

Optimization Design and Simulation Analysis of Booms System of High-Branch Pruning Machine

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Abstract—In order to design a rational and lighter booms system of the High-Branch pruning forestry machine and improve the stability, this paper mainly studies three aspects about the boom system of the High-Branch pruning forestry machine: parametric design, physical modeling and dynamic simulation. Firstly, this paper built the parameter model of the boom system by using the dynamics analysis software ADAMS of mechanical system. And this paper did the optimization design of the boom system. In the optimization design, the first manipulator of the boom system was taken as the research subject, minimizing the driving force of drive cylinder was taken as the optimization goal, the panel point's positions were taken as optimization variables, and the actual working conditions of the boom system were taken as certain constraints. The optimization result was that: the driving force of drive cylinder was reduced by 38.3% at most than before. Secondly, this paper built the model of the boom system by using the software of Solidworks, imported the model into ADAMS in the parasolid format, did dynamics simulation analysis, and thus obtain the load curve of fluid cylinders. This paper designs a better boom structure which can satisfy requirements of structural strength and stability. In addition, this paper provides a complete procedure including preliminary design, dynamic analysis.

Keywords—boom system; parametric design; dynamics simulation analysis.

I. INTRODUCTION

Forestry is one of the most important agricultural industry in China, and pruning trees is an important part of forest thinning projects in recent years. Mechanized pruning branches can greatly improve the quality and the operational efficiency. Precise agriculture, agricultural production automation and intelligent agricultural production are the important direction for future development of agriculture. The research of mechanical pruning is still in its infancy in China. The degree of automation of machinery is still in a low level, less variety and the work efficiency is not ideal[1].

The study of mechanical pruning started earlier in foreign countries, it is mainly high sticks pruning saws, which uses gasoline or diesel as a power and there is less vibration when it works. The pruning effect of them is good and the effective radius of pruning saws is about 21 cm. However, the working height of such a machine is only about six meters, and they can't meet the actual needs of Chinese forestry.

The new High-Branch pruning forestry machine in this paper can meet the requirements of this research work height and pruning radius, and it has a higher degree of automation, which filled a gap in the field of mechanical pruning to some extent. The boom system of the High-Branch pruning forestry machine is the most important part of the machine, because it not only affects the stability and flexibility of the whole machine, but also determines the operating height and quality of the machine. So the main task of this paper is to design and optimize the boom system of the High-Branch pruning forestry machine by the method of the parameters design, dynamics analysis and so on.

II. THE STRUCTURE AND WORKING PRINCIPLE OF HIGH PRUNING MACHINE FOR FORESTRY

In order to guarantee the precision and efficiency of the High-Branch pruning forestry machine to meet the branches of tall trees trim work requirements, the design of the High-Branch pruning forestry machine in this paper mainly includes four agencies actuators, which is rotating mechanism, elevating mechanism, booms system and terminal execution mechanism. Rotary mechanism is mainly used to ensure the machine in the horizontal plane rotational degrees of freedom, the elevating mechanism is used to lift the working height of the machine, booms system is mainly used for improving the flexibility of the end effector, and increasing the working space of the end of the execution. Boom system is the most important part of the whole machine, its structure and size is not only related to the hydraulic load and engine power, but also affects the complexity of the whole post-control program. Therefore, this paper mainly studies the boom system.

A. The structure

The structure shown in Figure 1, The boom system includes four manipulators, which are rolled together in the form of clockwise R-type folding considering the flexibility of the transport state. One end of the first manipulator is hinged to the mechanical base, the other end of the first manipulator and one end of the second manipulator are hinged through the first link mechanism, the other end of the second manipulator and one end of the third manipulator are hinged through the second link mechanism, the fourth manipulator and the other end of the

third manipulator are hinged through the third link mechanism as it shown in the H view of Figure 1.

B. Working principle

The first cylinder is used for driving the first link mechanism so that the first manipulator is rotated around the mechanical base, $0^\circ \leq \theta_1 \leq 120^\circ$. The second cylinder is used for driving the second link mechanism so that the second manipulator is rotated around the first manipulator, $0^\circ \leq \theta_2 \leq 180^\circ$. The third cylinder is used for driving the third link mechanism so that the third manipulator is rotated around the second manipulator, $0^\circ \leq \theta_3 \leq 180^\circ$. The fourth cylinder by driving the fourth link mechanism so that the fourth arm swings relative to the third manipulator side to side, $-30^\circ \leq \theta_4 \leq 30^\circ$.

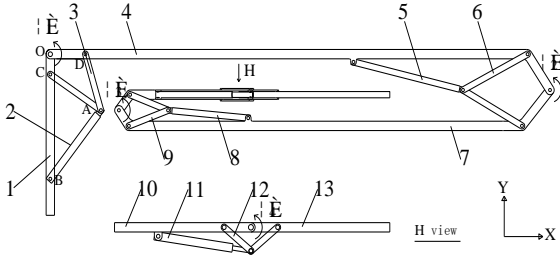


Figure 1: Boom system(1-mechanical base, 2-first cylinder, 3-first link

mechanism, 4-first manipulator, 5-second cylinder, 6-second link mechanism, 7-second manipulator, 8-third cylinder, 9-third link mechanism, 10-third manipulator, 11-fourth cylinder, 12-fourth link mechanism, 13-fourth manipulator)

III. OPTIMIZED DESIGN OF THE BOOM SYSTEM

In terms of the performance of the booms system of the High-Branch pruning forestry machine, the system of the cylinder luffing mechanism plays a vital role, the rationality of its structural parameters determines the reachable space of the end effector and the driving force of cylinders. In this section, this paper takes the position coordinates of the hinge points luffing mechanism as design variables, minimizes the driving force of drive cylinder as the optimization goal, optimize the design of the first luffing mechanism[2, 4-8], and thus obtain the more rational structure parameters of luffing mechanisms.

A. Position optimization of luffing mechanisms hinge point

As shown in Figure 1. In the first luffing mechanism, the first manipulator is connected with the mechanical base through the point O, AC-link is connected with the mechanical base through the point C, AD-link is connected with the first manipulator through the point D, AC-link and AD-link are connected with the cylinder piston rod through point A, the cylinder barrel is connected with the mechanical base through the point B. Optimization aims to optimize the location of the hinge point A B C D, to reduce the driving force of the hydraulic cylinder.

- The objective function

Under the premise of the design process to meet the work requirements, it is hoped to make a maximum of oil cylinder force as small as possible, so the force of one cylinder functions is regarded as one of the maximum targets, in order to make the force of hydraulic cylinders in the whole process small, this paper regards the cylinder force as another function of the two targets. This optimization has two optimization goals, and it is a multi-objective optimization problem. In the optimization process, the two goals can not simultaneously function tends to the optimal solution, instead the interaction of two optimal results, even contradictory phenomenon. Thus, according to the importance of the two points of the objective function, this paper processes the two functions by unity of purpose[3], and the final objective function was obtained as follows:

$$g = 0.3F_{(max)}/22774 + 0.7F_{(avg)}/6591.9 \quad (1)$$

In the above formula, $F_{(max)}$ and $F_{(avg)}$ represent the first sub-objective function and the second sub-objective function, and 22774 and 6591.9 respectively represent the maximum of two functions.

- Design variables

As shown in Figure 1. In the first luffing mechanism, the change of the position coordinates of all hinge points will affect hydraulic loading conditions except point O, because it is fixed. Therefore the Position coordinates of A B C D is set to variable parameters. As shown in Table 1.

TABLE I: THE POSITION COORDINATES OF HINGE POINTS

| POINT | Loc_X | Loc_Y |
|---------|--------|--------|
| POINT_A | (DV_1) | (DV_2) |
| POINT_B | (DV_3) | -0.8 |
| POINT_C | 0.0 | (DV_4) |
| POINT_D | (DV_5) | 0.0 |

- Restriction condition

There are many aspects of constraints need to be considered before the Parametric Design of the first luffing mechanism. According to the actual movement of first luffing mechanism, the constraints are mainly considered from the following two aspects.

Boundary Constraints. Take DV_1, DV_2, DV_3, DV_4, DV_5 as optimum design variables. According to external factors like cylinder stroke and machinery overall layout, defines the range of five design variables. As shown in Table 2.

TABLE II: THE RANGE OF DESIGN VARIABLES

| variables | Initial value | Minimum value | Maximum value |
|-----------|---------------|---------------|---------------|
| DV_1 | 0.3 | 0.26 | 0.36 |
| DV_2 | -0.3 | -0.36 | -0.26 |
| DV_3 | 0.3 | 0.01 | 0.35 |
| DV_4 | -0.2 | -0.25 | -0.13 |
| DV_5 | 0.2 | 0.13 | 0.25 |

Amplitude range constraints. In order to meet the design requirements, in the coordinate system shown in Figure 1, the angle θ_1 between the first manipulator and X-axis positive direction should be less than 120 degrees.

B. Result Analysis

The optimization objective is to minimize the function g , and obtain an optimal set of variable values by optimizing the design. As shown in Table 3.

TABLE III: OPTIMIZATION RESULTS

| DV_1 | DV_2 | DV_3 | DV_4 | DV_5 | Maximum value [N] | average value [N] |
|------|-------|------|-------|------|-------------------|-------------------|
| 0.36 | -0.36 | 0.35 | -0.13 | 0.25 | 3166.3 | 1969.1 |

In order to display the change of the load during the whole movement of the hydraulic cylinder before and after the optimization of the hinge points, the load change process of the first section of the mechanical arm cylinder is compared by using the scatter chart. Results are shown in Figure 2.

After comparing the two curves it can be seen that the average hydraulic load of the first manipulator is reduced to 1969.1N from 3192.46N after the optimum design, a drop of about 38.3%. After the optimum design, cylinder force curve gently, the maximum is reduced and the load fluctuation is reduced. Finally, the parameters of each part of the entire boom system were determine by using the same method.

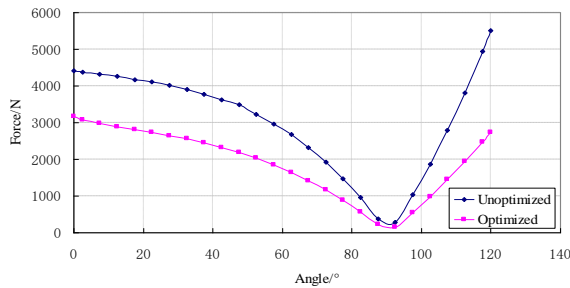


Figure 2: Cylinder load curve comparison

C. Dynamic Analysis of virtual prototype of the boom system

In order to obtain the load situation of each hydraulic cylinder, and provide the data to determine the parameters of the late hydraulic cylinder as basis, dynamics analysis was taken to the virtual prototype model of the boom system.

- Virtual prototype Under ADAMS environment

In SolidWorks software, use bottom-up assembly design technology for founding three-dimensional modeling of the boom system, (Individual parameters was slightly modified.) and import it into ADAMS software[6,9].

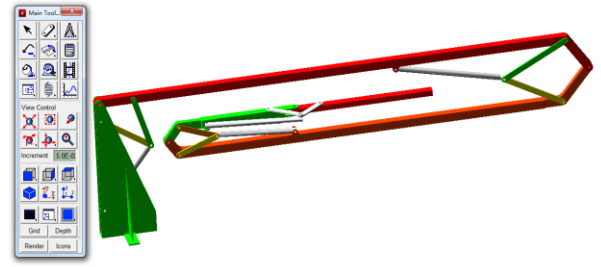


Figure 3: Virtual prototype of the boom system

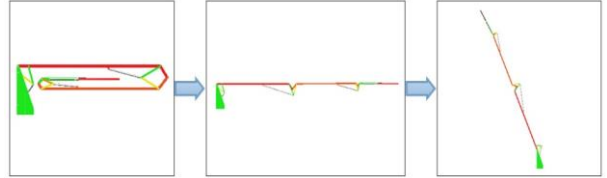


Figure 4: Dynamic simulation condition

It is easy to see that the most dangerous condition of the Booms system structure is showed in Figure 4. Because the fourth cylinder is in simple load cases, this section only makes a simulation analysis of the first three cylinders. Adding drive functions to the four cylinders. As shown in Table 4.

TABLE IV: CYLINDER DRIVE FUNCTION

| Name | position | driving function's expression |
|----------|-----------------|--|
| MOTION_1 | First cylinder | STEP(time, 0.0, 0.0, 20.0, 0.0) +STEP(time, 80.0, 0.0, 100.0, 0.48) |
| MOTION_2 | Second cylinder | STEP(time, 20.0, 0.0, 40.0, -0.4) |
| MOTION_3 | Third cylinder | STEP(time, 40.0, 0.0, 60.0, 0.36) |
| MOTION_4 | Fourth cylinder | STEP(time, 60.0, 0.0, 70.0, 0.03) +STEP(time, 70.0, 0.0, 80.0, -0.12) |

- Cylinder load analysis

Based on the virtual prototype dynamics analysis of booms system, it is concluded that the hydraulic cylinder shown in figure 4 under the condition of load change curve as shown in figure 5, we can see three cylinder load part is relatively flat before 90 seconds, even with the load change is slowly changing. Impact load mainly appeared in the 90-100 seconds, the reason is that the angle of first quarter mechanical arm and the horizontal is more than 90 degrees, so as to avoid the load impact of the hydraulic system, this kind of situation should be avoided in the actual operation.

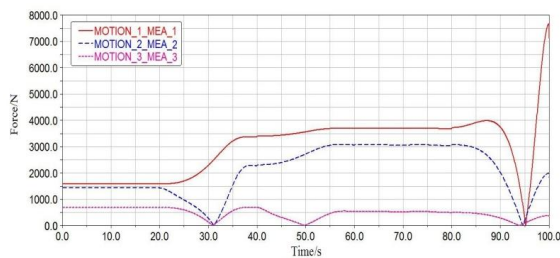


Figure 5: Cylinders load curve

IV. CONCLUSIONS

(1) The boom system of High-Branch pruning forestry machine is designed, and the parameters of the variable range mechanism of the boom system are modeled by using ADAMS software. By optimizing the position of the luffing mechanism, the average load of the cylinder is reduced by 38.3%, the pressure in the hydraulic circuit impact is reduced, and the optimum design of the variable amplitude mechanism is realized.

(2) Using SolidWorks for the boom system virtual prototype modeling, using the ADAMS software for its dynamic simulation, and analysis of the boom system load of each cylinder, which providing parameters for cylinder selection in the next step.

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