Study on Calculating Method of External Load on Digging Arm of Underground Excavator Truck

Shuping Zheng, Zengjie Yang and Zhongliang Yang *
Jiangsu XCMG Construction Machinery Research Institute Ltd organization, Jiangsu Xuzhou, China

*Corresponding author e-mail: zhengshuping123@126.com

Abstract. The digging arm of underground excavator truck has many movable joints, complex structure and variable operation properties which results in the complicated loading. So it is difficult to obtain the external working load directly through the test. For this problem, the calculation of the external load on the digging arm of the underground excavator truck is studied by combining test and virtual prototype in this paper. The details include: Firstly, the static test of the digging arm was conducted using pressure sensor and displacement sensor, and the parameters of the virtual prototype built by Motion View software were modified. Then the pressure and displacement curves of each hydraulic oil cylinder are measured during the operation on site. The working process of digging arm is simulated and reconstructed by taking the measured displacement curves of each cylinder as the driving inputs, and outputs the curves of centre of mass coordinate, curves of acceleration and corresponding curves of torque. The rotary torque of external forces and gravity on a rotation axis (rotation axes of movable arm and pitch) should be equal to the sum of torque of the product of each component’s mass and the acceleration of the centre of mass on that point according to the law of torque calculation. Thus, the non-homogeneous linear equations of external loads (horizontal, vertical and lateral loads) are derived under the actual conditions. And external loads were solved by substituting the above curve parameters into the equations. Finally, the obtained external load is reloaded into the virtual prototype of the digging arm to simulate the force of actual digging process. The error is less than 5% by comparing the calculated and test values of the oil cylinders force of rod and bucket and the effective strain error of the movable arm is less than 10%. So the correctness of the derivation method of external load and the dynamic model considering inertial force is verified. It provides theoretical and algorithm support for the next acquisition of load spectrum and fatigue analysis of digging arm.

1. Introduction
The underground excavator truck [1] is a new type of construction machinery integrating, which integrates the functions of excavation, milling, and bolting. It is one of the key supporting equipment for the mechanized operation of underground excavation tunnels, which is mainly used in the construction of subway tunnels and municipal pipeline projects. The underground excavator truck is a double arm structure. In this paper, the external load calculation method of the left arm (digging arm) is
There are few such studies due to the multi-link arm structure of the digging arm (figure 1). Most of the research is about dynamic simulation of excavator arm and finite element static simulation under typical dangerous working conditions, etc. [2-4]. In reference [5], the external load is approximately obtained and the transient dynamic simulation analysis is carried out, which takes the maximum theoretical impact force of the digging arm as the driving force. In reference [6], a load spectrum test method is proposed which can reflect the actual working conditions of excavator working device. In reference [7], the load spectrum of transmission system is studied. In reference [8], the parameters and posture control of hydraulic excavator are studied. In reference [9], the digging force under typical excavation attitude is calculated and analyzed. In reference [10]-[16], the virtual prototype model establishment and dynamic simulation of excavator working device were studied, and the motion and digging force performance of working device were calculated and analyzed. Based on the above analysis, it is found that there is little research on the calculation of external load in the whole working process of the digging arm. In the above literature, the model is greatly simplified to obtain approximate solutions, and the digging force is only calculated for some typical or dangerous working conditions, rather than for all the loads in the work engineering. The biggest difference of this paper is that the method of combining the real and the virtual is adopted. At the same time, the influence of lateral load and inertial force is considered in the calculation of external load, so that the bucket tip load obtained is more accurate, and the external load of each position in the working process of the digging arm is obtained.

The digging arm of the underground excavator truck is mainly composed of movable arm, transition joint, bucket bar, bucket and other parts, which can not only realize forward excavation but also horizontal and lateral excavation in front and vertical excavation on both sides of the core soil. Due to narrow working space and harsh geological environment, it is difficult to measure the external load on bucket end directly. Therefore, in this paper, the force condition of the digging arm under static conditions was first tested with a weighing sensor to modify the Motion View virtual prototype. Then, the data of each cylinder was obtained by testing in the user site which was imported to the Motion View virtual prototype. By substituting the motion parameters of each component output from the virtual prototype and the cylinder force obtained by the test into the derived external load matrix equation, the external load curve of bucket end under actual conditions is obtained. The oil cylinder forces of bucket and bucket rod obtained by experiment and simulation are compared by reloading the external load curve. At the same time, the strain values of the moving arm measuring points are also compared. So the correctness of the calculation method of the external load of the digging arm proposed in this paper is further verified.

Figure 1. The structure of underground excavator truck.
2. The verification of calculation method of external load of digging arm under static conditions and model modification

By testing the cylinder force and stroke of the digging arm under no-load condition, the model quality distribution of each component of the digging arm is adjusted, so that each cylinder force of the test and the calculation are consistent, and finally the purpose of model modification is achieved. In order to verify the correctness of the method of deducing the external load of digging arm by the cylinder force, the digging arm is pressed down under the non-working condition (as shown in figure 2) to achieve the force, length of each cylinder and the data of weighing sensor. Each cylinder forces of the digging arm are obtained by loading the specified external loads (as shown in figure 3) on the models with different postures. The forces of each cylinder and external load obtained by test and calculation under different postures are listed in table 1. It can be seen from the table that the cylinder force errors are all within 6%, which shows that the corrected virtual prototype meets the next test requirement.

![Figure 2](image2.png)

**Figure 2.** The external loading test of digging arm in position A under static condition.

![Figure 3](image3.png)

**Figure 3.** The model of digging arm in position A under static condition.
Table 1. Comparison of cylinder force, external load test and calculated in different poses.

| posture | movable arm oil cylinder | pitch oil cylinder | bucket rod oil cylinder | bucket oil cylinder | load |
|---------|--------------------------|-------------------|------------------------|---------------------|------|
| A       | test values              | -204.0            | 218.0                  | 71.0                | -78.0| 57.9 |
|         | calculated value         | -202.0            | 216.0                  | 66.9                | -74.3|      |
|         | error %                  | -1.0              | -0.9                   | -5.8                | -4.7 |
| B       | test values              | -204.5            | 215.0                  | 73.0                | -9.5 |
|         | calculated value         | -205.0            | 215.0                  | 68.8                | -75.8| 57.0 |
|         | error %                  | 0.3               | 0.0                    | -5.8                | -4.7 |
| C       | test values              | -210.0            | 218.0                  | 74.2                | -84.4|      |
|         | calculated value         | -209.5            | 217.7                  | 74.2                | -84.4| 56.9 |
|         | error %                  | -0.2              | -0.1                   | -5.9                | -4.4 |
| D       | test values              | -237.4            | 217.2                  | 135.6               | 46.8 |
|         | calculated value         | -233.0            | 211.0                  | 129.0               | 47.2 |
|         | error %                  | -1.8              | -2.9                   | -4.9                | 0.8  |

3. Study on calculation method of external load of digging arm under dynamic working conditions

3.1. The data acquisition of digging arm under construction conditions

Data collection is carried out under the working condition of underground excavator truck, as shown in figure 4. The excavation medium is sand pebble soil, which is loose and adherent. The big cavity pressure, small cavity pressure, displacement and other physical quantities of each cylinder are measured by using oil pressure sensors and pull rope displacement sensors. After processing the test data, the force and displacement curves of each cylinder are obtained respectively. The force curves of the pitching cylinder and the movable arm cylinder are shown in figure 5. Figure 6 shows the displacement variation curves of the above oil cylinders.

Figure 4. The load test of digging arm in working condition.
3.2. The establishment of virtual prototype of digging arm

The corresponding dynamic simulation model (as shown in Figure 7) was established in the Motion View according to the model modification parameters under the static working condition of the digging arm previously. The model is symmetric about the XY plane, and the revolute pairs, sliding pairs, fixed pairs and spherical hinges are established in the corresponding position according to the actual working state. A local coordinate system is established for each cylinder, and X axis is set along the axis of the cylinder. The other two directions are determined according to the right hand rule. In order to simulate the external load, three perpendicular forces (F1, F2 and F3) are established at the bucket end, where F3 is perpendicular to the outside of the paper. Figure 8 shows the summary of the results after the time synchronization of video and the simulation motion of digging arm. The figure on the right shows the displacement loading curve of the movable arm oil cylinder. The displacement curves of each cylinder obtained in the working conditions were taken as the cylinder motion of the virtual prototype to simulate the actual working process of the digging arm.

Figure 5. The cylinder force curves of pitch and movable arm in working condition.

Figure 6. The cylinder displacement curves of pitch and movable arm in working condition.
3.3. The calculation process of external load of digging arm

Each component of the digging arm is shown in figure 9. The bucket end is subjected to horizontal forward force $F_1$, vertical upward force $F_2$ and horizontal lateral force $F_3$. The torque of the external load, the gravity of each component and the corresponding cylinder force at the specific rotation point should be equal to the sum of the torque of the product of the mass of each component and the
acceleration of the center of mass at this point during the movement of the digging arm, as shown in the following formula:

$$\sum F_i l_i + \sum m_j g h_j + F_{oil} l_{oil} = \sum m_i a_i n_j$$

(1)

The detailed calculation process is as follows:

Firstly, the rotation of bucket, bucket rod, transition joint, each oil cylinder and movable arm around point A at the root of movable arm was analyzed, and the following items were outputted:

Output the vertical projection distance of the center of mass of each component relative to the rotation axis of point A, namely the moment arm $h_{Ai}$ generated by the gravity of each component relative to the rotation axis of point A, and records the weight $m_i$ of each component.

According to the center of mass of each component and the rotation axis of A at the root of the moving arm, the local coordinate system of the center of mass relative to A is established. The origin of the coordinate system is the center of mass, the X-axis points to the rotation axis of point A, the Z-axis lies in the plane composed of the center of mass and the rotation axis of point A, the Y-axis is determined by the right hand rule, and the counterclockwise rotation of the center of mass around the rotation axis of point A is positive. Output acceleration $a_{Ai}$ of the center of mass of each component along the Y axis of its local coordinate system, and output the coordinates of point A relative to each component along the X-axis in the local coordinate system, which is the moment arm $(n_{Ai})$ of the center of mass’s acceleration.

A local coordinate system along the direction of F1 and F2 and F3 at the end of the bucket is established. In the local coordinate system, output the moment arms ($l_{A1}$, $l_{A2}$ and $l_{A3}$) of the F1, F2 and F3 relative to the axis of rotation at point A.

Output the vertical distance of point A relative to the axis of movable arm oil cylinder, that is, the moment arm $(l_{Ad})$ of the movable arm cylinder’s force relative to point A.

Therefore, the moment equation at point A of the rotating center of the moving arm can be established:

$$F_1 l_{A1} + F_2 l_{A2} + F_3 l_{A3} + \sum m_i g h_{Ai} + F_d l_{Ad} = \sum m_i a_{Ai} n_{Ai}$$

(2)

Where: $F_d$ is the force of the movable arm cylinder obtained by field test.

Then, take the rotation point B between the transition joint and the moving arm as the reference point, and output the vertical projection distance of the center of mass of bucket, bucket rod, transition joint 1, transition joint 2, bucket cylinder, bucket rod cylinder and other components relative to the rotation axis of point B, that is, the force arm $h_{Bj}$ generated by the gravity of each component relative to the rotation axis of point B.

The local coordinate system is established according to the centroid point of each component and the rotation axis of the point B between the transition joint and the moving arm. The origin of the coordinate system is the center of mass, the X-axis points to the rotation axis of point B, the Z-axis lies in the plane composed of the center of mass and the rotation axis of point B, the Y-axis is determined by the right hand rule, and the counterclockwise rotation of the center of mass around the rotation axis of point B is positive. Output acceleration $a_{Bj}$ of the center of mass of each component along the Y axis of its local coordinate system, and output the coordinates of point B relative to each component along the X-axis in the local coordinate system, which is the moment arm $(n_{Bj})$ of the center of mass’s acceleration.

According to the established local coordinate system along the direction of F1, F2 and F3 at the bucket end, the moment arms ($l_{B1}$, $l_{B2}$ and $l_{B3}$) of the load of F1, F2 and F3 relative to the rotation axis of point B is output.

Output the vertical distance of point B relative to the axis of pitch oil cylinder, that is, the moment arm $(l_{Bf})$ of the pitch oil cylinder’s force relative to point B.

Therefore, the torque equation at point B of the rotation center can be established:

$$F_1 l_{B1} + F_2 l_{B2} + F_3 l_{B3} + \sum m_j g h_{Bj} + F_f l_{Bf} = \sum m_j a_{Bj} n_{Bj}$$

(3)

Where: $F_f$ is the force of pitch oil cylinder obtained by field test.
Finally, the local coordinate system is established on the basis of the centroid point of each component and the rotation axis of C point connected between the movable arm seat and the frame. The origin of the coordinate system lies at the center of mass of the component, the X-axis points to the rotation axis of point C, the Z-axis is located in the plane composed of the center of mass and the rotation axis described above, the Y-axis is determined by the right hand rule, and the counterclockwise rotation of the center of mass around the rotation axis of point C is positive. Output acceleration \( a_{ck} \) of the center of mass of each component along the Y axis of its local coordinate system, and output the coordinates of point B relative to each component along the X-axis in the local coordinate system, which is the moment arm \((n_{ck})\) of the center of mass’s acceleration.

According to the established local coordinate system along the direction of F1, F2 and F3 at the bucket end, the moment arms \((l_{c1}, l_{c2} \text{ and } l_{c3})\) of the load of F1, F2 and F3 relative to the rotation axis of point C is output.

Output the vertical distance of point C relative to the axis of the swinging arm cylinder, that is, the moment arm \((l_{bf})\) of the swinging arm cylinder’s force relative to point C.

Therefore, the torque equation at point C of the rotation center can be established:

\[
F_1 l_{c1} + F_2 l_{c2} + F_3 l_{c3} + F_b l_{cf} = \sum m_k a_{ck} n_{ck} \tag{4}
\]

Where: \( F_b \) is the force of movable arm swinging cylinder obtained by field test.

According to formula (2)-(4), the following non-homogeneous linear equations concerning F1, F2 and F3 are composed:

\[
\begin{bmatrix}
  l_{A1} & l_{A2} & l_{A3} & F_1 \\
  l_{B1} & l_{B2} & l_{B3} & F_2 \\
  l_{C1} & l_{C2} & l_{C3} & F_3
\end{bmatrix}
= \begin{bmatrix}
  \sum m_i a_{Al} n_{Al} - \sum m_i g h_{Al} - F_d l_{Ad} \\
  \sum m_j a_{Bl} n_{Bl} - \sum m_j g h_{Bl} - F_f l_{bf} \\
  \sum m_k a_{Ck} n_{Ck} - F_b l_{cf}
\end{bmatrix}
\tag{5}
\]

Where, I, j and k correspond to the serial number combination of different parts at points A, B and C.

The parameters previously output and the cylinder force curves obtained from test were substituted into formula (5), and the change curves of F1, F2 and F3 were obtained, as shown in figure 10:

(a). The curve of external loading \( F_1 \) of digging arm.
Figure 10. The curves of external loading $F_1$, $F_2$ and $F_3$ of digging arm in working condition.

4. The verification of external loads

4.1. The oil cylinder force verification

The calculated external loads $F_1$, $F_2$ and $F_3$ were loaded into the bucket end, and the simulation analysis of the digging process was carried out performed again. The oil cylinder forces of bucket rod and bucket obtained by the simulation are compared with that obtained by actual test. Due to the large amount of data, only part of the data is displayed for clarity, as shown in figure 11 and figure 12. The data between 107 s and 109 s are intercepted for magnification comparison in figure 11. The data between 176 s and 178s are intercepted for magnification comparison in figure 12. It can be seen from the figure that the simulated cylinder force curves of bucket rod and bucket are in good agreement with the test. Figure 13 and 14 respectively show the relative error curves of cylinder force of bucket rod and bucket of simulation and test, and the error values of both are within 5%, which proves the correctness of the calculation methods of external loads $F_1$, $F_2$ and $F_3$ of the digging arm.
Figure 11. The comparison of testing and simulation cylinder force of bucket rod between 107 s and 109 s.

Figure 12. The comparison of testing and simulation cylinder force of bucket between 176 s and 178 s.

Figure 13. The relative error of cylinder force of testing and simulation of bucket rod in working condition.
4.2. The strain validation

The structure strain of movable arm obtained from the test and simulation are also compared and analyzed except oil cylinder force of bucket rod and bucket, so as to further verify the correctness of the dynamic simulation model and the calculation method of external load. Figure 15 shows the strain test location of movable arm in working condition. Figure 16 is the rigid-flexible coupling simulation result after replacing the movable arm with a flexible body, from which the simulated strain data of the measuring point can be extracted and compared with the test. The strain-time history curves of movable arm between the simulation and test were compared, as shown in figure 17. The data between 190 and 200 s at a certain point of the moving arm are intercepted for comparison for clarity. It can be seen from the figure that the values of both are basically the same. Figure 18 shows the relative error curve of movable arm’s strain obtained by the simulation and test. In the figure, the error value at some small micro-strain is very large, which is caused by the large measurement error of the small strain by the strain testing system. Therefore, after considering the test system error and fatigue life requirements, measurement points with macrostrain larger than 80 are selected for relative error comparison (as shown in figure 19). As can be seen from the figure, when the strain of movable arm is large, the error is also large, but the overall error is within 10%. 

Figure 14. The relative error of cylinder force of testing and simulation of bucket in working condition.

Figure 15. The strain test of movable arm in working condition.
Figure 16. The rigid and flexible coupling simulation model of digging arm in working condition.

Figure 17. The comparison of movable arm’s strain between simulation and experiment in working condition.

Figure 18. The relative error curve of movable arm’s strain of simulation and experiment in working condition.
5. Conclusion

1) The method to calculate the external load of the digging arm of the underground excavator truck is studied by using the virtual and real combined technology of dynamic simulation and test in this paper. The matrix equation of external load is established by considering the influence of inertial force during the movement of digging arm. And then it was solved to obtain the external load data of digging arm in the actual working process by importing the relevant data obtained from test and simulation. The calculated oil cylinder force error is within 5% and the effective strain error within 10% by verifying the obtained external load data of cylinder force and structural strain. So the correctness of the external load calculation method is proved, which provides theoretical and algorithm support for the next step of load spectrum collection, fatigue analysis and structure design optimization of the digging arm.

2) This method can be further extended to the calculation of external loads of other mechanical structures with many kinematic joints, large acceleration and complex theoretical derivation process.

References

[1] Liu Gang. New techniques of underground excavation of XCMG promote new course on the construction of rail transit [J]. Construction Machinery and Equipment, 2016, (1).
[2] Li Xiangliang. Simulation of high altitude operational arm’s performance based on ADAMSS and Workbench [J]. Journal of Hebei University of Engineering, 2017, 99–108.
[3] Wan Yunlong, XU Hongxia. Dynamic simulation of robot manipulator [J]. Computer Simulation, 2016, 33(12):346–351.
[4] Peng Dan. Dynamic simulation and finite element analysis of loader working device [o] : (Master's thesis). Chang 'an university, 2015.
[5] Zhang Weiguo, QUAN Long, CHENG Heng, et al. Transient dynamic analysis of excavator working device based on real load [J]. Journal of Mechanical Engineering, 2011, 47(12):144–149.
[6] Xiang Qingyi, LV Pengmin, WANG Binhua et al. Load Spectrum Test Method of Hydraulic Excavator Working Device[J]. China J. Highw. Transp, 2017, 30(9):151–158.
[7] CHEN H, SUN Y, WU K. Load Spectrum Testing and Analysis for Transmission System of Closed High-speed Press[J]. Procedia Engineering, 2014, 81:1645–1650.
[8] Lv Pengmin, WU Yuwen, ZHANG Bei. Parametric Stance Control System of Hydraulic Excavator[J].Road Machinery & Construction Mechanization,2017,34(3):84–86.
[9] Zhou Xiaojin. Mechanical Properties Analysis of Key Components of Excavator [D].Xi’an:Chang’ an University,2015.
[10] HOUSTON R L, LIU Youwu. The Dynamics of Multibody System: I [M].Tianjin University Press, 1987.
[11] Jin Long, LI Yuqi, ZENG Hao. The Modeling and simulation analysis of a large hydraulic
excavator based on ADAMS [J]. Modern Manufacturing Engineering, 2017, (11):144–149.

[12] Liu Qiang, GE Weimin, WANG Xiaofeng. Dynamics Modeling and Simulation of Constrained Flexible Load with Manipulator Operation [J]. Mechanical transmission, 2018, 42(3):111–116.

[13] She Yini. A Research on the Dynamic Load Characteristics of a Large Face-shovel Hydraulic Excavator Attachment based on the Muti-body Dynamics [D]. Hangzhou: Zhejiang University of Technology, 2013.

[14] Wang Xiangbing. Research on Structure System Dynamics and Characteristics of Engineering Mechanical Arm [D]. Hangzhou:Zhejiang University, 2014.

[15] Li Xu. Research on Mechanical Properties of Key Structures on Boom-type Roadheader[D].Beijing:China University of Mining & Technology, 2017.

[16] Sun Keyi. Transient Dynamic Analysis and Simulation Research Of Hydraulic Excavator Working Device [D]. Lanzhou: LANZHOU UNIVERSITY OF TECHNOLOGY, 2014.