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The Simulation Analysis on Concrete Pump Truck Arm Device

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Abstract

The arm device is the important part of the concrete pump truck, for its high value in nature. It is convenient to establish virtual prototypes then design and optimize them. This paper establishes the virtual prototype of the concrete pump truck arm device both in Pro/E and ADAMS. By using computer technology, researching concrete pump truck arm device under various working circumstances, like running state, parametric design and mechanism optimization, are all become capable. In the virtual circumstance, we parametric design those coordinates of hinge points, which effected by the force-changing oil cylinders, to reduce the maximum stress of oil cylinders; using software to research operate coordinates of hinge points, then we target oil cylinders' coordinates which suffered seriously stress, through sensitivity analysis. We optimize those coordinates by using FEA, reduce the maximum stress that oil cylinder suffered, and gain the significant optimized effect.

Keywords*: concrete pump truck arm device; dynamics simulation; virtual prototype technology*

1. Introduction

The concrete pump truck is modified from the heavy loaded truck, equipment like power transport device, pumping and mixing device, arm device and some other auxiliary devices are installed on the chassis. Based on the work capability, designers choose the chassis type of truck, then they install delivery pump and extra devices. However, all of these installations should satisfy the stability requirement of the truck. The concrete pump truck arm device is consist of revolving-supporting structure, arms and tubes.

The pump truck arm use link mechanism, hydraulic cylinders which installed on the arm enable those commanded motion through their shorten and lengthen[1]. The pump truck performance parameters mainly reflect on the arm device performance parameters, which include the arm type, the maximum horizontal length, the maximum vertical height and the maximum depth. The arm device of truck is high technical and vital because of the working condition which the room is always limited, when the arms are expanding, space they occupied should as little as possible. Meanwhile, with the limited working condition and maintain condition, it is important to have a reliable design of arm device to ensure its working capability.

In the construction site, when pump truck has longer arms, which enable it delivers material to higher and broader place, not only does it benefits the project quality and efficiency, but reduces concrete consumption and pollution it brings at the same time[2][3]. Because of particularity of pump truck operation, the truck has high command of its overall balance performance and stability, thus, as the length of arm device is increase, the overall load of truck will also increase, then the swing resistance and stability will affected. Thus, both structure adjusting and optimizing should involved during the design process of arm device.

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Simulation of pump truck arm device is based on the virtual prototype technology, not only could it prove the rationality of the arm device's design scheme, but also verifies different units and optimizes the performance parameters during work. Applying Pro/E we establish the virtual prototype of pump truck arm, then we import prototype into ADAMS before simulating and optimizing. Thus, before establishing the real device, designers can simulate various kinds of working condition, to ensure if design and parameters setting are reasonable, which have vital practical sense.

2. Analysis of Arm Device

2.1 Jib Luffing Mechanism of Arm Device and Its Analysis

When cantilever cranes are expanding or tucking, or the pump truck is placement locating, the oil cylinder is the only driving link of the motion, then the cantilever crane is able to move through oil cylinder's extension or withdraw. Oil cylinder luffing mechanism is widely applied in mechanical engineering field, for it is compact-sized, light-weight and stable. Generally, for those jib lubbing mechanism, which range under 90 degree would adopt one-cylinder-three-hinge-point jib lubbing mechanism. However, in the pump truck arm device, angle between arms is always above 180 degrees, thus three-hinge-point jib lubbing mechanism is not proper apply.

If one mechanic arm of pump truck jibs to 230 degrees, we should cascade a four-bar linkage to a three-hinge-point jib lubbing mechanism. [4] When cantilever cranes folded, which require little space, thus multiple joint should be avoided during the design process of arm device. In order to avoid potential interference, and considering the particular design requirement, one-cylinder-six-hinge-point jib lubbing mechanism should be applied, as it solves the interference problem what three-hinge-point jib lubbing mechanism and four-bar linkage bring about. As shown in Figure 2.1.

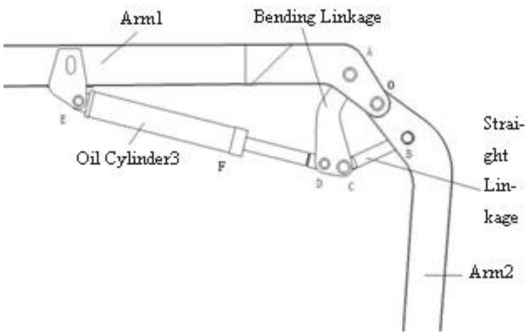


Figure 2.1. Schematic Diagram of the Luffing Mechanism

Diagram of the luffing mechanism of the first arm and the second arm as shown in Figure 2.1, this mechanism is consisted of OAE, OB, BC, ADC, FD and EF, and they combine into an one-cylinder-three-hinge-point jib lubbing mechanism ADFE and a four-bar linkage ACBD. After we simplify the luffing mechanism above, we have the simplified schematic diagram, as shown Figure 2.2.

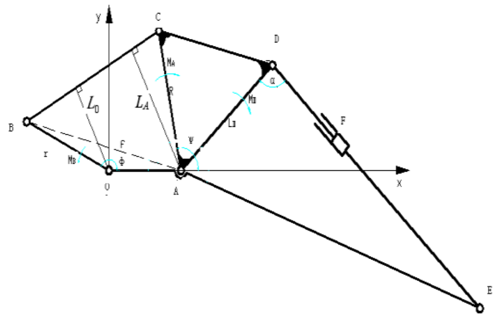


Figure 2.2. Boom Luffer Schematic Diagram

Based on the Figure 2.1 and Figure 2.2, OAE is the first arm, OB is the second arm, ACD is the bending linkage, BC is the straight linkage, DFE is the hydraulic cylinder, and this luffing mechanism is motivated by hydraulic cylinder DFE. Firstly, we analysis four-bar linkage OACB, which based on the Figure 2.2 can we infer it is double-rocker mechanism. In the OACB, AC is the active lever, BC is the connecting bar, OB is the driven lever, AO maintain fixed during the motion. To analyze it easily, We take O as the origin of coordinates, OA becomes the x axis direction, pointing to A; setting the length of OB as r, the length of AC as R, the length of BC as b, the angle between OB and x axis is φ, the angle between AC and x axis is Ψ.

The coordinates of four-bar linkage are: O (0,0), B (rcosφ, rsinφ), C (a+RcosΨ, RsinΨ), A (a,0).

Firstly, we analyze bar BC, assuming the distance between O to BC is $L_{o,}$ the distance between A to BC is L_A, M_c is the applied torque, M_B is the restoring torque, thus the equilibrium equation is:

$$M_B / FL_D = L_O \sin \alpha / L_A \tag{2.1}$$

Then, we analyze the mechanism AEFD. DF is the two-force bar, it endures the same pressure in opposite directions. And we assume the length of AD is L_D, the angle of ∠ADE is α, the pressure of oil cylinder is *F.* Based on the analysis of ADE, we infer the driving force torque is:

$$M_D = FL_D \sin \alpha \tag{2.2}$$

Based on the analysis of bending linkage, we ensure that $M_D = M_C$, by 2.1 and 2.2, we infer that:

$$M_C / L_A = M_B / L_O \tag{2.3}$$

Introducing coefficient $\lambda = L_O \sin \alpha / L_A$ which declares the relationship between friction torque and driving force torque, the coefficient is determined by parameters of cantilever cranes.

In the practical work, to ensure the mechanism to work normally, the driving force torque that oil cylinder provides should greater than friction torque, which combine the friction of oil cylinders, mechanism gravity and etc.

Thus:

$$M_{B\max} \leq L_D F_c \lambda_{\min} \tag{2.4}$$

M_B is the maximum friction torque, F_C is the maximum oil pressure of oil cylinder, is the minimum parameter determined by the mechanism parameters. By 2.4, we could infer:

$$F_c \leq M_{B\max} / L_D \lambda_{\min} \tag{2.5}$$

By equation 2.5, we are awared that, when friction torque is set, to increase the mechanism parameters, the oil pressure of oil cylinder will decrease as well.

2.2 The Analysis of Arm Prototype of Pump Truck

The arm device is consisted of 5 parts of arms, the length of first arm is 9180mm, the length of the second arm is 7976mm, the length of the third arm is 7960mm, the length of the forth arm is 7862mm, the length of the fifth arm is 7940mm. Based on relevant data and drawings, we named where the revolving table settled as the head end, and the other side named the tail side. To clarify different component, we list them from head to end, and named it as followed:

1. 5 parts of arm device called: arm1, arm2, arm3, arm4 and arm5;

2. The joint of each arm called: O_1, O_2, O_3, O_4;

3. Oil cylinders are called: oil cylinder 1, oil cylinder 2, oil cylinder 3, oil cylinder 4, oil cylinder 5 and oil cylinder 6. Oil cylinder 1 and 2 work together on the arm1.

The fig of arm device as shown in Figure 2.3

1. The base; 2. Piston rod1; 3. Piston rod2; 4. Oil cylinder1; 5. Oil cylinder2; 6. Arm1; 7. Oil cylinder3; 8. Piston rod3; 9. Bending linkage1; 10. Straight linkage1; 11. Arm2; 12. Oil cylinder4; 13. Piston rod4; 14. Straight linkage2; 16. Arm3; 17. Oil cylinder5; 18. Piston rod5; 19. Bending linkage3; 20. Straight linkage3; 21. Arm4; 22. Oil cylinder6; 23. Piston rod6; 24. Bending linkage4; 25.straight linkage4; 26. Arm5;

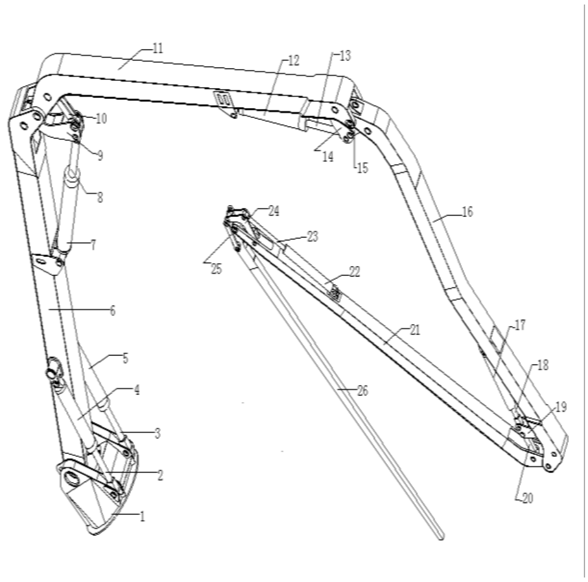


Figure 2.3. Shelf Structure of the Boom System

The arm of boom system is usually made up of welded box section structure which made of low alloy steel plate. The first arm is jointed with the revolving platform, and it is driven by a pair of oil cylinders through cylinders' extension and drawback. During work, the main pressure that the arm endures are self-weight and external applied load, thus the arm device is in the most dangerous position when they extend out horizontally[4][5]. Considering the paper mainly simulates and optimize the oil cylinder jib lubbing mechanism of arm device, so the self weight and load weight should both take into count when calculate the load of the arm. The self weight is consist of the weight of fixed part and the moving part, considering the effect of dynamic load, we multiple 1.2 on structural strength when calculate; and multiple 1.3 on the load weight when calculate the working load.

3. The Research of Dynamics of Multibody System

So far, the research method of multibody dynamics is well developed. [7-11] the method of analytic mechanics is represented by Lagrange's equations, vector learning method is represented by Newton - Euler equations, variational method and graph theory methods are more commonly used in the engineering.

Newton - Euler method and Lagrange method are traditional methods of classical mechanics, Robson - Wittenberg method and Kane - Houston method are for the analysis of modern mechanics.

Kane - Houston method by structuring the independent variable speed (pseudo) instead of not independent variable speed, then it assumpts to use ideal constraint to constraint system, and allows the exist of incomplete bind, thus the constraint force will not appear in the kinetic equation, and there is not necessary to calculate kinetic functions and its derivative. The concept of pseudo was introduced by Kane, in Kane's opinion, in multi-rigid-body-system, absolute velocity v_i and absolute angular velocity ω_i of any point is liner combination of pseudo.

$$v_i = \sum_{i=1}^{n} v_i^{(r)} + v_i^{(t)} \quad (i = 1, 2, ..., n) \qquad 3.1$$

$$(i = 1, 2, ..., \beta) \qquad 3.2$$

In above equations:

$v_i^{(r)}$, $\omega_i^{(r)}$ represents the partial velocity and angular velocity of the No.i rigid subject, which has no relevant with $v_i^{(t)}$ and $\omega_i^{(r)}$.

By Lagrange theorem:

$$\omega_i = \sum_{i=1}^{n} \omega_i^{(r)} + v_i^{(t)} \quad (i=1, 2, \ldots, n) \qquad 3.3$$

In the formula $F_i' = -m_i a_i$ represent the inertia force of rigid point i.

Setting $R_r = \sum_{i=1}^{n} F_i^a \cdot v_i^{(r)}$, $R_i^* = \sum_{i=1}^{n} F_i' \cdot v_i^{(r)}$, $\partial_{ri} = \sum_{i=1}^{n} v_i^{(r)} \cdot \delta_{\pi i}$.

thus:

$$\sum_{i=1}^{n} (F_i^a + F_i') \cdot \delta_{ri} = 0 \qquad 3.4$$

$R_r + R_i^* = 0$ represents the generalized active force. While R_i^* represents the generalized inertia force. Since the $\delta_{\pi r}$ has independence, thus:

$$\sum_{i=1}^{n} (R_r + R_i^*)\delta_{\pi r} = 0 \quad (i=1, 2, \ldots, n) \qquad 3.5$$

R_r This is the multi-rigid-body-system kinetic equation based on Kane method.

4. Simulationg and Optimizing of Pump Truck Arm Structure

4.1 Set the Basic Environment of Modeling

1. Set the coordinate system OXYZ, X axis is parallel to the ground, pointing to the frame, the Y axis is perpendicular to the ground, pointing up, Z axis along the center line of the origin point;
2. Set and adjust the work grid;
3. Set the gravity;
4. Choose the unit.

4.2. Add the Weight of Subject

Table 4.1 Steel Material Properties

Ultimate Strength (MPa)	Yield Stength (MPa)	Fatigue Limit (Mpa)	MOE (MPa)	Poisson Ratio	Density (kg/m³)
900-1100	900	600	2.1×10^6	0.3	7900

4.3 Set the Constraint and Drive to Work Device

4.4 Simulation Analysis Summarize

4.4.1 Simulation Type

1. Dynamic analysis: through solving series of nonlinear differential equations, simulate forces and movement of complex system that degree of freedom greater than zero.
2. Kinematic analysis: through solving series of algebraic equations, simulate system which degree of freedom is equal to zero, and has certain movement.
3. Static analysis: through the force equilibrium condition, to solve the equilibrium positions of each subject.
4. Assemble analysis: correct the inaccurate connections or inappropriate initial condition in the assembly and operating process.

4.5. Simulation of the Arm Fame Model under Various Working Condition

During the arm device movement, oil cylinder is the only driving link, thus it is necessary to identify the maximum pressure of oil cylinders when analyzing on the ADAMS, as optimizing the joint point to decrease the maximum pressure of oil cylinders.

Taking oil cylinder 5 as target subject, then discuss the force situation of oil cylinder 5 when the arm is in different positions. Since the oil cylinder 6 is the driving power between arm 4 and arm 5, and expect for the arm 4, changing position of arms will not affect oil cylinder. Thus, introducing length variable of oil cylinder 5, making oil cylinder 6 stretch out and withdraw completely as arms 4 changes position at a time. We can draw the stress curves diagram based on series movements mentioned above. To identify the motion of each arms, a STEP function of time will added to each oil cylinder. The STEP function is shown as:

$$\mathrm{STEP}(x,\ x0,\ h0,\ x1,\ h1) \qquad 4.1$$

STEP function is suppose to erase the velocity step from move to stop. Velocity step will cause incorrect interference peaks in oil cylinders force curve [12]. And, x is the independent variable, which could be time function or any time-relevant-functions; x1 is the end value of STEP function, x1 could be a constant, a function expression or any design variable; h0 is the initial value of STEP function, h0 could be a constant, a design variable or other function expressions; h1 is the end value of STEP function, h1 could be a constant, a design variable or other function expressions.

Based on the above simulation conditions, according to different working conditions, we define the driving force separately, and measure stress of different oil cylinders. Then we apply ADAMS to simulate arms from 2 to 5 separately.

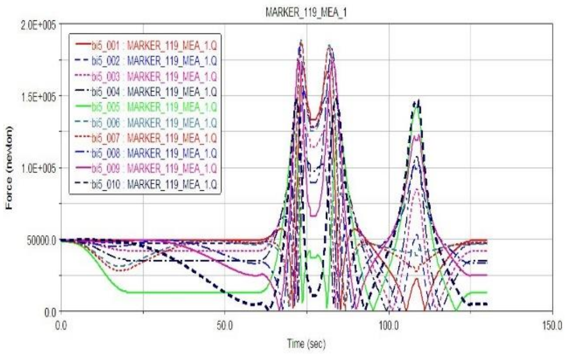


Figure 4.2 Cylinder 6 Working Diagram

The maximum and minimum stress of oil cylinder 6 as shown in table 4.3,

Table 4.3 Cylinder 6 under the condition of stress value

Test No.	Max Stress （N）	Mini Stress （N）
1	170 560	332
2	153 720	358
3	129 590	491
4	107 320	217
5	141 380	142
6	188 870	746
7	186 980	3129
8	184 230	783
9	174 670	677
10	147 480	200

By the Table 4.3, the maximum stress of oil cylinder6 is occur in the test No.6, the maximum stress is 188.87 KN.

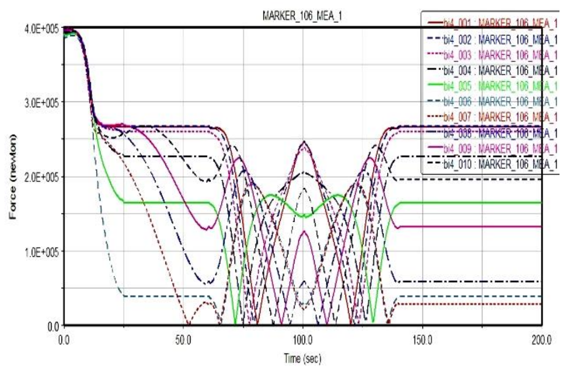


Figure 4.4 Cylinder 5 Working Diagram

The maximum and minimum stress of oil cylinder 5 as shown in Table 4.5

Table 4.5 Cylinder 5 under the Condition of Stress Value

Test No.	Max Stress （N）	Mini Stress （N）
1	396 060	1 656
2	395 160	362
3	394 340	1 096
4	393 110	330
5	391 670	1 292
6	389 450	257
7	399 220	845
8	397 210	320
9	395 860	1 421
10	394 630	1 200

The maximum stress is 399.22 KN, in the seventh test.

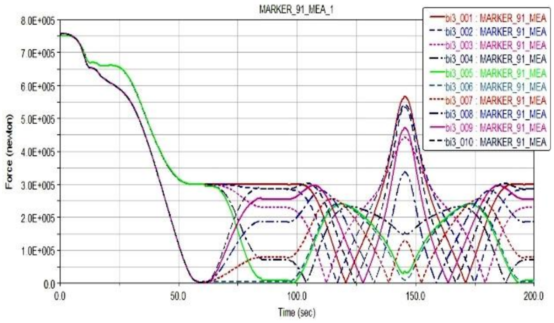


Figure 4.6 Cylinder 4 Working Diagram

The maximum and minimum stress of oil cylinder4 as shown in Table 4.7

Table 4.7 Cylinder 4 under the Condition of Stress Value

Test No.	Max Stress (N)	Mini Stress (N)
1	751 370	2 256
2	751 370	1 273
3	751 370	1 410
4	751 370	1 099
5	751 370	630
6	756 780	1 959
7	756 780	581
8	756 780	1 053
9	756 780	682
10	756 780	367

The maximum stress is 756.78 KN, happened in test 6 to 10

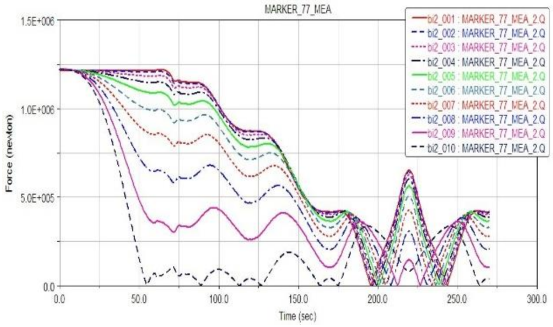


Figure4.8 Cylinder 3 Working Diagram

The maximum and the minimum stress of oil cylinder3 as shown in Table 4.9

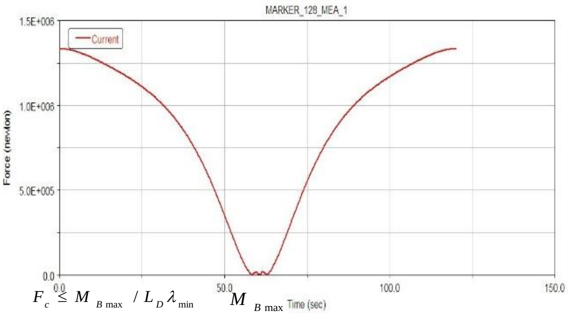
Table 4.9 Cylinder 3 under the Condition of Stress Value

Test No.	Max Stress (N)	Mini Stress (N)
1	1 220 000	2 191
2	1 216 660	2 511
3	1 217 200	4 025
4	1 217 800	5 677
5	1 218 500	6 930
6	1 219 400	3 995
7	1 220 300	5 466
8	1 221 400	6 220
9	1 222 600	4 106
10	1 223 800	385

The maximum pressure of oil cylinder3 occurred in text No.10, the pressure is 1223.8KN.

Since the oil cylinder1 and oil cylinder2 work together on the arm1, we only need to define one of the oil cylinders when measure and establish the actuation. Thus the paper defines the oil cylinder1.

The maximum pressure of oil cylinder occurs when the fifth arm of arm device lifts from horizontal position to vertical position. The simulation curve diagram of the maximum pressure as shown in fig 4.10



$F_c \le M_{B\max} / L_D \lambda_{\min}$ $M_{B\max}$ F_c

$\lambda_{\min}$

Figure 4.10 Cylinder 1 Working Diagram

The maximum pressure of oil cylinder1 is 1334.1 KN, the minimum pressure of oil cylinder1 is 1.23 KN.

After analyzing the force situation of series oil cylinders, it shows: under certain working area, some specific positions will cause oil cylinders suffer more pressure, which leads two points adverse effects:

1. When the cylinder-bore is determined, heavier weight leads to higher oil pressure, and higher pressure will cause serious oil leakage;

2. When the oil pressure is determined, increasing cylinder-bore in order to decrease the oil pressure, especially in limited space, will go against to the design of arm device.

4.6 Optimizing the Joints of Oil Cylinder Jib Lubing Mechanism.

4.6.1 Determine the Optimizing Subject

By section 2, $F = \min(\max \ |F_3|)$, F_3 is the maximum resisting torque, is the maximum oil pressure of derricking cylinder, is the minimum value determined by mechanism geometric parameter.

Parameters of jib lubbing mechanism are determined by geometric parameters of joints' coordinates. Once the joint's coordinate is determined, coordinates of relevant devices will determined, too. Based on the formula, the working pressure of oil cylinder is determined by relevant devices coordinates.

The oil cylinder between two connected arms is the only actuator that drives arms, thus the performance of oil cylinder directly determines the pump truck performance. When the load is heavy, oil cylinders should provide high oil pressure to satisfy the work command, and increasing oil pressure presents higher demands for oil cylinder itself. However, increasing cylinder-bore will influence arrangement of other devices, thus it is significant to optimize oil cylinders properly. This paper adopts the virtual prototype parametric technology to optimize the oil cylinders joints' coordinates.

Analyzing the joint position of oil cylinder3. In Figure 2.1, oil cylinder3 drives arm2 to stretch out and withdraw, and only arm1's position will determine the position of oil cylinder3. This paper aims at optimizing the stress problem of oil cylinder3. By table 4.9, we are aware, the maximum stress occurred in the No.10 test, thus the optimizing goal is to minimize the maximum stress of oil cylinder3.

4.1

: The force in No.10 test

4.6.2 Determine the Optimizing Variable

Based on the arm rigid prototype we optimize joints of oil cylinders. Components are constrained with linkage by revolute joint, we parameterize the relevant revolute joint marker to improve the design efficiency.

By fig 2.1, ACBO is a four-bar linkage, AEFD is jib lubbing mechanism, ACBO is joint with AEFD by bending linkage ADC, thus there are 6 joints in these mechanism in total. Taking no account of the joint O, which connect two arms, there are remain 5 joints.

In the plane-coordinate system, there are 10 coordinate parameters of 5 joints. Defining these 10 coordinates as variable value, through optimizing, we have sensibility value of each variables. As shown in table 4.10:

Table 4.10 Each Design Variable Sensitivity Value

Coordinates	Variables	Origin Value/mm	Minimum Value/mm	Maximum Value/mm	Sensibility
A_x	DV_143x	8 823.47	8 743.47	8 903.47	-2 446.5
A_y	DV_143y	942.48	842.48	1 042.48	-826.79
B_x	DV_81x	9 562.50	9 462.5	9 662.5	-30.702
B_y	DV_81y	996.12	896.12	1 096.12	252.92
C_x	DV_79x	9 607.19	9 557.19	9 657.19	-1 274.3
C_y	DV_79y	108.11	58.11	158.11	-223.11
D_x	DV_78x	9 410.81	9 360.81	9 460.81	593.92
D_y	DV_78y	-43.83	-93.83	6.17	1 060
E_x	DV_141x	5 379.43	5 329.43	5 429.43	-371.61
E_y	DV_141y	615.81	565.81	665.81	318.64

By table 4.10, it declares that A_y, A_x, C_x, D_x, D_y have high sensibility to the oil cylinder force changing.

Besides, based on the negative and positive condition of sensibility, we could assume the tendency of coordinate positions as oil cylinders' force changing. When the sensibility is negative, it declares that the target value decreases as the coordinate value increases; when the sensibility is positive, it declares that the target value decreases as the coordinate value decreases. Through the analysis of sensibility, we target the joint coordinate which is much more sensible to the changing force. Then we primarily confirm the direction of joint where it should move in order to optimize the oil cylinder force condition.

By table 4.11, the maximum pressure of oil cylinder3 is 1223.8 KN, occurred in No.3 test. Thus we rebuild this working condition, to analyze and optimize the link mechanism. Through the sensibility analysis, we choose variables which more sensible, thus coordinate variable of , , , , become design variable. The range of them are shown in table 4.10. Then after optimizing the joint coordinates in the ADAM, optimal solutions are shown in table 4.11

Table 4.11 Coordinate Changes in the Design Variables

Variable Name	Before Optimizing	After Optimizing
DV_143x	8 823.47	8903.5
DV_143y	942.48	1042.7
DV_79x	9 607.19	9657.2

DV_78x	9 410.81	9360.8
DV_78y	-43.83	-93.83

5. Analysis of Oil Cylinder Simulation

Curves before and after optimizing are shown in the fig 5.1:

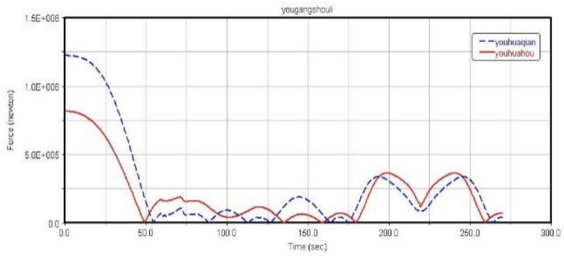


Figure 5.1 Luffing Cylinder 3 Optimized Stress Diagram

The movement of oil cylinders is accord with the law of STEP driving function. After optimizing joints stress of oil cylinder under different working conditions, we have the force curve diagrams of different oil cylinders under different working conditions. Then through comparing and analyzing simulated data, we target the oil cylinder3, which suffers the largest oil pressure under the most dangerous working condition.

By the diagram above, it shows, the stress condition of oil cylinder is improved after optimizing in majority, for example, the maximum stress of oil cylinder is decrease from 1223.8 KN to 816.8 KN.

6. Conclusion

Comparing with the former data, this paper parameterizes the joint coordinates of oil cylinder3, then set the optimizing target which is to decrease the maximum stress of oil cylinder. After simulating, calculating and analyzing the sensibility of pairs oil cylinders, we target some joints and relevant coordinates which seriously suffered of cylinder changing force. Then through optimizing coordinates, the maximum stress of oil cylinders are decreased efficiently. Since position of joints have changed, the maximum stress decreases from 1223.8 KN to 816.8 KN.

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2008-2009, 2009-2010 The outstanding postgraduate of Beijing Forestry University.
2008-2009, 2009-2010 The excellent academic award of Beijing Forestry University.
2008-2009, 2009-2010 The scholarship for outstanding postgraduate of Beijing Forestry University.

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1992-11. B.A undergraduate students in Central South University of Forestry and Technology. Majoring in Forestry Engineering by now.
Mr. Zhou received following awards:
2011~2012 The Merit Student of Central South University of Forestry and Technology.
2012~2013 Third grade scholarship of Central South University of Forestry and Technology.

