Optimization of garbage dumping mechanism of intelligent sanitation vehicle based on particle swarm algorithm

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Abstract. Aiming at the requirements of low power consumption and light weight of the intelligent garbage collection and transportation equipment of sanitation vehicles, this paper proposes an optimization method based on the particle swarm algorithm. In this paper, the kinematic equation and dynamic equation of the mechanical claw in the dumping action are derived by using the complex interpolation approach. The optimization objectives are the minimum maximum power, the minimum average power, the minimum maximum mutation of the instantaneous driving force and the minimum time-consuming of the dumping action. The garbage dumping angle is selected as constraints, and the arc radius of the dumping track, the centre distance of the mechanical claw’s groove wheels and the time-consuming of the dumping action are optimized. The optimization results show that the optimization objectives have been well optimized except for the increased time-consuming dumping action. And the driving force should act on the axis of the lower groove wheels.

1. Introduction
With the rapid development of cities, the disadvantages of low automation level and high labor cost of traditional sanitation vehicles have become more and more obvious. For this reason, many companies have developed garbage collection and transportation equipment with intelligent control system, which can identify the type and position of garbage bins, and complete the loading work automatically.

At present, intelligent garbage collection and transportation equipment can be divided into raising type and lifting type according to the mechanical structure. Among them, the raising type is relatively mature because its structure is similar to traditional industrial robots, and there are many research achievements on the optimization of the similar raising mechanism. For example, Yang Ping \cite{1} applied the response surface method to lightweight design of the raising mechanism. Li Zigui \cite{2} used ADAMS to optimize the working mechanism of the articulating boom crane with multiple objectives. Zhang Yan \cite{3} optimized the rod length and other parameters of the loader working mechanism based on the SQP algorithm. For the lifting type, Wang Zhongchang \cite{4} used ADAMS and ISIGHT and based on the genetic algorithm to optimize the traditional bucket lifting mechanism. However, such mechanism cannot meet the structural requirements of intelligent garbage collection and transportation equipment.

At present, there is a new type of intelligent garbage collection and transportation equipment \cite{5}, as shown in Figure 1. The guide rail in the figure is composed of linear section guide rail and circular section guide rail, and it is one of the components of the truss structure 3. The mechanical claw 2 moves...
along the guide rail under the drive of the hydraulic motor to realize the lifting and dumping function of load 1. The transmission mode of the equipment is chain transmission.

Figure 1  A new type of intelligent garbage collection and transportation equipment

Due to some unreasonable structural parameters in the initial design scheme, problems such as track deformation, large driving force and uneven operation are caused. In this paper, this equipment is taken as the research object. The kinematic and dynamic equation of the mechanical claw is established by complex interpolation approach. And the particle swarm algorithm is used to optimize the parameters of the arc radius of the dumping track, the center distance of the mechanical claw’s groove wheels and the time-consuming of the dumping action. Finally, ADAMS is used to verify the optimization results.

2. Kinematic and dynamic analysis of the mechanical claw

2.1. Establishment of the kinematic equation

The position relationship between the mechanical claw and the dumping track is shown in Figure 2. In the figure, \( P_1 \) is the center point of the upper groove wheel; \( P_2 \) is the center point of the lower groove wheel; \( P_{cm} \) is the equivalent center of mass of the mechanical claw and the load; \( r_1 \) is the arc radius of the dump track; \( l_1 \) is the center distance of the upper and lower groove wheels; \( l_2 \) is the distance from the center of the groove wheel to the back of the mechanical claw; \( \alpha \) is the angle that the upper groove wheel has rotated; \( \beta \) is the pitch angle of the mechanical claw.

\[
\beta = \arcsin \left( \frac{r_1 - r_1 \cos \alpha}{l_1} \right) \quad (1)
\]

As the mechanical claw cannot interfere with the track, the distance \( l_2 \) can be expressed as follows:

\[
l_2 = r_1 - \sqrt{r_1^2 - 0.25l_1^2} + 0.5d \quad (2)
\]

In the formula, \( d \) is the track width.

In this paper, the dumping process is divided into two sections of motion, as shown in Figure 3. The time intervals of the two sections are \([0, t_1]\) and \([t_1, t_2]\). Since the measurement of the chain linear
velocity is more convenient, the chain linear velocity \( v_0 \) is set to be constant. The expressions of the linear velocity \( v_0 \) and the time point \( t_1 \) are as shown in Eq. (3) and Eq. (4), respectively.

\[
v_0 = k_1 \frac{\pi r_1}{t_2} + k_2 \frac{l_1 + (\pi - \theta) r_1}{t_2}
\]

\[
t_1 = k_1 \frac{r_1 \theta}{v_0} + k_2 \frac{l_1}{v_0}
\]

The rotation angle \( \alpha \) of the upper groove wheel can be expressed as follows:

\[
\alpha = \begin{cases} 
  k_1 \frac{v_0 t}{r_1} + k_2 \left[ \arccos \left( \frac{2r_1^2 + (l_1 - v_0 t)^2 - l_1^2}{2r_1(r_1^2 + (l_1 - v_0 t)^2)^{\frac{3}{2}}} \right) - \arcsin \left( \frac{l_1 - v_0 t}{(r_1^2 + (l_1 - v_0 t)^2)^{\frac{3}{2}}} \right) \right], & 0 \leq t < t_1 \\
  \frac{v_0 (t - t_1)}{r_1} + \theta, & t_1 \leq t \leq t_2
\end{cases}
\]

According to the defined angles, the coordinates of point \( P_1 \), point \( P_2 \) and point \( P_{cm} \) in the first section of motion can be expressed as follows:

\[
\begin{align*}
P_1' &= (r_1 \cos \alpha, r_1 \sin \alpha) \\
P_2' &= (r_1, r_1 \sin \alpha - l_1 \cos \beta) \\
P_{cm}' &= (P_{cmx}', P_{cmy}')
\end{align*}
\]

In the formula, \( P_{cmx}' \) and \( P_{cmy}' \) can be specifically expressed as follows:

\[
\begin{align*}
P_{cmx}' &= \frac{1}{2} (r_1 \cos \alpha + r_1) + (l_2 + l_3) \cos \beta + l_4 \sin \beta \\
P_{cmy}' &= \frac{1}{2} (2r_1 \sin \alpha - l_1 \cos \beta) + (l_2 + l_3) \sin \beta - l_4 \cos \beta
\end{align*}
\]

The coordinates of these points in the second section of motion can be expressed in a similar way.

In this paper, the mechanical claw and the load are regarded as a rigid body with constant mass. And the relative position of the equivalent centre of mass remains unchanged. Then the speed and the acceleration of point \( P_1 \) and point \( P_2 \) can be obtained by Eq. (9) [6]:

\[
\begin{align*}
0 &= Re \left\{ \frac{(P_5 - P_1)^*(P_5 - P_1)}{|P_5 - P_1|^2} \right\} \\
\omega &= Im \left\{ \frac{(P_5 - P_1)^*(P_5 - P_1)}{|P_5 - P_1|^2} \right\} \\
-\omega^2 &= Re \left\{ \frac{(P_5 - P_1)^*(a_5 - a_1)}{|P_5 - P_1|^2} \right\}
\end{align*}
\]
In the formula, the superscript ‘c’ means that the quantity is in complex form; the superscript ‘*’ means the conjugate complex number of the quantity; \( v_1^c \) and \( v_2^c \) are the speed of point \( P_1 \) and point \( P_2 \); \( a_1^c \) and \( a_2^c \) are the acceleration of point \( P_1 \) and point \( P_2 \).

According to the complex interpolation approach, as shown in Eq. (10), the velocity and acceleration of the centre of mass \( P_{cm} \) can be obtained. This is the end of the establishment of the kinematic equation.

\[
\begin{align*}
    v_{cm}^c &= v_1^c + \frac{\hat{P}_{cm} - \hat{P}_1^c}{\hat{P}_2^c - \hat{P}_1^c} (v_2^c - v_1^c) \\
    a_{cm}^c &= a_1^c + \frac{\hat{P}_{cm} - \hat{P}_1^c}{\hat{P}_2^c - \hat{P}_1^c} (a_2^c - a_1^c)
\end{align*}
\]

**2.2. Validation of the kinematic equation**

In order to verify the correctness of the kinematic equation, this paper sets a set of parameters and calculates them through MATLAB and ADAMS respectively. The calculation results are shown in Figure 4 and Figure 5. These two results are basically the same. So, the kinematic equation is reliable.

2.3. Establishment of the dynamic equation

Based on the calculation of the speed and acceleration of the center of mass, the force situation of the mechanical claw is analyzed, as shown in Figure 6. In the figure, \( F_1 \) is the equivalent gravity of the mechanical claw and the load; \( F_{t1} \) and \( F_{t2} \) are the driving force; \( f_1 \) and \( f_2 \) are the friction force; \( N_1 \) and \( N_2 \) are the normal force.
When the mechanical claw moves in the linear section of the track, the speed and the acceleration of the mechanical claw remain unchanged. The driving force can be expressed as follows:

$$k_1 F_{t1} + k_2 F_{t2} = F_1 (2 \mu \frac{t_2 + t_5}{t_1} + 1)$$  \hspace{1cm} (11)$$

When the mechanical claw moves into the arc track, its speed and acceleration are changing at any time. According to the theorem of kinetic energy and the balance relationship of force, the dynamic equation of the mechanical claw in the first section of motion can be expressed as follows:

$$\begin{align*}
(k_1 F_{t1} - f_1) s_1 + (k_2 F_{t2} - f_2) s_2 - F_1 s_3 &= \Delta E_k \\
-(k_1 F_{t1} - f_1) \sin \alpha - N_1 \cos \alpha + N_2 &= ma_x \\
(k_1 F_{t1} - f_1) \cos \alpha - N_1 \sin \alpha + k_2 F_{t2} - f_2 - F_1 &= ma_y
\end{align*}$$  \hspace{1cm} (12)$$

In the formula, $s_1$ is the moving distance of the upper groove wheel in unit time; $s_2$ is the moving distance of the lower groove wheel in unit time; $s_3$ is the height change of the equivalent mass centre in unit time; $\Delta E_k$ is the kinetic energy change of the mechanical claw and the load in unit time. They can be expressed as follows:

$$\begin{align*}
s_1 &= r_1 (\alpha_{i+1} - \alpha_i) \\
s_2 &= r_1 (\sin \alpha_{i+1} - \sin \alpha_i) - l_1 (\cos \beta_{i+1} - \cos \beta_i) \\
s_3 &= P_{cm y,i+1} - P_{cm y,i} \\
\Delta E_k &= \frac{1}{2} m (v_{cm,i+1}^2 - v_{cm,i}^2)
\end{align*}$$  \hspace{1cm} (13)$$

Similarly, the dynamic equation of the mechanical claw in the second section of motion can be expressed as Eq. (14). This is the end of the establishment of the dynamic equation.

$$\begin{align*}
(k_1 F_{t1} - f_1) s_1 + (k_2 F_{t2} - f_2) s_1 - F_1 s_3 &= 0 \\
-(k_1 F_{t1} - f_1) \sin \alpha - N_1 \cos \alpha -(k_2 F_{t2} - f_2) \sin(\alpha - \theta) + N_2 \cos(\alpha - \theta) &= ma_x \\
(k_1 F_{t1} - f_1) \cos \alpha - N_1 \sin \alpha + (k_2 F_{t2} - f_2) \cos(\alpha - \theta) + N_2 \sin(\alpha - \theta) - F_1 &= ma_y
\end{align*}$$  \hspace{1cm} (14)$$

3. Parameter optimization based on particle swarm algorithm

3.1. Design variables

This paper takes the arc radius $r_1$ of the dump track, the center distance $l_1$ of the mechanical claw’s groove wheels and the time-consuming $t_2$ of the dumping action as design variables. Limited by the size and the total time cost, the value range of the design variables is shown as follows:

$$X = \left[r_1 \ l_1 \ t_2\right] \quad 50mm \leq r_1 \leq 200mm \\
80mm \leq l_1 \leq 210mm \\
1s \leq t_2 \leq 4s$$  \hspace{1cm} (15)$$

3.2. Constraints

In order to ensure the successful dumping of garbage, the rotation angle of the mechanical claw must be greater than or equal to 135°. So, the arc radius $r_1$ and the center distance $l_1$ must meet Eq. (16).

$$l_1 \leq r_1 \sqrt{2}$$  \hspace{1cm} (16)$$

The penalty equation is established as follows:

$$g(X) = \begin{cases}
0, & l_1 \leq r_1 \sqrt{2} \\
k, & l_1 > r_1 \sqrt{2}
\end{cases}$$  \hspace{1cm} (17)$$

In the formula, $k$ is a very large value.

3.3. Optimization objective function

In order to determine the optimization objective, this paper solves the instantaneous driving force required by the initial design scheme according to the established dynamic equation, as shown in Figure 7. The parameters of the initial design scheme are shown in Table 1. The total mass $m$ of the mechanical claw and the load is set to 318.08kg.
### Table 1 The initial design scheme parameter table

| Parameter | Value | Parameter | Value |
|-----------|-------|-----------|-------|
| $r_1$ (mm) | 120   | $k_3$     | 1     |
| $l_1$ (mm) | 150   | $k_2$     | 0     |
| $l_2$ (mm) | 51.3  | $\mu$    | 0.1   |
| $l_3$ (mm) | 386.89 | $d$ (mm)  | 50    |
| $l_4$ (mm) | 310.62 | $t_2$ (s) | 2     |

#### Figure 7 Instantaneous driving force required by the initial design scheme

During the dumping action, the maximum power is 3.76kW. The average power is 1.82kW. These two parameters will affect the mass and the cost of the hydraulic motor. At the same time, the maximum value of the instantaneous driving force mutation reaches 2442.92N, which is caused by the acceleration mutation. A large instantaneous driving force mutation will severely impact the hydraulic system and the control system. For the total time consumption of a dumping action, it determines the working efficiency.

Therefore, this paper takes the minimum maximum instantaneous power, the minimum average power, the minimum maximum mutation of the instantaneous driving force and the minimum time-consuming of the dumping action as the objectives. And the optimization objective function is established combined with the constraints, as shown in Eq. (18).

$$f(X) = g(X) + \min \left( \sum_{i=1}^{4} c_i \frac{Y_i}{Y_{i0}} \right)$$  \hspace{1cm} (18)

In the formula, $c_i$ is the weight coefficient; $Y_i$ is the optimization objective; $Y_{i0}$ is the value of the optimization objective under the initial design scheme.

#### 3.4 The basic principle of particle swarm optimization

Particle swarm optimization (PSO) was proposed by Kennedy and Eberhart in 1995 [7]. The basic idea of PSO is to solve the optimization problem by simulating the collective cooperative behavior of bird swarms flying for foods. Its core equations are shown as Eq. (19) and Eq. (20).

$$v_{in}(t + 1) = w v_{in}(t) + c_1 r_1 \left( P_{in, best} - x_{in}(t) \right) + c_2 r_2 \left( G_{in, best} - x_{in}(t) \right)$$  \hspace{1cm} (19)

$$x_{in}(t + 1) = x_{in}(t) + v_{in}(t + 1)$$  \hspace{1cm} (20)

In the formula, $v_{in}(t)$ is the velocity information of the nth dimension of particle $i$; $x_{in}(t)$ is the location information of the nth dimension of particle $i$; $w$ is the coefficient of inertia; $c_1$ and $c_2$ are the learning factors, generally $c_1 = c_2 = 2$; $r_1$ and $r_2$ are two random numbers with a value range of $(0,1)$.

In order to improve the performance of the algorithm, Y Shi [8] optimized the weight coefficient $w$, as shown in Eq. (21).

$$w(t) = w_{max} - (t - 1) \frac{w_{max} - w_{min}}{iterm}$$  \hspace{1cm} (21)

In the formula, $w_{max}$ is the final value of the weight coefficient; $w_{min}$ is the initial value of the weight coefficient; $iterm$ is the maximum number of iterations.
Finally, when the difference between the global extreme value of multiple iterations and the historical global extreme value is less than the set precision value or the number of iterations reaches the set maximum value, the iterative calculation of the particle swarm algorithm ends. And the global optimization result will be output.

3.5. **Optimization results**

In this paper, the scheme of driving force acting on the axis of the upper groove wheels and the scheme of driving force acting on the axis of the lower groove wheels were optimized respectively. The optimization results are shown in Table 2. Compared with the initial design scheme, the maximum instantaneous power was reduced by 1735.84W and 1827.46W, the average power was reduced by 789.18W and 701.61W, and the maximum instantaneous driving force mutation value was reduced by 1720.7N and 2093.6N, respectively. The dumping action time-consuming increased by 1.45s and 1.36s respectively, which is acceptable. It can be seen that the optimization effect of the lower side optimization scheme is the best. So, it should be applied as the optimization result.

| Table 2 Optimization results |
|-------------------------------|
| $r_1$ (mm) | $t_1$ (mm) | $t_2$ (s) | $P_{max}$ (W) | $\bar{P}$ (W) | $\Delta F_{max}$ (N) |
|-------------------------------|
| Initial design scheme | 120 | 150 | 2 | 3764.17 | 1817.43 | 2442.92 |
| Optimization scheme (upper side) | 130 | 157 | 3.45 | 2028.33 | 1028.25 | 722.22 |
| Optimization scheme (lower side) | 148.49 | 210 | 3.36 | 1936.71 | 1115.82 | 349.32 |

3.6. **Validation of the optimization result based on ADAMS**

The 3D model used for dynamic simulation in ADAMS is shown in Figure 8. And the parameter settings of the contact between the groove wheels and the track are shown in Table 3.

| Table 3 Contact parameter settings |
|-----------------------------------|
| Parameter | Value | Parameter | Value |
| Normal Force Impact | | Friction Force Static Coefficient | 0.15 |
| Stiffness 1.0E+05 | | Dynamic Coefficient | 0.1 |
| Force Exponent 1.5 | | Static Transition Vel. | 0.1 |
| Damping 100 | | Dynamic Transition Vel. | 10 |

The driving force was set in the form of component force, which was calculated by MATLAB. The comparison between ADAMS simulation results and MATLAB calculation results is shown in Figure 9. In the figure, the simulation results are very close to the calculation results, which proves that the dynamic equation and the optimization results are all reliable.
4. Conclusions

In this paper, the kinematic and dynamic equations of the mechanical claws in the dumping action are derived by the complex interpolation approach. And the particle swarm algorithm is used to optimize the arc radius of the dumping track, the center distance of the mechanical claw’s groove wheels and the time-consuming of the dumping action. The optimization results show that the performance of the scheme in which the driving force acts on the axis of the lower groove wheels is better. Specifically, the maximum power is reduced by 1827.46W and 48.55%. The average power is reduced by 701.61W and 38.6%. The maximum instantaneous driving force mutation is reduced by 2093.6N and 85.7%. And the time-consuming of the dumping action is increased by 1.36s. It can be expected that the garbage dumping action will be more stable and the power consumption, quality and cost of the garbage collection and transportation equipment will be reduced. Finally, ADAMS is used to verify the optimization results, which proves the feasibility of the optimization method in this paper.

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