Optimization Design and Program implementation for the Coveying Mechanism of Hot Sawing Machine

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Abstract

Hot sawing machine is an important and auxiliary equipment in hot rolling mill, the most important part is the coveying mechanism in hot sawing machine with four-bar linkage mechanism, this mechanism is the biggest influence on the quality of sawing steel. This subject goes on optimization design with the coveying mechanism of hot sawing machine in the optimization theory and method. It develops current software of optimization and analysis for the conveying mechanism of hot sawing machine with Visual Basic language, it uses this software to optimize the coveying mechanism of hot sawing machine and gains ideal size after optimizing, this offers the basis for improvement of hot sawing machine in the future, so this has offered a very good reference for optimization and design of hot sawing machines with four-bar linkage mechanism in other metallurgical enterprise.

Keywords

Hot sawing machine; four-bar linkage; coveying mechanism; optimization.

1. Introduction

Four link hot saw machine is an important equipment in the hot rolling mill, which is mainly composed of three parts: sawing mechanism, cutting mechanism and moving mechanism[1]. The saw mechanism is a four link mechanism. The design of the saw mechanism is closely related to the surface quality of the cutting steel [2]. At present, people usually only reference the hot sawing machine which be used in other similar manufacturers for hot sawing machine design, and there is no special optimization process for specific application, and the structure is not optimal. Therefore, it is necessary to study the optimization of the saw mechanism, so that the transmission is more reasonable, so as to achieve the purpose of high quality cutting, but also provide a scientific basis for the subsequent maintenance and transformation.

The optimal design of the four link saw mechanism firstly analyzes the mechanical mechanism of the saw mechanism, extract the parameters and analyzes the relationship between parameters, in order to transform the actual problem of mechanism motion into the mathematical model of optimum design[3] .It programms by penalty function method ,designs visual interface, completes parameter input, optimizes calculation, seeks optimal solution and so on, the results of the object function are finally represented in an intuitive graphical fashion.

2. Kinematics analysis of the coveying mechanism

Using the analytical method to analyze the kinematics of the covey mechanism [4][5]. The structure of four link mechanism in the covey mechanism is shown in Figure 1, O_1A is crank, O_1 is the center of crank rotation, AB is link, ABCD represents the upper frame of the saw, the CD segment is the reduced height of the saw shaft center, O_2B is a driven pendulum bar, O_2 is the center of the swing rod, the virtual circle is the position of the saw blade. When the O1A clockwise rotation around the O_1 point, the saw will advance.. The saw blade is driven by a special transmission mechanism. The O_1A 'in the graph is the left limit position of the crank in the working stroke of the sawing machine, O_1A'' is the right limit position of the crank in the working stroke of the saw machine, at the same time, the center of the saw blade reaches the corresponding position 'D' and 'D' respectively. The horizontal distance between D 'and D' is the working stroke of saw machine S. In order to correspond with the actual situation of the saw machine, take the XOY coordinate system. Set O_1A long l_1, AB rod length of l_2, swing rod O_2B length of l_3, BC length of l_4, CD length of l_5. y_{01} and x_{02} are the coordinate values of the crank O_1A and the swing center of the swing rod O_2B, θ_1, θ_2, θ_3 are respectively the position angle of the crank O_1A, the connecting rod AB, the swing rod O_2B, and the clockwise direction is positive [6] [7] [8] .

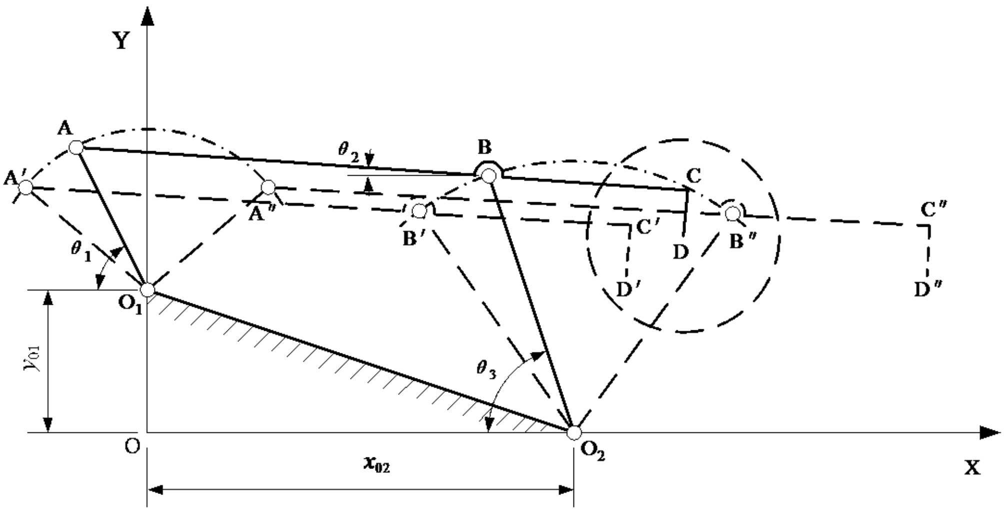

Fig. 1 Sketch map of the coveying mechanism of hot sawing machine

2.1 Determine the position angle of the driven pendulum

The following relations can be drawn from Figure 2:

$$\begin{cases} x_A = -l_1 \cos\theta_1 \\ y_A = y_{01} + l_1 \sin\theta_1 \end{cases} \tag{1}$$

$$\begin{cases} x_B = x_{02} - l_3 \cos\theta_3 \\ y_B = l_3 \sin\theta_3 \end{cases} \tag{2}$$

Four linkage mechanism has the following relations:

$$(x_B - x_A)^2 + (y_B - y_A)^2 = l_2^2 \tag{3}$$

The formula (1) and formula (2) is substituted into the formula (3), further consolidation, we can get the following formula:

$$\begin{aligned} &(\sin\theta_1 + y_{01}/l_1)\sin\theta_3 + (\cos\theta_1 + x_{02}/l_1)\cos\theta_3 \\ &= (x_{02}^2 + y_{01}^2 + l_1^2 - l_2^2 + l_3^2)/2l_1l_3 + (x_{02}/l_3)\cos\theta_1 + (y_{01}/l_3)\sin\theta_1 \end{aligned} \tag{4}$$

In order to solve the problem, and in order to facilitate computer calculation, the following relations will be used:

$A_1 = \sin\theta_1 + K_5$
$B_1 = \cos\theta_1 + K_4$
$K_4 = x_{02}/l_1$
$K_5 = y_{01}/l_1$
$C_1 = K_1 + K_2\cos\theta_1 + K_3\sin\theta_1$
$K_1 = (x_{02}^2 + y_{01}^2 + l_1^2 - l_2^2 + l_3^2)/2l_1l_3$
$K_2 = x_{02}/l_3$
$K_3 = y_{01}/l_3$

The formula (4) may be written in the following form:

$$A_1\sin\theta_3 + B_1\cos\theta_3 = C_1 \tag{5}$$

Further, we can use the following formula to simplify formula (5):

$x = \tan(\theta_3/2)$
$\sin\theta_3 = 2x/(1+x^2)$
$\cos\theta_3 = (1-x^2)/(1+x^2)$

Then formula (5) may be written in the following form:

$$(B_1 + C_1)x^2 - 2A_1x - (B_1 - C_1) = 0 \tag{6}$$

The following results can be obtained by solving this equation, According to the problem, the result is positive.

$$\theta_3 = 2\arctan\left[(A_1 \pm \sqrt{A_1^2 + B_1^2 - C_1^2})/(B_1 + C_1)\right] \tag{7}$$

2.2 Determine the position angle of the connecting rod

The following relationships can be found in Figure 1:

$$\begin{cases} x_B = l_2\cos\theta_2 - l_1\cos\theta_1 \\ y_B = y_{01} + l_1\sin\theta_1 - l_2\sin\theta_2 \end{cases} \tag{8}$$

The coordinates of the O1 and O2 points are (x01, y01), (x02, y02), so we can know:

$$y_{02} = 0 \tag{9}$$

$$(x_{02} - x_B)^2 + (y_{02} - y_B)^2 = l_3^2 \tag{10}$$

put (8), (9) into the formula (10) ,and we can get the formula as follow:

$$\begin{aligned} &(\sin\theta_1 + y_{01}/l_1)\sin\theta_2 + (\cos\theta_1 + x_{02}/l_1)\cos\theta_2 \\ &= (x_{02}^2 + y_{01}^2 + l_1^2 + l_2^2 - l_3^2)/2l_1l_2 + (x_{02}/l_2)\cos\theta_1 + (y_{01}/l_2)\sin\theta_1 \end{aligned} \tag{11}$$

For the convenience of computer calculation, the following relations are given:

$A_2 = \sin\theta_1 + K_5$
$B_2 = \cos\theta_1 + K_4$
$C_2 = K_6 + K_7\cos\theta_1 + K_8\sin\theta_1$
$K_6 = (x_{02}^2 + y_{01}^2 + l_1^2 + l_2^2 - l_3^2)/2l_1l_2$
$K_7 = x_{02}/l_2$
$K_8 = y_{01}/l_2$

Then formula (11) becomes:

$$A_2 \sin\theta_2 + B_2 \cos\theta_2 = C_2 \tag{12}$$

If the following formula is used, the formula (12) can be further simplified as formula (13).

$y = \tan(\theta_2/2)$

$\sin\theta_2 = 2y/(1+y^2)$

$\cos\theta_2 = (1-y^2)/(1+y^2)$

$$(B_2 + C_2)y^2 - 2A_2 y - (B_2 - C_2) = 0 \tag{13}$$

Solving the equation, get the formula (14), this is the value ofθ2, its negative sign .

$$\theta_2 = 2\arctan\left[\left(A_2 \pm \sqrt{A_2^2 + B_2^2 - C_2^2}\right)/(B_2 + C_2)\right] \tag{14}$$

2.3 Determining the rotation speed of the connecting rod and the driven swing rod

The derivative of formula (5) can be used to obtain the rotational speed of the pendulum bar O2B, which is denoted as ω 3 .

$$\omega_3 = \frac{\sin(\theta_3 - \theta_1) + K_2 \sin\theta_1 - K_3 \cos\theta_1}{\sin(\theta_3 - \theta_1) + K_4 \sin\theta_3 - K_5 \cos\theta_3}\omega_1 \tag{15}$$

The derivative of formula (12) can be used to obtain the rotational speed of the connecting rod AB, which is denoted asω_2 .

$$\omega_2 = \frac{\sin(\theta_2 - \theta_1) + K_7 \sin\theta_1 - K_8 \cos\theta_1}{\sin(\theta_2 - \theta_1) + K_4 \sin\theta_2 - K_5 \cos\theta_2}\omega_1 \tag{16}$$

2.4 To determine the trajectory and speed of the saw shaft center coordinates

According to figure 1, we can get the coordinates of the center of the saw axis (xD, yD)。

$$\begin{cases} x_D = (l_2 + l_4)\cos\theta_2 - l_1 \cos\theta_1 - l_5 \sin\theta_2 \\ y_D = l_3 \sin\theta_3 - l_4 \sin\theta_2 - l_5 \cos\theta_2 \end{cases} \tag{17}$$

For the derivative of the formula (17), the horizontal component vDX and the vertical component vDY of the center velocity of the saw axis are obtained。

$$\begin{cases} v_{DX} = -(l_2 + l_4)\omega_2 \sin\theta_2 + l_1\omega_1 \sin\theta_1 - l_5\omega_2 \cos\theta_2 \\ v_{DY} = l_3\omega_3 \cos\theta_3 - l_4\omega_2 \cos\theta_2 + l_5\omega_2 \sin\theta_2 \end{cases} \tag{18}$$

Then the center speed can be calculated:

$$v_D = \sqrt{v_{DX}^2 + v_{DY}^2} \tag{19}$$

3. Parameter optimization of coveying mechanism

3.1 Optimization mathematical model

3.1.1 Determination of design variables

According to the kinematic characteristics of the four link saw mechanism, taking the best approximation of the horizontal straight line of the center of the saw shaft of the saw as the goal, the geometric parameters of the four link saw mechanism are chosen as the design variables:

$$x = [l_1, l_2, l_3, l_4, l_5, x_{02}, y_{01}]^T = [x_1, x_2, x_3, x_4, x_5, x_6, x_7]^T \tag{20}$$

3.1.2 Determination of objective function

In the sawing process of hot sawing machine, the vertical wave momentum of the saw shaft center will influence the cutting quality, saw blade life and noise. Therefore, it is required that when the crank of the sawing machine rotates from the initial position angle θ_{11} to the stop position angle θ_{n1}, the center of the saw shaft needs to move from D′ to D″ , that is, the saw shaft moves a saw machine in the horizontal direction. According to the design requirements, the center of the saw shaft should be kept a certain height yD0 in the sawing process. At the same time, the movement path of the saw shaft center from D 'to D' should be as close as possible to the horizontal line. (i.e., the ordinate of the straight line is y_D0). In this paper, the aim is to minimize the error between the actual running track and the ideal running track, Therefore, the objective function is obtained, as shown in formula (21), The yDi is the longitudinal coordinates of the actual trajectory of the saw shaft center; yD0 is the longitudinal coordinate of the ideal trajectory of the saw shaft center; n is the number of points taken in the specified area.

$$F(x)=\sum_{i=1}^{n}(y_{Di}-y_{D0})^2 \tag{21}$$

3.1.3 Establishment of constraint conditions

(1) Equality constraint

According to figure 1, in the working range, each position in the movement of the saw mechanism must be guaranteed to meet the size requirements of the four bar linkage mechanism, thus the following equality constraints can be obtained,

$$\begin{cases} h_i(x)=x_6+x_1\cos\theta_{1i}-x_3\cos\theta_{3i}-x_2\cos\theta_{2i}=0 \\ h_{n+i}(x)=x_7+x_1\sin\theta_{1i}-x_3\sin\theta_{3i}-x_2\sin\theta_{2i}=0 \end{cases} \tag{22}$$

In the formula (22) ,n is the number of points taken in the specified area; θ_{1i} is the position angle of the crank in the i position; thetaθ_{2i} is the position angle of the connecting rod in the first i position; θ_{3i} is the position angle of the swing rod in the first I position. In the sawing process of hot sawing machine, the crank position angle θ_1 is rotated from 60 °to 120 °, and the angle is divided into 60 parts, so the value of n is 61, and there are a total of 122 equality constraints.

(2) Inequality constraint condition

a Constraint conditions of crank [9]

$$\begin{cases} g_1(x)=x_1-x_2\le 0 \\ g_2(x)=x_1-x_3\le 0 \\ g_3(x)=x_1-\sqrt{x_6^2+x_7^2}\le 0 \\ g_4(x)=\sqrt{x_6^2+x_7^2}+x_1-x_2-x_3\le 0 \\ g_5(x)=x_1+x_2-x_3-\sqrt{x_6^2+x_7^2}\le 0 \\ g_6(x)=x_1-x_2+x_3-\sqrt{x_6^2+x_7^2}\le 0 \end{cases} \tag{23}$$

b Constraint conditions for the requirements of transmission angle of planar four bar mechanism

$$\begin{cases} g_7(x)=\gamma_{\min}-\arccos\dfrac{x_2^2+x_3^2-(\sqrt{x_6^2+x_7^2}-x_1)^2}{2x_2x_3}\le 0 \\ g_8(x)=\arccos\dfrac{x_2^2+x_3^2-(\sqrt{x_6^2+x_7^2}+x_1)^2}{2x_2x_3}-\gamma_{\max}\le 0 \end{cases} \tag{24}$$

In the above formula, $\gamma_{\min}$ is the minimum transmission angle to meet the transmission performance; $\gamma_{\max}$ is the maximum transmission angle to meet the transmission performance.This paper takes $\gamma_{\min}$ = 40 °, $\gamma_{\max}$ =140 °.

c The constraint condition of the horizontal limit position of saw shaft center

During the sawing process of the hot saw machine, the crank moves from the initial position angle to the stop position angle, and the saw shaft also advances the distance of a stroke in the horizontal direction. n order to meet the requirements of production, the two limit positions of the center of the saw shaft in the horizontal direction should be limited, so that the difference between the horizontal distance of the two limit positions and the given value is within a certain range.

$$g_9(x) = \left|x_{D1} - x_{\min}\right| + \left|x_{Dn} - x_{\max}\right| - \Delta S \le 0 \tag{25}$$

In the above formula, the ΔS is the horizontal limit position error of the saw shaft center; x_{D1} is the horizontal coordinate of the actual minimum limit position of the saw shaft center; x_{Dn} is the horizontal axis of the actual maximum limit position of the saw shaft center; x_{min} is the minimum horizontal position of the center of the saw shaft;

x_{max} is the horizontal axis of the ideal maximum limit position of the saw shaft center.

d The constraint conditions of the velocity fluctuation of the saw shaft center

$$g_{10}(x) = \left|\frac{v_{D\max} - v_{D\min}}{v_{DA}}\right| - \delta \le 0$$

In the above formula, δ is the speed fluctuation rate allowed, $v_{D\max}$ is the maximum speed of the actual saw shaft center, $v_{D\min}$ is the minimum velocity of the saw shaft center, v_{DA} is the average speed of the saw shaft actual center.

e Constraints on the size of the mechanism

According to the actual production conditions, the optimized design variables should be within a certain range, so the upper and lower limits of each design variable should be specified。

The size of the optimized mechanism is larger than the lower limit, so the following constraints must be satisfied, $x_{i\min}$ is the lower limit of the design variable x_i , i=1,2,3,4,5,6,7.

$$\begin{cases} g_{11}(x) = x_{1\min} - x_1 \le 0 \\ g_{12}(x) = x_{2\min} - x_2 \le 0 \\ g_{13}(x) = x_{3\min} - x_3 \le 0 \\ g_{14}(x) = x_{4\min} - x_4 \le 0 \\ g_{15}(x) = x_{5\min} - x_5 \le 0 \\ g_{16}(x) = x_{6\min} - x_6 \le 0 \\ g_{17}(x) = x_{7\min} - x_7 \le 0 \end{cases} \tag{27}$$

After optimization, the size of the mechanism is less than the upper limit, so the following constraints must be satisfied, ximax is the upper limit of the design variable x_i ,i =1,2,3,4,5,6,7.

$$\begin{cases} g_{18}(x) = x_1 - x_{1\max} \le 0 \\ g_{19}(x) = x_2 - x_{2\max} \le 0 \\ g_{20}(x) = x_3 - x_{3\max} \le 0 \\ g_{21}(x) = x_4 - x_{4\max} \le 0 \\ g_{22}(x) = x_5 - x_{5\max} \le 0 \\ g_{23}(x) = x_6 - x_{6\max} \le 0 \\ g_{24}(x) = x_7 - x_{7\max} \le 0 \end{cases} \tag{28}$$

3.2 Selection of optimization method and program flow chart

In recent years, , many different optimization methods have been put forward for practical problems in the research of mechanical optimization methods.In order to solve the shortcomings of the traditional optimization method which is easy to fall into the local optimum, an improved particle swarm optimization algorithm is proposed[10].The advantages of genetic algorithm in mechanical optimization design are discussed by two aspects of genetic algorithm toolbox and genetic algorithm improved by genetic algorithm[11].The four link is optimized by matrix algorithm, and the accurate and reliable optimization parameters can be obtained[12] .

From the above analysis, the optimization design of hot sawing machine sawing mechanism is a nonlinear programming problem with multi variable, multi constraints and multi equation contains the inequality constraints, can be solved us

ing the mixed penalty function method, this method is most widely used, high computational efficiency and good stability. The solving process is to transform the constrained optimization problem into an unconstrained optimization problem, and then obtain the unconstrained optimal solution by Powell method. The specific solving program diagram is shown in figure 2.

According to the above analysis, the optimization problem of the saw mechanism is a mathematical model which contains 7 design variables, 122 equality constraints and 24 inequality constraints,the penalty function expression constructed by the hybrid method is as follows.

$$\phi(x,r^{(k)})=F(x)-r^{(k)}\sum_{u=1}^{24}\frac{1}{g_u(x)}+\frac{1}{\sqrt{r^{(k)}}}\sum_{v=1}^{122}\left[h_v(x)\right]^2 \tag{29}$$

$r^{(k)}$ is the penalty factor,$r^{(0)}> r^{(1)} > r^{(2)} >\ldots> r^{(k)} >\ldots$; $\lim_{k\to\infty} r^{(k)}=0$; $r^{(k+1)}=C\, r^{(k)}$, $C<1$。

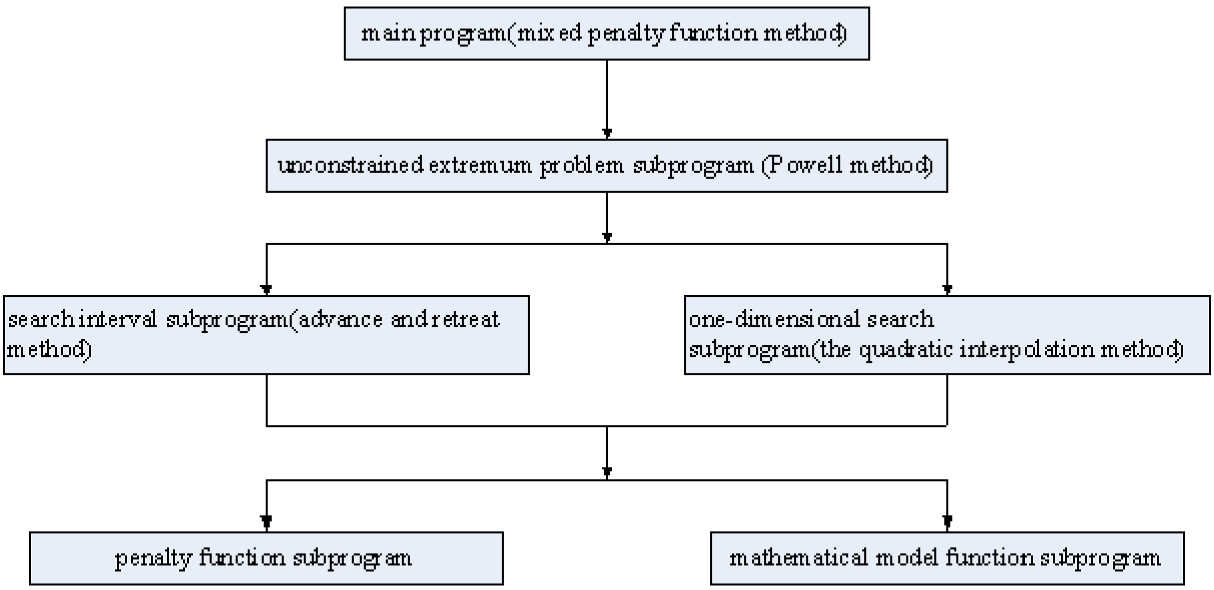

Fig.2 Basic optimum program structure of publishing function

4. Design and implementation of software architecture

Visual Basic is an integrated development environment from the Microsoft company launched, has the advantages of easy and powerful software, low expenses and quick[13]. the above advantages based on VB, develop the general optimization analysis software of saw feeding mechanism of hot sawing machine by using Visual Basic6.0 language, easy to produce excellent man-machine interface and powerful application program.

4.1 Software function structure diagram

The system software includes three modules: optimization design, calculation and analysis, curve display, each module contains a number of sub modules, the module structure shown in Figure 3, the main interface of the system program shown in figure 4.

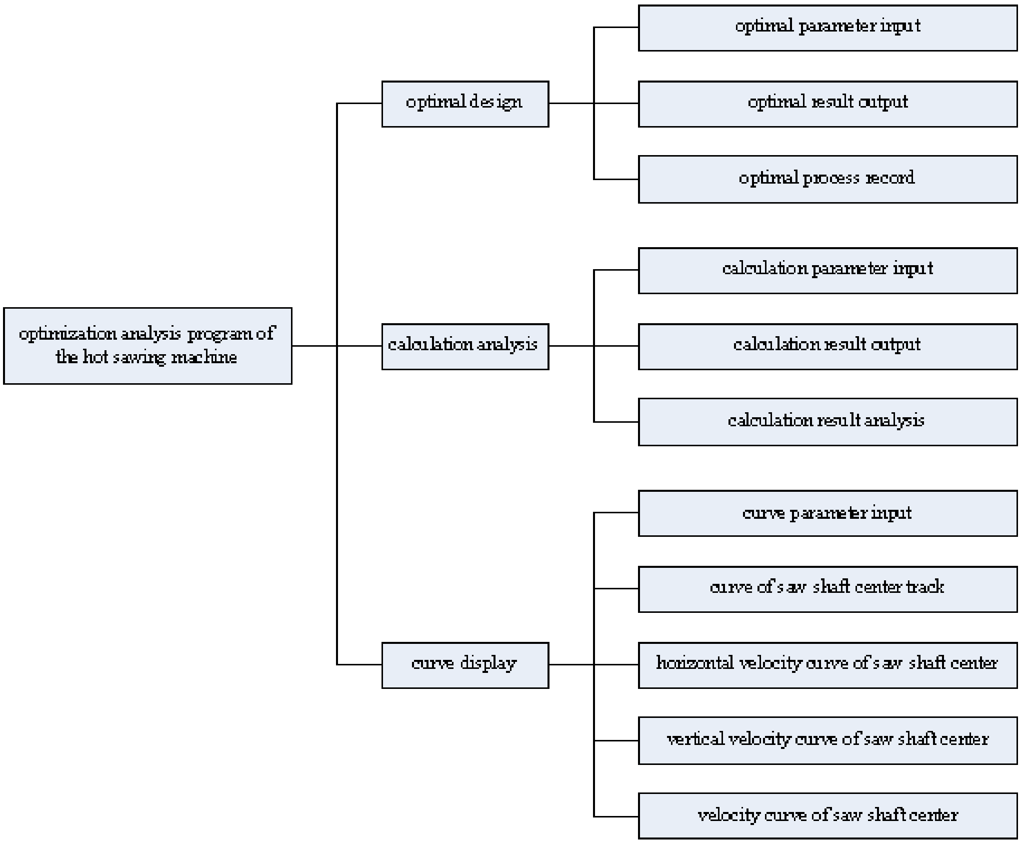

Fig.3 Structure of software

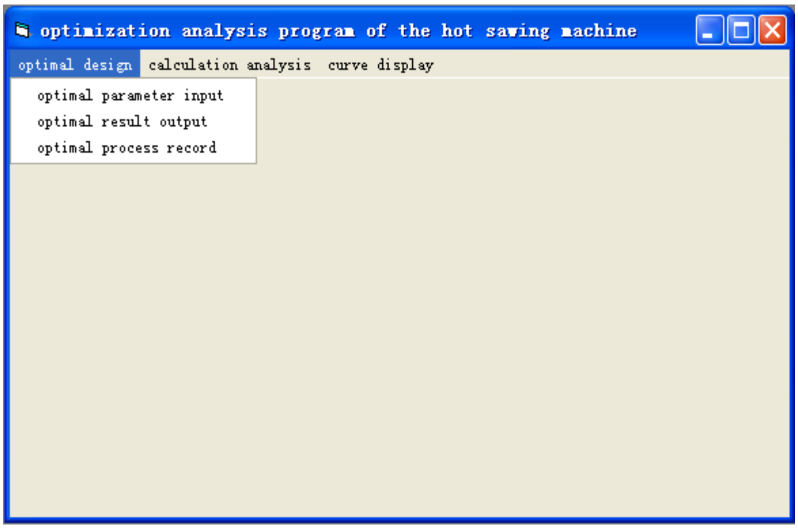

Fig.4 The main interface of program

5. Examples and results of optimization analysis

5.1 Program optimization process

On the main menu of Figure 4, click the " optimal parameter input " button, the pop-up interface is shown in figure 5. In the text box to enter a variety of optimization parameters and the accuracy of the optimization method, and enter the actual size and the upper and lower limits of the covey mechanism in a factory, then click the " optimization calculation" button, the program will output the optimization results, as shown in figure 6.

Of course, on the one hand, the nature of the objective function is not very understanding, on the other hand, due to the different initial point selection, the best is not necessarily the global best, is likely to be the local best. Therefore, the user needs to use this procedure to optimize the number of times, the results will be compared and analyzed, and then select the most consistent result。

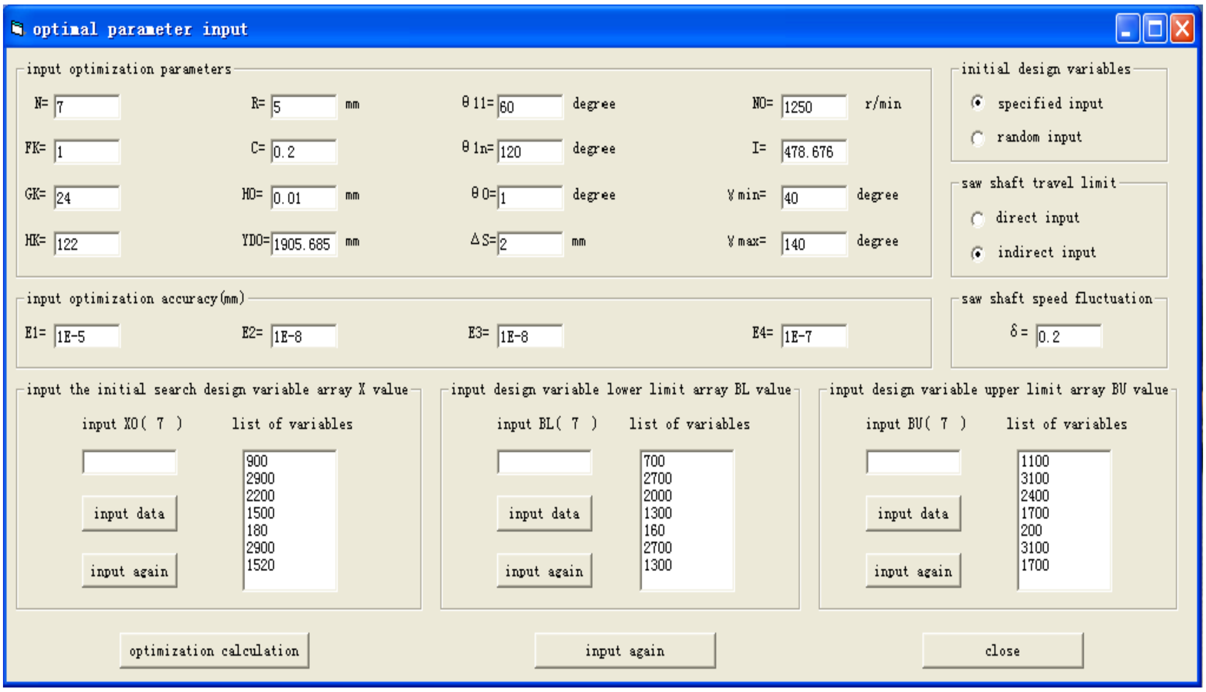

Fig.5 The interface one of program

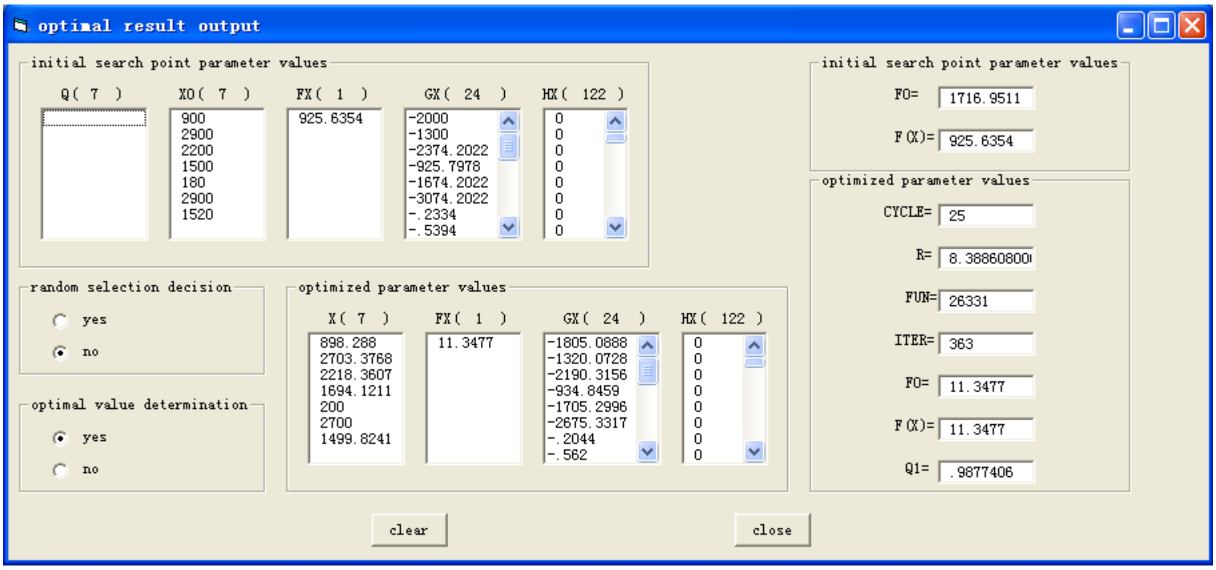

Fig.6 The interface two of program

5.2 Computational analysis process

After the optimization of the transport mechanism, it is necessary to analyze and compare the motion parameters of the center of the saw shaft, and find the best size that meets the requirements of the motion. On the main menu of Figure 4, click the "calculation parameter input" button, will pop up Figure 7 interface. Then the user entered the optimized parameters and dimensions, click on the "calculation result analysis" button, the program will pop up the results window, as shown in figure 8. The results show that the optimized parameters are analyzed by computer.

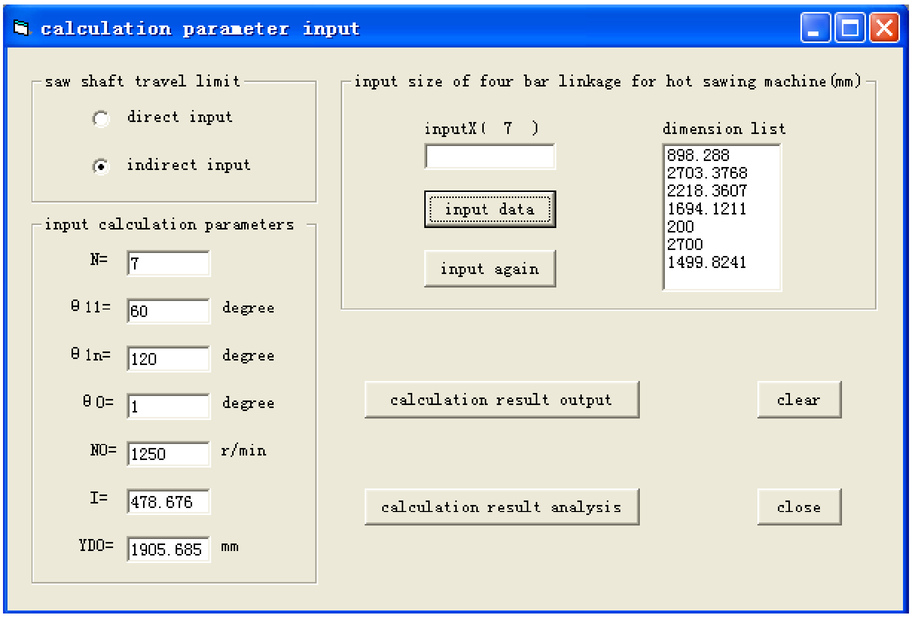

Fig.7 The interface three of program

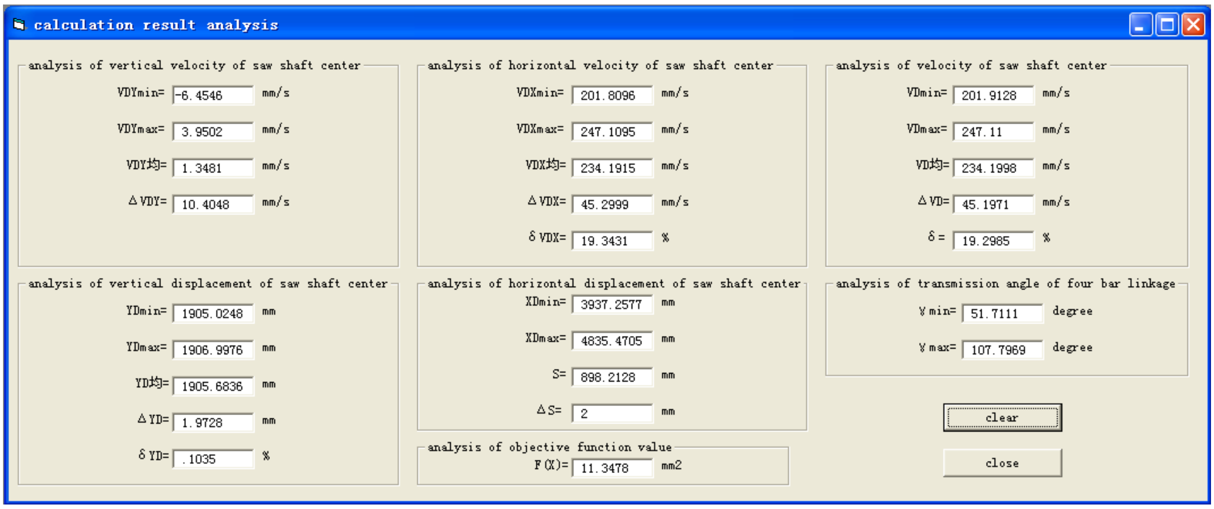

Fig.8 The interface four of program

In order to compare the size before and after optimization, we need to analyze the motion parameters before optimization. The results are shown in figure 9.

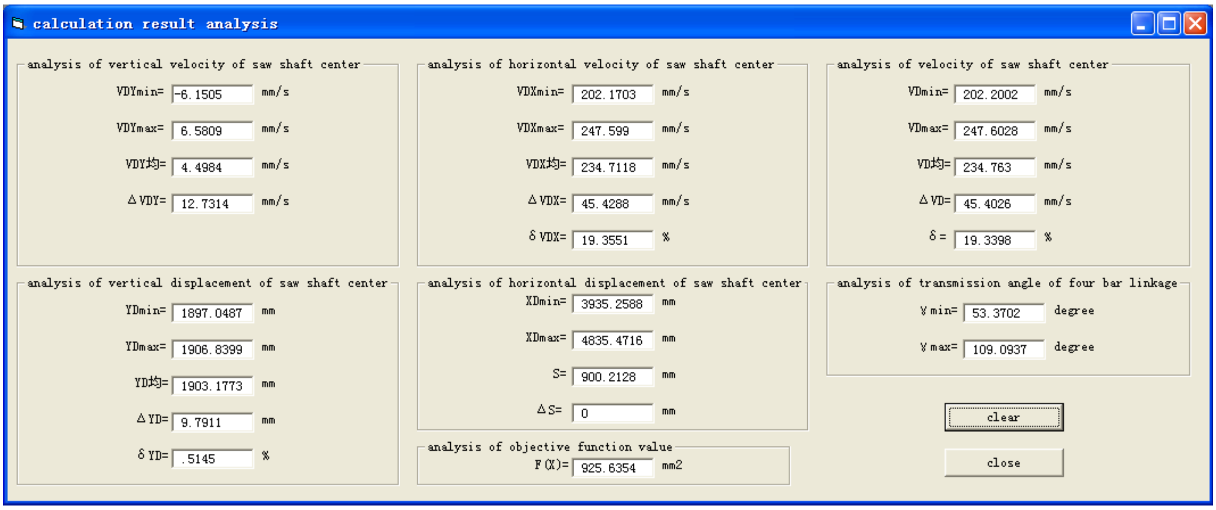

Fig.9 The interface five of program

Compared with the results of motion analysis before and after optimization, the results are as follows:

(1) the ΔS of the horizontal limit position error of the saw shaft center is increased by 2mm, but it is within the allowable error range.

(2) the optimal transmission angle ratio of the four bar mechanism is reduced a little, but it is also within the specified range.

(3) the average horizontal velocity of the saw shaft center is basically the same as that before optimization, and the fluctuation range and fluctuation rate are reduced, which is in line with the production requirements.

(4) the average value and the fluctuation range of the vertical velocity of the saw shaft center are much less than that before optimization.

(5) the average value of the center speed of the saw shaft is basically the same as that of the optimization before, and the fluctuation range and the fluctuation rate are reduced.

(6) saw shaft center vertical displacement is the most important optimization index, the optimized vertical displacement average and the ideal height deviation 0.0014mm, the value before optimization is 2.5077mm. after optimization, the range of vertical displacement is 1.9728mm, and the value before optimization is 9.7911mm. after optimization, the vertical displacement is 0.1035%, and the former is about 0.5145%.

Refer to the structural symbols shown in Figure 1,The specific size of the four bar linkage mechanism of a hot sawing machine before and after optimization is shown in table 1.

Table 1. The table of optimal result

The component symbol	the lower limit of the size (mm)	the upper limit of the size (mm)	Size before optimization (mm)	Size after optimization (mm)
l1	700	1100	900	898.288
l2	2700	3100	2900	2703.3768
l3	2000	2400	2200	2218.3607
l4	1300	1700	1500	1694.1211
l5	160	200	180	200
x02	2700	3100	2900	2700
y01	1300	1700	1520	1499.8241

6. Conclusion

Based on the research of the coveying mechanism of the hot saw in a factory, the general optimization software for the feeding mechanism of the hot saw is developed, and it is used to optimize and analyze the hot sawing machine in a factory, analysis results show that under the premise of meeting the requirements of the center of the saw axis, the trajectory of the center of the saw shaft is closer to the ideal straight line, and the optimized target is achieved, and the ideal size is obtained, which provides the basis for the improvement of the hot sawing machine in the future.

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