The Finite Element Analysis of the Boom of 20-ton Backhoe Hydraulic Excavator Based on ANSYS

Chao-ying Meng¹, Shan Fan¹a, and Lei-lei Han¹
¹Wenhua College, 430074 Wuhan, China

Corresponding author: 271732369@qq.com

Abstract. Taking the boom of 20-ton backhoe hydraulic excavator as the research object, this paper analyzed the strength of the boom by ANSYS software, simulated the load condition of the boom under the bucket excavation condition, and obtained the stress distribution cloud diagram of the boom. The analysis results show that the stress of the boom of the excavator is mainly concentrated on the hinge of the boom and the stick cylinder, which is the weakest link.

1. Introduction
Excavators are widely used in many industries of national economic construction. The working device is an important part of the excavator. It is mainly composed of the stick, the boom, the boom cylinder, the bucket cylinder and the bucket [1]. The boom is the main component of the excavator directly subjected to the working load, and its structural strength directly affects the reliability and working performance of the excavator. The analysis and research of the boom is the basis for the development of the whole machine. The stress on the boom is very complicated and constantly changing, and the working environment is also very bad. Therefore, it is difficult to get a real working situation analysis. At present, a large number of scholars have done a series of research on the force of the working device, including kinematics and dynamics analysis calculation, optimization design, reliability analysis and so on. [2-4]. There is very little analysis of the boom of the backhoe excavator. This article took the boom of a 20-ton backhoe excavator as the research object. The author established the parametric model of the excavator boom by Pro/E and imported it into the ANSYS software to analyze the force of the boom under the bucket excavation condition.

2. The force Analysis of Excavator Boom
The moving speed of each component in the excavator is small and the force is large. Therefore, the force of the boom is mainly analyzed from the static point of view. When the excavator is in the deepest excavation state, the boom is lowered to the lowest point, and the bucket arm has the largest force arm. The stick and the arm cylinder hinge point F, the stick and bucket hinge point Q and the bucket tip point V are three points on a lead line. The position of its working device is shown in Figure 1. From the analysis of various failures of the working device, the maximum stress of the dangerous section of the boom occurs in the bucket excavation condition. Therefore, this paper selected this working condition for force analysis. During work, the tooth resistance is affected by the cutting resistance and the soil resistance, and the sum of the two is called the total resistance of the excavation [5]. However, accurate calculation of the total resistance is more difficult. Therefore, the total drag W is generally decomposed into two
component forces in the calculation: the excavation resistance $F_{wt}$ in the tangential direction and the excavation resistance $F_{wn}$ in the normal direction. At the same time, the working device is also subjected to the force of the oil cylinder. There are three main types: the boom cylinder thrust $F_1$, the arm cylinder thrust $F_2$, and the bucket hydraulic cylinder force $F_3$. The force is shown in Figure 1.

![Figure 1. Excavator working device position and force analysis](image)

### 3. Load Calculation

#### 3.1. The Bucket Excavation Resistance

$F_{wt}$ is the component of the total digging force in the tangential direction, which is the excavation resistance; and $F_{wn}$ is the resistance generated when the bucket is inserted into the soil. In fact, the excavation resistance $F_{wt}$ consists of three parts, which are the additional resistance when loading soil, the frictional resistance of the soil to the bucket, and the resistance to soil cutting [6]. The empirical formula of $F_{wt}$ is as follows:

$$F_{wt} = C \left( r_1 \left[ 1 - \frac{\cos \emptyset_{\text{max}}}{\cos \emptyset_{\text{max}} - \emptyset} \right] \right)^{1.35} B A Z X + D \tag{1}$$

Where $C$ is the hardness coefficient of the soil, different values for different soil conditions, for grade II soil, $C=50 \sim 80$, for grade III soil, $C=90 \sim 150$, for grade VI soil, $C=160 \sim 320$. This paper assumes that the excavator is used for excavation of class III soil, its value can be 100; $r_1$ is the distance between the bucket and the stick hinge to the tip of the bucket, $r_1=1500\text{mm}$; $\emptyset_{\text{max}}$ is the half of the total corner of the bucket during the excavation; $\emptyset$ is the instantaneous corner of the bucket, take the empirical value $\emptyset_{\text{max}}=\emptyset=55$; $B$ is the cutting edge width influence coefficient, $B = 1 + 2.6b = 1 + 2.6 \times 1.04 = 3.7$; $A$ is the cutting angle variation influence coefficient, $A = 1.3$; $Z$ is the bucket coefficient, $Z = 0.75$; $X$ is the bucket sidewall thickness influence coefficient, $X = 1.15$; $D$ is the force of the cutting edge to squeeze the soil, according to the empirical statistics and the size of the bucket capacity, $D = 1.35 \times 104N$.

Bring the parameters into equation (1), we get:

$$F_{wt} = 168560N$$

The digging resistance of the bucket is generally less than that of the tangential excavation. The empirical formula is:

$$F_{wn} = 0.2 \times F_{wt} = 33712N$$
3.2. The Cylinder Thrust

There are three kinds of cylinders on the hydraulic excavator working device—the bucket cylinder, the stick cylinder and the boom cylinder. The determination of the force of these cylinders depends on the type of the excavator and the specific working conditions. Since only the force of the boom is analyzed here, it is only necessary to calculate the boom cylinder thrust $F_1$ and the arm cylinder thrust $F_2$. Calculate its size according to the torque balance formula.

Ignore the quality of the bucket and soil, the quality of each component, and the factors affecting the efficiency of the linkage mechanism \[7\]. According to the moment balance, the formula for calculating $F_1$ is as follows:

$$F_1 = \frac{F_{wt}l_A + F_{wn}l_A'}{l_1} \quad (2)$$

Where $l_1$ is the force arm of the boom cylinder thrust $F_1$ to the hinge point C on the boom; $l_A$ is the force arm of $F_{wt}$ on the hinge point C of the boom; $l_A'$ is the force arm of $F_{wn}$ on the hinge point C of the boom. The specific location is shown in Figure 1.

At this time, the arm and the boom cylinder are both locked, and we have:

$$F_2 = \frac{F_{wt}}{l_2} \quad (3)$$

Where $l_2$ is the arm of the arm cylinder thrust $F_2$ to the hinge point F; $l_B$ is the force arm of $F_{wt}$ to the hinge point F. The specific location is shown in Figure 1.

By calculation, we get:

$$F_1 = 1062N; F_2 = 256N$$

3.3 The calculation of hinge point reaction force

The force of the stick and the boom is shown in Figure 2. We decomposed the force at F into $F_{Fx}$ in the x direction and $F_{Fy}$ in the y direction. The stick is in equilibrium under the action of these forces, so we have:

$$\begin{cases}
F_{Fx} = -F_{wt} - F_2 \cos \alpha \\
F_{Fy} = -F_{wn} + F_2 \sin \alpha + G_2 + G_3 + G_4 + G_5
\end{cases} \quad (4)$$

Where $\alpha$ is angle between $F_2$ and x direction; $G_2, G_3, G_4, G_5$ is the gravity of the stick, bucket, bucket cylinder and connecting rod, its value can be 3.1 tons.

![Figure 2. Stick reaming force diagram](image1)

![Figure 3. Boom reaming force diagram](image2)
The boom was extracted for force analysis, and the force was shown in Figure 3. According to the relationship between the force and the reaction force, the magnitude and direction of the F-force of the boom hinge point could be obtained. According to the principle of static balance and torque balance, we could calculate the magnitude and direction of the load at the hinge points B, D and F of the boom under dangerous conditions. The specific values are shown in Table 1.

| F₁/KN | F₂/KN | Fₓ/KN | Fᵧ/KN | α/ (°) |
|-------|-------|-------|-------|-------|
| 1062  | 256   | 252   | -301  | 47    |

4. Finite Element Analysis

4.1 Finite Element Model

This paper used Pro/E software to parametrically model the boom. The boom is a welded part, and its structure is complicated. It has a lot of rounded corners and steps, which makes modeling difficult, so the necessary simplification of the boom was done before modeling. Small components such as rounded corners, steps, and ribs that do not affect the boom were omitted. Then we imported the 3D model into ANSYS software. The finite element model is shown in Figure 4.

4.2 Apply Constraints and Loads

4.2.1 The constraint

When the excavator was in the bucket excavation condition, the boom was hinged to the frame. So we applied full constraint to the inner wall of the boom and frame hinge C. Since the stick and the bucket hinge point and the bucket tip point were on a vertical line, the inner hole at F was restrained in the y direction.

4.2.2 The load condition

The components of the working device were hinged by the pin shaft. If the influence of the eccentric load was ignored in the calculation of the working condition, the boom could be considered to bear only the forces in the X and Y directions, and the entire boom is a statically fixed structure. In fact, the force at the hinge point is not a concentrated force or a uniformly distributed load. To simulate the actual situation, we assumed that the load was distributed by cosine in the range of 180° in the X-Y plane. The direction of the distributed force was the normal along the surface of the pin hole, and the load is evenly distributed in the z direction. The distribution of forces is shown in Figure 5. According to the above assumption, the distribution function F(θ) of the load is:

\[ F(\theta) = P \cos(\theta - \alpha) \] (5)
Where $\theta$ is the angle between the line connecting the point of the pin hole and the center of the circle and the $X$-axis direction, $\theta=0 \sim 180^\circ$; $P$ is the undetermined coefficient; $\alpha$ is the angle between the resultant force $R$ and the $X$ axis.

The resultant $R$ size is:

$$R = lr \int_{\frac{\pi}{2}}^{\frac{\pi}{2} + \alpha} \cos(\theta - \alpha) d\theta = lr \int_{\frac{\pi}{2}}^{\frac{\pi}{2} + \alpha} \cos^2(\theta - \alpha) d\theta$$

(6)

The resultant force $R$ is known, and the sum of the components of the distribution force $F$ in the direction of the resultant force should be equal to $R$, and the undetermined coefficient can be derived according to the formula (6):

$$P = \frac{2R}{\pi lr}$$

(7)

Where $l$ is the length of the pin hole; $r$ is the radius of the pin hole; $R$ is the resultant force of the cosine distribution load at the pin hole.

According to the formula (6), the distribution function of the load at the pin hole can be obtained, and the undetermined coefficient $P$ can be obtained according to the formula (7). For any hinge point, it is only necessary to know the magnitude and direction of the force it is subjected to apply the cosine-changing surface load.

Apply boundary constraints to the boom based on the force of the boom (see Figure 4) and the calculated load. Since the structure and force of each reaming are different, different loads must be applied to the three reaming holes of the boom. At the hinge hole B of the excavator boom and boom cylinder, the boom and the arm cylinder reaming D are at the pin joint, and the surface load was distributed as a cosine function with angle. Therefore, we was applied a surface load that varies by cosine function to the cylindrical surface. At the same time, we applied a concentrated force load in the $x$ and $y$ directions at the hinge point F.

5. Result Analysis

After pre-processing, the software can complete different preset commands in the module to view different results. The results of the analysis can be presented in a variety of forms in the software, enabling observation of the results from different angles [8-9]. After the solution, the overall deformation map and the stress cloud map were obtained. After solving, the deformation displacement cloud diagram is shown in Figure 6. The stress cloud diagram of the boom and the arm cylinder reaming D is shown in Figure 7. The stress diagram of the boom and the arm reaming F is shown in Figure 8.

[Images of Figures 6, 7, and 8]

As can be seen from Figure 6, the boom undergoes compression deformation. The amount of displacement of the boom from top to bottom and from left to right gradually increases. The maximum displacement deformation is 0.074245mm, and the maximum deformation occurs at the reaming hole
where the boom and the arm are connected, that is, the lowest point of the boom when the boom is in
the bucket excavation condition. The amount of deformation of the boom is in the elastic range, so the
design meets the requirements.
As can be seen from Figure 7-8, the maximum stress of the boom occurs at the joint of the boom and
the arm of the stick. The maximum stress is $188.462 MP$. The material of the boom is $Q235$ steel, and
the allowable stress is $235 MP$. The maximum stress of the boom is close to the allowable stress, but
less than the allowable stress, within the safe range, the analysis results meet the requirements.

6. Conclusion
This paper analysed the force of the excavator boom under excavation conditions and performed finite
element analysis in ANSYS software. The analysis results show that the stress of the boom of the
excavator is mainly concentrated in the hinge of the boom and the arm of the stick, which is the
weakest link. Therefore, in the design of the boom, it can be improved by adding reinforcing ribs
inside the boom, increasing the thickness of the guard plate, and adding a reinforcing plate at the arm
cylinder support. It also shows that the finite element analysis of the excavator boom with the useful
finite element software ANSYS is a reliable method, which provides a valuable reference for product
design.

References
[1] D. Kim, J. Kim, K. Lee, C. Park, J. Song, D. Kang, Excavator tele-operation system using a
human arm, Automat Constr. 18, 173–182 (2009)
[2] K. Awuah-Offei, S. Frimpong, Cable shovel digging optimization for energy efficiency, Mech
Mach Theory. 42, 995-1006 (2007)
[3] S. Frimpong, Cable shovel health and longevity and operat or efficiency in oil sands excavation,
Mining and Mineral Engineering. 1, 15-33 (2008).
[4] S. Frimpong, H. Yafei, K. Awuah-Offei, Mechanics of cable shovel-formation interactions in
surface mining excavations, J Terramechanics. 42 (2005)
[5] A.R.Dexter, E.A.Czyz, O.P.Gat, A method for prediction of soil penetration resistance, Soil
Till Res. 93, 412-419 (2007)
[6] D. Dopico, A. Luaces, M. González, A soil model for a hydraulic simulator excavator based on
real-time multibody dynamics, ACMD. 23-26 (2010)
[7] H. Takahashi, M. Hasegawa, E. Nakano, Analysis on the resistive forces acting on the bucket of
a Load-Haul-Dump machine and a wheel loader in the scooping task, Adv Robotics. 13, 97-
114 (1998)
[8] S. Kim, J Ha, S Jeong, Effect of joint conditions on the dynamic behavior of a grinding wheel
Spindle, Int J Mach Tool Manu. 41, 1749-1761 (2001)
[9] I. Cotton, Bradken Building, 5 Mining Monthly. 7, 122 (2006)