Design and optimization of automatic conveying equipment for cargo bags of cargo spacecraft

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Abstract: Cargo transportation equipment is commonly used for ground loading in aerospace. Taking flexible cargo bags as a transport carriers and taking a cargo ship hatch as the target container, a kind of cargo bag transportation system is designed according to the transportation process and function requirements. On the basis of the analysis of structure and working procedure, the working principle of the mechanical subsystem as well as the structure design of key components is proposed in detail. To improve the performance of support plate's stiffness, a topological optimization is implemented and the optimized result is hence achieved. Finite element simulation shows that the performance of the stiffness is improved significantly. The approach presented in this research benefits the design of cargo transportation equipment.

1. Introduction
Cargo spacecraft is a spacecraft specially used to transport goods to space beyond the earth's atmosphere[1-2]. It is an important means of transportation for space station supplies. The automatic conveying equipment for cargo bags in cargo spacecraft is an important device for conveying cargo bags from the ground to the spacecraft cabin [3-7]. Its design meets three requirements in terms of function:

- The automatic conveying equipment can transfer the goods stored in the designated position on the ground efficiently and safely to the interior of the cargo holds of the spacecraft.
- The automatic conveying equipment can transfer the goods with adjustable height, which is suitable for cargo holds of different heights and flexible packages of six different specifications.
- The automatic conveying equipment can minimize the space occupied and have high compactness.

On the other hand, each component should meet the performance indicators such as stiffness and strength. However, due to the limitation of the weight, size and hatch size of the cargo package, it is difficult to design. Topology optimization is a new structural design concept, which has been widely used in architecture, vehicle, medicine and other fields[8-10]. It provides a good design optimization tool for engineering designers and analysts. In this paper, the key parts are designed by topology optimization, which can not only meet the size limitation, but also have high stiffness and strength performance.

2. Functional Design of Baggage Conveying Intake Equipment
The equipment system designed in this paper is mainly divided into three parts: mechanical subsystem, control and drive subsystem and safety monitoring subsystem. As shown in Figure 1, the mechanical subsystem includes a base motion platform, a vertical lifting component of chain drive, a low level transmission component of chain power roller[11] drive and a high level transmission component of adjustable height. The base motion platform is used to adjust the horizontal position and direction angle.
of the hoist for alignment with the cabin. The bags on the low level transmission parts are transferred to
the vertical elevator parts through flexible pallets, and then transported to the high level transmission
parts, which are transported to the interior of the cabin by conveyor belts.

![3D schematic of cargo conveyer](image1)

The key to the design of this equipment is the high-level transmission component with adjustable
height. In addition to transporting the cargo package horizontally into the cargo hold, it should also meet
the two functions of horizontal position adjustment and height adjustment.

2.1. Horizontal Position Adjustment of High Bit Horizontal Transmission Components

Fig.2 is a high level transmission component, which uses aluminium alloy as supporting plate and uses
parallel sprocket power cylinder rollers to realize high level transmission of freight bags. Horizontal
position adjustment is realized by two sets of guide sliders and a set of gear rack mechanism. The slider
and the gear are mounted on the fixed platform, and the guide rail and the rack are mounted on the lower
surface of the transmission component along the x-axis. When the motor drives the gear to rotate, the
rack moves along the x-axis, thus driving the whole high-level transmission component to insert into
the hoist. When the motor rotates backwards, it penetrates into the cargo hold.

![Three-dimensional schematic of high level horizontal transmission component](image2)

2.2. Height Adjustment of High Bit Horizontal Transmission Components

In Figure 3, the height adjustment device of the high level transmission component is composed of two
pairs of synchronous screw and nut mechanisms, four sets of synchronous guide sliders and a set of gear
transmission system. Four sliders are fixed by connecting rods and nuts to form a vertical moving unit.
Four high sprockets of high horizontal transmission parts and vertical lifting parts are connected by
connecting rods to nuts or sliders. When adjusting the height, the motor drives the screw to rotate, the
nut moves up and down along the screw, and the slider moves along the guide rail, thus realizing the
height adjustment of the fixed platform, the high level transmission parts and the high level transmission
sprocket.
On the other hand, if the position of the lower sprocket remains unchanged, the transmission chain will be relaxed or stretched excessively when the high-level transmission components and the high-level sprocket are adjusted. Therefore, a low-level sprocket position adjustment device is designed, as shown in Fig.4. The device consists of a pair of bi-directional screw to save adjusting space: the left and right nuts are connected with a pair of sprockets 1 and 2, respectively. If the height of the high sprockets is reduced, the motor drives the bi-directional screw to rotate, and the left and right nuts drive the sprockets 1 and 2 to move left and right respectively, thus realizing the tension of the transmission chain.

3. Design and optimization of key components

3.1. Dimension Design of Key Components

In this design, the target delivery container of the cargo package is the cabin of a cargo spacecraft, with only one entrance at the front end. Flexible bags are required to smoothly pass through the entrance and enter the container.

- Considering the restrictions of the package specifications and equipment on hatches, the supporting structure can only be arranged on both sides of the high-level horizontal transmission components, and its cross-section size should be less than (width * height) 20 mm * 80 mm.
- The maximum width and height of the conveying bags are 570 mm and 470 mm respectively. Roller transmission is used for horizontal transmission components. For those parts requiring smooth transportation, the center distance of rollers is about 1/3 of the delivery length of bags, and the shortest length of delivery bags is 360 mm, while the center distance of rollers is 120 mm. The length of high-level horizontal transmission parts is 1800 mm, which requires about 15
rollers. Considering the size of hatch and the stability of transportation, the length of rollers should be 400 mm (70%~80%) and the width of package should be 400 mm, and the power stick with a diameter of 38 mm should be selected in combination with the above dimensions and the maximum mass of package 40kg.

- According to the bending moment of 40 kg cargo bag gravity load at the end of cantilever of support plate, SR35W guideway of THK company is selected.

3.2. Stress and restraint of supporting plate

(1) Stress condition of supporting plate

- Maximum weight of flexible terminal package is M=392N;
- Rod weight: The weight of the selected rollers is F1=9.8N, and the spacing of the supporting structure is 120 mm;
- Guide rail weight: single guide rail weight is q=62.72N/m;
- Rack weight: the choice of plastic rack rack, its weight is neglected;
- The gravity G of the supporting structure G≤3.4kg.

(2) Other constraints

- The supporting structure can only be arranged on both sides of the high-level horizontal transmission components, and its crosssection size (width * height) ≤20mm*80mm;
- Maximum deflection allowed at the end of cantilever structure with support plate when carrying the maximum package w≤10mm.
- Under the above conditions, the one-sided component of the supporting plate can be simplified to a cantilever beam. In addition to the gravity of the supporting plate itself, the other loads are shared by the two supporting plates, so the stress situation is shown in Figure 6.

3.3. Topological optimization of supporting plates

In the design process, it is found that if the existing aluminum profile is selected, it is difficult to satisfy the condition that the deflection of the free end of the support plate is less than 10 mm. Therefore, the support plate is designed by using topology optimization.

(1) Topology optimization problem description

This project takes the support plate as the optimization object, takes the maximum stiffness and minimum flexibility of the support plate as the optimization objective to optimize the structure topology, and chooses the density of each discrete element as the optimization variable. because SIMP (Solid Isotropic Material with Penalization), the material interpolation model can make the relative density of the element converge rapidly to 0 and 1\(^{[12-13]}\). In this paper, SIMP material interpolation model is chosen. \(E_0\) and \(E_e\) are set to the initial modulus of elasticity and the optimized modulus of elasticity. \(k_0\) is the elemental stiffness matrix before optimization, \(k_e\) is the elemental stiffness matrix after optimization. The stiffness matrices before and after optimization have the following relationships.

\[
E_e = \rho_e E_0, k_e = \rho_e k_0
\]  

\(\rho_e\) is relative density of materials in units, value in the range of (0,1); \(p\) is penalty factor.

On the basis of satisfying the structural equilibrium equation, this problem also needs to restrict the weight of the structure, that is, the volume of the solid part. For this reason, the mathematical model is described as follows:
Find:  \[ \rho_e = \begin{cases} 1 & \rho_e \in \Omega_e \\ 0 & \rho_e \notin \Omega_e \end{cases} \]  

Min:  \[ C = U^T K U = \sum_{e=1}^{N} u_e^T k_e u_e = \sum_{e=1}^{N} (\rho_e)^p u_e^T k_e u_e \]  

St.:  \[ \int_{\Omega} \rho_e d\Omega \leq fV_0 \]  

In the formula, the objective function \( C \) Overall flexibility defined as structure, \( K \) is Total Stiffness Matrix of Structures before Optimum Design, \( F_t \) is Total force vector, \( U \) is Displacement column vector, \( u_e \) is Element displacement column vector, \( N \) is Total number of structural discrete elements, \( V_0 \) is Initial volume of structure in the whole design domain, \( V \) is Optimized structure volume, \( f \) is Design volume ratio of structure, \( \Omega \) is Given initial design domain, \( \Omega_e \) is The area occupied by the cue material, \( \rho_{\text{min}} \) and \( \rho_{\text{max}} \) is the minimum and maximum limit values of the relative density of the element are introduced to prevent the singularity of the element stiffness matrix. Aiming at the specific optimization problems in this paper, the weight and size of the support are limited, Available volume ratio \( f \) is 45%; Unit size is \( 1\text{mm} \times 1\text{m} \), Initial design domain \( \Omega \) is \( 1800 \times 80 \); \( \rho_{\text{min}} \) and \( \rho_{\text{max}} \) Set them to 0.2 and 1.

In the process of solving the optimization algorithm, it is necessary to find the sensitivity of the objective function and the constraint function in order to achieve fast convergence. First, because it is constant, it is obtained from the first equation of equation (4).

\[ \frac{\partial K}{\partial \rho_e} U = -K \frac{\partial U}{\partial \rho_e} \]  

The sensitivity equation of the overall structural flexibility can be obtained from formula (3):

\[ \frac{\partial C}{\partial \rho_e} = U^T \frac{\partial K}{\partial \rho_e} U + 2U^T K \frac{\partial U}{\partial \rho_e} = -U^T \frac{\partial K}{\partial \rho_e} U \]  

3.4. Optimization criterion algorithm

After establishing the model of topology optimization, this paper uses OC (optimality criteria) optimization criterion method to solve the optimization problem. Lagrangian functions consisting of objective functions and constraints:

\[ L = C + \lambda_1(V - fV_0) + \lambda_2^T [KU - F_t] + \lambda_3(\rho_{\text{min}} - \rho + b^2) + \lambda_4(\rho - \rho_{\text{max}} + c^2) \]

In formula, \( \lambda_1 \), \( \lambda_2 \), \( \lambda_3 \), \( \lambda_4 \) are Lagrangian multiplier, \( \lambda_1 \) is scalar, \( \lambda_2 \), \( \lambda_3 \), \( \lambda_4 \) are column vectors composed by \( \lambda_2^1 \), \( \lambda_2^2 \), \( \lambda_2^3 \); \( b \), \( c \) are column vectors composed by \( b_1 \), \( c_1 \), \( b_2 \), \( c_2 \) are relaxation factor, \( \rho_{\text{min}} \), \( \rho_{\text{max}} \) are column vectors composed by \( \rho_{\text{min}}^e \), \( \rho_{\text{max}}^e \); \( \rho \) is column vector composed by \( \rho_e \). When \( \rho_e \) take extremum \( \rho_e^* \). The above formula needs to satisfy the following Kulm-Tucker condition:
In the formula, \( \nu_e \) is the unit volume. According to the upper and lower bounds of design variables, the following relationships can be obtained.

\[
\begin{align*}
\lambda_e^i = \lambda_e^i = 0, & \quad \text{if } \rho_{\text{min}} < \rho_e < \rho_{\text{max}} \\
\lambda_e^i \geq 0, & \quad \lambda_e^i = 0, \quad \text{if } \rho_e = \rho_{\text{min}} \\
\lambda_e^i = 0, & \quad \lambda_e^i \geq 0, \quad \text{if } \rho_e = \rho_{\text{max}}
\end{align*}
\]  

(7)

Considering the first case of equation (9), the second equation of equation (5)-(6), equation (8) and the first equation of equation (1) substituted for equation (8) can be obtained as follows:

\[-p(\rho_e)^{n-1} u_e^{*} K_{e0} u_e + \lambda_e v_e = 0 \quad (8)\]

The optimum design criteria can be obtained from the formulas:

\[C_e^i = \frac{p(\rho_e)^{n-1} u_e^{*} K_{e0} u_e}{\lambda_e v_e} = 1 \quad (9)\]

According to the formulas (9) and (11), the iteration formulas of optimization variables are obtained as follows:

\[
\begin{align*}
\rho_e^{k+1} &= \left( C_e^i \right)^{\eta} \rho_e^k & \quad \rho_e^{k+1} &< \left( C_e^i \right)^{\eta} \rho_e^k < 1 \\
\rho_e^{k+1} &= \rho_{\text{min}} & \quad \left( C_e^i \right)^{\eta} \rho_e^k &\leq \rho_{\text{min}} \\
\rho_e^{k+1} &= 1 & \quad \left( C_e^i \right)^{\eta} \rho_e^k &\geq \rho_{\text{max}}
\end{align*}
\]

(10)

In order to ensure the stability and convergence of the optimization calculation, the damping coefficient is introduced \( \eta \). Then, the convergence is determined by the following formula to determine the final topological structure.

\[
\frac{\max \rho_e^{k+1} - \max \rho_e^k}{\max \rho_e^k} < \varepsilon \quad (11)
\]

\( \varepsilon \) is the setting convergence tolerance.

3.5. Numerical instability in topology optimization

Porous and checkerboard grids may appear in the process of topology optimization calculation, which cannot be applied to high-level transmission components in this project. For this reason, the grid filtering method can be used to avoid the above phenomena\(^{[16]}\). Its purpose is to modify the sensitivity equation of overall flexibility with the following formula:

\[
\frac{\partial \tilde{C}}{\partial \rho_e} = \frac{1}{\rho_e \sum_{i=1}^{N} H_i \rho_e} \sum_{i=1}^{N} H_i \rho_e \frac{\partial C}{\partial \rho_e} \quad (12)
\]
\( \hat{C} \) is Object function after mesh filtering, 
\[
H_i = r_{\min} - \text{dist}(i,k), \ |i| \in N \ |\text{dist}(i,k) \leq r_{\min} \ , \ r_{\min} \text{ is half of the predefined minimum unit diameter, dist}(i,k) \text{ is the distance of unit } i \text{ and } k. \\
4. Topology optimization results and simulation verification
According to the structural model and boundary conditions of the supporting plate (Fig. 7), the OC optimization criterion method is used to optimize the topology of the supporting plate by using the topology optimization model proposed in the previous paper. The optimized results are shown in Fig. 6.

![Fig. 6 The result of topological optimization](image)

By comparing the aluminium profile of the support plate, the structure of the support plate after topology optimization is simpler than that of the aluminium profile. Combined with the analysis of the boundary conditions of the supporting plate, this distribution can distribute the force of the structure well, which can save material and enhance the stiffness of the supporting plate at the same time. It should be pointed out that the optimized structure is full of irregular lines and structures, and it is difficult to achieve mechanical processing. Therefore, it must be manufactured to meet the requirements of mechanical processing. The results of processing are shown in Figure 7.

![Fig. 7 Manufacturability process result](image)

In order to compare the performance, QY-8-2080W aluminium profile support plate (performance parameters are shown in Table 1 is selected. Its cross-section size meets the requirement of 20mm x 80mm. In order to verify the change of stiffness before and after optimization, the finite element simulation of aluminium profile and the support plate after topology optimization are carried out. In Ansys software, solid element SOLID92 is used to mesh. Both materials are 6063-T5 hard aluminium alloy (elastic modulus \( E=73 \text{GPa} \), density 2700kg/m\(^3\), tensile strength 205Mpa). The boundary conditions are shown in Fig.8. Fifteen concentrated loads \( F_{1/2} \) and \( q \), as well as their own gravity \( G \), are uniformly distributed on one end of the supporting plate and the other end of the supporting plate under a concentrated load \( M/2 \). Because the exterior dimensions, material parameters and boundary conditions of the structure obtained by the optimization of aluminium profile structure and topology are the same, only the maximum deflection of the structure under load is needed to be compared. The final stiffness comparison is shown in Table 1.

| Table1 Parameters of the chosen profile |
|-----------------|-----------------|-----------------|-----------------|
| \( E/\text{GPa} \) | \( \rho/\text{kg} \cdot \text{m}^{-1} \) | \( I_x/\text{cm}^4 \) | \( I_y/\text{cm}^4 \) |
| 0.73            | 2.65            | 25.6            | 3.64            |

![Fig. 8 Stiffness comparison between pre and post optimization](image)
From Fig.9 and Table 2, it can be seen that the maximum deflection of the optimized structure under the same load is only 51.4% of the deflection of the aluminum profile, and the maximum stress is only 93.1 MPa, which can meet the performance requirements of the supporting plate.

5. Conclusion

The equipment system for cargo transportation and loading on the ground of cargo spacecraft is designed, which realizes the horizontal and vertical lifting of cargo bags: the horizontal transmission function is realized by power rollers, and the vertical lifting function is realized by four synchronous transmission. Chains, which has high transportation efficiency and high automation level. On the basis of accomplishing the function of transporting cargo bags horizontally, the hoist can realize the adjustment of transporting height, which is suitable for cargo holds of different heights. When not working, it can also shrink into the hoist with compact structure. Minimize flexibility is the objective, SIMP material interpolation model and OC optimization criterion are used to optimize the topology of the support plate. The finite element simulation results show that the end deflection is reduced by 49%, and the stiffness performance of the support plate is improved.

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