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Finite element analysis of working device for hydraulic excavator

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ABSTRACT

According to the type of YC225LC-8 hydraulic backhoe excavator, mechanical analysis was carried out in three typical work condition of the working device by using the mechanical theory and method. The static strength finite element analysis of excavator boom was carried through by using ANSYS, from which, the stress and strain deformation contour diagrams of three typical work condition were obtained so that the hardness can be checked. The results of finite element analysis showed that the static intensity of the boom is enough. The maximum stress mainly occurred in the hinge point connected the boom cylinder with the boom and the hinge point connected the boom with the base, which played an important part in controlling the strength of the excavator working device. The study results are of certain guiding significance for working device's optimization design.

Key words: hydraulic excavator; working device; boom; finite element analysis; Pro/E; ANSYS; Workbench

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INTRODUCTION

Nowadays, hydraulic excavator is a kind of multi-functional engineering machinery which is widely used in transportation, water conservancy engineering, construction, mining, excavation, power engineering, mining areas and forestry applications [1]. In the process of operation, the type of YC225LC-8 hydraulic excavator was under complex stress condition such as tensile and compressing load, torsional load and shocking load, and therefore it required a higher intensity of working device. In the process of operation, the boom will often be subjected to a variety of force, so it is necessary to research the stress and strain distributions of the boom under loaded. In this paper, in order to identify structural weaknesses area of YC225LC-8 type hydraulic excavator's working device, we attempt to respectively analyze the stress and strain distribution under three typical work.

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Mechanical analysis of hydraulic excavator in the typical work condition

Composition of the working device of hydraulic excavator: The working device of YC225LC-8 type hydraulic backhoe excavator was composed of boom mechanism, bucket rod mechanism, bucket mechanism, linkage mechanism and so on, using the principle of link mechanism. The other parts of the movement such as the boom going up and down were achieved by the telescopic hydraulic cylinder to complete a variety of complex motion. The boom and boom oil's cylinder composed the boom linkage mechanism (short for boom mechanism), the bucket rod and bucket rod's oil cylinder composed the bucket rod linkage mechanism (short for bucket rod mechanism), and the bucket and bucket's oil cylinder and linkage mechanism composed bucket linkage mechanism (short for bucket mechanism), all of above linkage mechanism connected with each other by means of pin-hinged.[4] The Working device of hydraulic excavator is shown in Fig. 1 below.

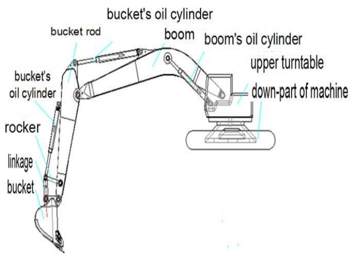


Fig.1: Working device of hydraulic excavator

Mechanical calculation of hydraulic excavator's working device under typical work condition:

Three typical work condition of hydraulic excavator:

Work condition 1: Tip of bucket teeth locates on the extension cord which is determined by one hinge point produced by bucket and bucket rod and the other hinge point produced by bucket rod and boom, and force applying dipper fluid cylinder pressure lever is largest. In this work condition, loads are gravity, lateral force and tangential force. The digging resistance reached the maximum. Work condition 2: Boom oil cylinder shrinks completely, Tip of bucket teeth locates on the extension cord which is determined by one hinge point produced by boom and boom rod and the other hinge point produced by bucket rod and bucket, the force arm of bucket rod's oil cylinder is largest, and bucket, boom and cylinder simultaneously closed. In this work condition, loads are gravity and tangential force. The digging resistance reached the maximum. Work condition3: Boom fluid cylinder shrinks completely, tip of bucket teeth locates on the extension cord which is determined by one hinge point produced by bucket and bucket rod and the other hinge point produced by bucket rod and boom, and lines which the three points subordinate locate on the plumb line pressure lever produced by boom and oil cylinder is largest. In this work condition, loads are gravity, lateral force and tangential force. The digging resistance reached the maximum.

Mechanical calculation of bucket digging resistance of hydraulic excavator: Taken work condition1 for example: for the boom will be bear large moment under such condition, the maximum stress of boom's dangerous section is typically produced when using the bucket digging. Therefore we can calculate on the work condition1 to get digging resistance of bucket.

When using bucket digging, the bucket digging resistance can be divided into lateral digging resistance and tangential digging resistance, here the maximum tangential resistance for bucket is:

$$W_{t\max} = 1.35C[R(1-\cos\varphi_{\max})]BAZX + D \tag{1}$$

where, C is coefficient of soil hardness, here we select the grade III soil, so C is 90; R is the distance between hinge point produced by bucket and bucket rod and tip of bucket teeth in the longitudinal symmetry plane, here R is 151 cm; $\varphi_{\max}$ is half of total angle when bucket full mining, here $\varphi_{\max}$ is 55°; B is coefficient of cutting edge width, B=1+2.6b, Where B is the average width of bucket, thus b is 13.1 cm，B is 4. 406; A is coefficient of cutting angle variation influence, here A is 1.3; Z is coefficient with bucket teeth, Z is 0.75(Z is 1 without bucket teeth); x is coefficient of side-bucket wall thickness influence, here x is 1.075; D is force of squeezing force from cutting edge to soil. According to the bucket capacity, here D is 15000 N. We substituted above parameters into Eq (1), then $W_{t\max} = 129975$ N

In general, the lateral digging resistance is smaller than the tangential digging resistance, by empirical formula:

$$W_{n\max} = 0.2W_{t\max} = 25995\,N \tag{2}$$

Loads calculation of hydraulic excavator's working device under typical work condition: Taken work condition1 for example, we calculate loads of hydraulic excavator's working device under typical work condition. The paths of the forces transferred through bucket oil cylinder, bucket rod oil cylinder, and boom rod oil cylinder. The thrust force F_2 for bucket rod's oil cylinder was shown in Fig.2. The torque balance equation of G point can be obtained as below:

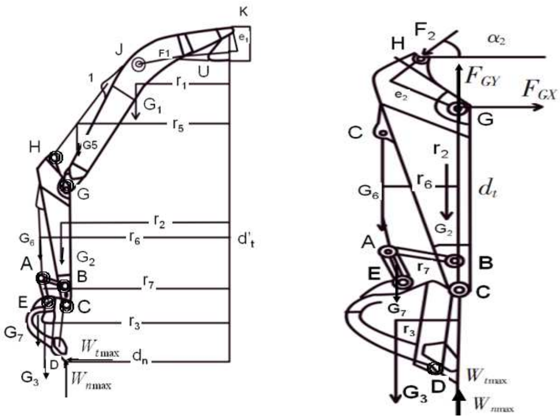


Fig.2: Calculation pressure F_1 and F_2 of oil cylinder pressure

$$\sum M_G = F_2 e_2 + G_2 r_2 + G_3 r_3 + G_6 r_6 + G_7 r_7 - W_{n\max} d_t = 0 \quad (3)$$

Where G_2, G_3, G_6, G_7 are respectively the mass of bucket rod, bucket, bucket cylinder, and linkage; e_2 is the force arm of F_2 to G point; r_2, r_3, r_6, r_7 are respectively the force arm of G_2, G_3, G_6, G_7 to G point; d_t is the force arm of $W_{n\max}$ to G point. Through measuring, we got that the mass of boom, bucket rod and bucket respectively are 2085 kg, 1175 kg, 1150 kg. Therefore the weight of boom, bucket rod and bucket respectively are 20.5KN, 11.7KN and 11.2KN when choosing 9.8 N/kg for the acceleration of gravity. [5]

The torque balance equation of K point can be obtained as below:

$$\begin{aligned}\sum M_K &= F_1 e_1 - G_1 r_1 - G_2 r_2 - G_3 r_3 - G_5 r_5 - G_6 r_6 \\ &- G_7 r_7 + W_{n\max} d_t + W_{n\max} d_n = 0\end{aligned} \quad (4)$$

Where G_1, G_5 are respectively the mass of boom's oil cylinder, bucket rod's oil cylinder; e_1 is the force arm of F_1 to K point; r_1, r_5 are respectively the force arm of G_1, G_5 to K point; d_t is the force arm of $W_{n\max}$ to K point; d_n is the force arm of $W_{n\max}$ to K point.

We can separately worked out forces of three oil cylinders, furthermore obtained forces of each hinge joints:

$$F_{GX} - W_{t\max} - F_2 \cos\alpha_2 = 0 \quad (5)$$

$$F_{GX} - W_{t\max} - F_2 \cos\alpha_2 - G_2 - G_3 - G_6 - G_7 = 0 \quad (6)$$

Where α_2 is the angle between F_2 and the horizontal direction. According to the Eq. (5), Eq.(6), the dimension and direction of F_{GX} and F_{GY} can be obtained, which applied on G point of bucket rod. Finally, by taking boom as research project, we can work out the dimension and direction of $F_{GX'}$ and $F_{GY'}$ applied on G point of boom, as Fig.3 shown below:

To sum up, according to Eq .(1) - Eq. (6), we can worked out load distributions of work condition for each hinge points such as hinge point G produced by boom and bucket

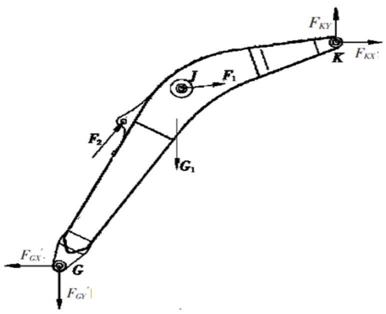


Fig. 3: Stresses condition of boom

and bucket rod's oil cylinder as well as hinge point J produced by boom and boom's oil cylinder. Similarly available load distributions of work condition can be acquired for each hinge points under work condition 2 and work condition 3, as shown in table 1.

Table 1: stresses of each hinge joint of boom

Work condition	direc-tion	Force of hinge point(KN)			
		F_1	F_2	F_{GX}	F_{GY}
Work condition 1	X	347	534	-226	-655
	Y	-252	65	316	-43
Work condition 2	X	428	413	-290	-575
	Y	-24	345	122	-422
Work condition 3	X	423	287	-340	-370
	Y	27	311	138	-43

FINITE ELEMENT ANALYSIS OF BOOM

Establishment of boom's geometrical model on Pro/E: The boom's geometrical model has been established according to the parameters of YC225LC-8 type hydraulic backhoe excavator by using 3D modeling software Pro/E. Boom is mainly welded together by thin plates of different thickness whose thickness is less than 16mm, and also it contains some other structural parts. During modeling process, threaded hole are omitted, transportation lifting lug, chamfers and other insignificant factors are also neglected [6] . Fig.4 shows the 3D geometrical model of boom.

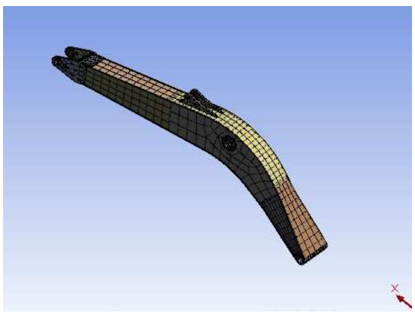


Fig. 4：Three-dimension (3D) model of boom

Establishment of boom's finite element (FE) model on workbench：Through establishing interface between workbench and Pro/E, we import the 3D geometrical model of boom into Workbench [7] . The finite element (FE) model of boom was shown in Fig.5.

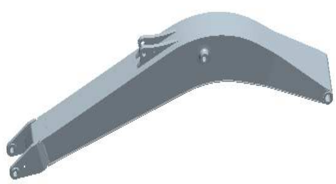


Fig. 5: Finite element (FE) model of boom

The finite element (FE) model shown in Fig.6. The boom of excavator is meshed into 9655 elements and 25075 nodes. The result is shown by following Fig. 6.

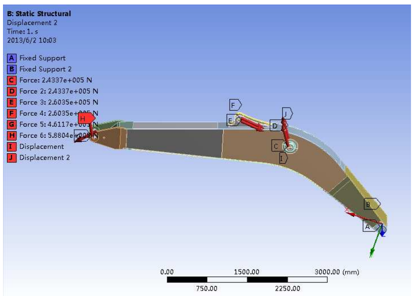


Fig.6: Meshing of boom

Constraint and load definition: The constraints are imposed on boom model according to actual work condition and the FEA requirement of workbench software. The constraint and load added as shown in Fig. 7.

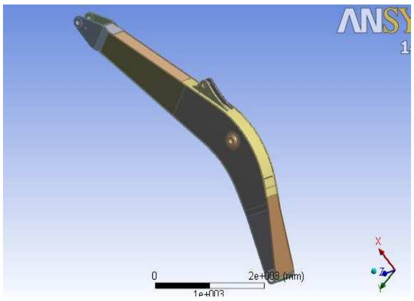


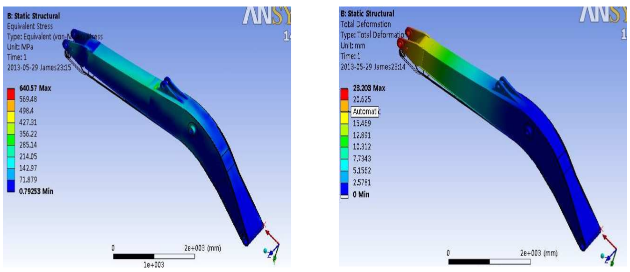
Fig. 7: Constraint and load definition of boom

Results of finite element analysis: Results of finite element analysis for three typical work conditions were shown in Fig. 9, including stress contour diagrams and strain contour diagrams.

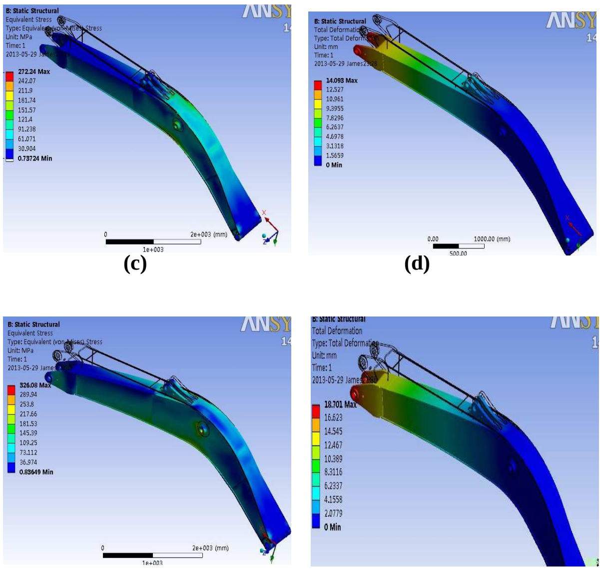
Fig.8(a-f) shows stress contour diagrams and strain contour diagrams of boom for three typical work conditions, it can be seen from the figure that work condition 1 is the most dangerous one. Boom back-end of primary structure appear small group of crack in the same location, which can conclude the fatigue crack, stress in the area go beyond combined stresses (230MPa) of Q345 parent metal, but also because it has a welded tube socket on the cover, weld may make material organization change and weld undercuts, which can decrease yield value of stress. This location may turn up fatigue failure prematurely.

CONCLUSION

Finite element modeling analysis showed that: the maximum stress does not exceed the material's yield and tensile strength, the boom of design is reasonable and does not need to be improved. Finite element analysis has great significance for the actual production process not only to study the excavator’s specific conditions using the actual data to simulate the working condition, which are close to the actual production results, but also to save the cost of research and development.



(a) **(b)**



(c) (d)

(e) (f)

Fig.8: Stress contour diagrams and strain contour diagrams of boom for three typical work conditions

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