Container built-in loading and unloading device design

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Abstract. Aiming at the problem of low efficiency of logistics loading and unloading tools at the present stage, in this paper, a container loading and unloading device with minimal container volume ratio is designed. The mechanical model of each part of the device is established, the local parameters of the device are designed by MATLAB software, and the device is simulated and verified by ANSYS software. The results show that the design of the device is reasonable. Under the working condition of ultimate load 577.98 kg, the telescopic arm is allowed to extend 1 m outward to the container, the range of transverse movement in the container is 1.55 m, and the range of longitudinal movement is 4.36 m, which meets the design requirements of rigidity and strength, and accounts for only 7.34% of the volume of the container, it can meet the requirement of on board.

1. Introduction
With the rapid development of e-commerce, the requirements for logistics and transportation efficiency are constantly improved. Loading, unloading and handling are the operation links with concentrated manpower and material resources, which directly affect the efficiency and cost of the entire logistics operation system [1]. The existing logistics loading and unloading are mainly carried out by combining tail plate with forklift or forklift with manual loading and unloading [2]. Although these two methods can complete the basic loading and unloading tasks, the loading and unloading efficiency is not high, especially when the height of the container from the ground is large and the container is long, the loading and unloading of heavy logistics parts is particularly laborious, and even requires additional auxiliary equipment for secondary loading and unloading, greatly increasing the loading and unloading time and cost. At present, for most logistics enterprises, saving cost and improving efficiency is one of the main goals of logistics and transportation [3]. In view of the general situation in reality, this paper designs a kind of built-in loading and unloading device which takes up a very small proportion of container volume.

2. Container built-in loading and unloading device scheme

2.1. Load-bearing beam and telescopic boom
In order to achieve the translation of telescopic boom in the carriage, the telescopic boom adopts the rack and pinion drive mode to ensure that there is enough stacking space for the goods in the container compartment. Its structure is shown in Figure 1.

The loading and unloading device track are composed of a channel steel 1 fixed on the inner wall of the carriage. The rack 2 is installed in the groove. The drive gear of the telescopic arm meshes with
it and can drive the telescopic arm to move forward and backward. The bottom of the telescopic arm is equipped with a guide wheel, which is used to reduce friction and withstand gravity [4].

(a) Structural drawing of crane telescopic boom. (b) Schematic diagram of telescopic arm drive device. (c) Telescopic boom structure of local structure schematic.

Figure 1. Schematic diagram of telescopic boom structure.
1- channel steel bearing beam 2- rack 3- drive gear 4- drive motor 5- reducer 6- drive gear wheel 7- telescopic arm 8- regulating wheel

2.2. Longitudinal crown block
As shown in Figure 2, the telescopic boom is equipped with rack 3, which is matched with the drive gear 2 installed at the middle and rear ends of the telescopic boom. The drive motor 5 is driven by the reducer 6 to drive the gear to rotate, thus driving the longitudinal crown block to move in the telescopic boom.

(a) Longitudinal crown assembly drawing. (b) Longitudinal crown structure drawing.

Figure 2. Diagram of longitudinal crown block.
1- channel steel bearing beam 2- drive gear 3- rack 4- drive gear 5- drive motor 6- reducer 7- longitudinal crown block rear channel steel beam 8- guide wheel

2.3. Transverse crown
Horizontal crown block diagram is shown in Figure 3. Transverse crowns can move in longitudinal crowns, and the principle of movement is the same as that of longitudinal crowns.

(a) Horizontal crown block structure drawing. (b) Horizontal crown block assembly diagram.

Figure 3. Diagram of horizontal crown block.
1- driving gear 2- electric hoist 3- motor and reducer 4- regulating wheel 5- bearing frame 6- rack 7- Channel steel bearing beam front end
3. Theoretical calculation and analysis

3.1. Force analysis of built-in loading and unloading devices of the container

(1) Transverse crane force model

Figure 4 shows the simplified force analysis diagram of transverse crown block.

![Figure 4. Transverse crown block diagram.](image)

The force analysis of the transverse crown block model shows that: \( F_H = F_I = G + G_1 \).
- \( G \) -- gravity of lifting load (assuming unbiased cargo);
- \( G_1 \) -- transverse crown block weight;
- \( F_H, F_I \) - longitudinal crown block supporting force to transverse crown block.

(2) Force model of longitudinal crown block

The simplified longitudinal crown block model and coordinate definition are shown in Figure 5.

![Figure 5. Definition sketch of vertical crane coordinates.](image)

1- transverse crown 2- longitudinal crown

Figure 6 shows the simplified force analysis diagram of the longitudinal crown block.

![Figure 6. Load diagram of longitudinal crown block.](image)
Force analysis of longitudinal crown block model:

\[
\begin{align*}
\sum F &= G + G_1 + G_2 \\
\sum M &= 0 \\
F_A L_1 - (G + G_1)(L_1 - X) - G_2 L_1^2 &= 0 \\
F_A' &= F_A \\
a_1 + a_2 &\leq X \leq L_1 - a_1 - a_2
\end{align*}
\]  

(1)

G2 -- longitudinal crown block dead weight; 
X -- the distance from the center point of transverse crown block to the center face of longitudinal crown block roller; 
a1 -- the distance between the inner end face of the longitudinal crown block and the center face of the longitudinal crown block roller; 
a2 -- the distance between the centerline of electric hoist rotation and the outer end face of transverse crown crane; 
FA, FB -- supporting force of telescopic beam fulcrum A and B to longitudinal crown block; 
FA' -- longitudinal crown block pressure on telescopic boom.

(2) Stress model of telescopic arm

The force diagram of the telescopic arm is shown in Figure 7.

\[\text{Figure 7. Force diagram of telescopic arm.}\]

3 longitudinal crown block roller

The force analysis of the telescopic boom model shows that:

\[
\begin{align*}
G_3 + F_A &= F_C + F_D \\
\sum M &= 0 \\
F_C (L_2 - y) - F_A' (L_2 - Z) - G_3 L_2^2 &= 0 \\
F_C' &= F_C \\
M_1 - F_C y &= 0 \\
Z &= a_3 + X
\end{align*}
\]  

(2)

G3 -- gravity of telescopic arm. 
Z -- the distance between the centerline of the electric hoist and the inner end face of the load-bearing beam. 
A3 -- the distance between the center face of the longitudinal crown roller and the inner end face of the heavy beam. 
Y -- width of bearing surface of external bearing beam. 
FC, FD -- supporting force of bearing beam fulcrum C and D on telescopic beam; 
FA' -- reaction force of telescopic boom on bearing beam fulcrum A; 
M1 -- moment of section of load-bearing beam.

The stress diagram of the telescopic arm is shown in Figure 8.
Figure 8. Schematic diagram of extension stress of telescopic boom.

The force analysis of the unilateral telescopic beam shows that:

\[
\begin{align*}
G_4 &= G_3 \frac{L}{L_4} \\
F_A + G_4 - F_G &= 0 \\
\sum M_F &= 0 \\
F_A L_3 - F_G (L_4 - L) + \frac{G_4}{2} \times \frac{L}{2} &= 0
\end{align*}
\] (3)

- \(L\) -- the extension length;
- \(L_3\) -- the distance from the inner end of the telescopic beam to the end of the container;
- \(L_4\) -- total length of telescopic arm;
- \(F\) -- box tail support point;
- \(G_4\) -- external extension weight.

When the elongation of the telescopic boom is the maximum, the bending moment of the telescopic boom reaches the maximum value.

(4) Calculation of container volume ratio

In order to meet the requirement of cargo-carrying, the volume ratio of built-in loading and unloading device is not appropriate. The calculation model of container volume ratio is shown as follows:

\[
\begin{align*}
\vartheta &= \frac{V_1}{V} \\
V_1 &= L' + H' + W' \\
V &= L + H + W
\end{align*}
\] (4)

- \(V_1\) -- volume of container handling device; \(L, H, W\) -- length, width and height of the container.
- \(V\) -- container contents volume; \(L', H', W'\) -- length, width and height of the handling device.

3.2. Check the parameters of built-in loading and unloading devices of the container

(1) Working condition requirements

In this paper, the commonly used 20-foot logistics container is taken as an example to define the loading and unloading device inside the container as shown in Table 1, considering the space layout requirements such as motor and drive shaft.

| Parameter                                      | Specification |
|------------------------------------------------|---------------|
| Maximum overhanging length of the overhanging beam | 1 m           |
| Payload target                                  | 500 kg        |
| Lateral crown block displacement range          | 155-1705 mm   |
| Maximum displacement of longitudinal crown-block | 320-4680 mm   |
| Maximum lifting height                          | 2500 mm       |

When the telescopic boom has the maximum elongation and the transverse crane moves to one side, it is a dangerous working condition. At this time, the telescopic boom and the load-bearing beam in the device are subject to the maximum bending moment and the maximum contact stress. Therefore, this paper mainly conducts parameter calculation and check on the bending moment and contact stress of the telescopic boom and the load-bearing beam.
(2) Check the bending moment of telescopic arm

Telescopic crane in loading and unloading device used for Q345 steel grade and models for 16 b (Channel steel 16b specification 160x 65 x 8.5mm, theoretical weight per meter 19.752 metric) of common channel, check steel allowable stress table shows steel grade Q345 yield strength limit for the \( \delta_s = 345 \text{MPa} \), Chad Q345 for material lifting machinery safety coefficient of girders for \( n = 1.4 \sim 1.6 \), \( n = 1.4 \), check the steel table shows model for 16 b hot rolled common channel steel bending section modulus of \( W_y = 17.6 \text{cm}^3 \), allowable stress and the maximum allowable bending moment. For [5]:

\[
\sigma = \frac{\delta_s}{S} = \frac{345}{1.4} = 246.43 \text{MPa}
\]

\[
M_{y_{\text{max}}} = 246.43 \times 10^6 \text{Pa} \times 17.6 \times 10^{-6} \text{m}^3
\]

\[
= 4337.16 \text{N} \cdot \text{m}
\]

According to Equations (1) and (3), the spatial curve equation of the distance between bending moment, lifting load and transverse crown block displacement to the central surface of the roller can be obtained:

\[
M = \frac{L_3 \times [G_2 \frac{L_1}{2} + (G + G_1) \times (L_1 - x)]}{L_1} + \frac{G_4 \times L}{2}
\]

MATLAB software is used to input the function model, and the limits \( M \) and boundaries \( x \) are as follows:

\[
0 \leq M \leq 4337.16 \text{N} \cdot \text{m}
\]
\[
0.155 \leq x \leq 1.705 \text{m}
\]

According to the design goal, substituting the outward expansion length =1 m, the distance between the inner end of the crane and the end of the container \( L_3=0.8\text{m} \), the curves of bending moment, lifting load and displacement are shown in Figure 9. It can be seen that when the distance decreases, the maximum allowable lifting load also decreases. Therefore, the maximum safe load bearing limit should be the allowable lifting load corresponding to the minimum distance and maximum bending moment, substitute \( x=0.155 \text{m}, \ M=4337.16 \text{N} \cdot \text{m} \), and get \( G = 5779.8 \text{N} \), that is, the lifting limit is 577.98kg.

![Figure 9. Relation diagram of bending moment, lifting load and displacement.](image)

(3) Parameter optimization and check of external bearing beam

Equations (2) and (3) are used to establish the equations about \( y \) and FC, where \( h_1 = 0.115 \text{m} \) limit \( y \leq 0.115 \text{m} \). The bearing limit \( G \), which does not affect the x limit, and the bearing contact
line peak value under the roller thickness of the telescopic beam are defined. When the failure probability is 1/100, the safety coefficient is 1, and the contact fatigue limit is [6]:

\[
[\sigma_z] = \frac{\sigma_H}{S} = \frac{345}{1} = 345 \text{MPa}
\]

The maximum compressive stress at the initial contact line represents the stress after contact between two parts, which is called contact stress and is expressed by symbol \( \sigma_H \). The calculation of contact stress is a problem of elasticity. According to the contact stress formula given by elasticity [7], the contact stress of the telescopic arm with 80 rollers on one side to the external load-bearing beam is deduced.

\[
\sigma_H = \frac{F_c \times r_1}{\pi \times 160y \left( \frac{1 - \mu_1^2}{E_1} + \frac{1 - \mu_2^2}{E_2} \right)}
\]

\( F_c \) -- Pressure acting on the contact surface  
\( r_1 \) -- radius of telescopic arm roller  
\( h_1 \) -- Roller thick  
\( \mu_1, \mu_2 \) -- Poisson's Ratio of Materials for Parts 1 and 2  
\( E_1, E_2 \) -- Elastic modulus of materials for parts 1 and 2

The contact surfaces are defined as Q345 materials.
MATLAB, programming input Formula (2), (3), computing in license
Under the limit of \( y \) area: \( 0.0617 \leq y \leq 1.8683 \), combined with the original scope \( y \)
To: \( 0.0617 \leq y \leq 0.115 \)
The relation diagram between \( y \) and \( F_c \) can be obtained and it was shown in Figure 10.
The relationship between \( \sigma_H \) and \( F_c \), \( y \) the three variables is shown in Figure 11.
It can be known that \( F_c \) is inversely proportional to \( y \), and the increase of \( y \) significantly decreases \( \sigma_H \).
In conclusion, the bearing section width \( y \) of the external load-bearing beam is 0.115mm, and the contact stress of the telescopic arm on the external load-bearing beam is obtained as follows:

\( \sigma_H = 256.38 \text{MPa} \)

Therefore, it is less than the contact fatigue limit \([\sigma_z]\). The moment of section of bearing beam is calculated as follows:

\( M_1 = 325.90 \text{N} \cdot \text{m} \)

Maximum allowable bending moment of section:

\[
M_{\text{max}2} = [\sigma] \times \left( \frac{y \times 10^2}{6} \right) = 345 \times 10^6 \text{pa} \times 1.91 \times 10^{-6} \text{m}^3 = 658.95 \text{N} \cdot \text{m}
\]

The results show that \( M_1 < M_{\text{max}2} \), bearing beam parameter design is reasonable, meet the strength requirements [8].

\[ F_c \]

\[ y \]

Figure 10. Relation curves of \( y \) and \( F_c \).
3.3. ANSYS simulation check

As an engineering analysis software, ANSYS is used to simulate the external load of mechanical structure system based on the finite element method, and then judge whether it meets the design requirements. This paper mainly conducts Static stress loading experiment on Static Structural. The steps in this experiment are: preparation - pretreatment - solving model - post-treatment result [9].

First, a simplified model for analysis is built in CATIA, and then the model is imported into ANSYS Workbench, the corresponding material characteristics of each part are defined in Engineering Data. See Table 2. Table 2 material parameters table

| Material | Elastic modulus (MPa) | Poisson's ratio (v) | Yield stress (MPa) | The main using |
|----------|-----------------------|--------------------|--------------------|----------------|
| Q235A    | 2.12E+05              | 0.288              | 235                | end beam       |
| Q345     | 2.06E+05              | 0.28               | 345                | main beam      |
| 20       | 2.13E+05              | 0.282              | 245                | Load hook      |
| 45       | 2.09E+05              | 0.269              | 355                | reducer shaft  |

Check and modify the model in the Design Modeler module, and perform the appropriate slicing operation, so as to partition the grid in the later period, reduce the distortion of the grid, and improve the efficiency high simulation accuracy. After finishing the operation in the Design Modeler module, enter Mechanical The module sets the contact type box to divide the model grid and define the indirect parts Contact type, applied load and set the solver type.

Where, the end face of the bearing beam of channel steel is defined as Fixed Support, the additional tension Force of lifting rope is 5779.8N, and the contact mode of pinion and rack is defined as Bonded.

The stress simulation cloud diagram is shown in Figure 12, and the strain simulation cloud diagram is shown in Figure 13.

It shows that the maximum shape variable is 0.001mm, the maximum stress is 214.48mpa, smaller than the allowable limit stress of 246.43mpa, and the design is reasonable.
Figure 12. Simulation stress cloud map.

Figure 13. Simulation of strain cloud map.

4. Calculation of unit ratio to container volume
According to Equation (4), take the commonly used internal dimensions of 20-foot container as an example to calculate the occupation volume ratio:

\[ \frac{V_1}{V} \times 100\% = \frac{5.69 \times 2.13 \times 0.16}{5.69 \times 2.13 \times 2.18} \times 100\% = 7.34\% \]

Based on the above calculation, it can be seen that the volume of the container occupied by the built-in loading and unloading devices is only 7.34%, which conforms to the on-board carrying condition.

5. Conclusions
(1) Under the extreme load condition, the telescopic arm was allowed to extend 1 m outwards to the container, and the transverse movement range in the container was 1.55 m, and the longitudinal movement range was 4.36 m;

(2) The maximum stress value of the device was 214.48mpa, the maximum shape variable was 0.001mm, and the allowable load limit of the cargo built-in loading and unloading device was 577.98kg;
(3) Compared with forklifts and tail plate devices, the total volume of loading and unloading devices was smaller, accounting for only 7.34% of the volume of the box, meeting the requirement of carrying on board.

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References
[1] Tao S and Hu Z H 2014 Systems engineering theory and practice 34(08) 1971-1985
[2] Huang F B, Yan Z and Chen X H 2016 Special Purpose Vehicle (04) 65-67
[3] Wang X C and Wang J Y 2014 An innovative research on a dumper that can automatically load and unload goods (Chong Qing: Chongqing University)
[4] Wang D 2015 Mechanical Design and Manufacturing (12) 45-47
[5] Lv H Y 2014 Fatigue life analysis of crane box girder based on equivalent structural stress method (Zheng Zhou: Zhengzhou University)
[6] Che D X 2007 Mechanical Design manual, Volume ii (BeiJing: Chemical Industry Press)
[7] Pu L G, Che G D and Wu L Y 2013 Mechanical Design (9th edition) Higher Education Press 34-35
[8] Wang X, Huang L, Gao Y, Gao S D and Wang Y H 2009 Journal of Dalian university of technology 49(03) 374-379
[9] Li X H and J L H 2014 Journal of Central South University of Forestry and Technology 34(01) 103-106