Development of a flexible roll forming machine for cutting curved parts with virtual prototyping technology

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Abstract
Predicting machine tool performance at the design stage is one way to resolve the time issue and achieve cost savings. Our objective in the present work was to develop a new roll forming machine for curved products comprising a servo motor, ball screw, and rolling-element linear guide, with the support of virtual prototyping technology. First, a multibody simulation model was developed based upon the machine concept to investigate machine dynamic behavior in real time. Then, we proposed an adaptive sliding-mode PID-based controller (ASMPID) to control the manufacturing process. We performed co-simulation between the mechanical structure and the virtual controller to investigate the head unit trajectory and to identify optimal control parameters. Finally, we manufactured a prototype machine to evaluate the machine accuracy and the benefits of the new system. The results indicated that the proposed model including the mechanical structure and the intelligent controller was effective and feasible for designing the roll forming machine. We expect this work to improve the prototyping efficiency of new rolling machines.

Keywords: Flexible rolling machine, Curved products, Virtual prototype technology, Adaptive sliding mode, Multibody simulation, Modeling and controlling

1. Introduction
Roll forming technology has been widely applied in the automotive industry due to its various advantages, including high production speed, reduced tooling cost, and improved quality (Bae and Huh, 2012). The roll forming process requires a cutting operation to separate a long formed strip at a desired length. Cutting devices were developed at a high level to produce final straight parts (Fig. 1). However, some roll forming products require curved paths, for which the cut-off process is more difficult. Moreover, integration of the roll forming and cutting operations in the machine can be considered as an appropriate choice to improve productivity. Therefore, the development of a flexible device that can manufacture various products with different cross sections and curvatures in the longitudinal direction is a worthwhile research direction.

Optimization of roll forming processes has attracted the attention of many researchers. Yan et al. (2013) investigated the deformation characteristics of flexible roll forming through a finite element (FE) model. A method has been proposed to reconfigure the roll forming process using adjustable punches (Yoon et al., 2014). Joo et al. (2015) designed the hat-channel profile to establish longitudinal strain during roll forming. However, recent research has not focused on the roll forming machine, and no one has investigated the interaction between the machine’s mechanical structure and its controller under working conditions. Consequently, approaches whereby machine performance can be predicted prior to physical prototyping are urgently needed.

Usually, a physical prototype is used in the design stage to produce and evaluate new machine models. However, this method is slow and expensive. Fortunately, many publications have indicated that virtual prototyping approaches using well-defined material properties, numerical models, and controllers can accurately analyze, simulate, and investigate real machine behavior (Altintas et al., 2005). (Xuewen et al. 2012) presented co-simulation between the
mechanical model and the controller to determine the important parameters for a linear hydraulic servo cylinder. Likewise, Catalin et al. (2010) developed a complex virtual prototyping platform for simulating the automotive suspension system. Guo et al. (2016) designed a mechanical structure and control system for a female fitting robot. Integration of multibody dynamics and control system has been used to precisely describe the behavior of a shaking table system (Seki and Iwasaki, 2015). These studies have demonstrated that virtual prototyping technology is a powerful tool for simulating and optimizing machine tool characteristics.

To increase the application potential of roll forming processes, the development of a flexible roll forming machine for manufacturing curved parts is considered herein. In this proposed machine, the rolling and cutting process are integrated to improve production speed. The curved products are separated to the desired length at the end of the rolling process. To evaluate machine characteristics, it is essential to have a reliable numerical model. Moreover, we found that the interaction between the mechanical structure and the controller, along with external disturbances in the processing time, contributed to the machine tool efficiency. Consequently, a robust approach for describing the machine tool behavior and generating optimal control parameters is an important research goal.

The remainder of this paper is structured as follows. First, we introduce the scientific methodology used. Next, we present our concept for a new roll forming machine and mechanical model, and develop a multibody simulation model and control system. Subsequently, we discuss the simulation results for the virtual cutting machine and physical prototyping. Finally, we draw conclusions and suggest future research.

2. Research methodology

In Fig. 2, we propose a framework for the development of a new rolling machine based on a multibody simulation model and an intelligent virtual controller. Based on specific requirements, we formulated the problem, designed the mechanical structure, and determined the working principles. Next, we analyzed the static and dynamic behavior to investigate the reliability of the machine structure. We developed a multibody simulation model of the new machine to represent the actual mechanical system, determining the body relationships, degrees of freedom, joint types, and geometric constraints in an integrative model. Subsequently, we conducted a dynamic simulation to identify the machine tool behavior in processing time. Then, we designed an intelligent controller and integrated it with the mechanical model to construct a virtual prototype of our new machine. A co-simulation was performed to evaluate the head unit trajectory and controller functionality. Finally, we manufactured a physical machine to investigate its accuracy and verify that our design requirements were met.

3. Concept of the roll forming machine

We developed a roll forming machine based on a 1.5-ton straight cutting version that is currently used to manufacture curved parts. For this purpose, the machine has five key systems: the head unit, slide unit, up-down unit, turn unit, and base unit. In the proposed concept, the base, turn, up-down, and slide units are fixed during the processing time. The head unit moves along a specific curved path (the rail) similar to the product profile; straight and curved rails can be used to manufacture straight and curved products, respectively. New product profiles can be generated by changing the rails.

The servo-driven axis, ball screw, and rolling-element linear guide are mounted on the machine slide unit to control the head unit motions. The closed-loop principle is applied to control the feed driving movements. The cutting movement
at the end point of the contour is conducted by a shearing die with the support of a hydraulic cylinder. The developed roll forming machine is a kind of moving cutting mechanism, which not only generates curved parts, but also cuts them accurately to the desired length. This machine can cut any kind of roll-formed product regardless of whether it has a straight or curved profile or what the cross section looks like. As a result, the new cutting machine offers high productivity, ensures stiffness, improves accuracy, and is more flexible compared to traditional ones. The configuration of the roll forming machine is shown in Fig. 3, and Table 1 lists its specifications.
The working principle of the roll forming machine can be divided into the following steps:

Step 1: The head unit is located at the initial position of the rail.
Step 2: During the roll forming process, the head unit moves along the rail to the end position.
Step 3: The formed product is cut by a shearing die to achieve the desired length.
Step 4: The head unit is returned to the initial position on the rail and a new roll forming cycle can be started.

4. Static analysis and dynamic analysis

To investigate the static and dynamic performance, finite element analysis was performed based on the initial design using the ANSYS Workbench platform. The machine model was imported into the simulation environment to define the material properties, the mesh, the boundary, and the working conditions. The objective of the static analysis is to find the displacement and stress distribution under the heaviest load. The small values of the maximum stress (approximately 71 MPa) and deformation (around 0.123 mm) indicated that the machine satisfied the design requirements for strength (Fig. 4).

The dynamic performance of the roll forming machine was solved in terms of modal and harmonic analyses by means of finite element analysis. Modal analysis has become a major technology for determining, improving, and optimizing the dynamic characteristics of engineering structures. A structural mode can be thought of as a shape and a frequency at which the structural shape resonates. Table 2 lists the modal analysis results considering six vibration modes. Fortunately, the natural frequencies were higher than the operating frequency in processing time. There was no resonant vibration and thus it was not necessary to perform harmonic analysis. Consequently, the machine under consideration was safe in terms of its dynamic behavior.

| Mode | 1     | 2     | 3     | 4     | 5     | 6     |
|------|-------|-------|-------|-------|-------|-------|
| Frequency (Hz) | 24.034 | 29.144 | 34.033 | 38.156 | 54.315 | 56.180 |

![Fig. 4. Static analysis results.](image)

5. Multibody system model

5.1 Analytical modeling of a constrained multi-body mechanical system

In the multi-body system (Shabana et al., 2014), the generalized coordinates associated with body $i$ are denoted by
with \( p, \epsilon \) being the position and the orientation of a body, respectively. For the system model containing \( nb \) bodies, the vector \( q \) is used to describe the position and orientation of each body at a given time in the system.

\[
q = [q_1^T \ q_2^T \ \ldots \ q_{nb}^T]^T
\]  

(2)

Joints are regarded as constraints in the multi-body system which are generally obtained as

\[
\Phi = \Phi(q,t) = 0
\]

(3)

The entire constraint equations imposed by the joints are showed as

\[
\Phi(q,t) = [\Phi_1(q,t) \ \Phi_2(q,t) \ \ldots \ \Phi_{nj}(q,t)]^T,
\]

(4)

where \( nj \) is the number of joints in the system. By taking one time derivative of the position kinematic constraint Eq.4, the velocity kinematic constraint equations are expressed as follows

\[
\Phi_q(q,t)\dot{q} = -\Phi_q(q,t).
\]

(5)

By taking one time derivative of the kinematic constraint equations in Eq. 5, the acceleration kinematic constraint equations are obtained as

\[
\Phi_q(q,t)\ddot{q} = -(\Phi_q\ddot{q}) - 2\Phi_q\dot{q} - \Phi_q(q,t) = \gamma.
\]

(6)

The Lagrange multipliers form of the constraints acting on the multi body system can be written as

\[
M\ddot{q} + \Phi_q^T\lambda = F,
\]

(7)

where \( M \) is the generalized mass matrix equivalent to \( \text{diag}[M_1, M_2, \ldots, M_n] \), \( F \) is the system force vector; and \( \lambda \) is the array of Lagrange multipliers. Combining Eqs. 6 and 7, the complete set of constrained equations of motion can be written as follows

\[
\begin{bmatrix}
M & \Phi_q^T \\
\Phi_q & 0
\end{bmatrix}
\begin{bmatrix}
\dot{q} \\
\dot{\lambda}
\end{bmatrix}
=
\begin{bmatrix}
F \\
\gamma
\end{bmatrix}.
\]

(8)

The mentioned principle is widely used in multi-body simulation software to design virtual models and investigate the working behavior of machines. In this paper, the ADAMS.MSC software, which allows the integration of theoretical formulations, was adopted to resolve the industrial issue. The current platform allows designers to identify mechanism functions without using an expensive physical model.

5.2 Modelling the multibody system of the roll forming machine

The framework for developing the multibody system analysis is illustrated in Fig. 5. First, we designed a 3D machine model with component characteristics based on the conceptual design. We imported the machine components into the Adams multibody dynamics software by means of a Parasolid file in the *.x_t format. The driving modes of the joints were then specified for predetermined motions. Finally, we performed a dynamic simulation to evaluate the characteristics of the conceptual design. The machine model used here takes several aspects into account, including gravity, contact constraints, joints, friction, and motion. The coordinate system for the machine model was chosen so that its center was the head unit in order to focus on the position of the head unit during the manufacturing process (Fig. 6). We conducted a dynamic simulation of the model to investigate the machine’s behavior under working conditions. A total of 1000 moving steps and a time interval of 3.5 s were used to increase the accuracy of the simulation results. The head unit position and the velocity during processing are shown in Fig. 7. The forward process of head unit is from the initial position to the end of the rail to determine the exactly position cutting of curved product. After that,
the returning process is moved to the start position for the new cycle manufacturing progress. In the simulation results, the head unit trajectories in the X-axis and Y-axis were reflected correctly in the product shape (Figs. 7a and 7b). Moreover, the head unit velocities during the forward and return movements were 200 and 300 mm/s, respectively. It was evident that the simulation outputs met the machine specifications. Therefore, the developed model can be used to simulate, analyze, and validate machine functionality.

![Flowchart modeling the multibody simulation.](image)

![Multi-body system model of the flexible cutting machine.](image)
6. Development of the control system

6.1. Interaction between the multibody system and the control system

The mechanical structure and control system communicate by means of exchanging state variables. In this work, we developed the control design based on ADAMS Controls and MATLAB Simulink, as depicted in Fig. 8. The input and output signals represent the main servo motor torque and head unit position, respectively. These parameters are first defined in the ADAMS model to generate the state variable. Then, the virtual mechanical model is exported to the MATLAB Simulink environment to complete the interaction (Fig. 9). As mentioned above, the new roll forming machine is considered to be a single-input, multi-output system.

6.2. Controller design

The PID control method is widely applied in linear systems due to its simplicity and effectiveness. This method is insensitive to parameter changes, including the proportional gain $K_p$, integral gain $K_i$, and derivative gain $K_D$. Furthermore, self-tuning controller gains can improve potential applications of the PID control (Tan et al., 2002, Chang et al., 2002). Also, sliding-mode control is a popular strategy to deal with uncertain control systems (Wang et al., 2014, Piltan et al., 2012). The main advantage of the sliding-mode control method is its robustness against parameter variations and external disturbances. Therefore, combining the adaptive sliding-mode control with a PID controller (ASMPID) is
an intelligent choice for updating PID gains and adapting to nonlinear systems (Kuo et al., 2008, Tsai et al., 2011). We propose an ASMPID controller to investigate the head unit position in our flexible cutting machine (Fig. 10).

The flexible cutting machine is considered as a single-input, multiple-output system. The state-space equation describing the head unit position can be expressed as follows:

\[
\begin{align*}
\dot{x}_1(t) &= x_2(t) \\
\dot{x}_2(t) &= f_1(X) + b_1(X)u_x + d_1(t) \\
\dot{x}_3(t) &= x_4(t) \\
\dot{x}_4(t) &= f_2(X) + b_2(X)u_y + d_2(t)
\end{align*}
\]

where \( X = (x_1, x_2, x_3, x_4) \) is a state variable vector representing the position and velocity of the head unit along the X-axis and Y-axis. The terms \( f_1(X), b_1(X), f_2(X), \) and \( b_2(X) \) are nonlinear functions. The terms \( d_1(t) \) and \( d_2(t) \) represent the bounded lumped disturbances, including the system parameter variations and external disturbances. The terms \( u_x \) and \( u_y \) are the control inputs used to control the position of the head unit along the X-axes and Y-axes, respectively.

From (9), the mechanical model has two subsystems: the positions of the head unit in the X-axis and Y-axis. Therefore, the sliding surfaces of the two subsystems are defined as follows:

\[
\begin{align*}
s_x &= \dot{x}_2 + \lambda_x e_x \\
s_y &= \dot{x}_4 + \lambda_y e_y
\end{align*}
\]

In equation (10), \( e_x = x_2 - x \), \( x_d \) is the desired position, \( x \) is the measured position, and \( \lambda \) is a positive constant. In this paper, a control input \( u \) was designed to control the position of the head unit in the X-axes and Y-axes simultaneously. The design methods for these two controllers are similar so we will only present the algorithm used to drive the head unit’s position along the X-axis. Taking the derivative of (10) and substituting \( \dot{x}_2 = \dot{x} \) gives the following:

\[
\dot{s}_x = \ddot{x}_2 - f_1(X) - b_1(X)u_x - d_1(t) + \lambda_x \dot{e}_x.
\]

The control input of the PID controller is designed based on (12) as follows:

\[
u_{PID} = \frac{1}{b_1(X)}[\ddot{x}_2 - f_1(X) - d_1(t) + \lambda_x \dot{e}_x] = AB + c,
\]

where \( A = \begin{bmatrix} K_p & K_f & K_d \end{bmatrix} \) is the vector expressing the gain of the PID controller, \( B = [s \int s \frac{ds}{dt}]^T \) is a basic vector of the PID controller, and \( c \) is an appropriate error term.

The control signal \( u_x \) of the controller is determined as follows:

\[
u_x = u_{PID} + u_b = \hat{A}B + u_b
\]

where \( \hat{A} = [\hat{K}_p \hat{K}_f \hat{K}_d] \) is the estimated value of vector \( A \) and \( u_b \) is the control signal of the auxiliary controller.

Substituting (14) into (12) yields

\[
\dot{s}_x = \ddot{x}_2 - f_1(X) - b_1(X)[\hat{A}B + u_b] - d_1(t) + \lambda_x \dot{e}_x
\]

where \( A = A - \hat{A} \) is the estimation error. To prove stability, Lyapunov’s function can be used:

\[
V = \frac{1}{2} s_x^2 + \frac{1}{2\gamma} A^2.
\]

Taking the derivative of (16) yields:
\[
\dot{V} = s_x s_x + \frac{1}{\gamma} A A \\
= s_x (b_1(X)AB + b_1(X)e - b_1(X)u_x) + \frac{1}{\gamma} A A \\
= (s_x b_1(X)B + \frac{1}{\gamma} A)A + s_x b_1(X)e - b_1(X)u_x s_x \leq 0
\] (17)

From (17), we have \((s_x b_1(X)B + \frac{1}{\gamma} A) = 0\). Hence, \(A = -\dot{A} = -\gamma s_x b_1(X)B\).

The three PID gains \((K_P, K_I, \text{ and } K_D)\) are updated online by using the following adaptive laws:

\[
\begin{align*}
\dot{K}_P &= \gamma s_x b_1(X) s_x, \\
\dot{K}_I &= \gamma s_x b_1(X) \int s_x, \\
\dot{K}_D &= \gamma s_x b_1(X) \frac{ds_x}{dt}
\end{align*}
\] (18)

Considering (17), \(\dot{V} = s_x b_1(X)e - b_1(X)u_x s_x \leq 0\). With the auxiliary controller \(u_h = \eta \text{sgn}(s)\), where the sign function is defined as

\[
\text{sgn}(s) = \begin{cases} 
1 & \text{if } s > 0 \\
0 & \text{if } s = 0 \\
-1 & \text{if } s < 0
\end{cases}
\] (19)

we have the following:

\[
\begin{align*}
\dot{V} &= s_x b_1(X)e - b_1(X)\eta \text{sgn}(s)s_x \\
&= s_x b_1(X)e - b_1(X)\eta |s_x| < b_1(X) |s_x| (|e| - \eta) < 0 \\
\Rightarrow \eta &> |e|
\end{align*}
\] (20)

Equation (17) proves that the sliding surface is stable. The control input for controlling the flexible cutting machine is composed of the position control inputs \(u_x\) and \(u_y\) as follows:

\[u = u_x + u_y.\] (21)

7. Simulation results

We performed a co-simulation integrating the multibody simulation model and the ASMPID controller. The tracking performance of the head unit is shown in Fig. 11. It can be stated that both the PID and ASMPID controllers had an overshoot and settling time of nearly zero, and that the steady state error satisfied the stable control criteria. External disturbances perfectly eliminated very fast responses to overcome the initial error conditions. Furthermore, the error tracking of the ASMPID controller was smaller than that of PID control in both the X- and Y-directions. Consequently, the proposed ASMPID controller is suitable for the roll forming machine and robust regardless of the system parameters and disturbance variations strategy.
8. Experiment

Based on the CAD model and the results described in Section 4 and Section 6, prototype hardware of the roll forming machine was fabricated. The primary components of the base unit, turn, up-down, slide, and head units were manufactured (Fig. 12). The machine controller was implemented with the algorithm proposed in Section 6 (Fig. 13).

The physical prototype was connected with the hydraulic system to carry out manufacturing to allow investigation of the machine accuracy (Fig. 14). Significantly, a high-precision 3D portable coordinate measuring machine FARO laser tracker was used in conjunction with a SMR sensor to record the head unit position during processing. Table 3 lists the measured results; \((X_1, Y_1, Z_1)\) and \((X_2, Y_2, Z_2)\) respectively represent the start and end head unit positions. The head unit considered the intended positions with an accuracy within the allowed tolerance of \(\pm0.1\) mm. The results showed that the proposed ASMPID controller is a robust approach to optimizing the system parameters, and the roll forming machine for curved products adequately met the quality criterion as well as the design specifications (Table 4).
Fig. 12. Prototype mechanical parts.

Fig. 13. Control system.

Fig. 14. Experimental facilities.
Table 3. Coordinate data of the head unit during the manufacturing process

| No. | X₁ (mm) | Y₁ (mm) | Z₁ (mm) | X₂ (mm) | Y₂ (mm) | Z₂ (mm) |
|-----|---------|---------|---------|---------|---------|---------|
| 1   | 2290.197 | 504.757 | −398.440 | 1977.760 | 525.385 | −399.181 |
| 2   | 2290.199 | 504.758 | −398.439 | 1977.849 | 526.378 | −399.208 |
| 3   | 2290.202 | 504.754 | −398.440 | 1977.862 | 526.381 | −399.207 |
| 4   | 2290.203 | 504.754 | −398.442 | 1977.947 | 526.364 | −399.208 |
| 5   | 2290.208 | 504.756 | −398.442 | 1978.083 | 525.344 | −399.212 |
| 6   | 2290.207 | 504.755 | −398.438 | 1978.112 | 525.344 | −399.212 |
| 7   | 2290.209 | 504.757 | −398.435 | 1978.174 | 526.335 | −399.217 |
| 8   | 2290.211 | 504.758 | −398.441 | 1978.241 | 526.326 | −399.220 |
| 9   | 2290.209 | 504.756 | −398.435 | 1978.269 | 525.332 | −399.236 |
| 10  | 2290.209 | 504.755 | −398.435 | 1978.269 | 526.330 | −399.219 |
| 11  | 2290.249 | 504.763 | −398.447 | 1978.400 | 526.315 | −399.222 |
| 12  | 2290.251 | 504.760 | −398.445 | 1978.440 | 526.308 | −399.235 |
| 13  | 2290.246 | 504.769 | −398.443 | 1978.509 | 526.300 | −399.245 |
| 14  | 2290.243 | 504.755 | −398.448 | 1978.499 | 526.301 | −399.240 |
| 15  | 2290.242 | 504.754 | −398.447 | 1978.518 | 526.299 | −399.243 |
| 16  | 2290.241 | 504.751 | −398.449 | 1978.581 | 526.294 | −399.255 |
| 17  | 2290.240 | 504.750 | −398.449 | 1978.594 | 526.292 | −399.259 |
| 18  | 2290.247 | 504.748 | −398.448 | 1978.562 | 526.292 | −399.265 |
| 19  | 2290.244 | 504.759 | −398.449 | 1978.632 | 526.281 | −399.267 |
| 20  | 2290.247 | 504.752 | −398.452 | 1978.760 | 526.271 | −399.271 |
| 21  | 2290.247 | 504.754 | −398.450 | 1978.767 | 526.264 | −399.277 |
| 22  | 2290.244 | 504.759 | −398.451 | 1978.586 | 526.277 | −399.275 |
| 23  | 2290.249 | 504.755 | −398.449 | 1978.530 | 526.281 | −399.272 |
| 24  | 2290.250 | 504.760 | −398.454 | 1978.562 | 526.279 | −399.277 |
| 25  | 2290.249 | 504.758 | −398.451 | 1978.647 | 525.276 | −399.278 |
| 26  | 2290.249 | 504.758 | −398.453 | 1978.770 | 525.268 | −399.287 |
| 27  | 2290.250 | 504.753 | −398.454 | 1978.710 | 525.277 | −399.283 |
| 28  | 2290.252 | 504.756 | −398.450 | 1978.747 | 526.273 | −399.291 |
| 29  | 2290.252 | 504.786 | −398.450 | 1978.698 | 526.277 | −399.281 |
| 30  | 2290.572 | 504.796 | −398.450 | 1978.830 | 526.260 | −399.288 |

| RMS error | 0.077 | 0.059 | 0.046 | 0.073 | 0.036 | 0.056 |

Table 4. Experimental confirmation

| Performance specifications | Unit | Desired values in the machine | Measured values | Standard |
|---------------------------|------|-------------------------------|----------------|----------|
| Forward speed             | mm/s | 200                           | 200            | Test report |
| Return speed              | mm/s | 300                           | 300            | Test report |
| Product tolerance         | mm   | ±2                            | −0.524         | KS B ISO 9283:2001 |
| Repeatability             | mm   | ±1                            | 0.855          | KS B ISO 9283:2001 |

9. Conclusions

In summary, a particular approach toward developing a new roll forming machine for curved products was presented based upon multi-body simulation and the implementation of modern control technology. A multi-body simulation model was developed to investigate the machine’s dynamic behavior during processing. An adaptive sliding-mode PID-based controller (ASMPID) was proposed and integrated into the synthesis model. Co-simulation was performed to investigate
the interaction between the mechanical structure and the controller and to determine optimal control parameters for eliminating system disturbances. A prototype machine was implemented to evaluate the machine tool accuracy. The following conclusions can be drawn from this investigation.

1. We developed a new roll forming machine using a servo motor, a ball screw, and a rolling-element linear guide to manufacture curved parts. In the proposed machine, the curved products are cut at the end of the roll forming process.

2. Simulation results indicate that the virtual prototyping model is safe in terms of dynamic behavior, satisfies the specifications and the strength design requirements, and has stable control criteria.

3. The ASMPID controller effectively eliminates external disturbances in this nonlinear system during the roll forming process.

4. The experimental results show that the new roll forming machine can eliminate the disadvantages of a traditional machine and increase the product quality.

5. The results indicated that the proposed approach using SOLIDWORK, ANSYS, ADAMS and MATLAB can be considered as a powerful method for machine tool development in the design stage. Furthermore, the current method could help manufacturing engineers and researchers identify the working behaviors of multi-body systems without resorting to time-consuming and costly experiments.

Unfortunately, the hydraulic system has many disadvantages such as noisy operation, high energy consumption, low productivity, and large space required for installation. Therefore, use of the proposed machine should consider these drawbacks.

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