

Xiangwei Liu

Professor
Shanghai Maritime University
Logistics Engineering College, China

Changyu Wang

Postgraduate Student
Shanghai Maritime University
Logistics Engineering College
China

Bing Zhao

Engineer
Shandong Port Rizhao Port Group Co., Ltd
China

Yandi Wang

Engineer
Shandong Port Rizhao Port Group Co., Ltd
China

Jingui Luo

Engineer
Shanghai Maritime University
Logistics Engineering College
China

Le Kuai

Instructor
Shanghai Maritime University
Logistics Engineering College
China

Luigi Solazzi

Professor
University of Brescia
Department of Mechanical and Industrial
Engineering
Italy

Nenad Zrnic

Professor
University of Belgrade, Faculty of
Mechanical Engineering
Serbia

A Passive Mechanical Anti-Sway System for Automated Container Transfer Cranes Using Spiral Torsion Springs: Modeling, Simulation and Experimental Validation

Container sway limits positioning accuracy, safety, and handling efficiency in automated container terminals. Existing anti-sway solutions often involve complex structures, high maintenance costs, or limited robustness under varying loads. This study proposes a passive anti-sway system using spiral torsion spring modules and clutch-assisted intermittent engagement. The system converts sway-induced kinetic energy into elastic strain energy and dissipates residual energy through a braking unit, avoiding continuous active control. A coupled crane-payload dynamic model is established using the Lagrange method, and the spring modules are designed according to engineering standards. Multibody simulations under rated-load and no-load conditions show that the system reduces unilateral sway amplitude to below 100 mm within 2.5 oscillation cycles under rated load. Prototype tests confirm an approximately linear relationship between spring rotation angle and restoring torque, with a 15.30% average deviation from theoretical predictions. The system provides an energy-efficient, low-maintenance solution for short-lift container handling equipment.

Keywords: Anti-sway control; Spiral torsion spring; Container crane; Mechanical damping system; Multibody dynamics.

1. INTRODUCTION

During container crane operations, the swaying spreader acts as a moving payload that induces time-varying inertial forces on the crane structure, potentially exciting structural vibrations, amplifying dynamic stresses, and consequently degrading positioning accuracy and compromising operational safety [1-3]. With the advancement of container terminal automation, the working efficiency of automated cranes has been further improved, which has put forward new requirements for the anti-sway control of crane spreaders. Existing anti-sway technologies for crane spreaders cover multiple dimensions, including structural optimization, auxiliary devices, and control systems [4]. They mainly include solutions such as cross-winding of lifting wire ropes, eight-rope systems, mechanical and hydraulic devices,

electronic control, and rigid guide frames. Among them, cross-winding of lifting wire ropes adjusts the tension and anti-sway torque by increasing the angle between the wire rope and the horizontal plane [5], but it will significantly increase the bending frequency of the wire rope, which will easily lead to fatigue damage of the wire rope and shorten its service life. The eight-rope anti-sway system is designed based on the set principle of triangular stability [6]. This method replaces the pulley structure with wire rope fixed point control. Although it reduces pulley interference, it increases the complexity of the structure, and the anti-sway performance is greatly affected by the lifting height [7,8]. Mechanical anti-sway devices achieve tension control by winding and unwinding the wire rope through a motor. They have a good anti-sway effect during operation, but they have problems such as slow response, poor adaptability, and high maintenance costs. Hydraulic anti-sway devices dynamically adjust the tension of the wire rope through a hydraulic buffer to maintain system balance [9]. Although they can effectively suppress the swaying of the spreader, the anti-sway effect is greatly affected by the lifting weight and the

Received: August 2026 Accepted: October 2026

Correspondence to: Dr Xiangwei Liu
Logistics Engineering College, 1550 Haigang Avenue,
Pudong New Area, Shanghai, China.

E-mail: xwliu@shmtu.edu.cn

doi: 10.5937/fme2604731L

© Faculty of Mechanical Engineering, Belgrade. Allrights reserved

FME Transactions (2026) 54, 731-740 731

arrangement of the hydraulic system and pulleys. Therefore, the hydraulic anti-sway device has poor applicability to different working conditions [10]. Electronic anti-sway systems use sensors to monitor the status of the spreader in real time and actively correct the deviation of the spreader through a motor control algorithm. They have the advantages of fast response and high control accuracy, but they have high requirements for system integration and algorithm stability [11,12]. Rigid guide frame anti-sway uses physical structure to forcibly restrict the degree of freedom of the spreader's movement, suppressing swaying from the source. The anti-sway effect is significant and is not affected by the lifting height, but it has problems such as bulky structure, poor adaptability, and limited installation space. Solihin [13] proposed a fuzzy tuned PID control method, which effectively enhances the robustness of traditional PID control in nonlinear systems by introducing fuzzy logic to adjust the PID parameters. Wang [14] constructed a novel global equivalent sliding mode controller for bridge crane systems and verified its effectiveness through simulation. Dongtian[15] designed a control method to suppress the nonlinear resonance of wire ropes based on the motor output torque compensation mechanism, which effectively reduced the interference of wire rope swaying on the performance of the mechanical anti-sway system. Zhong [16] proposed a passive anti-sway system for hydraulic container cranes based on structural parameter optimization. The results showed that the anti-sway effect of the system was best under critical damping conditions. Dong [17] proposed a hierarchical sliding mode control method based on the Lagrange dynamics model, which effectively suppressed the swaying of container gantry cranes during transportation and achieved a significant improvement in high-precision trajectory tracking and system robustness. Zhang [18] proposed a crane positioning anti-sway control method based on fuzzy sliding mode control, which significantly reduced the dependence on the accurate system model, so that the crane positioning and anti-sway control still showed good robustness and adaptability in complex environments. Zrnić et al. [19] reviewed the automation technologies and anti-sway systems of ship-to-shore container cranes, pointing out that the increase in crane operating speed and automation level makes spreader sway suppression more critical for operational safety and efficiency. Zrnić et al. [20] further analyzed the moving-load problem in crane structural dynamics and demonstrated that the moving oscillator model can effectively describe the dynamic interaction among the trolley, payload, and crane structure. In addition, Zrnić et al. [21] established a dynamic model of a gantry crane subjected to a moving oscillator and showed that trolley speed, acceleration, and structural damping significantly affect the vibration response and sway characteristics of the crane system.

To address the above problems, a mechanical damping anti-sway system using spiral torsion springs is proposed. This system absorbs the kinetic energy of the spreader's swing through a spiral torsion spring buffer box, suppressing container sway and achieving rapid sway control. A dynamic model of the mechanical dam-

ping anti-sway system is established, and the feasibility and effectiveness of the proposed anti-sway system are verified through multibody dynamics simulation. The linear relationship between the number of spiral torsion spring rotations and the output reverse torque is experimentally verified. Compared with traditional active control systems, this system effectively reduces the adverse effects of lifting height changes on anti-sway performance by adjusting the number of spiral torsion spring buffer box rotations, improves the positioning accuracy of the spreader, simplifies the anti-sway control strategy, and reduces operating energy consumption.

2. MECHANICAL STRUCTURE OF SPIRAL TORSION SPRING DAMPING ANTI-SWAY SYSTEM

In the design of container transfer cranes, the spreader anti-sway system is a crucial element in ensuring smooth transfer, efficient loading and unloading, and operational safety. Considering the characteristics of transfer cranes small lifting capacity and low lifting height this paper designs a mechanical damping anti-sway system as shown in Figure 1(a), which mainly consists of a steel structure, a mechanical damping system, and a mechanical anti-sway system. This anti-sway mechanism is based on the principle of passive damping, controlling the retraction and extension of the spiral torsion spring to dissipate the kinetic energy of the spreader, thereby suppressing spreader swaying.

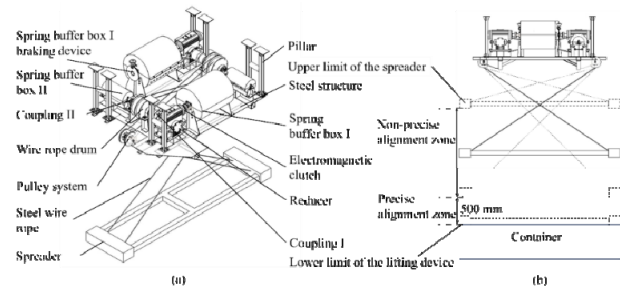


Figure 1. Mechanical damping anti-sway system and operating range. (a) Schematic diagram of mechanical damping anti-sway system, (b) Schematic diagram of operating range

The mechanical damping system mainly consists of a spring buffer box I, a reducer, and a spring buffer box II. Spring buffer box I provides anti-sway torque to ensure smooth loading and unloading of the crane. A braking device is installed on spring buffer box I to dissipate the kinetic energy absorbed by the spring. Spring buffer box II prevents the anti-sway wire rope from slackening when spring buffer box I is disengaged. The reducer reduces the input torque of the mechanical damping system and amplifies the output torque. The mechanical anti-sway system mainly consists of a wire rope drum, wire rope, and pulley block. To improve the energy efficiency and control accuracy of the mechanical damping anti-sway system, the vertical working space is divided into a precise alignment zone and a non-precise alignment zone based on the distance between the spreader and the transfer truck, as shown in Figure 1(b). The precise alignment zone refers to the space within 500 mm above the lower limit of the spreader. Within this zone, spring buffer box I is always

connected to the mechanical anti-sway system, and the anti-sway force continuously increases as the spreader de-scends, thereby significantly suppressing end-effector sway and improving positioning accuracy. The non-precise alignment area is the space between 500 mm above the lower limit and the upper limit of the spreader. In this area, the spiral torsion spring buffer box I intermittently engages the mechanical anti-sway system according to the swaying of the spreader, thereby achieving the purpose of dissipating kinetic energy.

3. PRINCIPLE OF SPIRAL TORSION SPRING DAMPING ANTI-SWAY SYSTEM

When the spreader is in a non-precise alignment area and undergoes downward movement, the spiral torsion spring buffer box II rotates synchronously with the wire rope drum and stores elastic potential energy. This stored potential energy is used to maintain the tension of the wire rope, preventing slack. The spiral torsion spring buffer box I intermittently engages or disengages from the mechanical anti-sway system according to the spreader's swing state, as shown in Figure 2(a) and 2(b). When the spreader begins to swing to the left, the left electromagnetic clutch engages, and the left spiral torsion spring buffer box I engages the mechanical anti-sway system, absorbing the spreader's swing kinetic energy and forming reverse damping. When the spreader swings to its highest point on the left, the left electromagnetic clutch disengages, and the left spiral torsion spring buffer box I disengages from the mechanical anti-sway system. At this time, the braking device of the left spiral torsion spring buffer box I intermittently engages, dissipating the absorbed kinetic energy through frictional damping.

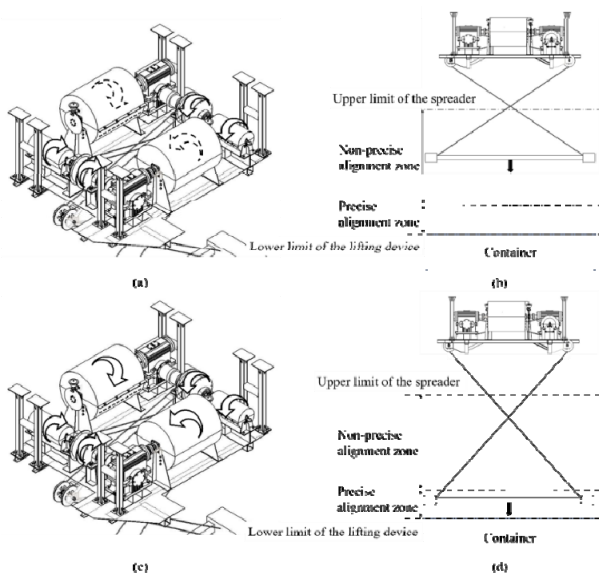


Figure 2. Descending phase. (a) Schematic diagram of spring steering during descending phase 1, (b) Descending of the spreader in the non-precise alignment area during descending phase 1, (c) Schematic diagram of spring steering during descending phase 2, (d) Descending of the spreader in the non-precise alignment area during descending phase 2.

When the spreader begins to swing to the right, the mechanical anti-sway system operates in the same mode

as on the left. The two sets of spiral torsion springs operate alternately, forming a periodic and symmetrical anti-sway control process until the spreader achieves stable downward movement. When the spreader is in the precise alignment zone and begins its downward movement, as shown in Figure 2(c) and 2(d), the electromagnetic clutch remains closed, and both the spiral torsion spring buffer box I and the spiral torsion spring buffer box II rotate synchronously with the wire rope drum. When the spreader descends to its lower limit, the spiral torsion spring buffer box I absorbs the maximum kinetic energy.

When the spreader is in the precise alignment zone and undergoing upward movement, as shown in Figure 3(a) and 3(b), the electromagnetic clutch remains closed, and both the spiral torsion spring buffer box I and spiral torsion spring buffer box II rotate synchronously with the wire rope drum. When the spreader moves upward away from the precise alignment zone, the spiral torsion spring buffer box I completely re-releases the absorbed kinetic energy. When the spreader is in the non-precise alignment zone and undergoing upward movement, its anti-sway mechanism is similar to that during the downward movement phase in the non-precise alignment zone, as shown in Figure 3(c) and 3(d). When the spreader begins to swing to the left, the left electromagnetic clutch closes, the left spiral torsion spring buffer box I engages the mechanical anti-sway system, and begins to absorb the swing kinetic energy of the spreader. When the spreader swings to its highest point on the left, the left electromagnetic clutch disengages, the left spiral torsion spring buffer box I disengages from the mechanical anti-sway system, and the braking device of the left spiral torsion spring buffer box I intermittently closes to dissipate the kinetic energy absorbed by the spiral torsion spring buffer box I. When the spreader begins to swing to the right, the working mode of the mechanical anti-sway system is the same as on the left. The spring buffer boxes on both sides work in cycles until the spreader is lifted stably.

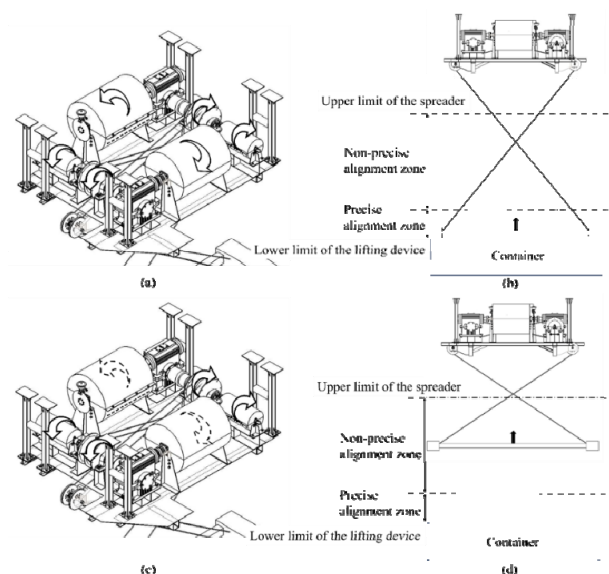


Figure 3. Mechanical damping anti-sway system and operating range. (a) Schematic diagram of mechanical damping anti-sway system, (b) Schematic diagram of operating range.

4. DESIGN CALCULATION AND DYNAMIC MODELING OF SPIRAL TORSION SPRING ANTI-SWAY STRUCTURE

4.1 Design Calculation

Spiral torsion springs can be divided into two types: contact type and non-contact type [22]. Contact type spiral torsion springs have adjacent coils in contact during operation, resulting in a higher effective working number of coils and thus a higher energy storage capacity. However, their characteristic curve changes nonlinearly with the increase in the number of rotations, so they are mainly used in energy storage applications. In contrast, non-contact type spiral torsion springs maintain a gap between coils during operation, and their characteristic curve is approximately linear, providing stable reverse torque and suitable for control systems requiring linear recovery characteristics [23],[24]. Therefore, this paper selects non-contact type spiral torsion springs as the core component of the crane lifting anti-sway system.

The overall structure of the spiral torsion spring is shown in Figure 4(a). The spiral torsion spring mainly consists of a spiral torsion spring shell, a spindle, and the spiral torsion spring itself. The inner and outer ends of the spiral torsion spring are fixed to the spindle and the spiral torsion spring buffer shell respectively by bolts, as shown in Figure 4(b). When the spiral torsion spring buffer is working, the inner end of the spiral torsion spring rotates with the spindle, converting rotational kinetic energy into spiral torsion spring elastic potential energy, thereby achieving the purpose of kinetic energy conversion. Based on the application of the spiral torsion spring buffer box, spiral torsion spring buffers are divided into spiral torsion spring buffer I and spiral torsion spring buffer II. The spiral torsion spring buffer I and spiral torsion spring buffer box I are used to absorb and dissipate the kinetic energy of the spreader and provide anti-sway torque for the spreader. The spiral torsion spring buffer II is used to prevent the anti-sway wire rope from slackening during operation.

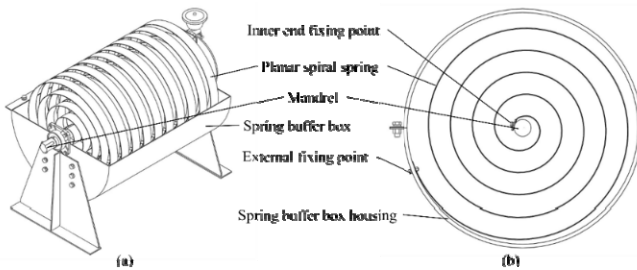


Figure 4. Spring buffer. (a) Structural diagram, (b) Installation diagram.

The calculation process for the main design parameters of the spiral torsion spring is as follows: First, considering the overall size of the spiral torsion spring buffer, the total torque (T_1), etc., the torque (T_{ii}) of a single spiral torsion spring in the spiral torsion spring buffer is calculated. Among them, the total torque (T_1) of spiral torsion spring buffer I is calculated based on the weight of the spreader, the weight of the load, and the diameter of the wire rope drum, as shown in Eq. (1).

$$T_1 = f_1 f_2 FL \quad (1)$$

where L is the radius of the wire rope drum, mm; f_1 is the anti-sway component coefficient; f_2 is the transmission ratio of the reducer; F is the total tension on the wire rope on a single drum, N , as shown in Eq. (2).

$$F = \frac{P_H}{2 \cdot \cos \alpha} \quad (2)$$

where P_H is the horizontal inertial force when the container moves, N ; α is the preset maximum value of the angle between the wire rope and the plane of the spreader when the spreader descends to the lowest point, °. The P_H is defined in Eq. (3).

$$P_H = 1.5(m + m_0) \times a_H \quad (3)$$

where m is the weight of the crane spreader, kg; m_0 is the weight of the container being loaded or unloaded, kg; a_H is the acceleration of the container's horizontal movement, m/s^2 .

The total torque of the spring buffer II (T_2) is calculated based on the weight of the spreader, the load weight, the diameter of the wire rope drum, and the number of working coils of the spring, as shown in Eq. (4).

The calculation process for T_2 is as follows:

$$T_2 = m_1 g L \quad (4)$$

where m_1 is the weight of the wire rope, kg.

The specific calculation process is as follows: According to the relevant provisions of the standard "Design Calculation of Planar Spiral Spring" (JB/T 7366-94) [25], and combined with the calculation results of Eq. (1)-(4), the key geometric parameters of the spiral torsion spring are further solved, including the thickness of the spiral torsion spring steel plate h_i , the working coil unfolding length of the spiral torsion spring material l_i , the inner coil radius $R_{i(i+1)}$, the outer coil radius R_{ii} , the pitch t_i and the number of free state coils n_i . The specific calculation formulas of the above parameters are shown in Eq. (5)-(10),

$$h_i = \sqrt{\frac{6K_2 T_{ii}}{b_i [\sigma]}} \quad (5)$$

where h_i is the cross-sectional dimension of the spiral torsion spring material in spiral torsion spring buffer i , mm; K_2 is the fitting coefficient, K_2 is 1 when the outer end is fixed, and K_2 is 2 when the outer end rotates; T_{ii} is the spiral torsion spring torque in spiral torsion spring buffer i , $N \cdot mm$; b_i is the width of the spiral torsion spring material in spiral torsion spring buffer i , mm; $[\sigma]$ is the allowable stress, $[\sigma] = 0.6 \sigma_b$, N/mm^2 ; σ_b is the tensile strength limit of the material, N/mm^2 .

$$l_i = \frac{Eb_i h_i^3 \psi}{12K_1 T_{ii}} \quad (6)$$

where l_i is the unfolded length of the working coil of the spiral torsion spring material in spiral torsion spring buffer i , mm; E is the elastic modulus of the material, N/mm^2 ; ψ is the deformation angle of the spiral torsion

spring under the action of torque T , rad; K_1 is the fitting coefficient, K_1 is 1 when the outer end is fixed, and K_1 is 1.25 when the outer end rotates.

$$R_{i(i+1)} = (8 \sim 15) h_i \quad (7)$$

where $R_{i(i+1)}$ is the inner radius of the spiral torsion spring in spiral torsion spring buffer i , mm.

$$R_{ii} = \frac{2l_i}{\psi} R_{i(i+1)} \quad (8)$$

where R_{ii} is the outer radius of the spiral torsion spring in spiral torsion spring buffer i , mm.

$$t_i = \frac{\pi(R_{ii}^2 - R_{i(i+1)}^2)}{l_i} \quad (9)$$

where t_i is the spring pitch in spring buffer i , mm.

$$n_i = \frac{R_{ii} - R_{i(i+1)}}{l_i} \quad (10)$$

where n_i represents the number of free coils of the spiral spring in spiral torsion spring buffer i .

Taking a 50t load as an example, **Table 1** lists the main technical parameters of the mechanical damping anti-sway system.

Table 1. Main Technical Parameters.

Parameter	Numeric al values	Unit
Load capacity	50	t
Horizontal inertial force of containers	15	kN
Wire rope drum diameter	500	mm
Total torque of spiral torsion spring buffer I	750	N·m
Total torque of spiral torsion spring buffer II	48	N·m
Number of spiral torsion springs I	10	/
Number of spiral torsion springs II	5	/
Thickness of spiral torsion spring I	2.5	mm
Thickness of spiral torsion spring II	1	mm
Width of spiral torsion spring I	75	mm
Width of spiral torsion spring II	60	mm
Material unfolded length of spiral torsion spring I	7695	mm
Material unfolded length of spiral torsion spring II	8800	mm
Inner radius of spiral torsion spring I	37.5	mm
Inner radius of spiral torsion spring II	15	mm
Outer radius of spiral torsion spring I	473	mm
Outer radius of spiral torsion spring II	191	mm
The pitch of spiral torsion spring I	91	mm
The pitch of spiral torsion spring II	11	mm

spring II

Number of working turns of spiral torsion spring I	3	/
Number of working turns of spiral torsion spring II	13.5	/
Speed ratio of the reducer	10	/
Allowable torque of electromagnetic clutch	1000	N·m

4.2 Anti-sway system dynamic equations

The dynamic model of the mechanical damping anti-sway system mainly consists of a trolley, a mechanical damping anti-sway platform, lifting wire ropes, anti-sway wire ropes, and a container, as shown in Figure 5

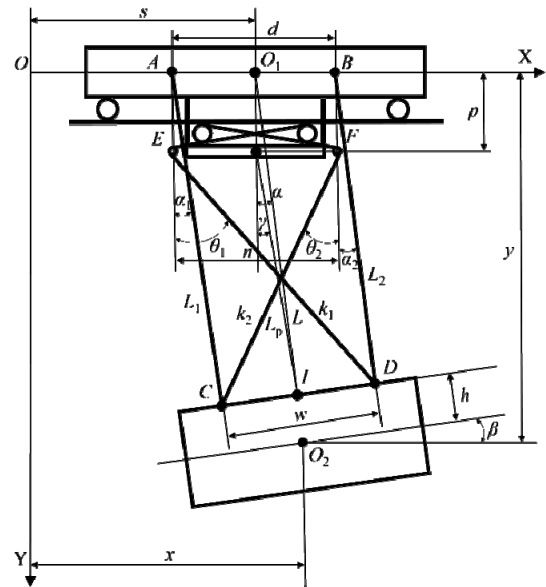


Figure 5. Dynamic model of mechanical damping anti-sway system.

where d is the horizontal distance between the lifting pulleys on the trolley and between the redirecting pulleys on the anti-sway platform; p is the vertical distance between the anti-sway platform and the trolley; α and γ are the swing angles of the container relative to the trolley and the anti-sway platform, respectively; α_1 and α_2 are the angles between lifting wire ropes 1 and 2 and the vertical direction, respectively; θ_1 and θ_2 are the angles between anti-sway wire ropes 1 and 2 and the vertical direction, respectively; β is the angle of the container relative to the horizontal direction; x represents the horizontal position of the container's center of mass; y is the vertical position of the container's center of mass; and L_1 and L_2 are the vertical positions of lifting wire ropes 1 and 2, respectively. The lengths of the anti-sway wire ropes are given by K_1 and K_2 respectively. L and L_p are the vertical distances between the trolley and the anti-sway platform and the container in the stationary state. δ , δ_1 and δ_2 represent the changes in the vertical distance between the trolley and the container and the elongation of the lifting wire ropes 1 and 2, respectively, during the lifting analysis. During the anti-sway analysis, they represent the changes in the vertical distance between the anti-sway platform and the container and the elongation of the anti-sway wire ropes 1 and 2, respectively. w is the

horizontal distance between the lifting pulley and the anti-sway pulley on the container, and h is the distance from the centroid of the container to the bottom of the container.

In Figure 5, the arrows for X and Y form squares in the horizontal and vertical directions, respectively. The counterclockwise direction of a , a_1 , a_2 , γ , θ and β is the positive direction, and the clockwise direction of θ_2 forms a square. Based on the geometric relationships of the pulley-spreader system, the following kinematic constraints can be derived, as presented in Eq. (11)-(14).

$$L_i \cdot \sin a_i = L \sin a + (-1)^i \frac{w}{2} \cos \theta - (-1)^i \frac{d}{2}, (i=1,2) \quad (11)$$

$$L_i \cdot \cos a_i = L \cos a + (-1)^i \frac{w}{2} \sin \theta, (i=1,2) \quad (12)$$

$$K_i \sin \theta_i = d - (-1)^i L \sin a + \frac{w}{2} \cos \beta - \frac{d}{2}, (i=1,2) \quad (13)$$

$$K \cos \theta_i = L \cos a - p + (-1)^i \frac{w}{2} \sin \beta, (i=1,2) \quad (14)$$

Find the sum of squares of Eq. (11) and (12), as well as Eq. (13) and (14), as shown in Eq. (15) and (16).

$$L_i^2 = \left(L \sin a + (-1)^i \frac{w}{2} \cos \theta - (-1)^i \frac{d}{2} \right)^2 + \left(L \cos a - (-1)^i \frac{w}{2} \sin \theta \right)^2, (i=1,2) \quad (15)$$

$$K_i^2 = \left(d - (-1)^i L \sin a + \frac{w}{2} \cos \beta - \frac{d}{2} \right)^2 + \left(L \cos a - p + (-1)^i \frac{w}{2} \sin \beta \right)^2, (i=1,2) \quad (16)$$

Considering the elastic deformation of the wire rope, the length relationships are given by Eq. (17)-(19).

$$L_i = L_0 + \delta_i, (i=1,2) \quad (17)$$

$$K_i = K_0 + \delta_i, (i=1,2) \quad (18)$$

$$L = L_0 + \delta \quad (19)$$

Substituting Eq. (17) and (19) into Eq. (15), and Eq. (18) and (19) into Eq. (16), the elongation expressions are obtained as Eq. (20)-(22).

$$\delta_i = \frac{w^2 t^2}{8L_0} + (-1)^{i+1} \frac{w}{2} (ta + \theta) + \delta_i, (i=1,2) \quad (20)$$

$$\delta_i = \frac{w^2 t^2}{8L_0} + (-1)^{i+1} \frac{w}{2} \left[a(t+2) - \beta \left(1 - \frac{p}{L_0 + \delta} \right) \right] + \delta, (i=1,2) \quad (21)$$

$$\frac{d-w}{w} \quad (22)$$

The kinetic and potential energy of the container are shown in Eq. (23) and (24).

$$E_k = \frac{1}{2} m (\dot{x}^2 + \dot{y}^2) + \frac{1}{2} J \dot{\beta}^2 \quad (23)$$

$$E_p = mgy + \frac{1}{2} k (\delta_1^2 + \delta_2^2) \quad (24)$$

The horizontal and vertical coordinates of the container centroid are expressed in Eq. (25) and (26).

$$x = s + L \sin a + h \sin \beta \quad (25)$$

$$y = L \cos a + h \cos \beta \quad (26)$$

If we define the generalized coordinates of this dynamic system as $x = [\delta \ \theta \ \beta]^T$, then according to the Lagrange equations, the mass matrix and stiffness matrix of the system are shown in Eq. (27) and (28), respectively:

$$M = \begin{bmatrix} m & 0 & 0 \\ 0 & mL_0^2 & mhL_0 \\ 0 & mhL_0 & mh^2 + J \end{bmatrix} \quad (27)$$

$$K = \begin{bmatrix} 2k & 0 & 0 \\ 0 & mgL_0 + \frac{1}{2} kw^2 t^2 & \frac{1}{2} kw^2 t \\ 0 & \frac{1}{2} kw^2 t & mgh + \frac{1}{2} kw^2 \end{bmatrix} \quad (28)$$

Meanwhile, the external force applied to the container in this system can be defined as shown in Eq. (29) and (30):

$$f = \begin{bmatrix} mg \times \frac{kw^2 t^2}{4L_0} - mL_0 \ddot{s} - mh \ddot{s} \end{bmatrix} \quad (29)$$

$$M\ddot{x} + Kx = f \quad (30)$$

In summary, the dynamic equation of the mechanical damping anti-sway system is shown in Eq (31).

$$M \begin{bmatrix} \ddot{\delta} \\ \ddot{\theta} \\ \ddot{\beta} \end{bmatrix} + K \begin{bmatrix} \delta \\ \theta \\ \beta \end{bmatrix} = f \quad (31)$$

5. DYNAMICS SIMULATION AND EXPERIMENTAL VERIFICATION OF ANTI-SWAY SYSTEM

5.1 DYNAMICS SIMULATION

To preliminarily verify the feasibility of the proposed mechanical damping anti-sway system, a simulation model of the system was established in the dynamic simulation software Recurdyn, as shown in Figure 6. It mainly includes: a trolley, an anti-sway platform, lifting wire ropes, anti-sway wire ropes, a spiral torsion spring buffer box, and a container. To shorten the simulation time, the operation of the crane trolley was simplified. In the simulation, the trolley was set to be stationary, and an initial horizontal velocity of 1.2 m/s was applied to the container. The unilateral sway amplitude of the container under full load (50 t) and empty load (13 t) conditions was recorded, as shown in Figure 7 and Figure 8.

As shown in Figure 7, under the conditions of design torque and 1.2 and 1.5 times the design torque, the sway amplitude of the container can be reduced to within 100 mm within 2.5 cycles, meeting safety regulations. In contrast, when the torque is reduced to 0.8 times the design value, the time required for the sway amplitude

to decrease to within 100 mm is extended to approximately 3 cycles, failing to meet safety regulations.

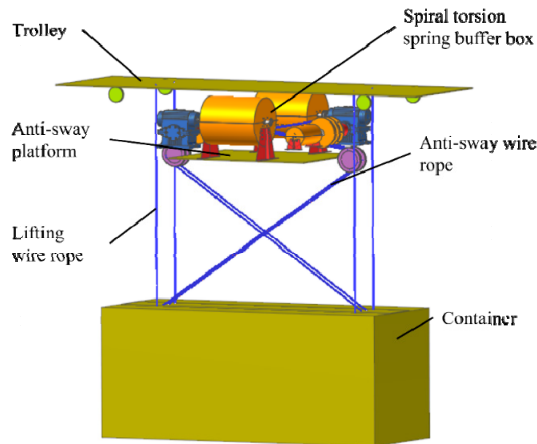


Figure 6. Dynamic simulation model.

Simultaneously, with the increase of the anti-sway torque, the maximum unilateral sway amplitude of the container decreases from 604 mm to 538 mm, improving anti-sway performance. Therefore, in the design of a mechanical damping anti-sway system, it should be ensured that the anti-sway torque is not lower than the specified value. Furthermore, the anti-sway torque can be appropriately increased according to specific operating conditions to improve the robustness of the system. However, excessive torque may cause abrupt rope tension variation and local impact loading, which could reduce mechanical durability. Therefore, torque optimization rather than simple torque maximization is recommended.

The improved anti-sway performance with increasing spring torque is mainly attributed to the enhancement of the restoring moment transmitted through the anti-sway wire rope system. A larger spring torque increases the lateral constraint force acting on the spreader, thereby accelerating the conversion of sway-induced kinetic energy into elastic strain energy stored in the torsion spring. Meanwhile, the braking mechanism dissipates part of the stored energy, resulting in faster attenuation of oscillatory motion.

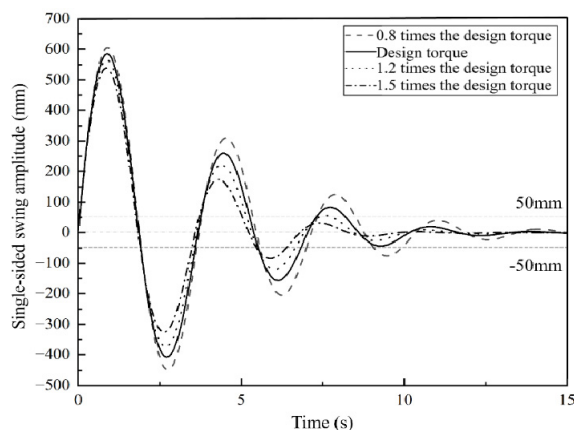


Figure 7. Single-sided sway amplitude of a fully loaded container under different anti-sway torques.

As shown in Figure 8, under the action of the design torque and 0.3 and 0.4 times the design torque, the container sway can decay to within 100 mm within 2.5 cycles, meeting safety regulations. In contrast, when the

torque drops to 0.2 times the design value, the time required for the sway to decay to within 100 mm increases to approximately 4.5 cycles, failing to meet safety regulations. Simultaneously, with the increase of the anti-sway torque, the maximum sway of the container on one side decreases from 487 mm to 305 mm, improving anti-sway performance. Since the crane load under no-load conditions is approximately 26% of the full-load load, and applying 0.3 times the design torque is sufficient to meet the minimum safety limit requirements, the anti-sway torque can be adjusted according to the actual load during system design and operation to reduce energy consumption and improve operational efficiency while ensuring operational safety. Under no-load conditions, both the translational inertia and rotational inertia of the payload system are significantly reduced. Therefore, the resisting torque required to suppress sway motion decreases accordingly. This explains why only 0.3 times the rated torque is sufficient to achieve the same attenuation target under light-load operation.

In addition, applying excessive torque under no-load conditions may over-constrain the payload motion and induce secondary oscillation near the equilibrium position [26]. Hence, adaptive torque regulation is beneficial for both energy efficiency and dynamic stability.

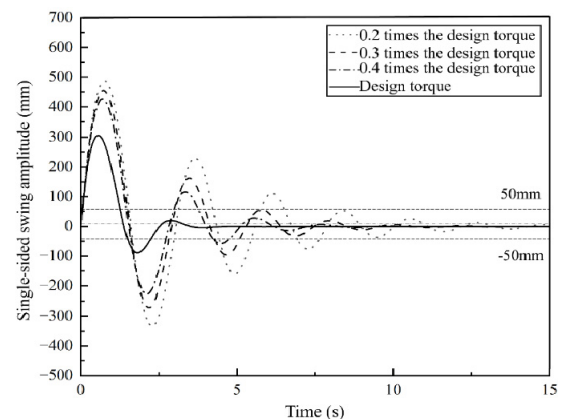


Figure 8. Single-sided sway amplitude of empty container under different anti-sway torques.

5.2 Experimental verification

To verify the quantitative relationship between the number of rotations of the spiral torsion spring and the reverse torque it generates, the experimental setup shown in Figure 9 was designed. This setup simplifies the mechanical damping anti-sway system while retaining key functional components, keeping only the wire rope drum, reducer, and spiral torsion spring.

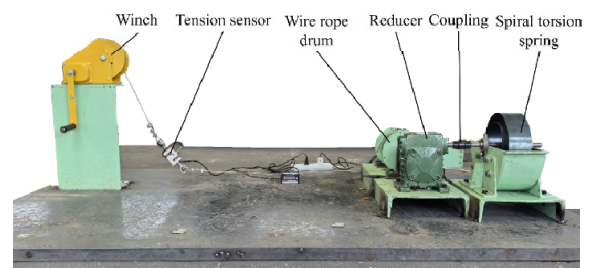


Figure 9. Experimental setup diagram of mechanical damping anti-sway system.

In the experiment, a winch was used to simulate the dynamic tension on the wire rope during the swing of the spreader. Tension sensors placed on the wire rope were used to collect real-time tension data corresponding to different numbers of spiral torsion spring rotations. The anti-sway torque provided by the spiral torsion spring at different numbers of rotations was further calculated, revealing the functional relationship between its elastic energy storage and output torque. This provides a theoretical basis for the mechanical modeling and parameter optimization of the anti-sway mechanism. The main technical parameters of the spiral torsion spring used in the experiment are shown in Table 2.

Table 2. Main Technical Parameters of Experimental Spiral torsion springs.

Parameter	Numerical values	Unit
Number of rotations in free state	2.48	/
Theoretical value of torque	16000	N·mm
Material thickness	1	mm
Material width	100	mm
Material unfolded length	1598	mm

The experiment recorded the number of rotations of the spiral torsion spring when the tension of the wire rope reached the preset value, and compared it with the theoretical calculation value. The experimental results are shown in Table 2.

Table 2. Comparison of Experiments and Theories.

Comparison items	Wire rope tension (N)		
	8000	12000	16000
Test rotation number of revolutions	1.08	1.63	2.12
Theoretical number of revolutions	1.24	1.86	2.48
Relative deviation (%)	14.81	14.11	16.98

Table 2 shows that the average deviation of the three experimental groups was 15.30%. The largest deviation (16.98%) occurred when the wire rope tension was 16000 N, while the smallest deviation (14.11%) occurred when the wire rope tension was 12000 N. Overall, the number of rotations of the spiral torsion spring in the experiments was generally lower than the theoretical calculation value. This may be due to the use of a larger safety factor in the design to improve service life. Furthermore, the need to round down parameters such as the width and thickness of the spiral torsion spring according to specifications during the design process also leads to deviations between actual and theoretical values. When the number of rotations approached the theoretical design value, the deformation of the experimental spiral torsion spring approached or exceeded the allowable value, further amplifying the deviation. The deviation indicates that the theoretical model slightly overestimates the effective winding angle under

practical conditions. This is mainly because real systems involve assembly clearance, transmission friction, and material stiffness tolerance, which reduce the actual elastic deformation participating in torque generation.

In addition, the experimental results also show a linear relationship between the number of rotations of the spiral torsion spring and the reverse torque it generates, as shown in Figure 10. This characteristic is consistent with the theoretical model prediction, verifying that the reverse torque of the non-contact spiral torsion spring is generated solely by the elastic deformation of the spiral torsion spring material. The experimental results verify the effectiveness and feasibility of using spiral torsion springs in the anti-sway design of cranes. This linearity confirms that the restoring torque is predominantly generated by elastic bending deformation of the spring strip rather than contact friction between adjacent coils. Therefore, the non-contact spring configuration is advantageous for predictable torque control and stable anti-sway performance.

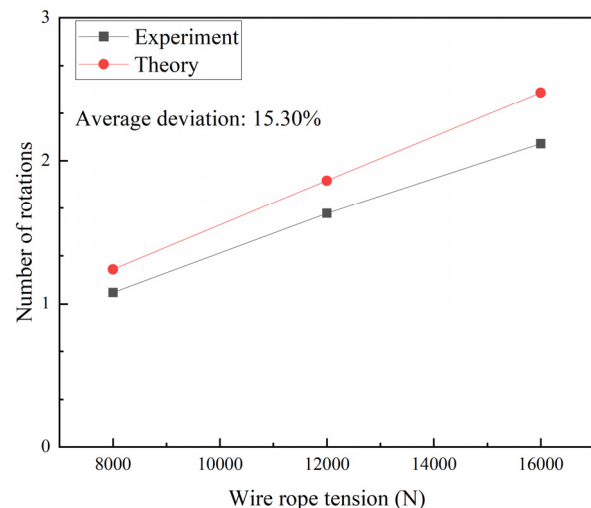


Figure 10. Comparison of Experiment and Theory.

6. CONCLUSIONS

To reduce energy consumption and simplify the mechanical structure of anti-sway systems for container cranes in automated terminals, a mechanical damping anti-sway system utilizing spiral torsion springs was proposed. The effectiveness and feasibility of applying the spiral torsion springs to crane anti-sway devices were verified through simulation and experiments. Based on this, the following conclusions were drawn:

(1) The proposed passive anti-sway mechanism can effectively attenuate spreader sway by converting oscillatory kinetic energy into elastic strain energy and dissipating residual motion through braking. Simulation results demonstrated that the unilateral sway amplitude could be reduced below 100 mm within 2.5 oscillation cycles under rated load conditions, indicating satisfactory operational safety and positioning capability.

(2) The anti-sway performance is strongly dependent on the output torque of the spiral torsion spring module. Increasing the restoring torque enhances the lateral constraint force and accelerates vibration energy decay. However, excessively large torque may introduce unnecessary rope tension fluctuation and mechanical impact.

Therefore, torque matching according to load condition is more desirable than simple torque maximization.

(3) Under no-load operation, the required anti-sway torque was significantly lower than that under rated load due to the reduction in system inertia. This suggests that adaptive torque configuration can reduce energy consumption while maintaining sway suppression performance.

(4) Experiments confirmed an approximately linear relationship between spring rotation angle and restoring torque. This validates the suitability of the non-contact spiral torsion spring for predictable passive control applications. The average deviation between theoretical and measured values was 15.30%, which is acceptable for engineering design and mainly attributed to manufacturing tolerance, friction loss, and conservative safety margins.

(5) Compared with conventional hydraulic or actively controlled systems, the proposed design offers the advantages of simple structure, low maintenance demand, reduced sensor dependence, and potential retrofit feasibility for short-lift automated transfer cranes.

Although the present study verified the feasibility of the concept, several issues require further investigation, including long-term fatigue life of the spring module, intelligent clutch timing optimization, dynamic response under wind disturbance, and full-scale field validation in real terminal environments.

Overall, the proposed passive mechanical anti-sway system provides a practical and energy-efficient alternative for next-generation automated container handling equipment.

REFERENCES

- [1] Solazzi, L.; Zrnić, N. Dynamic analyses of gantry crane under several trolley and payload movements. *FME Transactions*. 2020, 48(2), 281–286.
- [2] Solazzi, L.; Zrnić, N. Design of a high capacity derrick crane considering the effects induced by the application and the release of the load. *Journal of Applied Engineering Science*. 2017, 15(1).
- [3] Tomasi, I.; Solazzi, L. Dynamic Analysis of an Offshore Knuckle-Boom Crane Under Different Load Applications Laws. *Applied Sciences*. 2025, 15(14), 8100.
- [4] Ramli, L.; Mohamed, Z.; Abdullahi, A.M.; et al. Control Strategies for Crane Systems: A Comprehensive Review. *Mech. Syst. Signal Process*. 2017, 95, 1–23.
- [5] Zhang, Q.; Ge, Y.F.; Chen, M.; et al. Active Anti-Sway Method of ARMG Spreader Auxiliary Wire-Rope Traction System. *J. Harbin Inst. Technol.* 2020, 52, 36–42. (in Chinese)
- [6] Wang, H.F.; Wang, J. Anti-Sway Technology for Eight-Rope Hoisting Mechanism of RTG. *China Spec. Equip. Saf.* 2013, 29, 20–22. (in Chinese).
- [7] Wang, Z.S. Study on Connection Mode between Wire Rope and Spreader of Eight-Rope Anti-Sway System. *Hoisting Convey. Mach.* 2021, 21, 74–76. (in Chinese).
- [8] Zhang, Q.; Li, Q.; Chen, M.; et al. Analysis of Anti-Sway Influencing Factors and Dynamic Simulation of RTG Eight-Rope Anti-Sway System. *Mech. Des. Res.* 2018, 34, 57–61. (in Chinese).
- [9] Cheng, W.M.; Zhong, B.; Zhang, Z.Q.; et al. Dynamic Analysis of Hydraulic-Cylinder Anti-Sway System for Container Crane. *China Railw. Sci.* 2007, 28, 105–109. (in Chinese).
- [10] Zhong, B. Study on Structural Parameters of Hydraulic Anti-Sway System for Railway Container Crane. *China Railw. Sci.* 2010, 31, 133–137. (in Chinese).
- [11] Huang, K.; Zheng, J.Q.; Li, X.D. Research and Development of Electronic Anti-Sway Technology for Container Crane. *Hoisting Convey. Mach.* 2006, 12, 5–8. (in Chinese).
- [12] Sun, L.J.; Tang, W.Q.; Huang, X.L.; et al. Application of Embedded System in Spreader Anti-Sway Control of Crane. *J. Lanzhou Univ. Technol.* 2020, 46, 97–102. (in Chinese).
- [13] Solihin, M.I.; Wahyudi; Legowo, A. Fuzzy-Tuned PID Anti-Swing Control of Automatic Gantry Crane. *J. Vib. Control* 2010, 16, 127–145.
- [14] Wang, T.; Tan, N.; Qiu, J.; et al. Global-Equivalent Sliding Mode Control Method for Bridge Crane. *IEEE Access* 2021, 9, 160372–160382.
- [15] Dongtian, L.; Huikang, L.; Lin, C. Research on Harmonic Elimination and Anti-Swing of Crane Wire Rope Based on Motor Output Torque Compensation. In *Proceedings of the 2022 34th Chinese Control and Decision Conference (CCDC)*, Hefei, China, 15–17 June 2022; pp. 3950–3955.
- [16] Zhong, B.; Ma, L.; Dong, H. Dynamic Analysis and Structure Parameter Research on a Hydraulic Anti-Swaying System for Container Cranes. *Appl. Sci.* 2022, 12, 12537.
- [17] Dong, J.; Sun, D.; Jia, T.; et al. Research on Anti-Swing of Container Gantry Crane as Hierarchical Sliding Mode Control. *J. Fail. Anal. Prev.* 2023, 23, 1741–1751.
- [18] Zhang, Q. et al. Fuzzy Sliding Mode Control on Positioning and Anti-Swing for Overhead Crane. *Int. J. Precis. Eng. Manuf.* 2023, 24, 1381–1390.
- [19] Zrnić, N.; Petković, Z.; Bošnjak, S. Automation of Ship-To-Shore Container Cranes: A Review of State-of-the-Art. *FME Trans.* 2005, 33, 111–121.
- [20] Zrnić, N.; Gašić, V.; Bošnjak, S.; Đorđević, M. Moving Loads in Structural Dynamics of Cranes: Bridging the Gap Between Theoretical and Practical Researches. *FME Trans.* 2013, 41, 291–297.
- [21] Zrnić, N.Đ.; Gašić, V.M.; Bošnjak, S.M. Dynamic Responses of a Gantry Crane System Due to a Moving Body Considered as Moving Oscillator. *Arch. Civ. Mech. Eng.* 2015, 15, 243–250.
- [22] Ponomarev, S.D.; Andreeva, L.E. *Calculation of Elastic Elements of Machines and Instruments*; Chemical Industry Press: Beijing, China, 1987. (in Chinese).
- [23] Muñoz-Guijosa, J.M.; Caballero, D.F.; de la Cruz, V.R.; et al. Generalized Spiral Torsion Spring Model. *Mech. Mach. Theory* 2012, 51, 110–130.

- [24] Muñoz-Guijosa, J.M.; Caballero, D.F.; de la Cruz, V.R.; et al. On the Use of Variable Bending Stiffness Clothoidal Strips for the Analysis and Synthesis of Low Variability Torque-Angle Turned Curves in Spiral Torsion Springs. *Mech. Mach. Theory* 2013, 67, 32–46.
- [25] JB/T 7366-94. Design and Calculation of Plane Spiral Spring; Standards Press of China: Beijing, China, 1994. (in Chinese).
- [26] Yang, C.; Huang, J.; Singhose, W. Dynamic Modeling and Oscillation Control of Industrial Cranes Transporting Upright Slender Flexible Payloads. *Mech. Syst. Signal Process.* 2024, 220, 111676.

ПАСИВНИ МЕХАНИЧКИ СИСТЕМ ЗА ПРИГУШЕЊЕ ЊИХАЊА КОД АУТОМАТИЗОВАНИХ ДИЗАЛИЦА ЗА ПРЕТОВАР КОНТЕЈНЕРА СА СПИРАЛНИМ ТОРЗИОНИМ ОПРУГАМА: МОДЕЛОВАЊЕ, СИМУЛАЦИЈА И ЕКСПЕРИМЕНТАЛНА ВЕРИФИКАЦИЈА

К. Лиу, Ч. Ванг, Б. Жао, Ј. Ванг, Ј. Луо, Ј. Куан, Л. Солаци, Н. Зрнић

Њихање контејнера ограничава тачност позиционирања, безбедност и ефикасност руковања на ау-

томатизованим контејнерским терминалима. Постојећа решења за пригушење њихања често подразумевају сложене конструкције, високе трошкове одржавања или ограничену робусност при променљивим оптерећењима. Ова студија предлаже пасивни систем за пригушење њихања који користи модуле са спиралним торзионим опругама и систем за повремено ангажовање уз помоћ квачила. Систем претвара кинетичку енергију насталу услед њихања у енергију еластичне деформације и расипа преосталу енергију путем кочионе јединице, чиме се избегава потреба за континуалним активним управљањем. Динамички модел спрегнутог система дизајлине и терета успостављен је применом Лагранжовог метода, а модули са опругама пројектовани су у складу са инжењерским стандардима. Симулације динамике више тела, спроведене у условима номиналног оптерећења и рада без терета, показују да систем смањује амплитуду једностраног њихања на мање од 100 mm у року од 2,5 циклуса осциловања при номиналном оптерећењу. Тестирања прототипа потврђују приближно линеарну зависност између угла ротације опруге и повратног момента, уз просечно одступање од 15,30% у односу на теоријска предвиђања. Овај систем представља енергетски ефикасно решење које захтева минимално одржавање и намењено је опреми за руковање контејнерима са малом висином дизања.