A_i(t) = \omega_i^2 \left[ \left( \dot{r}_i(0) + \zeta_i \omega_i r_i(0) \right)^2 + (r_i(0) \omega_{di})^2 \right]^{1/2} \]
\[ B_i(t) = \sqrt{\left( \zeta_i \omega_i f_i / \omega_{di} + \zeta_i \omega_{di} f_i / \omega_i \right)^2 + \left( \zeta_i f_i + \omega_{di}^2 f_i / \omega_i^2 \right)^2} \]
\[ \phi_i = \tan^{-1} \frac{r_i(0) \omega_{di}}{f_i(0) + \zeta_i \omega_i r_i(0)}, \quad \psi_i(t) = \tan^{-1} \frac{-2 \zeta_i \omega_i r_i(0)}{\zeta_i \omega_i^2 - \omega_{di}^2}, \quad \varphi_i(t) = \tan^{-1} \frac{-\zeta_i f_i + \omega_{di}^2 f_i / \omega_i^2}{\zeta_i \omega_i f_i / \omega_{di} + \zeta_i \omega_{di} f_i / \omega_i} \]
where the matrix \( P \) is composed of the orthonormal eigenvectors and \( r_i(t) \) is the solution of the modal equation, and \( \omega_i, \omega_{di}, \zeta_i \) and \( f_i \) are the natural frequency, the damped natural frequency, the damped ratio, and the external force of each mode, respectively.

The stiffness of the SLM as a function of displacement can be obtained from iterative calculations.
\[ k_{SLM}(t) = \frac{k}{x_i(t) - x_2(t)} \left[ \cos \gamma(0) \left( 1 + \tan \gamma(0) \sqrt{\frac{(l \cos \gamma(0) - (x_1(t) - x_2(t)))^2}{l^2 - (l \cos \gamma(t) - (x_1(t) - x_2(t)))^2}} \right) - \Delta x(t) \right] \quad \text{for } t > 0 \]

(16)

In this analysis, when \( v = 4 \text{ m/s}, \quad F_{E1} = 50 \text{ N}, \quad m_1 = 4 \text{ kg}, \quad m_2 = 1 \text{ kg}, \quad m_{arm} = 5 \text{ kg}, \quad m_{head} = 2.5 \text{ kg}, \quad k_e = 30 \text{ kN/m}, \quad k_n = 1 \text{ kN/m} [14], \quad k_{SLM}(0) = 200 \text{ kN/m}, \quad l = 0.16 \text{ m}, \quad k = 0.1 \text{ kN}, \quad g(0) = 165, \) the initial conditions are \( x = [0 \quad 0 \quad 0]^T \) and \( \dot{x} = [v \quad 0 \quad 0]^T \), the head acceleration is plotted in Fig. 13.

Substituting the head acceleration into Eq. (12), the HIC value can be calculated as follows:
\[ \text{HIC} = T \left( \frac{1}{T} \int \frac{x_{head}}{g} \, dt \right)^{2.5} \]

(17)

where \( x_{head} \) is \( x_3 \) in the case of the robot arm with SLM, and \( g \) is the gravitational acceleration (=9.8 m/s\(^2\)).

In the case of the robot arm without SLM, as shown in Fig. 13a, the head acceleration reached a peak value of 35 g and the HIC value was computed as 258, which indicates a high risk of injury to human. However, when the robot arm is equipped with SLM, the HIC value decreased to 20 and the complete duration of acceleration also substantially decreased. It was verified that the robot arm with SLM provides much higher safety for human–robot contact than that without SLM.

4. Experiments for safe link mechanism

4.1. Prototype of SLM

The prototype of SLM shown in Fig. 14 was constructed to conduct various experiments related to the performance of the SLM. Most components are made of duralumin which can endure the shock exerted on SLM. The moving plate can translate relative to the fixed plate by means of the linear bushing guides.

Fig. 13. Analytical results showing head acceleration as a function of time during collision with robot arm (a) without and (b) with SLM.
As shown in Fig. 15a, the initial transmission angle can be adjusted physically by inserting some thin plates between the slider and the shaft end block. The contact surface of one module has a convex hemisphere, while that of another module has a concave shape, to prevent the twist between the modules. The wire is made of stainless steel, which can endure to the shock. The wire tension can be adjusted by means of the worm and worm gear.

4.2. Experimental results

Fig. 16 shows an experimental setup in which the SLM is installed at the 1-DOF robot arm. A force/torque sensor is installed at the end-effector of the arm to measure the collision force. The displacement of SLM is measured by an encoder attached to SLM.

In the experiment for static collision, the spring constant was 10 kN, its initial length was 34 mm, and the transmission angle was 165°. The end-effector of the robot arm was initially placed to barely touch a fixed wall, and its joint torque provided by the motor was increased slowly. The static collision force between the robot and the wall was measured by a force/torque sensor. Experiments were conducted for the robotic arms with and without SLM.

The robot arm without the SLM delivered a contact force that increased up to 70 N to the wall, as shown in Fig. 17a. However, the contact force of only up to 40 N was transmitted to the wall for the robot arm with SLM, as shown in Fig. 17b. In other words, the contact force above the pain tolerance does not occur because the excessive force is absorbed by SLM. In Fig. 17d, virtually no displacement of the robot arm occurs when
the contact force is below the critical impact force of 28 N. Therefore, the robot arm with the SLM can accurately handle a payload up to approximately 2 kg as if it were a very stiff link. As the contact force rises above the critical impact force, SLM stiffness quickly diminishes, thus maintaining the robot arm in the safe region. In summary, the SLM provides high positioning accuracy of the robot arm in the working region, and guarantees safe human–robot contact by absorbing the contact force above 50 N in the unsafe region.

![Experimental setup for robot arm with SLM.](image)

Fig. 16. Experimental setup for robot arm with SLM.

![Experimental results for static collision for robot arm: collision force vs. time (a) without SLM, (b) with SLM, (c) transmission angle vs. collision force, and (d) collision force vs. displacement of end-effector.](image)

Fig. 17. Experimental results for static collision for robot arm: collision force vs. time (a) without SLM, (b) with SLM, (c) transmission angle vs. collision force, and (d) collision force vs. displacement of end-effector.
Next, some experiments of dynamic collision were conducted for the robot arm equipped with the SLM. The experimental conditions including the spring constant, the initial length of the spring, and the initial transmission angle were set to the same values as those of static collision experiments. For dynamic collision, a urethane ball of 2.5 kg moving at a velocity of 3 m/s was forced to collide with the end-effector of the robot arm. The acceleration of the ball was measured by the accelerometer mounted at the ball.

The experimental results are shown in Fig. 18. At the instant the ball contacts the end-effector, the acceleration of the ball reached a peak value of 10g, but immediately after collision, the collision force delivered to the ball dropped rapidly because of the operation of SLM. The dynamic collision safety of the robot arm with SLM can be verified in terms of HIC defined by Eq. (12). The HIC value was computed as 12, which is far less than 100. Therefore, the safe human–robot contact can be achieved even for this harsh dynamic collision.

Fig. 18b shows the experimental results for the dynamic collision of the robot arm without SLM. The peak value of the acceleration is almost 5 times that of the robot arm with SLM, and the HIC value reached as high as 532, which indicates high risk of injury to a human. Therefore, the robot arm with SLM provides much higher safety for human–robot contact than that without SLM.

It is difficult to apply the proposed SLM to an industrial manipulator. This safety mechanism was developed to be used for the safe arm of a service robot which undergoes frequent physical contact with humans and environment. Service robot arms usually require a smaller payload (for example, 2 kg) and less positioning accuracy than industrial manipulators. The proposed SLM can be used for the manipulator whose
payload is below 4–5 kg without sacrificing the positioning accuracy during the manipulation task. We experimentally verified that the proposed SLM installed at the 2 DOF robot arm with a payload of 1 kg can provide both positioning accuracy and collision safety, as shown in Fig. 19 [15,16].

5. Conclusions

In this research, the safe link mechanism (SLM) was proposed. The SLM maintains very high stiffness up to the pre-determined critical impact force, but provides very low stiffness above this critical value, at which point the SLM absorbs the impact acting on the robot arm. From the analysis and experiments, the following conclusions are drawn:

1. SLM has very high stiffness like a rigid arm when the external force acting on it is less than the critical impact force. Therefore, high positioning accuracy of the robot arm can be achieved in normal operation.
2. When the external force exceeds the critical impact force, the stiffness of SLM abruptly drops. As a result, the robot arm acts as a flexible arm with high compliance. Therefore, human–robot collision safety can be attained even for a high-speed dynamic collision.
3. The critical impact force of SLM can be set accurately by adjusting the initial transmission angle of the double-slider mechanism, the spring constant and the initial spring length.
4. The proposed SLM is based on passive compliance, so it shows faster response and higher reliability than that based on the active compliance having sensors and actuators.

Currently, a simpler and lightweight safe link mechanism not using the wire-module system is under development. Furthermore, the research on the safe joint mechanism possessing the similar characteristics is under way.

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