Energy-saving Design and Simulation Analysis of the Baggage Diversion Lifting Mechanism

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Abstract. The use of springs to reduce energy consumption and the requirements on the drive element is proposed. The design of the energy saving mechanism is completed, and the dynamics simulation of the mechanism is carried out. The driving force characteristics are obtained and compared with the original driving force characteristics to verify the energy saving effect of the mechanism. The spring energy storage system is optimized, and the spring stiffness coefficient that comprehensively considers the driving force characteristics and energy saving requirements is obtained.

Introduction

Mechanical energy storage system plays a very important role in the field of engineering machinery, which can reduce energy consumption and improve the efficiency and performance of mechanical system [1]. The mechanical energy storage system has the characteristics of dynamic energy absorption and timely release, and has some advantages such as large energy storage capacity, high efficiency, low cost, and no pollution [2]. Common mechanical energy storage system mainly includes compressed air energy storage system, flywheel energy storage system and spring energy storage system [3], and these energy storage systems have been widely used in wind energy integration and volatility control [4, 5], energy saving of refrigeration system [6], improvement of power quality and stability [7, 8, 9], and recovery of vehicle braking energy [10, 11]. Spring energy storage system has been extensively studied in the recent years [12], and the research contents mainly include the study of spring energy model [13,14], the low-cost recovery of automotive braking energy [15], the fluid kinetic energy storage of fluid turbine [16], the energy storage and rapid release of micro-machines [17].

In the field of engineering machinery, lifting mechanism is widely used [18, 19]. In order to solve the airport baggage diversion problem, Du et al. designed a new baggage diversion lifting mechanism [20]. However, the driving force of the lifting mechanism completely does negative work in the reset process, which causes a waste of energy. And both the starting value and the peak value of the driving force are large, which causes high requirements for the driving element. In this paper, the spring energy storage system is introduced on the basis of the original baggage diversion lifting mechanism for solving the energy waste problem, the energy saving design, simulation and optimization of the baggage diversion lifting mechanism will be studied.

Structural Characteristics and Dynamic Analysis of the Lifting Mechanism

Mechanical Structure and Working Principle

The baggage diversion lifting mechanism is driven by an electronic cylinder and a parallelogram mechanism to realize the lifting and reset function, and the kinematic sketch of mechanism as shown in Fig. 1. The lifting mechanism consists of triangular arm ABC and EFG, electronic cylinder, Push rod BE, and support plate I₁-I₁. Points A and G are the revolute pairs of the triangular arm and the
frame. Points B and E are the revolute pairs of the Push rod and the triangular arm. Points C and F are the revolute pairs of the support plate and the triangular arm. Point O is the revolute pair of the electronic cylinder and the frame.

The basic dimensions in the mechanism are: the length of the support plate is $L_1=2390\text{mm}$, and the width is $640\text{mm}$; the distance from the revolute pair C to the left end of the supporting plate is $L_5 = 520\text{mm}$; the distance from the revolute pair F to the right end of the supporting plate is $L_6 = 470\text{mm}$; the distance from the revolute pair C to the bottom surface of the support plate is $H_4 = 30\text{ mm}$; in the reset state, the distance between the revolute pair A and the support plate is $H_1 = 21\text{ mm}$; the distance between the revolute pair A and G is $L_2=1400\text{mm}$; the length of the triangular arm ABC is $L_{AB}=100\text{mm}$, $L_{AC}=80\text{mm}$, $L_{BC}=128\text{mm}$; the arm of force that the revolute pair B(E) to the revolute pair A(G) is $H_2=99.2\text{mm}$; the arm of force that the revolute pair C(F) to the revolute pair A(G) is $L_4=79.4\text{mm}$; the length of the push rod is $1400\text{ mm}$; the horizontal distance between the rotating pair O and the rotating pair A is $L_3=600\text{mm}$ and the vertical distance is $H_2=99.2\text{mm}$; the distance between the revolute pair O and the ground is $189.8\text{mm}$; the total stroke of the lifting process or reset process is $24\text{mm}$.

When the lifting mechanism is working, the electronic cylinder rod pushes the push rod to move first. Then the push rod pushes the triangular arm ABC and the triangular arm EFG to rotate around the revolute pairs A and G, respectively. Finally, the support plate completes the lift and reset functions under the action of the triangular arm. When the tray and baggage needs to be diverted, the lifting mechanism raises to the design position in advance. After the tray and baggage passes, the lifting mechanism is quickly reset.

![Figure 1. The kinematic sketch of mechanism. 1 triangular arm, 2 electronic cylinder, 3 Push rod, 4 support plate.](image1)

![Figure 2. Characteristics of the driving force on the cylinder rod.](image2)

**Dynamic Analysis**

**Input Movement.** The cylinder rod drives the push rod directly, so the input motion can be expressed by the movement of the cylinder rod. In order to make the mechanism run smoothly both in the process of lifting and reset, the cylinder rod does a uniform acceleration first, then a uniform motion, and finally a uniform deceleration. During the entire movement, the acceleration time, uniform time and deceleration time of the cylinder rod are all $0.1\text{s}$, and the accelerations are $1500\text{ mm/s}^2$, $0\text{ mm/s}^2$ and $-1500\text{ mm/s}^2$, respectively.

**Analysis of Driving Force Characteristics.** The driving force characteristics obtained according to the input movement set of the system are as follows:

As shown in Fig. 2, the first $0.3\text{s}$ is the lifting process, and the remaining $0.3\text{s}$ is the reset process. During the lifting process, the direction of the electronic cylinder output force is consistent with the
direction of movement, the electronic cylinder does positive work. During the reset process, the direction of the electronic cylinder output force is opposite to the direction of movement, the electronic cylinder does negative work. The starting values of the driving force for the lifting process and the reset process are 1712.8N and 1307.2N respectively, and the peak value of the driving force is 1781.6N. According to the characteristics of the driving force, it can be found that the mechanical energy in the reset process is dissipated by converted into thermal energy, which causes a waste of energy. The starting value and the peak value of the driving force are high, which makes high requirements on the driving element.

**Energy Saving Design of the Baggage Diversion Lifting Mechanism**

**Design of the Energy Saving Mechanism**

In this paper, a spring energy storage device is added to the original structure of the lifting mechanism. The spring energy storage device is installed at both ends of the support plate 4, and the structure diagram of the energy saving mechanism as shown in Fig. 3.

![Figure 3. The structure diagram of the energy saving mechanism. 1 triangular arm, 2 electronic cylinder, 3 push rod, 4 support plate.](image)

Compared to the model in section Mechanical Structure and Working Principle, except for the two spring energy storage devices with the same stiffness $k$, the remaining moving members, connection methods and the mechanism dimensions are the same. Due to the space limitation, in the reset state, the length of the spring is taken as $H_5=H_6=100\text{mm}$, and in order to ensure the force balance of the support plate, the distance between the center line of the spring and the support plate is $L_7=L_8=300\text{mm}$.

When the lifting mechanism is lifted, the spring energy storage system releases energy, and the parallelogram mechanism realizes the lifting function under the combined action of the spring force and the driving force of the cylinder rod. When the lifting mechanism is reset, the spring is compressed, the spring energy storage system brakes the parallelogram mechanism and stores energy, and the parallelogram mechanism realizes the reset function under the combined action of the spring force and the driving force of the cylinder rod.

**Determination of Compression Length and Stiffness Coefficient of the Spring**

**The Compression Length of the Spring**. From the section Design of the Energy Saving Mechanism, it is known that the compression length $X$ of the spring and the lifting height $S$ of the lifting mechanism should meet $X \geq S$. In order to ensure the stability of the mechanism, assuming that
the spring is free when the lifting mechanism reaches the lifting height. So the compression length of the spring can be taken as \(X=S=24\text{mm}\).

**The Stiffness Coefficient of the Spring.** Assuming that the gravitational potential energy \(E_p\) of the support plate is all converted into elastic potential energy \(E_k\) of the spring energy storage device during the reset stroke. We get \(E_k=E_p=mg\cdot h\), which \(m\) is the mass of the support plate, \(g\) is the acceleration of gravity, and \(h\) is the lifting height of the support plate. When \(m=190\text{kg}\), \(g=10\text{m/s}^2\), \(h=S=24\text{mm}\), \(E_k=E_p=4.56\times10^4\text{N}\cdot\text{mm}\).

The elastic potential energy of the spring is \(E_k=0.5k_1x^2\), which \(k_1\) is the total stiffness coefficient of the spring, \(x\) is the compression length of the spring, \(x=24\text{mm}\). So we can get \(k_1=158.33\text{N/mm}\), here we take \(k_1=158\text{N/mm}\). Because there is a spring on both sides of the support plate, the stiffness coefficient \(k\) of a single spring is \(k=0.5k_1=79\text{N/mm}\).

**Simulation and Optimization of the Energy Saving Structure of the Baggage Diversion Lifting Mechanism**

Simulation analysis of a mechanism can be realized by using ADAMS software [21,22]. According to the design of energy saving mechanism, 3D model of the energy saving structure of the lifting mechanism is built and multi-body dynamics simulation analysis is carried out.

**Simulation Results and Comparative Analysis**

Add motion pairs at each joint of all components, set the mass of each unit, set the spring stiffness coefficient and preload, and perform dynamic simulation according to the motion input in section Input Movement. The results are as follows:

Fig. 4 shows the characteristics of the driving force on the cylinder rod of the energy saving structure. The starting value of the driving force for the lifting process (0~0.3s) is -1320.8N. The starting value of the driving force for the reset process (0.3~0.6s) is 1309.6N. The peak value of the driving force is 1320.8N.

Comparing with Fig. 2, the starting value of the driving force of the lifting process is decreased by 22.9%, and the starting value of the driving force of the reset process is basically unchanged, the peak value of the driving force is decreased by 25.9%. The energy saving design improves the drive characteristics of the mechanism. Obviously, it can be seen that the mechanical energy of the support plate is mostly stored by the spring energy storage system, and the negative work done by the driving force during the entire movement is greatly reduced. Although the driving force of the lifting and reset process has done negative work, the total work that done by the driving force is reduced, so the mechanism achieves the purpose of energy saving.

**Optimization of the Spring Stiffness**

Although the driving force characteristics have been improved and the energy saving has been achieved, the starting value of the driving force and the negative work performed are still relatively large. Therefore, it is necessary to further study the minimum starting value and power consumption of the driving force. From the above research, it can be seen that the optimization analysis can be carried out by changing the value of the spring stiffness coefficient \(k\). The results are as follows:
As shown in Fig. 5, with the increase of spring stiffness coefficient $k$, the starting value of the driving force increases during the lifting process. Because the spring is in the free state when the reset process starts, the device is completely braked by the driving force. Therefore, the starting value of the driving force is unchanged.

As shown in Fig. 6, the area enclosed by the power curve of the driving force and the time axis reflects the magnitude of work done by the driving force. Therefore, it can be seen that as the spring stiffness coefficient $k$ increases, both the negative work and the total work, which done by the driving force, are decreased first and then increased. So there is a minimum value in both negative work and total work.

After analysis, it can be found that when the starting value of the driving force is the minimum, the spring stiffness coefficient $k$ is 44 N/mm, and the starting value and peak value of the driving force is 23.2 N and 1308.5 N, respectively. Comparing with Fig. 2, it can be seen that both the negative work and total work, which done by the driving force, are decreased by 57.1%, and the starting value and peak value of the driving force are decreased by 98.6% and 23.8%, respectively. When the negative work and total work done by the driving force are the minimum, the spring stiffness coefficient $k$ is 59 N/mm, and the starting value and peak value of the driving force is -552.8 N and 1309.0 N, respectively. Comparing with Fig. 2, it can be seen that the negative work and total work, which done by the driving force, are decreased by 69.6%, and the starting value and peak value of the driving force are decreased by 67.8% and 23.8%, respectively. Combining the requirements of optimized driving force characteristics and energy saving, the spring stiffness coefficient is finally taken as $k=44$ N/mm.

**Conclusion**

The energy saving design of the mechanism is carried out with spring energy storage system, and stiffness coefficient of the spring is determined.

Dynamic simulation of the energy saving mechanism is carried out. The peak value of the driving force is reduced by 25.9%. The starting value of the driving force of the lifting process is decreased by
22.9%. During the entire process, both the negative work and the total work, which done by the driving force, were reduced to some degree. The purpose of energy saving is achieved.

The optimization analysis is performed by changing the stiffness coefficient of the spring. According to the requirements of optimized driving force characteristics and energy saving, the spring stiffness coefficient is finally taken as \( k = 44 \text{N/mm} \). At this time, the negative work and total work, which done by the driving force, are decreased by 57.1%, and the starting value and peak value of the driving force are decreased by 98.6% and 23.8%, respectively.

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