Research on an Active Seat Belt System*

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Abstract
In a car crash, permanent injury can be avoided if deformation of an occupant’s rib cage is maintained within the allowable value. In order to realize this condition, the occupant’s seat belt tension must be instantaneously adjusted by a feedback control system. In this study, a seat belt tension control system based on the active shock control system is proposed. The semi-active control law used is derived from the sliding mode control method. One advantage of this proposed system is that it does not require a large power actuator because the seat belt tension is controlled by a brake mechanism. The effectiveness is confirmed by numerical simulation using general parameters of a human thorax and a passenger car in a collision scenario with a wall at a velocity of 100 km/h. The feasibility is then confirmed with a control experiment using a scale model of about 1/10 scale. The relative displacement of the thorax model approaches the allowable value smoothly along the control reference and settles near this value. Thus, the proposed seat belt tension control system design is established.

Key words: Motion Control, Nonlinear Control, Automobile, Safety Engineering, Safety Device, Seat Belt, Impact, Shock, Deformation Control, Thorax Compression

1. Introduction
In a car crash, permanent injury can be avoided if deformation of the occupant’s rib cage is maintained within the allowable value. In order to realize this condition, the occupant’s seat belt tension must be adjusted within a millisecond by a feedback control system. In other words, an active seat belt system is required.

From the viewpoint of occupant safety in an automobile, a crushable zone is designed in the car body, and seat belts and air bags are installed. With regard to an active seat belt, Balandin, Bolotnik and Pilkey deduced calculation methods for seat belt tension that held the acceleration of the thorax, deformation of the thorax, and excursion of the occupant in the vehicle within the allowable value for a nonlinear human-vehicle system [1]. However, it is not possible to completely eliminate the effect of disturbances because these are open-loop systems. A control system that instantaneously provides feedback on the current state is required to ensure the control effect.

In order to realize an active seat belt system using a feedback control system, it is necessary to develop an actuator that generates instantaneously a large force corresponding to the impulsive force. This is a formidable problem given the typical requirement of the automobile industry for light-weight, high fuel efficiency vehicles.

In this study, an active seat belt system is proposed, and the control system is derived by applying a semi-active shock control system [2], which the author has developed. Next, in order to confirm the active seat belt system’s effectiveness, it is illustrated by numerical
simulations that thorax compression is maintained within the allowable value during a car crash while the occupant excursion in the vehicle is reduced. An actuator that adjusts the seat belt tension by friction within a millisecond of the crash is then developed for the semi-active shock control. Finally, an experimental model of about 1/10 scale is produced, and through this model experiment, it is shown that the thorax deformation of the experimental model converges to the desired value naturally and is maintained near this value in order to confirm the feasibility of the proposed active seat belt system.

2. Nomenclatures

The main symbols used for the active seat belt system and numerical values are shown below. The numerical values for the human thorax used are those of a standard adult, and the numerical values used for the vehicle are those of a general passenger car, which has the crushable zone deformed elastically in the first 2 cm and then deformed plastically at a constant deceleration of 45 G in the crush.

\[ c_{\text{compress}} = 0.525 \text{kN/s} \] damping coefficient of the thorax in compression

\[ c_{\text{restore}} = 1.23 \text{kN/s} \] damping coefficient of the thorax in restoration

\[ f_{\text{fric}} = 453 \text{kN} \] force for coulomb friction in the crushable zone of the vehicle

\[ k_1 = 22.9 \text{MN/m} \] spring constant of the crushable zone

\[ k_2 = 359.2 \text{kN/m} \] spring constant of the seat belt

\[ k_3 = 52.5 \text{kN/m} \] additional spring constant of the thorax, with a strong affecting force

\[ m_1 = 1000 \text{kg} \] mass of the vehicle

\[ m_2 = 27.2 \text{kg} \] effective mass of the remaining portion of the thorax and the part of the human body mass attached to the thorax by the vertebral column

\[ \delta_1 = 0.0318 \text{m} \] displacement in which the additional spring begins to work

\[ \mu = -0.075 \text{m} \] allowable value of thorax compression

3. Semi-active Shock Control System

In collision protection, generally a buffer material is inserted in the collision plane to decrease the deformation and internal stress in each object. However, in press working, it is desirable to minimize the internal stress of the die to maximize the life, while the workpiece is made to plastically deform by the collision. This implies that a unidirectional transmission of the mechanical energy from the colliding element to the collision-receiving element is required. T. Shimogo found that this collision concept, termed a “dynamic semiconductor,” was reported in a Soviet Union information magazine in the 1960s, but the details remain unknown. In response to this article, he concluded that an actuator had to be inserted instead of a buffer material to allow for the adjustment of the buffer effect in each object, and an active control system that provides feedback on the current status of the system was necessary. Subsequently, he advanced original research on a shock control system that enabled a unidirectional transfer. The control system for the collision between two objects was developed by applying the optimal control theory and assuming the deformation characteristic of each object in a simple elasto-plastic deformation model. A numerical simulation confirmed the following: (1) the characteristics of the control system change with the weights of the performance index and (2) a peak appears in the deformation ratio of the two objects. That is to say, an optimum condition exists. However, the reason for the ratio peak was not clarified, and the optimum condition had to be obtained by numerical simulation on a per-case basis.
In contrast, the present author has developed a shock control system based on the sliding mode control technique, in which the control law for a nonlinear system can easily be derived and the optimal path of the state as a hyper-plane can be defined, because the elasto-plastic deformation model is strongly nonlinear. The system performance is then confirmed by a numerical solution\(^ 8\)). One feature of this study is the proposition of a design technique for a hyper-plane in consideration of the deformation response of a colliding object so that the collision-receiving object deforms plastically and the deformation of the colliding object is minimized. In related studies, the H-infinity control system\(^ 9\)) and the feedforward system\(^ 10\)) have been developed for active shock control, and the features have been examined experimentally. There is also a study that applies the active shock control to a knee bolster\(^ 11\)).

In these shock control systems, the actuator inserted between colliding objects has to generate a large force, which corresponds to the impulsive force; therefore, the development of such an actuator is a big challenge. The main action of the actuator is to control the impulsive force, that is, the optimal force transmission from the colliding object to the collision-receiving object. Therefore, a semi-active control system that controls the force transmission by a brake system would be suitable for a shock control system, although the control performance is inferior. The author has proposed a semi-active shock control system in which the actuator controls the tension within a millisecond using a friction device. The effectiveness of the proposed system has been confirmed by a numerical simulation and model experiment\(^ 2\)). In this study, by adapting this semi-active shock control, the active seat belt system is developed.

4. Analytical Model of the Active Seat Belt System

The active seat belt system is simplified for a collision system of two objects---a thorax of an occupant and a vehicle, as shown in Fig. 1. The state equation is then derived for the numerical simulation and the development of control system.

4.1 Assumptions

An analytical model is obtained using the following assumptions;

(1) Two bodies, namely, a thorax (Body A) and a vehicle (Body B), collide with a rigid wall.

(2) The vehicle is modeled as a mass. The crushable zone is represented by a perfect elasto-plastic deformation model.

(3) The seat belt is represented by an elastic deformation model. The force holding the thorax is controlled by the tension of the seat belt.

(4) The thorax (Body A) of the human is modeled as two masses, and is represented by a nonlinear visco-elasticity deformation model with a broken-linear characteristic.
The collision is analyzed until the plastic deformation of the vehicle (Body B) stops.

4.2 State Equation

The displacement of the mass \( m_1 \) modeled as the vehicle, that is, the deformation of the crushable zone is designated \( x_1 \); the displacements of the masses \( m_2 \) and \( m_3 \) modeled as the thorax are designated \( x_2 \) and \( x_3 \), respectively. Under the above assumptions, the equations of motion on the displacements \( x_1 \), \( x_2 \), and \( x_3 \) are derived. For the active seat belt system, the displacement between the dashboard of the vehicle and thorax of the occupant, and the compression of the thorax are important. Therefore, the excursion of the occupant, that is, the relative displacement between \( m_1 \) and \( m_2 \), is designated \( y_1 \), and the deformation of the thorax, that is, the relative displacement between \( m_2 \) and \( m_3 \), is designated \( y_2 \). The equations of motion are transformed into equations for the displacements \( x_1 \), \( y_1 \), and \( y_2 \). Then, the state equation on the state vector \( \{x\} = \{x_1 \ y_1 \ y_2 \ x_1 \ y_1 \ y_2 \}^T \) is obtained as follows:

\[
\frac{d\{x\}}{dt} = \{f\} + \{B\}u,
\]

where \( u \) designates the force affecting the thorax through the seat belt, \( f_1 = \dot{x}_1 \), \( f_2 = \dot{y}_1 \), \( f_3 = \dot{y}_2 \), \( f_4 = -\frac{P_1}{m_1} \), \( f_5 = -\frac{P_1}{m_1} + \frac{P_2}{m_2} + \frac{C_2}{m_2} \dot{y}_2 \), \( f_6 = -\frac{m_2 + m_3}{m_1 m_3} (P_2 + C_3 \dot{y}_2) \), \( B_1 = B_2 = B_3 = 0 \), \( B_4 = \frac{1}{m_1} \), \( B_5 = \frac{m_1 + m_2}{m_1 m_2} \), and \( B_6 = -\frac{1}{m_2} \). \( P_1 \) designates the force for the perfect elasto-plastic deformation model in the crushable zone of the vehicle. It consists of the restoring force and the coulomb friction and is defined as the function of displacement \( x_1 \) as follows:

\[
-x_u \leq x_1 \leq x_u : P_1 = k_1 x_1 \quad \text{and} \\
x_u < x_1 : P_1 = f_{r1},
\]

where \( x_u = f_{r1} / k_1 \) designates the proportional limit. \( P_2 \) and \( C_2 \) designate the nonlinear restoring force and the damping coefficient of the visco-elasticity deformation model, respectively, as the thorax has a broken-linear characteristic. They change according to the relative displacement \( y_2 \) and relative velocity \( \dot{y}_2 \), respectively, as follows:

\[
-\delta_2 \leq y_2 : P_2 = k_3 y_2 \\
y_2 < -\delta_2 : P_2 = (k_3 + k_4) y_2 + k_4 \delta_2,
\]

\[
\dot{y}_2 \leq 0 : C_3 = c_{3\text{compress}} \quad \text{and} \\
0 < \dot{y}_2 : C_3 = c_{3\text{restore}}
\]

where \( \delta_2 \) is the deformation of Body A in which the spring coefficient increases.

5. Semi-Active Control Law

On the Basis of the active shock control system\(^2\), the semi-active control law for the active seat belt system is derived by applying the sliding mode control theory. The sliding mode control has two features: (1) a robust control law for the disturbance, which can be developed if the state can be accurately observed and the input can be switched quickly and (2) the control law for the nonlinear system, which can be easily derived.
5.1 State Equation for the Control System

The purpose of this study is to maintain the compressive deformation of the thorax within the allowable value. Therefore, thorax deformation \( y_2 \) and the velocity \( \dot{y}_2 \) are made to be controlled variables. Then, state equation (1) is simplified into a state equation in the state vector \( \{x_c\} = \{y_2 \ \dot{y}_2\} \) as follows:

\[
d\{x_c\}/dt = \{f_c\} + \{B_c\}u,
\]

where \( f_{c1} = \dot{y}_2 \), \( f_{c2} = -\frac{m_2 + m_3}{m_2 m_3} (P_2 + C_3 \dot{y}_2) \), \( B_{c1} = 0 \), and \( B_{c2} = -\frac{1}{m_2} \).

5.2 Design of the Hyper-Plane

The hyper-plane that defines how the states converge to the desired values will be set to the state variables \( y_2 \) and \( \dot{y}_2 \) as follows:

\[
s = (y_2 - y_{2s}) + \frac{1}{\omega_s} (\dot{y}_2 - \dot{y}_{2s}),
\]

where \( \omega_s = \sqrt{(1/m_2 + 1/m_1) k_3} \) is the non-damped circular natural frequency (assuming the spring constant of the thorax being \( k_3 \)), and \( y_{2s} \) and \( \dot{y}_{2s} \) are the desired variables of \( y_2 \) and \( \dot{y}_2 \), respectively. Because the duration of elastic deformation of the crushable zone shown in Fig. 1 is \( 1/30 \) of the collision duration in the case of a collision at 60 km/h and is \( 1/85 \) of the collision duration in the case of a collision at 100 km/h, the plastic deformation is only considered in the design of the desired variables. The damping characteristic of the thorax is over-damping. Therefore, the desired variables are set as a step response of the over-damped mass-spring system, in which the displacement changes to a set point smoothly and is constant at the point as follows:

\[
y_{2s} = \mu \frac{\omega_s}{2} e^{-\omega_s t} \left(1 - \frac{\zeta}{\sqrt{\omega_s^2 - 1}} e^{\omega_s t} - \frac{\zeta}{\sqrt{\omega_s^2 - 1}} e^{-\omega_s t}\right)
\]

and

\[
\dot{y}_{2s} = \frac{\omega_s \mu}{2\sqrt{\omega_s^2 - 1}} e^{-\omega_s t} \left(e^{\omega_s t} - e^{-\omega_s t}\right),
\]

where \( \zeta = (1/m_2 + 1/m_1) c_1 / (2\omega_s) \) and \( \omega_s = \sqrt{\omega_s^2 - 1} \omega_s \). By making the step response to equal the desired variables, it is possible to miniaturize the actuator because the control input in which the thorax is deformed impossibly is not required. The desired variables are focused in the increase of deformation in the initial stage because the step response has been derived in the case that the force affecting the thorax is weak.

5.3 Sliding Mode Control Law

If the nonlinear state equation is derived as in Eq. (5) and the hyper-plane is set as in Eq. (6), a sliding mode control law is obtained as follows\(^{12}\):

\[
u = -\alpha [u_{eq}] g S + F_c,
\]

where \( \alpha > 1 \), \( g = [G]^{-1} B_c \neq 0 \), \( [G] = \partial s/\partial \{x_c\} \), and \( u_{eq} = -[G] \{f_c\} / g \) is an equivalent
control input that slides the state to the origin on the hyper-plane. \( F_r = -k_1 \mu (1 + m_z/m_x) \) is a force deforming the thorax to the allowable value \( \mu \) when the spring constant of the thorax is assumed to be \( k_1 \), and it is added in Eq. (8). Further, the smooth function \( \gamma \cdot s/(\gamma \cdot s + \varepsilon) \) is used instead of the sign function (also termed the signum function) in order to suppress the chattering by the switching of the control input, although the control performance is inferior.

5.4 Semi-Active Control Law

The control input \( u_a \) of Eq. (8) becoming negative is rare because the force \( F_r \) deforming the thorax to the allowable value \( \mu \) is added as a feedforward input in the control law. Additionally, the seat belt being pulled back is rare because this study focuses on the duration of the collision, that is, the duration of the crushable zone compressing. Therefore, a semi-active control law \( u \) can be defined using the sliding mode control law of Eq. (8) as follows:

\[
\begin{align*}
\text{if } u_a \geq 0, & \quad \text{then } u = u_a, \quad (9a) \\
\text{if } u_a < 0, & \quad \text{then } u = 0. \quad (9b)
\end{align*}
\]

6. Numerical Simulations

Numerical simulations using the Runge-Kutta method is carried out to confirm the effectiveness of the active seat belt system applying the semi-active shock control system.

Figure 2 shows the result in the case of a general passenger car driven by a standard adult that collides into the rigid wall from an initial velocity of 100 km/h using the parameters described in Section 2. From the top, these figures show the time histories of the vehicle displacement, that is, the deformation of crushable zone \( x_1 \); the relative displacement between the dashboard and the occupant, that is, the excursion of occupant \( y_1 \); the deformation of thorax \( y_2 \); the desired variable \( y_2^* \) indicated by an alternating long and short dashed line; and the dimensionless control input \( u/F_r \), that is, the dimensionless force affecting the thorax in the case of \( \alpha = 1.05 \) and \( \varepsilon = 0.01 F_r \). The tension is only generated by the actuator while the seat belt is extended (\( F_r < 0 \)) for the semi-active control. Therefore, there is a delay of the thorax deformation \( y_2 \) in the initial stage until the seat belt begins to extend and the tension becomes stable. Moreover, the control law requires a large control input \( u/F_r \) in this duration because the seat belt tension generated is not stable. However, the subsequent deformation of the thorax increases naturally along the desired variable and approaches the allowable value \( \mu \). The control input is chattering in the interval between 0.004 s and 0.04 s, which is a result of the generation of rapid changes in the control input, the convergence of the thorax deformation to the allowable value, and the sequential change in thorax rigidity within this short period. In this case, the maximum force affecting the thorax is 4100 N, and the occupant approaches 60.5 cm to the dashboard.

Next, Fig. 3 shows the simulation result where the desired variable is uniformly set to the allowable value from the beginning in order to decrease the amount of occupant excursion to the dashboard. The approach of the occupant to the dashboard is reduced to 54.9 cm, although the maximum force affecting the thorax increases to 6200 N. In this case, the chattering of the control input has not occurred because the change in thorax rigidity, the rapid change in control input, and the convergence of the thorax deformation toward the safe value are not generated in the short period.

Further, Fig. 4 shows the simulation result where the rigidity of the thorax is increased to 1.5 times in order to confirm the robustness. Although the deformation is slightly smaller than the set point for the shortage of the seat belt tension (because \( \alpha \) in Eq. (8) is set
small in order to suppress the chattering of the control input), the thorax deforms according to the desired variable. The robustness for the parameter change can then be confirmed.

From the above results, the effectiveness of the proposed active seat belt system can be confirmed.

7. Model Experiments

In order to confirm the feasibility of the active seat belt system (in particular, to examine that the tension can be controlled during a collision of about 60 ms), a real-time scale model in which the length and mass are scaled to 1/10 is designed, and the control experiments are carried out.

7.1 Actuator for Tension Control

For an actuator that changes the friction of a contact plane quickly, a piezo-electric actuator and a giant magneto-strictive actuator can be used. A giant magneto-strictive

Fig. 2 Simulation results of the active seat belt system (in the standard parameters case)  
Fig. 3 Simulation results of the active seat belt system (in the changed target case)
actuator is more suitable for the actuator of the active seat belt system in the vehicle because it can be controlled by a current from the batteries on the board, and it is not brittle like a piezo-electric actuator. However, a multilayer piezo-electric actuator is used in this experiment since a giant magneto-strictive actuator having the performance suitable for this experimental model of 1/10 scale was not found.

Figure 5 shows the actuator for tension control, with a length of 120 mm and diameter of 52 mm. The structure in the tension control component is shown in Fig. 6. A stainless strand steel wire (φ 0.27) modeled as the seat belt and a wire for rewinding are wound on a pulley. The pulley is held between two GaeaDrives (Gaeatech Co.), and supported through a thin type ball bearing in the shaft of the upper and lower press plate in order to freely rotate the pulley. This component is pressed by a multilayer piezo-electric actuator (NEC TOKIN Co., AE1010D44H40, resonant frequency: 34 kHz), and the friction of the rollers in the GaeaDrives is changed in order to control the wire tension. In the GaeaDrive, it is expected that the static friction can smoothly shift to the kinetic friction and the torque can be
changed quickly, since 12 rollers are obliquely placed and slipped and rotated.

The tension control actuator is adjusted to focus on the smallest tension for the semi-active control. As a result, the static characteristic, in which the minimum tension is 0.13 N, the maximum tension is 2.00 N, and the relationship between tension and input voltage is almost linear, is obtained as shown in Fig. 7.

![Fig. 7 Static characteristic of the tension control actuator](image)

### 7.2 Experimental Apparatus

As shown in Fig. 8, the collision phenomenon of the vehicle is reproduced by a linear motor of 900 mm overall length (Shicoh Engineering Co., MML020A-KMP01, MML020-MP300, maximum thrust: 69N). The moving coil plate supported by linear guides is considered as the vehicle model. The tension control actuator, in which the wire (modeled as the seat belt) is wound around the pulley, and two aluminum blocks restricted on the linear guide of 150 mm in length (modeled as the thorax) are mounted on it. As shown in Fig. 9, two blocks are connected with a spring, and another hard spring of short length is inserted between the blocks in order to reproduce the nonlinear restoration characteristic. The wire drawn from the actuator is hooked in a U-shape on the front face of the front block, and the end of the wire is fixed at the opposite side of the actuator. In the rear block, a laser displacement sensor (Keyence Co., LB-02) is installed to measure the relative displacement between the blocks. The mass of the rear block $m_3$ is smaller than the contraction scale mass for the constraint of the linear motor. The position of the coil plate is observed using a linear scale, which is installed parallel with magnet plates of the linear motor on the base. The coil plate is controlled by the driver (Servoland Co., SVEM4A) to stop at the constant deceleration of $-3 \text{G}$ after it is accelerated and run at the constant speed (over 2.78 m/s). The semi-active control input is calculated in a digital signal processing (DSP) system with AD and DA converters (mtt Co., s-BOX). The tension of the wire is adjusted in the sampling frequency of 1000 Hz according to the signal measured in the sampling frequency of 2000 Hz by the laser displacement sensor. The main parameters of the experimental apparatus and the numerical values used are as follows:

$$c_{\text{spring}} = c_{\text{tension}} = 0 \text{ Ns/m}, \quad k_1 = 265 \text{ N/m}, \quad k_2 = 1040 \text{ N/m},$$

$$m_2 = 0.04572 \text{ kg}, \quad m_3 = 0.1433 \text{ kg}, \quad \delta_2 = 0.00333 \text{ m}, \text{ and } \mu = -0.0045 \text{ m}.$$

### 7.3 Control Law and Hyper-Plane for Model Experiment

Although the thorax is considered to be an over-damped system, it is difficult to reproduce the over-damped characteristic in the experimental model. Thus, the damping of the thorax model is ignored in the model experiment, and the desired variables in the hyper-plane and the control law are derived under the condition of non-damping.
The approach that makes the desired variable to be a step response is not changed. The desired variables for the model experiment are then set following the step response of a non-damped mass-spring system, instead of the step response of an over-damped system shown in Eq. (7):

\[ 0 \leq t \leq \frac{\pi}{\omega_n} : \quad y_{21} = \frac{\mu}{2}(1 - \cos \omega_n t), \quad \dot{y}_{21} = \frac{\mu}{2} \omega_n \sin \omega_n t \quad \text{and} \quad (10a) \]

\[ \frac{\pi}{\omega_n} < t : \quad y_{21} = \mu, \quad \dot{y}_{21} = 0. \quad (10b) \]

Although the semi-active control law of Eq. (9) does not change, the feedforward input in Eq. (8) is set considering the conversing performance to the set point (modeled as the allowable value of the thorax) as follows:

\[ 0 \leq t \leq \frac{\pi}{\omega_n} : \quad u_s = -\alpha' \left[ \frac{y \cdot s}{\sqrt{y \cdot s} + \varepsilon} \right] F'_s + \frac{F_s}{2} \quad \text{and} \quad (11a) \]

\[ \frac{\pi}{\omega_n} < t : \quad u_s = -\alpha' \left[ \frac{y \cdot s}{\sqrt{y \cdot s} + \varepsilon} \right] F'_s. \quad (11b) \]

where \( \alpha = 15, \quad \varepsilon = 0.001, \quad F_s = -k_3 \mu (1 + m_2 / m_1), \quad \alpha' = 1.1, \) and \( F'_s = -(k_3 + k_4) \mu + k_4 \varepsilon (1 + m_2 / m_1). \)
Although the feedforward input should be changed from $F_r$ to $F'_r$ on the condition that the additional spring becomes to operate at $y_2 = -\delta_2$, it is changed on the condition that the desired variables become constant by assuming the robustness of the sliding mode control, since the time difference is extremely small. The effectiveness of these changes has been confirmed by the numerical simulation using the parameters of the experimental model\textsuperscript{13}).

7.4 Experimental Results

Figure 10 shows a typical experimental result. From the top, these figures show the time histories of the thorax deformation $y_2$, and the desired variable $y_{2t}$ (indicated by an alternating long and short dashed line), the semi-active control input $u$, and the velocity of vehicle $\dot{x}_1$ and the theoretical value (indicated by an alternating long and short dashed line) in case that the deceleration is changed stepwise to $-3G$. For comparison, time histories without the control are also shown in the top figure of $y_2$. A dashed line indicates the maximum tension fixing, and a broken line indicates the minimum tension fixing. Although about 8 ms are required to reach the deceleration of vehicle model to $-3G$, the thorax model deforms along the desired variable. Furthermore, although thorax deformation overshoots, it is maintained near the set point. Therefore, the feasibility of the active seat belt system can be confirmed.

![Fig. 10 Result for the model experiment](image-url)
8. Conclusions

In order to maintain the deformation of the thorax within the allowable value, the active seat belt system is proposed by applying the semi-active shock control system, and the effectiveness of this system is confirmed by the numerical simulation. The experimental model of 1/10 scale is then produced, together with the tension control actuator for the model, and control experiments are carried out. The experimental results confirmed the feasibility of the proposed active seat belt by illustrating that the thorax model is compressed naturally and maintained along the desired variable.

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