Optimization Design of Execution Mechanism for Detachable Container

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Abstract. In this paper, optimization design is carried out comprehensively for execution mechanism of the detachable container. Firstly, formulae are derived for kinematics analysis and force analysis of the execution mechanism. Secondly, a mathematical model is established for optimization design of the execution mechanism. Thirdly, half-penalty function method is selected as the optimization method. A C++ program is compiled for optimization design of the execution mechanism. Finally, an example of the execution mechanism is optimized by the compiled C++ program. Motion simulation of the optimized execution mechanism is carried out. The simulation result shows that the optimization result is satisfactory.

Introduction

Rapid global economic growth in the past decade has brought about the increase in urban waste. Detachable container truck is widely used to collect and transport urban garbage and other cargos.

The execution mechanism is the core component of the detachable container truck. It determines the performance of the detachable container truck. Several scholars have carried out optimization design of the execution mechanism for detachable container [1-4] by using MathCAD and ADAMS software, where the design variables are the coordinates of revolute joints of execution mechanism. However, as of today, there were no reports of optimization design of the execution mechanism, in which the dimension parameters of detachable container and carframe are considered. These parameters play a decisive role in the optimization design for the execution mechanism.

Kinematics Analysis

Figure 1 depicts the execution mechanism of detachable container. x axle is set on horizontal ground surface, and y axle is at the center of left side roller of carframe. Let l and h denote the length and height of carframe respectively. Let $l_x$ and $h_x$ denote the length and height of the container respectively. Let $r$ denote the radius of the rolling wheel. Let G, H denote the center of rolling wheels respectively, and let J denote the center of gravity of the container and cargo.

![Figure 1. Execution mechanism of detachable container.](image)

The parameters of the execution mechanism are the coordinates of revolute points A and B, the distance from point B to point C, the angle between segment BC and segment CD, etc.
From the geometry in figure 1, the angle $\gamma$ between segment BA and x axle is calculated by

$$\gamma = \arctan\left[\frac{(y_A - y_B)}{(x_A - x_B)}\right]$$  \hspace{1cm} (1)

The angle $\varphi$ between segment BC and x axle is calculated by

$$\varphi = \arccos\left[\left(\frac{l_{AB}^2 + l_{BC}^2 - l_{AC}^2}{2l_{AB}l_{BC}}\right)\right] + \gamma$$  \hspace{1cm} (2)

where $l_{AB} = \sqrt{(x_A - x_B)^2 + (y_A - y_B)^2}$.

The coordinates of point C satisfy

$$\begin{cases} 
x_C = x_B + l_{BC} \cos \varphi \\
y_C = y_B + l_{BC} \sin \varphi
\end{cases}$$  \hspace{1cm} (3)

The angle $\phi$ between segment CD and x axle is

$$\phi = \varphi - \pi + \alpha$$  \hspace{1cm} (4)

The coordinates of point D satisfy

$$\begin{cases} 
x_D = x_C + l_{CD} \cos \phi \\
y_D = y_C + l_{CD} \sin \phi
\end{cases}$$  \hspace{1cm} (5)

The angle $\theta$ between segment DE and x axle is

$$\theta = \phi + \pi - \beta$$  \hspace{1cm} (6)

The coordinates of point E satisfy

$$\begin{cases} 
x_E = x_D + l_{DE} \cos \theta \\
y_E = y_D + l_{DE} \sin \theta
\end{cases}$$  \hspace{1cm} (7)

There are two cases for the coordinates of rest points. The first case is that the container is in contact with the ground as shown in figure 1(a), and the second case is that the container leaves off the ground as shown in figure 1(b).

From the geometry in figure 1(a), the distance $l_{EH}$ is

$$l_{EH} = \sqrt{(l_x - l_H)^2 + h_e^2}$$  \hspace{1cm} (8)

The coordinates of point H satisfy

$$\begin{cases} 
x_H = x_E - \sqrt{l_{EH}^2 - (y_E - r)^2} \\
y_H = r
\end{cases}$$  \hspace{1cm} (9)

The angle $\psi$ between segment HE and x axle is

$$\psi = \arctan\left[\frac{(y_E - y_H)}{(x_E - x_H)}\right]$$  \hspace{1cm} (10)

The angle $\varphi_1$ between segment HF and x axle is

$$\varphi_1 = \psi - \arctan\left[\frac{h_e}{(l_x - l_H)}\right]$$  \hspace{1cm} (11)

The coordinates of point F, I and J satisfy
\[
\begin{align*}
    x_F &= x_H + l_{HF} \cos \varphi_i , \quad x_I = x_F - h \sin \varphi_i , \quad x_J = x_I - l \cos \varphi_i \\
    y_F &= y_H + l_{HF} \sin \varphi_i , \quad y_I = y_F + h \cos \varphi_i , \quad y_J = y_I - l \sin \varphi_i \\
\end{align*}
\]

From the geometry in figure 1(b), the angle \( \psi \) between segment GE and x axle is

\[
\psi_i = \begin{cases} 
\arctan \left( \frac{(y_E - h) / x_E} \right) & x_E \geq 0, \quad y_E - h \geq 0 \\
\arctan \left( \frac{(y_E - h) / x_E} \right) + \pi & x_E < 0 \\
\arctan \left( \frac{(y_E - h) / x_E} \right) + 2\pi & x_E > 0, \quad y_E - h \leq 0 
\end{cases}
\]

In order to save space, the rest formulae for kinematics analysis are not shown explicitly.

**Force Analysis**

There are two cases for the force analysis of container. The first case is that the container is in contact with the ground as shown figure 2(a), and the second case is that the container leaves off the ground as shown figure 2(b).

![Diagram of container](image)

**Figure 2. Force analysis of container.**

According to the moment balance about point E and force balance in figure 2(a), we can obtain

\[
F_{H_y} = (x_e - x_J)W / (x_E - x_H) 
\]

\[
\begin{align*}
    F_{E_x} &= 0 \\
    F_{E_y} &= W - F_{H_y} 
\end{align*}
\]

According to container force analysis as shown figure 2(b), we can obtain

\[
\begin{align*}
    F_{G_x} &= -F_G \sin \varphi_2 \\
    F_{G_y} &= F_G \cos \varphi_2 
\end{align*}
\]

where \( \varphi_2 = \psi_i - \arcsin \left( \frac{h_e}{l_{EG}} \right) \).

In light of the moment balance about point E and force balance in figure 2(b), we can obtain

\[
F_G = \frac{W(x_E - x_J)}{(y_E - h) \sin \varphi_2 + x_E \cos \varphi_2}
\]

\[
\begin{align*}
    F_{E_x} &= F_G \sin \varphi_2 \\
    F_{E_y} &= W - F_G \cos \varphi_2 
\end{align*}
\]

According to the geometry in figure 1, the angle \( \varphi_3 \) between segment AC and x axle is
\[
\varphi_3 = \begin{cases} 
\arctan \left( \frac{y_C - y_A}{x_C - x_A} \right) & x_C - x_A \geq 0, \ y_C - y_A \geq 0 \\
\arctan \left( \frac{y_C - y_A}{x_C - x_A} + \pi \right) & x_C - x_A < 0 \\
\arctan \left( \frac{y_C - y_A}{x_C - x_A} + 2\pi \right) & x_C - x_A > 0, \ y_C - y_A \leq 0 
\end{cases}
\]

(19)

The two components of \( F_{AC} \) in the \( x \) and \( y \) directions as shown in figure 3 can be determined by

\[
\begin{align*}
F_{C_Ax} &= F_{C_A} \cos \varphi_3 \\
F_{C_Ay} &= F_{C_A} \sin \varphi_3
\end{align*}
\]

(20)

In light of the moment balance about point B and force balance in figure 3, we can obtain

\[
F_{C_A} = \frac{\frac{F_{E_x} (y_E - y_B)}{y_C - y_B} - \frac{F_{E_y} (x_E - x_B)}{y_C - y_B} \cos \varphi_3 - (x_C - x_B) \sin \varphi_3}{(y_C - y_B) \cos \varphi_3 - (x_C - x_B) \sin \varphi_3}
\]

(21)

\[
\begin{align*}
F_{B_x} &= F_{E_x} - F_{C_A} \cos \varphi_3 \\
F_{B_y} &= F_{E_y} - F_{C_A} \sin \varphi_3
\end{align*}
\]

(22)

![Figure 3. Force analysis of lifting arm.](image)

**Mathematical Model**

**Design Variables:** In this optimization design, design variables are the coordinates of revolute joints A and B, all dimension parameters of the execution mechanism, and the extension ratio \( \lambda_{AC} = \frac{l_{AC_{max}}}{l_{AC_{min}}} \) of hydraulic cylinder.

\[
X = (x_1, x_2, \ldots, x_{12})^T = (x_A, y_A, x_B, y_B, l_{AC_{min}}, l_{BC}, l_{CD}, l_{DE}, h, \lambda_{AC}, \alpha, \beta)^T
\]

(23)

**Objective Function:** In order to minimize hydraulic cylinder force, the maximum force \( F_{C_A} \) needs to be minimized, that is

\[
\min f_1(X) = \min \left[ \max F_{C_A}(X) \right]
\]

(24)

In order to reduce the construction size of the execution mechanism, we introduce the sub-objective function

\[
\min F_2(X) = \min (l_{AC_{min}} + l_{BC} + l_{CD} + l_{DE})
\]

(25)

In sum, the objective function of optimization design of execution mechanism for detachable container is

\[
\min F(X) = w_1 f_1(X) + w_2 F_2(X)
\]

(26)

where \( w_1 \) and \( w_2 \) are weighting factors.
**Constraint Conditions:** The conditions of the sub-mechanism for hydraulic cylinder are

\[
I_{AC_{\text{min}}} + I_{BC} > I_{AB}, \quad I_{AC_{\text{min}}} + I_{AB} > I_{BC}, \quad I_{BC} + I_{AB} > \lambda_{AC} I_{AC_{\text{min}}}
\]  
(27)

\[
\lambda_{AC}^2 I_{AC_{\text{min}}} + I_{BC}^2 - I_{AB}^2 - 2 \lambda_{AC} I_{AC_{\text{min}}} I_{BC} \cos \gamma_{\text{min}} \leq 0
\]
(28)

\[
I_{AC_{\text{min}}} + I_{BC}^2 - I_{AB}^2 - 2 I_{AC_{\text{min}}^2} I_{BC} \cos (\pi - \gamma_{\text{min}}) \geq 0
\]
(29)

where \(\gamma_{\text{min}}\) is the minimum transmission angle.

Let the upper and lower bounds of variable \(x_i\) be \(x_{i_{\text{min}}}\) and \(x_{i_{\text{max}}}\). Constraints of the upper and lower bounds of variables are

\[
x_{i_{\text{min}}} \leq x_i \leq x_{i_{\text{max}}} \quad (i = 1, 2, \ldots, 12)
\]  
(30)

**Example**

In this optimization design, the dimension parameters of carframe are \(l=4000\)mm, \(h=1450\)mm. The dimension parameters of container are \(l_c=4000\)mm, \(h_c=1700\)mm, \(l_f=150\)mm, and \(r=100\)mm. The weight of container and goods is \(W=10000\)kg, and the center \(J\) of gravity of the container and cargo is the geometric center of the container. Let \(\gamma_{\text{min}}=16^\circ\), \(w=1\) and \(w_2=0.4\). The upper and lower bounds of variables are listed in table 1. A half-penalty function method [5,6] is selected as the optimization method, and a feasible point is found by the efficient method in reference [7]. A C++ program is compiled for the optimization design of the execution mechanism. And the optimal results of this example are listed in table 1.

<table>
<thead>
<tr>
<th>variable (x_i)</th>
<th>lower bound (x_{i_{\text{min}}})</th>
<th>upper bound (x_{i_{\text{max}}})</th>
<th>optimal solution (x_i)</th>
<th>lower bound (x_{i_{\text{min}}})</th>
<th>upper bound (x_{i_{\text{max}}})</th>
<th>optimal solution (x_i)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(x_A)</td>
<td>3000</td>
<td>4100</td>
<td>4100</td>
<td>800</td>
<td>1700</td>
<td>1679.89</td>
</tr>
<tr>
<td>(y_A)</td>
<td>1000</td>
<td>1300</td>
<td>1289.94</td>
<td>1300</td>
<td>1700</td>
<td>1697.93</td>
</tr>
<tr>
<td>(x_B)</td>
<td>1000</td>
<td>2800</td>
<td>1528.05</td>
<td>1200</td>
<td>1700</td>
<td>1641.23</td>
</tr>
<tr>
<td>(y_B)</td>
<td>1000</td>
<td>1300</td>
<td>1000</td>
<td>1.33</td>
<td>1.75</td>
<td>1.7429</td>
</tr>
<tr>
<td>(l_{AC_{\text{min}}})</td>
<td>1000</td>
<td>2000</td>
<td>1858.83</td>
<td>150</td>
<td>185</td>
<td>159.88</td>
</tr>
<tr>
<td>(l_{BC})</td>
<td>450</td>
<td>850</td>
<td>769.972</td>
<td>85</td>
<td>96</td>
<td>90.347</td>
</tr>
</tbody>
</table>

\(F_{i}(X)\approx6006.62\) \quad \(F_{i}(X)\approx40066\) \quad \(F(X)\approx22033.02\)

\(F_{\text{Gmax}}\approx6881\text{kg}\) \quad \(F_{\text{Gmax}}\approx5303\text{kg}\) \quad \(F_{\text{Gmax}}\approx40493\text{kg}\) \quad \(F_{\text{Cmax}}\approx40066\text{kg}\)

Figure 4 shows motion simulation of the optimized execution mechanism. Figure 4(a) is shows beginning motion. Figure 4(b) shows that the container is about to leave off the ground. Figure 4(c) shows that the container has leaved off the ground, and at this position, the hook force and the roller G force have reached maximal value, that is \(F_{\text{Emax}}\approx6881\text{kg}\) and \(F_{\text{Gmax}}\approx5303\text{kg}\). Figure 4(d) shows that the container has finished motion position, and at this position, the minimum transmission angle of the sub-mechanism for hydraulic cylinder is \(22.1689^\circ\), the maximum force of the hinge point B is \(F_{\text{Bmax}}\approx40493\text{kg}\), and the maximum force of hydraulic cylinder is \(F_{\text{Cmax}}\approx40066\text{kg}\).
Summary

This paper conducts a comprehensive study on the optimization design of execution mechanism for detachable container. Formulae of motion and force are derived of execution mechanism for detachable container. To minimize the pressure force for the hydraulic cylinder and to reduce the size of the execution mechanism, a mathematical model is established of the execution mechanism for the detachable container. This paper provides theoretical basis for the execution mechanism design of the detachable container.

References


